



CONTRIBUTION TO THE VALIDATION AND VERIFICATION OF DRIVE-NEUTRAL-REVERSE SUBSYSTEM OF A CVT TRANSMISSION

PEDRO NUNO DOS SANTOS ALMEIDA

dezembro de 2017

CONTRIBUTION TO THE VALIDATION AND VERIFICATION OF DRIVE-NEUTRAL-REVERSE SUBSYSTEM OF A CVT TRANSMISSION

Pedro Nuno dos Santos Almeida

2017

Instituto Superior de Engenharia do Porto – School of Engineering, Polytechnic of Porto

Mechanical Engineering Department

CONTRIBUTION TO THE VALIDATION AND VERIFICATION OF DRIVE-NEUTRAL-REVERSE SUBSYSTEM OF A CVT TRANSMISSION

Pedro Nuno dos Santos Almeida

1120666

Dissertation presented to ISEP – School of Engineering to fulfill the requirements necessary to obtain a Master's degree in Mechanical Engineering, carried out under the guidance of Doctor Francisco José Gomes da Silva, Adjunct Professor.

2017

Instituto Superior de Engenharia do Porto – School of Engineering, Polytechnic of Porto

Mechanical Engineering Department

JURY

President

Manuel Jorge Does de Castro

Adjunct Professor, Department of Mechanical Engineering, ISEP

Advisor

Francisco José Gomes da Silva

Adjunct Professor, Department of Mechanical Engineering, ISEP

Arguing

Ricardo José Alves de Sousa

Auxiliar Professor, Department of Mechanical Engineering, University of Aveiro

ACKNOWLEDGEMENTS

I would like to start by thanking to all who contributed directly and indirectly to my personal and academic life course.

To my family and close friends who were always present even when the distance was a barrier ready to support and motivate me.

To my girlfriend, Beatriz, who was always an unconditional supporter of all my decisions and a source of courage and admiration to make me embrace new challenges.

To my university's mentor, Professor Francisco J. G. Silva, for all the academic assistance, the inexhaustible willingness to provide feedback and the motivation to me facing this challenge.

I could not forget the person who introduced me to the automotive industry, Amadeu Bastos, his knowledge, experience and advices, with no fear of sharing his vast knowledge and experience, in the internship at Simoldes Plásticos during over a year.

I would like to thank Gert Heirman who provided me the internship opportunity at Punch Powertrain NV and believed me and my potential. I would also like to thank my team at Punch Powertrain, Rudi Aerts, Wouter Tits and Ronald Vanderloo for all the experience and what they have taught me. The readiness, availability and willingness to help me to get better results and learn as much as possible are some of the factors that made this stage so important and constructive.

Finally, I would like to thank you Luiz Filipe Pereira for all the advices and to Carolien Duser for the support.

KEYWORDS

Continuously Variable Transmission, Drive-Neutral-Reverse subsystem, Planetary Gear Set, Wet clutches, Drag losses, pre-load conditions, biting-point conditions, transmissible torque, Tolerance chain stack up

ABSTRACT

Continuously Variable Transmissions (CVT) are the synonymous for fuel economy and smooth ride. Due to these major features of this type of automatic transmission (AT) it can be easily the future of automatic transmissions since the restrictions regarding to the fuel emissions are getting stricter.

The knowledge by the student related to this thematic was developed in an internship with a duration of six months at a specialized company in the development and production of CVT transmissions. By being incorporated in the department of R&D, specifically on the team of the DNR subsystem (Drive-Neutral-Reverse subsystem), it was possible to be introduced to CVT. Punch Powertrain NV in Belgium, Sint-Truiden, was the perfect opportunity to enrich the academic experience of the student, to introduce the student to development field of automotive transmissions industry from concept to testing and also to consolidate the experience in the automotive industry previously started at Simoldes Plásticos during the master degree. Firstly, it was suggested to assist to the prototype assembly process of a complete CVT followed by the analyses to the flow chart of the DNR subsystem and the manufacturability of all the components of the subsystem in order to get introduced to the subsystem. Secondly it was proposed to develop a showcase model which would be a great tool to explain people the working principle of a DNR subsystem in detail. At the same time, the student decided to start some calculations related to the DNR subsystem not only to better understand the subsystem and apply the knowledge obtained during the graduation but also to learn more specific subjects related to this field of study. This task would later allow the student to verify the mechanical requirements. After asking for quotations and getting in contact with several suppliers it was time to assemble the entire showcase getting in contact with different manufacturing processes and difficulties. At the end, it is irrefutable the amount of knowledge acquired during the internship. Undoubtedly,

the support from all the colleagues was very important by transmitting me all their knowledge and experience.

PALAVRAS CHAVE

Transmissão Continuamente Variável, Subsistema Drive-Neutral-Reverse, Conjunto de engrenagens planetárias, Pratos de embraiagem húmidos, Perdas por arraste, Condições de pré-carga, condições de biting-point, torque transmissível, Cadeia de tolerância em série

RESUMO

Transmissões Continuamente Variáveis (CVT) são sinónimo de economia de combustível e condução suave. Devido a estas grandes características deste tipo de transmissão, podemos estar facilmente perante o futuro das transmissões automáticas visto que as restrições relativamente às emissões de combustível se estão a tornar mais apertadas.

O conhecimento adquirido pelo estudante nesta temática foi desenvolvido num estágio com duração de seis meses numa empresa especializada no desenvolvimento e produção de transmissões CVT. Ao estar incorporado no departamento de I&D, mais especificamente na equipa de desenvolvimento do subsistema DNR, foi possível familiarizar-se com as Transmissões de Continuamente Variáveis. Punch Powertrain NV na Bélgica, Sint-Truiden, foi a oportunidade perfeita para enriquecer a experiência do estudante, introduzi-lo à área de desenvolvimento da indústria das transmissões automóveis desde o conceito até à fase de testes, e também para consolidar a experiência na indústria automóvel previamente iniciada num estágio na Simoldes Plásticos durante o percurso de mestrado. Em primeiro lugar, foi sugerido assistir à montagem da fase protótipo de uma CVT seguida da análise do esquema do fluxo do óleo do subsistema DNR e os processos de fabrico de todos os componentes do subsistema com vista a integrar-se no subsistema. Em segundo lugar foi proposto desenvolver um equipamento de amostra que seria uma ótima ferramenta para explicar detalhadamente às pessoas o princípio de funcionamento de um subsistema DNR. Ao mesmo tempo, o estudante decidiu começar alguns cálculos relativos ao subsistema DNR não só para melhor compreender o subsistema, mas também para aplicar o conhecimento adquirido durante o seu percurso académico com vista a aprender campos mais específicos sobre este tema de estudo. Esta tarefa iria mais tarde permitir ao estudante a verificação dos requisitos. Após de alguns pedidos de orçamentação e de

ter entrado em contacto com alguns fornecedores, era tempo da montagem do equipamento de amostra estando diretamente em contacto com diferentes processos de fabrico e respetivas dificuldades. No final, é irrefutável a proporção de conhecimento adquirido durante este estágio. Sem dúvida alguma o apoio dos colegas foi bastante importante ao transmitirem todo o seu conhecimento e experiência.

LIST OF SYMBOLS AND ABBREVIATIONS

List of abbreviations

AMT	Automated manual transmission
Assy	Assembly
AT	Automatic Transmission
BOM	Bill of materials
CAFE	Corporate Average Fuel Economy
CC	Clutch Carrier
CO	Carbon monoxide
CO ₂	Carbon dioxide
CP	Clutch plates
CVT	Continuously Variable Transmission
D	Drive Gear
DCT	Dual Clutch Transmission
DNR	Drive-Neutral-Reverse subsystem
EOP	Engine operating point
EPA	Environmental Protection Agency
EPT	Epicyclic Gear Train
EU	European Union
EUDC	Extra urban cycle
EV	Electric vehicle
FBD	Free body diagram
FEA	Finite Elements Analysis
FFOR	Fully Functional Operating Range
FL	Free length
FWD	Forward
HC	Hydrocarbons
IC	Internal combustion
ICE	Internal Combustion Engine
ID	Inner diameter
ISO	International Standard Organization
LCV	Low carbon vehicle

MT	Manual Transmission
N	Neutral Gear
NO _x	Oxides of nitrogen
NV	<i>Naamloze Vennootschap</i>
OD	Outer diameter
PGS	Planetary Gear Set
PGT	Planetary Gear Train
PHEV	Plug-in hybrid electric vehicle
R	Reverse Gear
REV	Reverse
RSS	Root Sum Square
SF	Safety Factor
SP	Spring pack
TRH	Transmission Housing
UK	United Kingdom
USA	United States of America

List of units

°C	Celsius degrees
cm	Centimetre
cP	Centipoise
g	Gram
hp	Horsepower
Hz	Hertz
K	Kelvin degrees
kg	Kilogram
km	Kilometre
m	Meter
mm	Millimetre
MPa	Megapascal
mpg	Miles per gallon
N	Newton
Nm	Newton meter (torque)
Pa	Pascal
rad	Radian
rpm	Revolutions per minute
s	Second
t	Tons

List of symbols

A	Elongation
A_F	Friction area
A_H	Hydraulic area
A_n	Extension of a bar up to the point of fracture measured over an initial gauge length of n times the bar diameter
b	Thickness
b_w	Radial wall (width) (wavy washer)
C	Spring index (compression spring)
d	Deflection
d_c	Coil diameter (compression springs)
d_m	Limit spring deflection
D_m	Mean diameter
d_{max}	Limit spring deflection
D_p	Fluid entry diameter
E	Young's Modulus or Modulus of Elasticity
F	Load
F_a	Amplitude force
F_{ax}	Axial force
F_C	Centrifugal force
F_H	Hydraulic force
FL	Free length (springs)
F_m	Midrange force
F_n	Load at height n
F_s	Load at solid height
G	Material shear modulus
h	Height
H	Unloaded spring height (Belleville washer)
h_0	Initial cone height of disc springs without ground ends (equal to free overall height, FL, minus b)
h_B	Unloaded height of truncated cone of free spring (Belleville washer)
h_p	Clearance between consecutive clutch plates
i	Ratio (gears)
ID	Inner diameter
ID_c	Compression springs inner diameter
ID_F	Inner friction surface diameter
$ID_{F\ tol}$	Inner friction surface diameter tolerance
ID_H	Inner hydraulic diameter
$ID_{H\ tol}$	Inner hydraulic diameter tolerance
IR	Inner radius
IR_F	Inner friction surface radius
IR_H	Inner hydraulic radius

K_1, K_2, K_3, K_4	Coefficients used for the calculation of fatigue disc springs
K_B	Bergsträsser factor
K_C	Wahl correction factor (compression spring)
K_W	Multiple wave factor (wavy washer)
L	Length
N	Number of springs
N_A	Number of active coils
N_P	Number of clutch pads
N_S	Number of frictional surfaces (clutches)
N_T	Total number of coils
N_W	Number of waves per turn (wavy washer)
OD	Outer diameter
OD_c	Compression springs outer diameter
OD_F	Outer friction surface diameter
$OD_{F\ tol}$	Outer friction surface diameter tolerance
OD_H	Outer hydraulic diameter
$OD_{H\ tol}$	Outer hydraulic diameter tolerance
OR	Outer radius
OR_F	Outer friction surface radius
OR_H	Outer hydraulic radius
p	Pitch
Q	Needing feed flow rate
Q_i	Input flow rate
r	Slope line
R	Radius
R_1	Inner radius of the friction plates
R_2	Outer radius of the friction plates
R_F	Mean friction radius
RF	Reaction force
r_m	Mean radius
R_m	Tensile strength
r_p	Fluid entry radius
R_p	Yield strength
$R_{p\ 0.2\%}$	0.2% offset yield strength
R_s	Equivalent radius of oil film
R_W	Effective wave radius (wavy washer)
SF_f	Safety factor against fatigue
SH	Solid height
SR	Spring rate/stiffness
S_{sa}	Shear stress amplitude limit
S_{sa}^{Gerber}	Shear stress amplitude limit according to Gerber
$S_{sa}^{Goodman}$	Shear stress amplitude limit according to Goodman

S_{sa}^{Sines}	Shear stress amplitude limit according to Sines
S_{se}	Shear endurance limit
S_{se}^{Gerber}	Shear endurance limit according to Gerber
S_{se}^{Goodman}	Shear endurance limit according to Goodman
S_{sm}	Maximum endurance strength component for infinite life
S_{su}	Torsional modulus of rupture
T	Temperature
t	Time instant
T_C	Transmissible torque
T_D	Drag torque
t_w	Wire thickness (wavy washer)
w	Width
W_H	Half wave free height (wavy washer)
W_L	Half wave length (wavy washer)
X	Fatigue stress ratio (wavy washer)
Z	Number of teeth (gears)
Z_w	Number of turns (wavy washer)
$\gamma_1, \gamma_2, \gamma_3$	Calculation coefficients (Belleville washer)
δ	Diameter ratio
ζ	Friction coefficient
$\zeta_{\max}$	Max. Friction coefficient
$\zeta_{\min}$	Min. Friction coefficient
Θ_w	Included wave angle (wavy washer)
λ	Fluid's kinematic viscosity
μ	Fluid's dynamic viscosity
ν	Poisson's ratio
ρ	Density
ρ	Density
σ	Stress
$\sigma_I, \sigma_{II}, \sigma_{III}, \sigma_{IV}, \sigma_{OM},$	Calculated material stresses at the point I to OM of the disc spring cross section
σ_s	Stress at solid height
τ_a	Shear stress amplitude
τ_m	Midrange shear stress
ω	Rotating speed

GLOSSARY OF TERMS

Allow for set	Spring is supplied longer than specified to compensate for length loss when fully compressed in assembly by customer. Recommended for large quantity order to reduce cost (Spring, 2011)
Barge	A long boat with a flat bottom, used for carrying heavy objects on rivers or canals (Cambridge Dictionary, n.d.)
Biting point	Biting point is known as the moment when there is contact between clutch plates starting to transmit torque
Commercial vehicle	Motor vehicle designed for the purpose of: – transporting people – Bus; – for transporting goods and pulling trailers – Truck; or just for pulling trailers – Tractor. This excludes passenger cars. (Naunheimer, Bertsche, Ryborz, & Novak, 2010)
Crawling	To move at an unusually slow pace
Engine Idling	Engine running with no load and the vehicle is steady
Fully functional operating range	The range of temperatures between -20 and 120 °C. This is the range that maximum torque and maximum efficiency need to be guaranteed. From -25 to -20 °C it needs to be driveable as so from 120 to 140 °C (Punch Powertrain NV)
Gradeability	Gradeability is defined as the highest grade a vehicle can ascend maintaining a particular speed (Akilesh Yamsani, 2014)
Gridlock	a situation where roads in a town become so blocked by cars that it is impossible for any traffic to move (Dictionary, n.d.)
Hybrid (vehicle)	A vehicle that can be powered by both electricity and an internal-combustion engine (Hillier, 2012)
Passenger car	Motor vehicle designed and equipped mainly for transporting people, with a maximum of nine seats. (Naunheimer <i>et al.</i> , 2010)
Preset	Presetting consists of coiling the spring to a longer length than the ultimate desired free length and then compressing it beyond its elastic limit. Presetting allows the use of higher design stresses with the added cost of the secondary operation. Some of this added cost,

	however, is sometimes offset by using less material within a given envelope. (Corp, n.d.)
Set	Length loss in operation due to high stress condition in the spring (Spring, 2011)
Tensile strength	The maximum engineering stress, in tension, that may be sustained without fracture. Often termed ultimate (tensile) strength. (William D. Callister & David G. Rethiwsch, 2010)
Torque converter	a device that transmits or multiplies the torque generated by an engine (Hillier, 2012)
Yield strength	The stress required to produce a very slight yet specified amount of plastic strain; a strain offset of 0.002 is commonly used. (William D. Callister & David G. Rethiwsch, 2010)

INDEX OF FIGURES

FIGURE 1 - DEVELOPMENT OF GOODS TRAFFIC AND POPULATION IN GERMANY; FIGURES FOR THE WHOLE OF GERMANY FROM 1990 (NAUNHEIMER ET AL., 2010).....	3
FIGURE 2 – BREAKDOWN OF JOBS DEPENDENT ON THE MOTOR TRANSPORT INDUSTRY IN GERMANY (NAUNHEIMER ET AL., 2010)	4
FIGURE 3 – OVERALL ENERGY CONVERSION PROCESS IN VEHICLE TRANSPORTATION – THE WELL-TO-WHEELS IDEA (MASHADI & CROLLA, 2012)	5
FIGURE 4 – CLASSIFICATION OF ROAD VEHICLES AND MOTOR VEHICLES TO GERMAN STANDARD DIN 70010 (NAUNHEIMER ET AL., 2010)	5
FIGURE 5 – INCREASE IN THE NUMBER OF MOTOR VEHICLES WORLDWIDE (NAUNHEIMER ET AL., 2010)	6
FIGURE 6 - COMPARISON OF PASSENGER TRANSPORT SUPPLY AND DEMAND RELATED TO LENGTH OF JOURNEY (NAUNHEIMER ET AL., 2010)	9
FIGURE 7 – EXAMPLE OF TYPICAL ENERGY FLOWS DURING URBAN (A) AND HIGHWAY (B) DRIVING (MASHADI & CROLLA, 2012)	12
FIGURE 8 – THREE TYPES OF TYPICAL HYBRID/ELECTRIC VEHICLE ARCHITECTURES AVAILABLE IN 2011 (MASHADI & CROLLA, 2012)	22
FIGURE 9 – DEFINITION OF DIRECTION OF ROTATION IN A POWERTRAIN (NAUNHEIMER ET AL., 2010) .	24
FIGURE 10 - SIGN RULES FOR ROTATIONAL SPEED, TORQUE AND POWER (NAUNHEIMER ET AL., 2010)	26
FIGURE 11 – WOLF TRANSMISSION SYMBOLS (NAUNHEIMER ET AL., 2010)	26
FIGURE 12 – MANUAL TRANSMISSION ASSEMBLY (MASHADI & CROLLA, 2012).....	28
FIGURE 13 – SYNCHRONIZING CONCEPT (MASHADI & CROLLA, 2012).....	29
FIGURE 14 – SLIDING MESH AND CONSTANT-MESH GEAR MESHING METHODS (MASHADI & CROLLA, 2012)	29
FIGURE 15 – SHIFTING MECHANISM (MASHADI & CROLLA, 2012)	30
FIGURE 16 - AUTOMATIC GEARBOX INTERNALS (HILLIER, 2012)	31
FIGURE 17 - OVERALL CONSTRUCTION OF A CONVENTIONAL AUTOMATIC (MASHADI & CROLLA, 2012)	32
FIGURE 18 – ACTUATORS FOR AMT (MASHADI & CROLLA, 2012)	33
FIGURE 19 – SCHEMATIC OF A 6-SPEED DCT (MASHADI & CROLLA, 2012).....	35
FIGURE 20 – CONCEPTUAL CVT (MASHADI & CROLLA, 2012)	36
FIGURE 21 – AUTOMOTIVE CVTS’ CLASSIFICATION (ADAPTED FROM (MASHADI & CROLLA, 2012))	37

FIGURE 22 – GEOMETRY OF A TYPICAL BELT CVT (MASHADI & CROLLA, 2012)	38
FIGURE 23 – ELEMENTS OF A THRUST LINK CHAIN AND PRINCIPLE OF OPERATION OF THE VARIATOR (NAUNHEIMER ET AL., 2010)	39
FIGURE 24 – A) SECONDARY MAP OF AN ICE WITHOUT A GEARBOX; B) SECONDARY MAP OF AN ICE WITH REAR-MOUNTED 4-SPEED GEARBOX: TRACTION DIAGRAM (NAUNHEIMER ET AL., 2010)	41
FIGURE 25 – PLANETARY GEAR SET (D. A. CROLLA, 2009).....	43
FIGURE 26 – TYPES OF SPRING WASHERS (JUVINALL & MARSHEK, 2012)	44
FIGURE 27 – SIMPLE FORM OF A DIAPHRAGM SPRING (MASHADI & CROLLA, 2012)	45
FIGURE 28 – TYPES OF ENDS FOR COMPRESSION SPRINGS (SHIGLEY ET AL., 2002)	46
FIGURE 29 – COUNTING COILS FOR COMPRESSION SPRINGS (SPRING, 2011)	47
FIGURE 30 – INFLUENCE OF THE TYPES OF ENDS OF COMPRESSION SPRINGS (SHIGLEY ET AL., 2002)	47
FIGURE 31 – TERMINOLOGY OF A WAVE SPRING (SPRING, 2011)	59
FIGURE 32 – STACKED BELLEVILLE WASHERS (JUVINALL & MARSHEK, 2012).....	64
FIGURE 33 – MAIN DIMENSIONS FOR BELLEVILLE WASHERS (SPRING, 2011)	64
FIGURE 34 – CROSS SECTION OF A SINGLE DISC SPRING, INCLUDING THE RELEVANT POINTS OF LOADING ("DIN 2093," 1992).....	67
FIGURE 35 – GRAPHICAL REPRESENTATION OF ENDURANCE LIFE OF SPRINGS WHERE THICKNESS IS LESS THAN 1,25 MM (DIN 2093, 1992)	68
FIGURE 36 – GRAPHICAL REPRESENTATION OF ENDURANCE LIFE OF SPRINGS WHERE 1,25 MM ≤ THICKNESS ≤ 6 MM (DIN 2093, 1992).....	69
FIGURE 37 – GRAPHICAL REPRESENTATION OF ENDURANCE LIFE OF SPRINGS WHERE 6 MM < THICKNESS < 14 MM (DIN 2093, 1992)	69
FIGURE 38 - EFFECT OF CENTRIFUGAL FORCE ON A FLUID (WHITE, 2010)	72
FIGURE 39 – PLATE-VIEW FLOW PATTERN IN THE GAP AT DIFFERENT ROTATING SPEED (SHIHUA & ZENGXIONG, 2010)	75
FIGURE 40 – FLOW PATTERN IN THE CLEARANCE AT DIFFERENT ROTATING SPEED (SHIHUA & ZENGXIONG, 2010)	76
FIGURE 41 – COMPARISON BETWEEN DIFFERENT MATHEMATICAL MODELS AND EXPERIMENTAL TESTS RESULTS (SHIHUA & ZENGXIONG, 2010)	77
FIGURE 42 – SCHEMATIC OF DISENGAGED WET CLUTCH (YUAN SHIHUA, PENG ZENGXIONG, 2010).....	78
FIGURE 43 – NEEDING FEED FLOW RATE CURVE FOR FULL OIL FILM (HEYAN ET AL., 2013)	80
FIGURE 44 – SCHEMATIC DIAGRAM OF PARTIAL OIL FILMING IN THE CLEARANCE (HEYAN ET AL., 2013)	81

FIGURE 45 – EQUIVALENT RADIUS CURVE WITH ROTATING SPEED (HEYAN ET AL., 2013)	82
FIGURE 46 – DRAG TORQUE OF SINGLE FRICTION PAIR (HEYAN ET AL., 2013).....	83
FIGURE 47 – GEAR SET WITH FIXED AXLES AND EPICYCLIC GEAR (NAUNHEIMER ET AL., 2010)	84
FIGURE 48 – THE MECHANICS OF A CAM OF ARBITRARY CONTOUR (HENRIKSEN, 1973)	86
FIGURE 49 – THE MECHANICS OF AN ARCHIMEDES SPIRAL CAM (HENRIKSEN, 1973)	89
FIGURE 50 - THE MECHANICS OF THE ECCENTRIC CAM (HENRIKSEN, 1973)	90
FIGURE 51 - CROSS SECTION OF THE CVT (LEFT SIDE) (COURTESY OF PUNCH POWERTRAIN NV).....	95
FIGURE 52 – DNR LOCATION INSIDE THE CVT (COURTESY OF PUNCH POWERTRAIN NV)	96
FIGURE 53 – LOCATION OF EXTERNAL COMPONENTS IN THE TRANSMISSION (COURTESY OF PUNCH POWERTRAIN).....	97
FIGURE 54 - DNR SCHEMATICS (COURTESY OF PUNCH POWERTRAIN NV).....	98
FIGURE 55 – PGS SCHEME FOR DRIVE GEAR	99
FIGURE 56 – PGS SCHEME FOR REVERSE GEAR.....	100
FIGURE 57 - PGS SCHEME FOR NEUTRAL GEAR.....	100
FIGURE 58 – EXPLODED VIEW OF THE TURBINE SHAFT ASSY (COURTESY OF PUNCH POWERTRAIN NV)	102
FIGURE 59 – EXPLODED VIEW OF THE PGS (COURTESY OF PUNCH POWERTRAIN NV)	103
FIGURE 60 – BOM OF THE DNR COMPONENTS TO BE ASSEMBLED IN THE CVT (COURTESY OF PUNCH POWERTRAIN NV)	105
FIGURE 61 – TORQUE CONVERTER AND DNR INSIDE THE CVT (COURTESY OF PUNCH POWERTRAIN NV)	106
FIGURE 62 - PISTON ASSEMBLED IN THE CC (COURTESY OF PUNCH POWERTRAIN NV)	107
FIGURE 63 - SPLINES MATCHING BETWEEN THE FWD SEPARATOR PLATE (WHITE) AND THE CC (BLACK) (COURTESY OF PUNCH POWERTRAIN NV).....	107
FIGURE 64 – FWD CLUTCH PLATE (COURTESY OF PUNCH POWERTRAIN NV).....	108
FIGURE 65 – MATCHING BETWEEN SUN GEAR (RED) AND CLUTCH PLATES (COURTESY OF PUNCH POWERTRAIN NV)	108
FIGURE 66 – SUN GEAR (COURTESY OF PUNCH POWERTRAIN NV)	108
FIGURE 67 – MATCHING BETWEEN FORWARD ENDPLATE AND CLUTCH PLATES (COURTESY OF PUNCH POWERTRAIN NV)	109
FIGURE 68 – PGS: CROSS SECTION (SIDE AND FRONT VIEW) (COURTESY OF PUNCH POWERTRAIN NV)	109
FIGURE 69 – REV PISTON LOCATION (COURTESY OF PUNCH POWERTRAIN NV).....	110

FIGURE 70 – SPLINE MATCHING BETWEEN TRANSMISSION HOUSING AND REV SEPARATOR PLATE (COURTESY OF PUNCH POWERTRAIN NV).....	110
FIGURE 71 – SPLINE MATCHING BETWEEN REV CLUTCH PLATES AND PGS CARRIER (COURTESY OF PUNCH POWERTRAIN NV)	111
FIGURE 72 - SPLINE MATCHING BETWEEN ENDPLATE REVERSE CLUTCH AND TRANSMISSION HOUSING (COURTESY OF PUNCH POWERTRAIN NV).....	111
FIGURE 73 – HYDRAULIC CIRCUIT SCHEME (COURTESY OF PUNCH POWERTRAIN NV).....	112
FIGURE 74 – CHANNELS DEDICATED TO DNR OIL FLOW OF THE STATOR PLATE (COURTESY OF PUNCH POWERTRAIN NV)	113
FIGURE 75 – LOCATION OF CROSS SECTION A (COURTESY OF PUNCH POWERTRAIN NV)	114
FIGURE 76 – CROSS SECTION A (COURTESY OF PUNCH POWERTRAIN NV)	114
FIGURE 77 – LOCATION OF CROSS SECTION B (COURTESY OF PUNCH POWERTRAIN NV)	115
FIGURE 78 – CROSS SECTION B (COURTESY OF PUNCH POWERTRAIN NV)	115
FIGURE 79 – STATOR PLATE’S LUBRICATION OUTPUTS (COURTESY OF PUNCH POWERTRAIN NV)	116
FIGURE 80 - LOCATION OF THE STATOR PLATE'S OIL OUTPUTS (COURTESY OF PUNCH POWERTRAIN NV)	116
FIGURE 81 – CROSS SECTION G (COURTESY OF PUNCH POWERTRAIN NV)	117
FIGURE 82 – LOCATION OF CUTTING SECTION G (COURTESY OF PUNCH POWERTRAIN NV)	117
FIGURE 83 – DETAIL OF THE SPROCKET (COURTESY OF PUNCH POWERTRAIN NV)	118
FIGURE 84 - LOCATION OF CUTTING SECTION H (COURTESY OF PUNCH POWERTRAIN NV).....	119
FIGURE 85 – CROSS SECTION H (COURTESY OF PUNCH POWERTRAIN NV)	119
FIGURE 86 – TURBINE SHAFT ASSY (PART OF) (COURTESY OF PUNCH POWERTRAIN NV)	120
FIGURE 87 - OIL FLOW ON CUTTING SECTION H (FRONT AND PERSPECTIVE VIEW RESPECTIVELY) (COURTESY OF PUNCH POWERTRAIN NV).....	120
FIGURE 88 – FLOW EXITS PRESENT ON CROSS SECTION I (COURTESY OF PUNCH POWERTRAIN NV)	121
FIGURE 89 - OIL FLOW INSIDE COMPENSATION CHAMBER (COURTESY OF PUNCH POWERTRAIN NV) ..	121
FIGURE 90 – COMPENSATION CHAMBER FLOW HOLES (COURTESY OF PUNCH POWERTRAIN NV).....	122
FIGURE 91 – LOCATION OF CROSS SECTION I (COURTESY OF PUNCH POWERTRAIN NV)	122
FIGURE 92 – OIL FLOW INSIDE COMPENSATION CHAMBER (COURTESY OF PUNCH POWERTRAIN NV) ..	123
FIGURE 93 – CUTTING SECTION J	123
FIGURE 94 - DETAIL OF PERSPECTIVE VIEW OF CUTTING SECTION J (COURTESY OF PUNCH POWERTRAIN NV)	124

FIGURE 95 - OIL FLOW ON CUTTING SECTION J (COURTESY OF PUNCH POWERTRAIN NV)	124
FIGURE 96 – LOCATION OF CROSS SECTION J (COURTESY OF PUNCH POWERTRAIN NV)	125
FIGURE 97 - GAP BETWEEN COMPENSATION PISTON AND SNAP RING (COURTESY OF PUNCH POWERTRAIN NV)	125
FIGURE 98 – OIL FLOWING THROUGH THE SUN GEAR (COURTESY OF PUNCH POWERTRAIN NV).....	126
FIGURE 99 - SUN GEAR OIL OUTPUTS: AXIAL FLOW DIRECTION OUTPUTS (A) AND RADIAL FLOW DIRECTION OUTPUTS (B) (COURTESY OF PUNCH POWERTRAIN NV)	127
FIGURE 100 – CROSS SECTION K (COURTESY OF PUNCH POWERTRAIN NV)	127
FIGURE 101 – PERSPECTIVE VIEW OF CUTTING SECTION K (COURTESY OF PUNCH POWERTRAIN NV) ..	128
FIGURE 102 – LOCATION OF CUTTING SECTION K (COURTESY OF PUNCH POWERTRAIN NV)	128
FIGURE 103 - OIL FLOW IMMEDIATELY AFTER LEAVING SUN GEAR (COURTESY OF PUNCH POWERTRAIN NV)	129
FIGURE 104 – CUTTING SECTION L (COURTESY OF PUNCH POWERTRAIN NV).....	130
FIGURE 105 – PERSPECTIVE VIEW OF CROSS SECTION L (COURTESY OF PUNCH POWERTRAIN NV)	130
FIGURE 106 – LOCATION OF CUTTING SECTION L (COURTESY OF PUNCH POWERTRAIN NV).....	131
FIGURE 107 - OIL FLOW WHEN LEAVING PGS: INCORRECT OIL FLOW (A) AND CORRECT OIL FLOW (B) COURTESY OF PUNCH POWERTRAIN NV)	132
FIGURE 108 – OIL FLOW ON THRUST BEARINGS (COURTESY OF PUNCH POWERTRAIN NV).....	132
FIGURE 109 - OIL PATH WHICH COMES FROM THE PRIMARY (OUTPUT) SHAFT (COURTESY OF PUNCH POWERTRAIN NV)	133
FIGURE 110 - HIGH PRESSURE CHAMBER BETWEEN TURBINE SHAFT AND PRIMARY SHAFT (COURTESY OF PUNCH POWERTRAIN NV)	133
FIGURE 111 – RESUME OF THE LUBRICANT FLOW (COURTESY OF PUNCH POWERTRAIN NV).....	134
FIGURE 112 - STATOR PLATE’S CHANNELS FOR FORWARD CLUTCH OIL CIRCUIT (COURTESY OF PUNCH POWERTRAIN NV)	135
FIGURE 113 – STATOR PLATE’S CHANNELS FOR FORWARD CLUTCH OIL CIRCUIT (COURTESY OF PUNCH POWERTRAIN NV)	135
FIGURE 114 – CROSS SECTION D (COURTESY OF PUNCH POWERTRAIN NV)	136
FIGURE 115 – LOCATION OF CROSS SECTION D (COURTESY OF PUNCH POWERTRAIN NV)	136
FIGURE 116 – OIL FLOW ON CROSS SECTION D (COURTESY OF PUNCH POWERTRAIN NV)	137
FIGURE 117 - HOLES FOR OIL FLOW IN STATOR PLATE (A) AND TURBINE SHAFT ASSEMBLY (B) (COURTESY OF PUNCH POWERTRAIN NV)	137

FIGURE 118 - PERSPECTIVE VIEW OF TURBINE SHAFT ASSY AND STATOR PLATE (COURTESY OF PUNCH POWERTRAIN NV)	138
FIGURE 119 - CHAMBER BETWEEN TURBINE SHAFT ASSEMBLY AND FORWARD PISTON (COURTESY OF PUNCH POWERTRAIN NV)	139
FIGURE 120 – REVERSE PISTON HOUSING (COURTESY OF PUNCH POWERTRAIN NV)	140
FIGURE 121 - PERSPECTIVE VIEW OF THE INLETS/EXITS OF TRANSMISSION HOUSING: CHANNEL AVAILABLE ONLY FOR PROTOTYPE (A) AND SERIAL CHANNEL (B) (COURTESY OF PUNCH POWERTRAIN NV) .	140
FIGURE 122 – OIL PATH INSIDE REVERSE CHAMBER (COURTESY OF PUNCH POWERTRAIN NV).....	141
FIGURE 123 – ILLUSTRATION OF THE PRESSURE INSIDE REVERSE CHAMBER (COURTESY OF PUNCH POWERTRAIN NV)	141
FIGURE 124 – DRIVE GEAR (COURTESY OF PUNCH POWERTRAIN NV)	143
FIGURE 125 – NEUTRAL GEAR (COURTESY OF PUNCH POWERTRAIN NV)	144
FIGURE 126 – REVERSE GEAR (COURTESY OF PUNCH POWERTRAIN NV).....	145
FIGURE 127 – THRUST BEARINGS’ CROSS SECTION.....	146
FIGURE 128 - THRUST BEARINGS CORRECT ORIENTATION	147
FIGURE 129 – GRAPH OF PRE-LOAD CONDITIONS FOR FWD CLUTCH	153
FIGURE 130 – GRAPH OF PRE-LOAD CONDITIONS OF REV CLUTCH SET	162
FIGURE 131 – RESULTING GRAPH FOR FATIGUE CYCLE LIFE STUDY OF THE DISC SPRING (ADAPTED FROM (“DIN 2093,” 1992))	166
FIGURE 132 – GRAPH OIL DENSITY VS TEMPERATURE.....	168
FIGURE 133 - GRAPH OIL KINEMATIC VISCOSITY VS TEMPERATURE	168
FIGURE 134 - OIL DYNAMIC VISCOSITY VS TEMPERATURE	171
FIGURE 135 – SIGNS CONVENTION FOR THE CALCULATIONS RELATED TO FWD CLUTCH SET	172
FIGURE 136 - FWD SPRINGS ASSOCIATION SCHEMATIC	174
FIGURE 137 – GRAPH RELATING ASSOCIATION SPRINGS’ STIFFNESS AND TOTAL TRAVELLING	176
FIGURE 138 - GRAPH OF TRAVELLING VS REACTION FORCE	177
FIGURE 139 - GRAPH OF PRESSURE VS AXIAL FORCE OF NOMINAL SCENARIO FOR FWD PISTON	178
FIGURE 140 - GRAPH OF TRANSMISSIBLE TORQUE VS PRESSURE FOR THE NOMINAL CASE OF FWD CLUTCH	182
FIGURE 141 - GRAPH OF TRANSMISSIBLE TORQUE VS PRESSURE FOR THE SECOND CASE OF FWD PISTON	183

FIGURE 142 - GRAPH OF TRANSMISSIBLE TORQUE VS PRESSURE FOR THE THIRD CASE OF FWD PISTON	184
FIGURE 143 - GRAPH OF TRANSMISSIBLE TORQUE VS PRESSURE FOR THE FOURTH CASE OF FWD PISTON	185
FIGURE 144 – TRANSMISSIBLE TORQUE SCENARIO COMPARISON FOR FWD CLUTCH SET	186
FIGURE 145 – TRAVELLING VS PRESSURE COMPARISON BETWEEN SCENARIOS	187
FIGURE 146 – GRAPH FORCE VS TRAVELLING OF THE SPRINGS ASSOCIATION FOR 3 RD AND 4 TH SCENARIOS OF FWD CLUTCH SET	188
FIGURE 147 - RESULTING CHART OF TRANSMISSIBLE TORQUE FOR A SLIPPING SPEED OF 100 RPM	190
FIGURE 148 – RESULTING CHART OF TRANSMISSIBLE TORQUE FOR A SLIPPING SPEED OF 200 RPM	190
FIGURE 149 – RESULTING GRAPH OF THE COMPARISON OF TRANSMISSIBLE TORQUE FOR DIFFERENT SLIPPING SPEEDS	192
FIGURE 150 – GRAPH OF THE EFFECT OF ROTATING SPEED ON BALANCE DAM CENTRIFUGAL FORCE ..	193
FIGURE 151 - EFFECT OF ROTATING SPEED ON FWD PISTON CENTRIFUGAL FORCE	194
FIGURE 152 – EFFECT OF ROTATING SPEED ON RESULTING CENTRIFUGAL FORCE	194
FIGURE 153 – INFLUENCE OF OIL DENSITY IN THE CENTRIFUGAL FORCE OF FWD PISTON	195
FIGURE 154 – INFLUENCE OF FLUID ENTRY RADIUS IN THE CENTRIFUGAL FORCE OF FWD PISTON	196
FIGURE 155 – INFLUENCE OF FLUID ENTRY RADIUS IN THE CENTRIFUGAL FORCE OF BD	196
FIGURE 156 – GRAPH OF MINIMUM TRANSMISSIBLE TORQUE FOR NOMINAL SCENARIO OF FWD CLUTCH	197
FIGURE 157 - GRAPH OF AXIAL FORCE VS PRESSURE FOR NOMINAL SCENARIO OF FWD CLUTCH	198
FIGURE 158 - GRAPH OF AXIAL FORCE VS PRESSURE FOR SECOND SCENARIO OF FWD CLUTCH	199
FIGURE 159 - GRAPH OF MINIMUM TRANSMISSIBLE TORQUE FOR SECOND SCENARIO OF FWD CLUTCH	199
FIGURE 160 - GRAPH OF MINIMUM TRANSMISSIBLE TORQUE FOR THIRD SCENARIO OF FWD CLUTCH	200
FIGURE 161 - GRAPH OF AXIAL FORCE VS PRESSURE FOR 3 RD SCENARIO OF FWD CLUTCH	200
FIGURE 162 - GRAPH OF MINIMUM TRANSMISSIBLE TORQUE FOR FOURTH SCENARIO OF FWD CLUTCH	201
FIGURE 163 - GRAPH OF AXIAL FORCE VS PRESSURE FOR 4 TH SCENARIO OF FWD CLUTCH	201
FIGURE 164 - EQUIVALENT RADIUS CURVE WITH SPEED OF DRAG LOSSES' MODEL VALIDATION	203
FIGURE 165 - NEEDING FEED FLOW RATE CURVE FOR FULL OIL FILM OF DRAG LOSSES' MODEL VALIDATION	203

FIGURE 166 - DRAG TORQUE OF SINGLE PAIR OF DRAG LOSSES' MODEL VALIDATION	204
FIGURE 167 – GRAPH OF NEEDING FEED FLOW RATE FOR FULL OIL FILM.....	206
FIGURE 168 - PRESSURE PORTIONS VS ROTATING SPEED FOR 0,3 L/MIN INPUT FLOW RATE.....	206
FIGURE 169 - PRESSURE PORTIONS VS ROTATING SPEED FOR DIFFERENT INPUT FLOW RATES	207
FIGURE 170 – DRAG LOSSES FOR ALL PLATES OF FWD CLUTCH.....	208
FIGURE 171 – DRAG LOSSES OF A SINGLE PLATE ON FWD CLUTCH.....	208
FIGURE 172 – NEEDING FEED FLOW RATE FULL OIL FILM FOR DIFFERENT ΔP	209
FIGURE 173 - EQUIVALENT RADIUS VS ROTATING SPEED FOR DIFFERENT FEED FLOW RATES	210
FIGURE 174 – EQUIVALENT RADIUS VS ROTATING SPEED FOR DIFFERENT VALUES OF ΔP	211
FIGURE 175 - NEEDING FEED FLOW RATE FOR FULL OIL FILM VS ROTATING SPEED FOR DIFFERENT OIL TEMPERATURES	212
FIGURE 176 - DISSIPATED POWER VS ROTATING SPEED FOR FWD CLUTCH	213
FIGURE 177 – INFLUENCE OF OIL TEMPERATURE IN THE DRAG TORQUE	214
FIGURE 178 – GRAPH OF COMPARISON OF CLUTCH PLATES' PRESSURE OF FWD CLUTCH SET	216
FIGURE 179 - GRAPH OF COMPARISON OF CLUTCH PLATES' PRESSURE OF REV CLUTCH SET.....	217
FIGURE 180 – GRAPH RELATING SPRINGS' ASSOCIATION'S STIFFNESS AND TOTAL TRAVELLING	219
FIGURE 181 – GRAPH RELATING TRAVELLING AND REACTION FORCE OF THE SPRINGS' ASSOCIATION .	219
FIGURE 182 – GRAPH OF TRANSMISSIBLE TORQUE VS PRESSURE FOR THE NOMINAL SCENARIO OF REV CLUTCH SET.....	223
FIGURE 183 - GRAPH OF TRANSMISSIBLE TORQUE VS PRESSURE FOR THE SECOND SCENARIO OF REV CLUTCH SET.....	224
FIGURE 184 - GRAPH OF TRANSMISSIBLE TORQUE VS PRESSURE FOR THE THIRD SCENARIO OF REV CLUTCH SET.....	225
FIGURE 185 – GRAPH SPRING STIFFNESS VS TRAVELLING FOR ALL THE SCENARIO OF REV CLUTCH SET	226
FIGURE 186 – TRANSMISSIBLE TORQUE SCENARIOS' COMPARISON FOR REV CLUTCH SET	226
FIGURE 187 – GRAPH FORCE VS TRAVELLING OF THE SPRINGS' ASSOCIATION FOR ALL THE SCENARIOS OF REV CLUTCH SET.....	227
FIGURE 188 – GRAPH OF MINIMUM TRANSMISSIBLE TORQUE VS HYDRAULIC PRESSURE FOR NOMINAL SCENARIO OF REV CLUTCH SET	228
FIGURE 189 – GRAPH OF AXIAL FORCE VS HYDRAULIC PRESSURE FOR NOMINAL SCENARIO OF REV CLUTCH SET.....	229

FIGURE 190 - GRAPH OF MINIMUM TRANSMISSIBLE TORQUE VS HYDRAULIC PRESSURE FOR SECOND SCENARIO OF REV CLUTCH SET	230
FIGURE 191 - GRAPH OF AXIAL FORCE VS HYDRAULIC PRESSURE FOR SECOND SCENARIO OF REV CLUTCH SET.....	230
FIGURE 192 - GRAPH OF AXIAL FORCE VS HYDRAULIC PRESSURE FOR THIRD SCENARIO OF REV CLUTCH SET.....	231
FIGURE 193 - GRAPH OF MINIMUM TRANSMISSIBLE TORQUE VS HYDRAULIC PRESSURE FOR THIRD SCENARIO OF REV CLUTCH SET	231
FIGURE 194 – GRAPH OF NEEDING FEED FLOW RATE FOR FULL OIL FILM OF REV CLUTCH	233
FIGURE 195 – PRESSURE PORTIONS VS ROTATING SPEED FOR 0,45 L/MIN INPUT FLOW RATE OF REV CLUTCH	234
FIGURE 196 – PRESSURE PORTIONS VS ROTATING SPEED FOR DIFFERENT INPUT FLOW RATES	235
FIGURE 197 – GRAPH OF DRAG LOSSES FOR A SINGLE PLATE CALCULATED ON THE REV CLUTCH	236
FIGURE 198 – GRAPH OF DRAG LOSSES FOR A SINGLE PLATE CALCULATED ON THE PRIMARY SHAFT ...	236
FIGURE 199 – GRAPH OF DRAG LOSSES OF ALL THE PLATES CALCULATED ON THE REV CLUTCH SET.....	237
FIGURE 200 – GRAPH OF DRAG LOSSES FOR ALL THE PLATES CALCULATED ON THE PRIMARY SHAFT ...	237
FIGURE 201 – NEEDING FEED FLOW RATE FOR DIFFERENT VALUES OF ΔP	238
FIGURE 202 – EQUIVALENT RADIUS VS ROTATING SPEED FOR DIFFERENT FEED FLOW RATES OF REV CLUTCH	239
FIGURE 203 – EQUIVALENT RADIUS VS ROTATING SPEED FOR DIFFERENT VALUES OF ΔP FOR REV CLUTCH SET.....	240
FIGURE 204 – DRAG LOSSES FOR ALL PLATES IN REV CLUTCH	241
FIGURE 205 – NEEDING FEED FLOW RATE TO ENSURE FULL OIL FILM AFFECTED BY DIFFERENT WORKING TEMPERATURES IN REV CLUTCH.....	241
FIGURE 206 – DRAG LOSSES FOR ALL PLATES CALCULATED ON THE PRIMARY SHAFT FOR REV CLUTCH	242
FIGURE 207 – DISSIPATED POWER THROUGH DRAG LOSSES FOR ALL THE PLATES CALCULATED ON THE PRIMARY SHAFT FOR REV CLUTCH.....	243
FIGURE 208 - TOLERANCE CHAIN STACK UP OF FWD CLUTCH SET	245
FIGURE 209 - TOLERANCE CHAIN STACK UP OF REV CLUTCH SET	247
FIGURE 210 – PGS CONTACT POINTS BETWEEN PLANET GEARS AND SUN AND RING GEARS.....	249
FIGURE 211 – LINEAR SPEED RELATIVE TO THE CENTRE OF THE PGS IN D GEAR.....	250
FIGURE 212 – ROTATING SPEEDS ON D RELATIVE TO THE CENTRE OF THE PGS IN D GEAR	250

FIGURE 213 – GRAPH OF ROTATING SPEED OF PGS GEARS VS ROTATING SPEED OF THE TURBINE SHAFT IN N.....	252
FIGURE 214 – GRAPH OF ROTATING SPEED OF PGS GEARS VS THE ROTATING SPEED OF THE TURBINE SHAFT IN N.....	252
FIGURE 215 - GRAPH OF ROTATING SPEED OF PGS GEARS VS ROTATING SPEED OF THE TURBINE SHAFT IN N.....	253
FIGURE 216 – ANGULAR SPEEDS RELATIVE TO GEARS' CENTRES OF R	254
FIGURE 217 – ANGULAR SPEEDS RELATIVE TO POINT O IN R.....	254
FIGURE 218 – PLANET GEARS' LINEAR SPEED RELATIVE TO POINT O IN R.....	255
FIGURE 219 - ILLUSTRATION OF TORQUE ORIENTATION ON THE PGS IN D.....	256
FIGURE 220 – SCHEMATIC OF THE RESULTING TORQUES ON R IN THE PGS.....	257
FIGURE 221 – FBD OF PLANETARY GEAR.....	258
FIGURE 222 – COMPARISON OF AVERAGE TRANSMISSIBLE TORQUE BETWEEN FWD AND REV CLUTCH SETS	260
FIGURE 223 – COMPARISON OF DRAG LOSSES BETWEEN FWD AND REV CLUTCH SETS FOR NOMINAL FEED FLOW RATE	261
FIGURE 224 – COMPARISON OF DRAG LOSSES CALCULATED ON THE PRIMARY SHAFT BETWEEN FWD AND REV CLUTCH SETS FOR NOMINAL FEED FLOW RATE	262
FIGURE 225 - COMPARISON OF DRAG LOSSES BETWEEN FWD AND REV CLUTCH SETS FOR MAXIMUM FEED FLOW RATE	263
FIGURE 226 - COMPARISON OF DRAG LOSSES CALCULATED ON THE PRIMARY SHAFT BETWEEN FWD AND REV CLUTCH SETS FOR MAXIMUM FEED FLOW RATE	263
FIGURE 227 – COMPARISON OF CLUTCH PLATES' PRESSURE BETWEEN FWD AND REV CLUTCH SETS ...	264
FIGURE 228 – RESULTS OF THE RAMP-DOWN TEST FOR FWD CLUTCH INSTALLED.....	266
FIGURE 229 – RESULTS OF THE RAMP-DOWN TEST FOR REV CLUTCH INSTALLED	267
FIGURE 230 – RESULTS OF THE RAMP-DOWN TEST FOR BOTH CLUTCHES INSTALLED.....	268
FIGURE 231 – FWD CLUTCH DRAG TORQUE FOR COMPARISON WITH RAMP-DOWN TEST	269
FIGURE 232 - REV CLUTCH DRAG TORQUE FOR COMPARISON WITH RAMP-DOWN TEST	269
FIGURE 233 – DRAG LOSSES ON FWD CLUTCH IN D FOR THE VALIDATION OF THE RESPECTIVE REQUIREMENT.....	280
FIGURE 234 – GRAPH OF THE DRAG LOSSES ON REV CLUTCH CALCULATED ON THE INPUT SHAFT FOR A SINGLE PLATE	280

FIGURE 235 – GRAPH OF TRAVELLING OF THE CAMS VS TIME OF THE SHOWCASE	286
FIGURE 236 – TERMINOLOGY FOR THE CALCULATIONS OF FWD LEVER OF THE SHOWCASE	287
FIGURE 237 – ILLUSTRATION OF TRANSMISSION OF MOTION TO THE CAM.....	289
FIGURE 238 - PERSPECTIVE VIEW OF FWD STOPPER.....	290
FIGURE 239 – FWD CAM MECHANISM.....	290
FIGURE 240 – FBD OF FWD PRESSING PLATE SYSTEM	292
FIGURE 241 – REV CAM MECHANISM	293
FIGURE 242 – FBD OF REV PRESSING PLATE SYSTEM.....	295
FIGURE 243 – FITTING BETWEEN FWD CAM AND BEARING	298
FIGURE 244 – ILLUSTRATION OF FITTING BETWEEN BEARING AND MAIN PLATE	299
FIGURE 245 – FITTING BETWEEN BUSHING SUPPORT AND HANDWHEEL.....	300
FIGURE 246 – FITTING BETWEEN BALL TRANSFER UNIT’S ADAPTER AND DNR FWD PISTON.....	301
FIGURE 247 – FITTING BETWEEN GEAR SELECTOR’S JOINING AND SHOULDER SCREW	301
FIGURE 248 – FITTING BETWEEN GEAR SELECTOR’S SUPPORT AND SHOULDER SCREW	302
FIGURE 249 - FITTING BETWEEN RING AND FWD PRESSING PLATE.....	303
FIGURE 250 – FITTING BETWEEN BUSHING SUPPORT AND BUSHING	303
FIGURE 251 – REV PRESSING PLATE	305
FIGURE 252 – CAVITIES OF REV PISTON	305
FIGURE 253 – SCHEMATIC OF SIMULATION CONDITIONS OF THE COMPONENT PNSA3401_2	306
FIGURE 254 – LOCATION OF VON MISES MAX. STRESS AREA	307
FIGURE 255 – BALL TRANSFER UNITS’ ADAPTER (LEFT) AND APPLICATION (RIGHT)	308
FIGURE 256 – BOUNDARY CONDITIONS OF COMPONENT PNSA1203_3	309
FIGURE 257 – DEFORMED SHAPE OF THE ADAPTER (SCALE: 2.3×10^4).....	309
FIGURE 258 – FWD PRESSING PLATE ASSY.....	311
FIGURE 259 – BOUNDARY CONDITIONS OF COMPONENT PNSA3302_5	311
FIGURE 260 – MAX. STRESS REGIONS OF FEA OF COMPONENT PNSA3302_5 (SCALE: 2.2×10^2)	312
FIGURE 261 - REV PRESSING PLATE ASSY	313
FIGURE 262 – BOUNDARY CONDITIONS OF THE FEA OF COMPONENT PNSA3401_1.....	314
FIGURE 263 - MAX. STRESS REGIONS OF THE FEA OF COMPONENT PNSA3401_1.....	315
FIGURE 264 – FRICTION SURFACES OF AS CLUTCH PLATE (MASHADI & CROLLA, 2012).....	333

FIGURE 265 – MAJOR COMPONENTS OF WET CLUTCH SYSTEM (D. CROLLA ET AL., 2014)	335
FIGURE 266 – SHAPES OF GROOVES OF FRICTION PLATES (D. CROLLA ET AL., 2014)	336
FIGURE 267 – SCHEMATIC OF A FLUID COUPLING (MASHADI & CROLLA, 2012)	337
FIGURE 268 – STRUCTURE OF A RADIAL NEEDLE ROLLER BEARING (D. CROLLA ET AL., 2014)	338
FIGURE 269 – DIFFERENT DESIGN VARIANTS OF AXIAL NEEDLE ROLLER BEARINGS AND CYLINDRICAL ROLLER THRUST BEARINGS (D. CROLLA ET AL., 2014)	339
FIGURE 270 – SHOWCASE ASSEMBLY PROCESS: STEP 1	414
FIGURE 271 – SHOWCASE ASSEMBLY PROCESS: STEP 3	415
FIGURE 272 – SHOWCASE ASSEMBLY PROCESS: STEP 5	415
FIGURE 273 – SHOWCASE ASSEMBLY PROCESS: STEP 6	416
FIGURE 274 – SHOWCASE ASSEMBLY PROCESS: STEP 7	416
FIGURE 275 – SHOWCASE ASSEMBLY PROCESS: STEP 8	417
FIGURE 276 – SHOWCASE ASSEMBLY PROCESS: STEP 9	417
FIGURE 277 – SHOWCASE ASSEMBLY PROCESS: STEP 10	418
FIGURE 278 – SHOWCASE ASSEMBLY PROCESS: STEP 11	418
FIGURE 279 – SHOWCASE ASSEMBLY PROCESS: STEP 12	419
FIGURE 280 – SHOWCASE ASSEMBLY PROCESS: STEP 13	419
FIGURE 281 – SHOWCASE ASSEMBLY PROCESS: STEP 14	420
FIGURE 282 – SHOWCASE ASSEMBLY PROCESS: STEP 15	420
FIGURE 283 – SHOWCASE ASSEMBLY PROCESS: STEP 16	421
FIGURE 284 – SHOWCASE ASSEMBLY PROCESS: STEP 17	421
FIGURE 285 – SHOWCASE ASSEMBLY PROCESS: STEP 18	422
FIGURE 286 – SHOWCASE ASSEMBLY PROCESS: STEP 19	423
FIGURE 287 – SHOWCASE ASSEMBLY PROCESS: STEP 20	423
FIGURE 288 – SHOWCASE ASSEMBLY PROCESS: STEP 21	424
FIGURE 289 – SHOWCASE ASSEMBLY PROCESS: STEP 23	424
FIGURE 290 – SHOWCASE ASSEMBLY PROCESS: STEP 24	425
FIGURE 291 – SHOWCASE ASSEMBLY PROCESS: STEP 25	426
FIGURE 292 – SHOWCASE ASSEMBLY PROCESS: STEP 27	427
FIGURE 293 – SHOWCASE ASSEMBLY PROCESS: STEP 26	427

FIGURE 294 - SHOWCASE ASSEMBLY PROCESS: STEP 29	428
FIGURE 295 - SHOWCASE ASSEMBLY PROCESS: STEP 30	429
FIGURE 296 - SHOWCASE ASSEMBLY PROCESS: STEP 31	429
FIGURE 297 - SHOWCASE ASSEMBLY PROCESS: STEP 32	430
FIGURE 298 - SHOWCASE ASSEMBLY PROCESS: STEP 33	431
FIGURE 299 - SHOWCASE ASSEMBLY PROCESS: STEP 34	431
FIGURE 300 - SHOWCASE ASSEMBLY PROCESS: STEP 37	432
FIGURE 301 - SHOWCASE ASSEMBLY PROCESS: STEP 36	432
FIGURE 302 - SHOWCASE ASSEMBLY PROCESS: STEP 39	433
FIGURE 303 - SHOWCASE ASSEMBLY PROCESS: STEP 40	433
FIGURE 304 - SHOWCASE ASSEMBLY PROCESS: STEP 41	433
FIGURE 305 - SHOWCASE ASSEMBLY PROCESS: STEP 42	433
FIGURE 306 - SHOWCASE ASSEMBLY PROCESS: STEP 43	434
FIGURE 307 – SHOWCASE ASSEMBLY PROCESS: FINAL STEP	434

INDEX OF TABLES

TABLE 1 – TRANSMISSION-ORIENTED CLASSIFICATION OF MOTOR VEHICLES BY TYPE OF VEHICLE AND TYPE OF USE (NAUNHEIMER ET AL., 2010)	11
TABLE 2 – GEAR RATIOS FOR DIFFERENT POWER SOURCES (MASHADI & CROLLA, 2012)	41
TABLE 3 – SPRING INDEX OF COMPRESSION SPRINGS (ADAPTED FROM (SPRING, 2011))	50
TABLE 4 – MULTIPLE WAVE FACTOR (SPRING, 2011)	61
TABLE 5 – WAVE SPRINGS DESIGN STRESSES FOR STATIC/LOW CYCLES (SPRING, 2011)	62
TABLE 6 – WAVE SPRINGS DESIGN STRESSES FOR DYNAMIC SERVICE APPLICATIONS (SPRING, 2011)	63
TABLE 7 - WAVY SPRING ESTIMATED CYCLE LIFE (SPRING, 2011)	63
TABLE 8 – CONDITIONS OF CLUTCH SIMULATION (HEYAN ET AL., 2013)	80
TABLE 9 – STATES OF MOTION AND RATIOS OF A SIMPLE PGS (NAUNHEIMER ET AL., 2010)	84
TABLE 10 – COEFFICIENTS OF FRICTION μ FOR WEDGES, CAMS , PIVOTS AND BEARINGS MADE OF HARDENED STEEL (HENRIKSEN, 1973)	86
TABLE 11 – TURBINE SHAFT ASSY BOM	101
TABLE 12 - PGS ASSY BOM (BILL OF MATERIALS)	101
TABLE 13 - BOM TO ASSEMBLE IN THE TRANSMISSION HOUSING (COMPLETE DNR)	104
TABLE 14 - CALCULATION SCENARIOS FOR FWD CLUTCH SET	148
TABLE 15 – CALCULATIONS' SCENARIOS FOR REV CLUTCH SET	149
TABLE 16 - RESULTS OF GENERAL CALCULATIONS FOR FWD SPRING PACK	150
TABLE 17 – RESULTS OF GENERAL CALCULATIONS FOR FWD SPRING PACK (3 RD SCENARIO)	150
TABLE 18 - RESULTS OF GENERAL CALCULATIONS FOR FWD SPRING PACK (4 TH SCENARIO)	151
TABLE 19 – FWD SPRING PACK SUPPLIER'S VERIFICATION	151
TABLE 20 – RESULTS OF PRE-LOAD CONDITIONS CALCULATIONS FOR NOMINAL SCENARIO OF FWD CLUTCH	153
TABLE 21 – RESULTS OF PRE-LOAD CONDITIONS CALCULATIONS FOR SECOND SCENARIO OF FWD CLUTCH	153
TABLE 22 - RESULTS OF PRE-LOAD CONDITIONS CALCULATIONS FOR THIRD SCENARIO OF FWD CLUTCH	154

TABLE 23 - RESULTS OF PRE-LOAD CONDITIONS CALCULATIONS FOR FOURTH SCENARIO OF FWD CLUTCH	154
TABLE 24 - RESULTS OF FATIGUE CALCULATIONS OF FWD SPRING PACK.....	155
TABLE 25 - RESULTS OF GENERAL CALCULATIONS OF WAVY WASHER.....	157
TABLE 26 – WAVY WASHER SUPPLIER’S VERIFICATION	157
TABLE 27 - RESULTS OF GENERAL CALCULATIONS OF WAVY WASHER (3 RD SCENARIO)	158
TABLE 28 - RESULTS OF GENERAL CALCULATIONS OF WAVY WASHER (4 TH SCENARIO)	158
TABLE 29 - RESULTS OF FATIGUE CALCULATIONS OF THE WAVY SPRING.....	158
TABLE 30 – RESULTS OF GENERAL CALCULATIONS FOR REV SPRING PACK (NOMINAL SCENARIO)	159
TABLE 31 - RESULTS OF GENERAL CALCULATIONS FOR REV SPRING PACK (SECOND SCENARIO).....	160
TABLE 32 - RESULTS OF GENERAL CALCULATIONS FOR REV SPRING PACK (THIRD SCENARIO)	160
TABLE 33 – REV SP SUPPLIERS’ VERIFICATION (NOMINAL SCENARIO).....	160
TABLE 34 - REV SP SUPPLIERS’ VERIFICATION (SECOND SCENARIO)	160
TABLE 35 - REV SP SUPPLIERS’ VERIFICATION (THIRD SCENARIO)	161
TABLE 36 – HYDRAULIC AREA OF EACH SCENARIO FOR PRE-LOAD CALCULATIONS OF REV CLUTCH SET	161
TABLE 37 - RESULTS OF PRE-LOAD CONDITIONS CALCULATIONS FOR NOMINAL SCENARIO OF REV CLUTCH	162
TABLE 38 - RESULTS OF PRE-LOAD CONDITIONS CALCULATIONS FOR SECOND SCENARIO OF REV CLUTCH	162
TABLE 39 - RESULTS OF PRE-LOAD CONDITIONS CALCULATIONS FOR THIRD SCENARIO OF REV CLUTCH	163
TABLE 40 - RESULTS OF FATIGUE CALCULATIONS OF REV SPRING PACK	163
TABLE 41 – GENERAL CALCULATIONS’ RESULTS OF BELLEVILLE WASHER	164
TABLE 42 – INPUT DATA FOR THE CALCULATIONS OF THE SPRING STIFFNESS OF THE DISC SPRING (2 ND AND 3 RD SCENARIOS)	165
TABLE 43 – GENERAL CALCULATIONS’ RESULTS OF THE DISC SPRING	165
TABLE 44 - RESULTS OF FATIGUE CALCULATIONS OF DISC SPRING.....	165
TABLE 45 - DYNAMIC VISCOSITY OF TRANSMISSION OIL	167
TABLE 46 - RESULTS OF GENERAL CALCULATIONS OF FWD CLUTCH	177
TABLE 47 - GENERAL CALCULATIONS RESULTS FOR FWD CLUTCH (NOMINAL CASE)	178
TABLE 48 - GENERAL CALCULATIONS RESULTS FOR FWD CLUTCH (SECOND SCENARIO)	179

TABLE 49 – GENERAL CALCULATION’S DATA INPUT FOR FWD CLUTCH SET (3 RD SCENARIO)	179
TABLE 50 - GENERAL CALCULATIONS RESULTS FOR FWD CLUTCH (3 RD SCENARIO)	180
TABLE 51 - GENERAL CALCULATION’S DATA INPUT FOR FWD CLUTCH SET (4 TH SCENARIO)	180
TABLE 52 - GENERAL CALCULATIONS RESULTS FOR FWD CLUTCH (4 TH SCENARIO)	181
TABLE 53 – INPUT DATA FOR TRANSMISSIBLE TORQUE CALCULATION (3 RD SCENARIO)	184
TABLE 54 – INPUT DATA FOR TRANSMISSIBLE TORQUE CALCULATION (4 TH SCENARIO)	185
TABLE 55 – RESULTS OF BITING-PINT CONDITIONS OF FWD CLUTCH SET (NOMINAL SCENARIO)	198
TABLE 56 - RESULTS OF BITING-PINT CONDITIONS OF FWD CLUTCH SET (SECOND SCENARIO).....	199
TABLE 57 - RESULTS OF BITING-PINT CONDITIONS OF FWD CLUTCH SET (THIRD SCENARIO)	201
TABLE 58 - RESULTS OF BITING-PINT CONDITIONS OF FWD CLUTCH SET (FOURTH SCENARIO).....	202
TABLE 59 - FLOW RATE CONDITIONS OF FWD CLUTCH.....	205
TABLE 60 – RESULTS FOR FWD CLUTCH PLATES’ PRESSURE	215
TABLE 61 - RESULTS FOR REV CLUTCH PLATES’ PRESSURE.....	216
TABLE 62 – RESULTS OF GENERAL CALCULATION OF THE REV CLUTCH SET (NOMINAL SCENARIO)	220
TABLE 63 – RESULTS OF GENERAL CALCULATIONS FOR REV CLUTCH SET (NOMINAL SCENARIO)	220
TABLE 64 – GENERAL CALCULATIONS DATA INPUT FOR REV CLUTCH SET (SECOND SCENARIO)	221
TABLE 65 - RESULTS OF GENERAL CALCULATIONS FOR REV CLUTCH SET (SECOND SCENARIO).....	221
TABLE 66 - GENERAL CALCULATIONS DATA INPUT FOR REV CLUTCH SET (THIRD SCENARIO).....	221
TABLE 67 - RESULTS OF GENERAL CALCULATIONS FOR REV CLUTCH SET (THIRD SCENARIO)	222
TABLE 68 – BITING POINT CONDITIONS OF NOMINAL SCENARIO OF THE REV CLUTCH SET	229
TABLE 69 - BITING POINT CONDITIONS OF SECOND SCENARIO OF THE REV CLUTCH SET.....	229
TABLE 70 - BITING POINT CONDITIONS OF THIRD SCENARIO OF THE REV CLUTCH SET	231
TABLE 71 – FLOW RATE CONDITIONS FOR REV CLUTCH	232
TABLE 72 – RESULTS OF TOLERANCE CHAIN STACK UP ANALYSIS OF FWD CLUTCH SET	246
TABLE 73 - RESULTS OF TOLERANCE CHAIN STACK UP ANALYSIS OF REV CLUTCH SET	247
TABLE 74 – REV RATIO RANGE	271
TABLE 75 – REQUIREMENT’S RESUME	275
TABLE 76 - REQUIREMENTS' VALIDATION RESUME	276
TABLE 77 - PRESSURE DIFFERENCE BETWEEN BITING AND KISS POINT FOR D	282

TABLE 78 – PRESSURE DIFFERENCE BETWEEN BITING AND KISS POINT FOR R.....	282
TABLE 79 – FWD PISTON VOLUMES	283
TABLE 80 - REV PISTON VOLUMES	284
TABLE 81 – FWD LOAD INPUT DATA OF SHOWCASE.....	291
TABLE 82 – OBTAINED RESULTS FOR FWD LOAD OF SHOWCASE	291
TABLE 83 – REV LOAD INPUT DATA FOR SHOWCASE DESIGN.....	294
TABLE 84 – OBTAINED RESULTS FOR FWD LOAD OF SHOWCASE	294
TABLE 85 – INPUT DATA FOR THE CALCULATIONS OF THE FORCES ON THE CAM	297
TABLE 86 – FITTING BETWEEN CAM AND BEARING	298
TABLE 87 – FITTING BETWEEN BEARING AND MAIN PLATE.....	299
TABLE 88 - ABAQUS® SYSTEM OF UNITS' CONVENTION	304
TABLE 89 – ABAQUS® RESULTING UNITS	304
TABLE 90 – INPUT DATA OF FEA OF THE COMPONENT PNSA3401_2.....	306
TABLE 91 - RESULTS OF FEA OF THE COMPONENT PNSA3401_2.....	307
TABLE 92 - INPUT DATA OF FEA OF THE COMPONENT PNSA1203_3.....	310
TABLE 93 – RESULTS OF FEA OF THE COMPONENT PNSA1203_3.....	310
TABLE 94 - INPUT DATA OF FEA OF THE COMPONENT PNSA3302_5.....	312
TABLE 95 - INPUT DATA OF FEA OF THE COMPONENT PNSA3302_5.....	313
TABLE 96 - INPUT DATA OF FEA OF THE COMPONENT PNSA3401_1.....	315
TABLE 97 – RESULTS OF FEA OF THE COMPONENT PNSA3401_1.....	315
TABLE 98 – RESULTING CENTRIFUGAL FORCE OF NOMINAL AND SECOND SCENARIO FOR FWD CLUTCH	339
TABLE 99 – AUXILIARY TABLE FOR THE STUDY OF SPRINGS' ASSOCIATION OF FWD CLUTCH.....	342
TABLE 100 - AUXILIARY TABLE FOR CALCULATIONS OF THE LOAD ALONG THE TRAVELLING OF THE FWD SPRINGS' ASSOCIATION	343
TABLE 101 – RESULTS OF TRANSMISSIBLE TORQUE OF THE NOMINAL CASE FOR FWD CLUTCH	345
TABLE 102 - RESULTS OF TRANSMISSIBLE TORQUE OF THE SECOND SCENARIO FOR FWD CLUTCH	346
TABLE 103 - RESULTS OF TRANSMISSIBLE TORQUE OF THE THIRD AND FOURTH SCENARIOS FOR FWD CLUTCH	348
TABLE 104 - RESULTING CENTRIFUGAL FORCE OF THIRD AND FOURTH SCENARIOS FOR FWD CLUTCH.....	351

TABLE 105 – AUXILIARY TABLE TO PLOT THE GRAPH OF ERROR! REFERENCE SOURCE NOT FOUND.....	353
TABLE 106 – AUXILIARY TABLE TO THE CALCULATIONS OF PRE-LOAD CONDITIONS.....	355
TABLE 107 – CREEPING EFFECT ON FWD CLUTCH SET FOR SLIPPING SPEED OF 200 RPM	356
TABLE 108 – RESULTING TRANSMISSIBLE TORQUE FOR A CONSTANT SLIPPING SPEED OF 200 RPM.....	358
TABLE 109 - RESULTING TRANSMISSIBLE TORQUE FOR A CONSTANT SLIPPING SPEED OF 100 RPM.....	360
TABLE 110 – INPUT DATA FOR THE CALCULATION OF THE INFLUENCE OF FLUID ENTRY RADIUS IN THE CENTRIFUGAL FORCE OF FWD PISTON	362
TABLE 111 - INPUT DATA FOR THE CALCULATION OF THE INFLUENCE OF FLUID ENTRY RADIUS IN THE CENTRIFUGAL FORCE OF BD	362
TABLE 112 – RESULTS FOR PLOTTING THE GRAPHS OF FIGURE 154 AND FIGURE 155	362
TABLE 113 – EXCERPT OF FIRST SERIES OF TABLES FOR THE CALCULATION OF FWD DRAG LOSSES.....	363
TABLE 114 – EXCERPT OF SECOND SERIES OF TABLES OF FWD DRAG LOSSES.....	367
TABLE 115 - EXCERPT OF THIRD SERIES OF TABLES OF FWD DRAG LOSSES.....	369
TABLE 116 - DRIVE CLUTCH ZERO-SLIP CURRENT-TO-TORQUE SENSITIVITY EXPRESSED IN NM/BAR FOR THE NOMINAL SCENARIO.....	374
TABLE 117 - DRIVE CLUTCH ZERO-SLIP CURRENT-TO-TORQUE SENSITIVITY EXPRESSED IN NM/BAR FOR THE SECOND SCENARIO	375
TABLE 118 - AUXILIARY TABLES OF GRAPHS OF FIGURE 180 AND FIGURE 181	376
TABLE 119 – RESULTS OF THE TRANSMISSIBLE TORQUE FOR REV CLUTCH SET (NOMINAL SCENARIO) .	377
TABLE 120 - RESULTS OF THE TRANSMISSIBLE TORQUE FOR REV CLUTCH SET (SECOND SCENARIO).....	378
TABLE 121 - RESULTS OF THE TRANSMISSIBLE TORQUE FOR REV CLUTCH SET (THIRD SCENARIO)	380
TABLE 122 – EXCERPT OF FIRST SERIES OF TABLES OF DRAG LOSSES CALCULATIONS OF REV CLUTCH ..	382
TABLE 123 - REVERSE CLUTCH ZERO-SLIP CURRENT-TO-TORQUE SENSITIVITY EXPRESSED IN NM/BAR FOR THE NOMINAL SCENARIO	386
TABLE 124 - REVERSE CLUTCH ZERO-SLIP CURRENT-TO-TORQUE SENSITIVITY EXPRESSED IN NM/BAR FOR THE SECOND SCENARIO	387
TABLE 125 - REVERSE CLUTCH ZERO-SLIP CURRENT-TO-TORQUE SENSITIVITY EXPRESSED IN NM/BAR FOR THE THIRD SCENARIO.....	388
TABLE 126 – AUXILIARY TABLE TO THE DESIGN OF FWD SHOWCASE SHIFTING LEVER (LOWER LEVER) .	389
TABLE 127 - AUXILIARY TABLE TO THE DESIGN OF FWD SHOWCASE SHIFTING LEVER (UPPER LEVER)...	389
TABLE 128 – RESUME AUXILIARY TABLE TO THE DESIGN OF FWD SHOWCASE SHIFTING LEVER	390

TABLE 129 - AUXILIARY TABLE TO THE DESIGN OF REV SHOWCASE SHIFTING LEVER (LOWER LEVER) ...	391
TABLE 130 - AUXILIARY TABLE TO THE DESIGN OF REV SHOWCASE SHIFTING LEVER (UPPER LEVER)	391
TABLE 131 - RESUME AUXILIARY TABLE TO THE DESIGN OF REV SHOWCASE SHIFTING LEVER	392
TABLE 132 - NECESSARY FORCE TO ACTUATE THE CAM	393
TABLE 133 – APPLIED LOAD ON THE SHIFTING LEVER	394
TABLE 134 – AUXILIARY TABLE FOR THE CALCULATION OF THE TRENDLINE OF THE KINEMATIC VISCOSITY VS TEMPERATURE	397
TABLE 135 - TRANSMISSION OIL PROPERTIES (COURTESY OF PUNCH POWERTRAIN NV).....	398
TABLE 136 – FWD SPRINGS DATA.....	399
TABLE 137 – FWD SPRING PACK PLATES DATA	399
TABLE 138 – FWD SPRING PACK DATA	399
TABLE 139 - WAVY WASHER DATA.....	400
TABLE 140 – REV SPRING PACK PLATE DATA.....	401
TABLE 141 – REV SPRING PACK DATA	401
TABLE 142 - DISC SPRING DATA.....	401
TABLE 143 - FWD PISTON DATA	402
TABLE 144 - BALANCE DAM DATA.....	402
TABLE 145 – REV PISTON DATA.....	403
TABLE 146 - FWD CLUTCH PLATES DATA.....	403
TABLE 147 - REV CLUTCH PLATES DATA	404
TABLE 148 - FWD CLUTCH SET DATA.....	404
TABLE 149 – REV CLUTCH SET DATA	404
TABLE 150 – RING GEAR DATA	405
TABLE 151 – SUN GEAR DATA	405
TABLE 152 – PLANETARY GEAR DATA.....	405
TABLE 153 – TRANSMISSION HOUSING FOR REV CLUTCH SET HOUSING DATA	406
TABLE 154 – 20CRMOH EQUIVALENT GRADES (ADAPTED FROM (NATIONAL STANDARD OF THE PEOPLE’S REPUBLIC OF CHINA, 2004) AND (KYOCERA, 2017)).....	406
TABLE 155 – 18CRMO4 (DIN)MECHANICAL PROPERTIES (ADAPTED FROM (DATENBLATT SAARSTAHL, N.D.)).....	407

TABLE 156 – SAPH310 (JIS G 3113) MECHANICAL PROPERTIES (ADAPTED FROM (NIPPON STEEL & SUMITOMO METAL, 2012)).....	407
TABLE 157 – HC 380 (EN 10 268) MECHANICAL PROPERTIES (ADAPTED FROM (BAOSTEEL, N.D.))	407
TABLE 158 – DC 01 (EN 10130) MECHANICAL PROPERTIES (ADAPTED FROM (TERMS, N.D.)).....	408
TABLE 159 – EN-GJS-900-2 (EN 1563:2012) MECHANICAL PROPERTIES (ADAPTED FROM (FOUNDERIES DE BROUSSEVAL ET MONTREUIL, N.D.))	408
TABLE 160 – S45C (JIS G 4051) MECHANICAL PROPERTIES (ADAPTED FROM (RODS ET AL., 1991))	409
TABLE 161 – 51CRV4 (DIN EN 10083) MECHANICAL PROPERTIES (ADAPTED FROM (SAARSTAHL, 2011A))	409
TABLE 162 – SAE 1060-10 (DIN EN 10083) (ADAPTED FROM (SAARSTAHL, 2011B))	410
TABLE 163 – 100CR6 (DIN) MECHANICAL PROPERTIES (ADAPTED FROM (SHEET, 2006))	410
TABLE 164 – SAPH 440 (JIS G 3113-1990) MECHANICAL PROPERTIES (ADAPTED FROM (CORPORATION, N.D.)).....	410
TABLE 165 – ALSI9CU3 (EN 1706) MECHANICAL PROPERTIES (ADAPTED FROM (STENA ALUMINIUM, 2010))	411
TABLE 166 – S355MC (EN 10149) MECHANICAL PROPERTIES (ADAPTED FROM (TATA STEEL, 2010)) ...	411
TABLE 167 – EQUIVALENT GRADES OF SAE 1008 (ASTM A568M) (ADAPTED FROM (PRODUCTS, N.D.))	412
TABLE 168 – DC 03 (EN 10130) MECHANICAL PROPERTIES (ADAPTED FROM (PRODUCTS, N.D.))	412
TABLE 169 – SAE 1010 (ASTM A568M) EQUIVALENT GRADES (ADAPTED FROM (PRODUCTS, N.D.)).....	412
TABLE 170 – ASTM A401 MECHANICAL PROPERTIES (ADAPTED FROM (DENDOFF SPRINGS, N.D.))	413

INDEX

1	INTRODUCTION	3
1.1	Contextualization	3
1.1.1	Fundamental Principles of Traffic and Vehicle Engineering	3
1.1.2	The significance of motor vehicles in our mobile world	3
1.1.3	Statistics	7
1.1.4	Trends in Traffic Engineering	8
1.1.5	Passenger and Goods Transport Systems	9
1.1.6	Classification of vehicles and vehicle use	10
1.1.7	Vehicle performance	11
1.2	Main goals	14
1.3	Methodology	15
1.4	Thesis structure	15
1.5	Welcoming company	16
2	LITERATURE REVIEW	19
2.1	Theoretical aspects	19
2.1.1	Power sources	19
2.1.2	Vehicle powertrain concepts	20
2.1.3	Transmissions	22
2.1.4	Planetary gear set	43
2.2	Calculations' methods and expressions	44
2.2.1	Springs	44
2.2.2	Centrifugal force in rotating cylinder	71
2.2.3	Transmissible torque	73
2.2.4	Drag torque of wet clutches	73
2.2.5	Planetary Gear Set ratio	83

2.2.6	PGS Kinematics	85
2.2.7	Cams	85
3	THESIS DEVELOPMENT	95
3.1	CVT transmission	95
3.2	External components	96
3.3	DNR working principle	97
3.3.1	DNR schematics	97
3.3.2	List of components	101
3.3.3	Rotational kinematics of components	106
3.3.4	Hydraulic circuit analysis	112
3.3.5	Kinematics of clutches	142
3.3.6	Gears	143
3.3.7	Thrust bearings orientation	146
3.4	Calculations data	148
3.5	Analytical Calculations	148
3.5.1	Calculation scenarios	148
3.5.2	Springs	149
3.5.3	Transmission oil	166
3.5.4	Clutch slipping	171
3.5.5	FWD clutch set	172
3.5.6	Clutch plates	214
3.5.7	REV clutch set	217
3.5.8	Tolerance chain stack up analysis	244
3.5.9	PGS	247
3.5.10	Planetary pin	259
3.6	Results comparison between FWD and REV clutch	259
3.6.1	Transmissible torque	260
3.6.2	Drag losses	261

3.6.3	Clutches' pressure	264
3.7	Ramp-down test	264
3.7.1	Description	264
3.7.2	Procedure	265
3.7.3	Calculations	265
3.7.4	Results	265
3.8	Comparison between tests and analytical calculations	268
3.8.1	FWD clutch set installed	268
3.8.2	REV clutch set installed	269
3.9	Requirements description	270
3.9.1	Requirement #1 - Max Clamping Torque in FWD Clutch	270
3.9.2	Requirement #2 - Max Clamping Torque in REV Clutch	270
3.9.3	Requirement #3 - Drive ratio range	270
3.9.4	Requirement #4 - Reverse ratio range	270
3.9.5	Requirement #5 - Drive clutch zero-slip current-to-torque sensitivity	271
3.9.6	Requirement #6 - Maximum guaranteed clutch pressure to reach maximum clamping torque in clutches	271
3.9.7	Requirement #7 - Oil compatibility	271
3.9.8	Requirement #8 - Maximum input torque in Drive	271
3.9.9	Requirement #9 - Cooling/lubrication budget	272
3.9.10	Requirement #10 - Safety Factor Torque at primary shaft	272
3.9.11	Requirement #11 - Drag torque of Drive clutch (in Reverse and Drive/Open)	272
3.9.12	Requirement #12 - Drag torque of Reverse clutch (in Drive and Reverse/Open)	272
3.9.13	Requirement #13 - Drag torque in Neutral	272
3.9.14	Requirement #14 - Safe state 1 - Open all DNR clutches	273
3.9.15	Requirement #15 - Sufficient separation between Open mode and preload pressure	273
3.9.16	Requirement #16 - Normally Neutral	273
3.9.17	Requirement #17 - Sufficient separation between preload pressure and biting point	273
3.9.18	Requirement #18 - Reverse clutch zero-slip current-to-torque sensitivity	273

3.9.19	Requirement #19 - Maximum input torque in Reverse	273
3.9.20	Requirement #20 - Max allowed clutch pressures	274
3.9.21	Requirement #21 - Piston Stroke Volume D	274
3.9.22	Requirement #22 - Piston Stroke Volume R	274
3.9.23	Requirement #23 - Max. Pressure on FWD clutch plates	274
3.9.24	Requirement #24 – Max. pressure on REV clutch plates	274
3.9.25	Requirement #25 – Max. rotational speed of the Needle bearing	274
3.9.26	Requirement #26 – Max. rotating speed of the needle roller thrust bearing	274
3.9.27	Resume	275
3.10	Design validation	276
3.10.1	Requirements verification	276
3.11	Development of the showcase model	285
3.11.1	Showcase function	285
3.11.2	Design	285
3.12	Preliminary assembly feasibility study	304
3.13	Showcase FEA	304
3.13.1	Abaqus® system of units	304
3.13.2	Pin (PNSA3401_2)	305
3.13.3	Ball transfer units' adapter (PNSA1203_3)	308
3.13.4	FWD pressing plate (PNSA3302_5)	310
3.13.5	REV pressing plate (PNSA3401_1)	313
3.14	Drawings	316
3.15	Poster	316
4	CONCLUSIONS AND FUTURE WORKS PROPOSAL	319
4.1	CONCLUSIONS	319
4.2	FUTURE WORKS PROPOSAL	322
5	REFERENCES AND OTHER SOURCES OF INFORMATION	325

5.1	SCIENTIFIC JOURNALS' ARTICLES and LITERATURE	325
5.2	CATALOGUES	326
5.3	STANDARDS	327
5.4	WEBSITES	327
6	APPENDIX	333
6.1	Extra theoretical aspects	333
6.1.1	Clutches	333
6.1.2	Torque converter	337
6.1.3	Bearings	338
6.2	Table 98 – Resulting centrifugal force of nominal and second scenario for FWD clutch	339
6.3	Table 99 – Auxiliary table for the study of springs' association of FWD clutch	342
6.4	Table 100 - Auxiliary table for calculations of the load along the travelling of the FWD springs' association	343
6.5	Table 101 – Results of transmissible torque of the nominal case for FWD clutch	345
6.6	Table 102 - Results of transmissible torque of the second scenario for FWD clutch	346
6.7	Table 103 - Results of transmissible torque of the third and fourth scenarios for FWD clutch	348
6.8	Table 104 - Resulting centrifugal force of third and fourth scenarios for FWD clutch	351
6.9	Table 105 – Auxiliary table to plot the graph of Error! Reference source not found.	353
6.10	Table 106 – Auxiliary table to the calculations of pre-load conditions	355
6.11	Table 107 – Creeping effect on FWD clutch set	356
6.12	Table 108 – Resulting transmissible torque for a constant slipping speed of 200 rpm	358
6.13	Table 109 - Resulting transmissible torque for a constant slipping speed of 100 rpm	360
6.14	Auxiliary tables for the calculation of the influence of fluid entry radius in the centrifugal force	362
6.15	Table 113 – Excerpt of first series of tables for the calculation of FWD drag losses	363

6.16	Table 114 – Excerpt of second series of tables of FWD drag losses	367
6.17	Table 115 - Excerpt of third series of tables of FWD drag losses	369
6.19	Table 116 - Drive clutch zero-slip current-to-torque sensitivity expressed in Nm/bar for the nominal scenario	374
6.20	Table 117 - Drive clutch zero-slip current-to-torque sensitivity expressed in Nm/bar for the second scenario	375
6.21	Table 118 - Auxiliary tables of graphs of Figure 180 and Figure 181	376
6.22	Table 119 – Results of the transmissible torque for REV clutch set (nominal scenario)	377
6.23	Table 120 - Results of the transmissible torque for REV clutch set (second scenario)	378
6.24	Table 121 - Results of the transmissible torque for REV clutch set (third scenario)	380
6.25	Table 122 – Excerpt of first series of tables of drag losses calculations of REV clutch	382
6.26	Table 123 - Reverse clutch zero-slip current-to-torque sensitivity expressed in Nm/bar for the nominal scenario	386
6.27	Table 124 - Reverse clutch zero-slip current-to-torque sensitivity expressed in Nm/bar for the second scenario	387
6.28	Table 125 - Reverse clutch zero-slip current-to-torque sensitivity expressed in Nm/bar for the third scenario	388
6.29	Auxiliary tables of the design of FWD showcase shifting lever	389
6.30	Auxiliary tables of the design of REV showcase shifting lever	391
6.31	Table 132 - Necessary force to actuate the cam	393
6.32	Table 133 – Applied load on the shifting lever	394
6.33	Table 134 – Auxiliary table for the calculation of the trendline of the kinematic viscosity vs temperature	397
6.34	Calculations data	398
6.34.1	Transmission oil	398
6.34.2	Springs	399

6.34.3	Pistons	402
6.34.4	Clutch plates	403
6.34.5	Clutch sets	404
6.34.6	Gears	404
6.34.7	External components	405
6.34.8	Materials	406
6.35	Preliminary assembly feasibility study	414
6.36	Datasheet LOCTITE® 638™	435
6.37	Datasheet LOCTITE® 648™	436
6.38	Showcase model's BOM	437
6.39	Poster	438
6.40	Showcase photos	439
6.41	Technical drawings of individual parts of the showcase model	440
6.42	Technical drawings of assemblies of the showcase model	441

INTRODUCTION

1.1 Contextualization

1.2 Main goals

1.3 Methodology

1.4 Thesis structure

1.5 Welcoming company

1 INTRODUCTION

1.1 Contextualization

1.1.1 Fundamental Principles of Traffic and Vehicle Engineering

The interrelations between traffic and traffic engineering and the economy is not only close but also fundamental, when considered a whole. An example of this interrelation is illustrated on Figure 1.

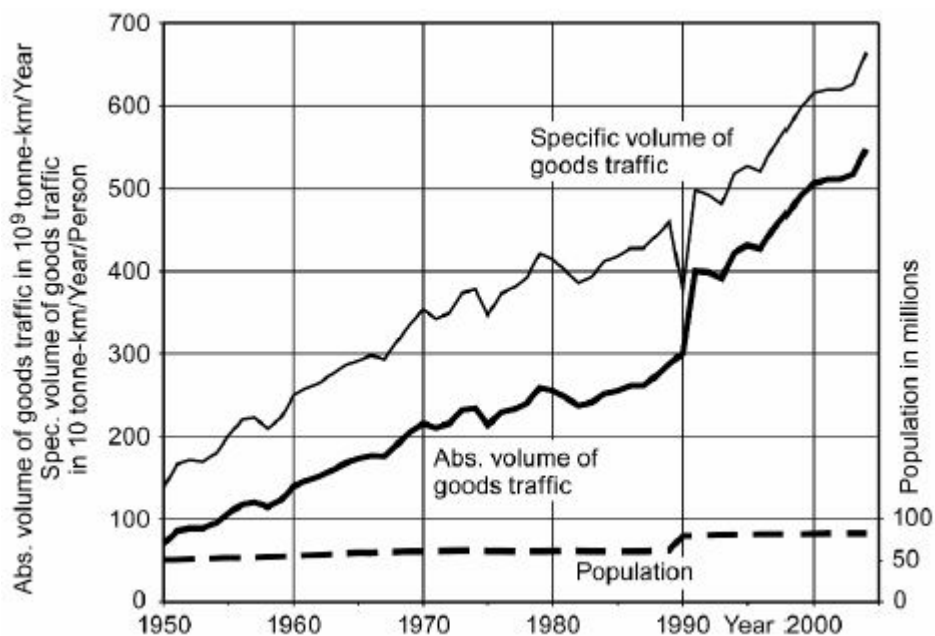


Figure 1 - Development of goods traffic and population in Germany; figures for the whole of Germany from 1990 (Naunheimer *et al.*, 2010)

Figure 1 illustrates a constant increase in goods traffic performance, both universally and with reference to the population of Germany. The largest part of this goods traffic takes place on the road (Naunheimer *et al.*, 2010).

1.1.2 The significance of motor vehicles in our mobile world

Mobility is assumed as a basic and human need. There are two factors which have an influence on human choice of means of transport. The first one is the satisfaction of his objective needs like transportation performance, door-to-door access and destination

attainability. The second one is the satisfaction of perceived subjective needs such as comfort, convenience and freedom to decide the means, destination and timing of the journey (Naunheimer *et al.*, 2010).

Automotive industry has quite an important significance in the economy regarding to employment and human sustenance. As an example, in 2005, one in seven German citizens worked for the automobile industry. Furthermore, the sales volume of the automotive industry in Germany is twelve times greater than the industry of machine tool (Naunheimer *et al.*, 2010).

In Figure 2, it is presented the portion of jobs that are direct or indirectly dependent on the transport industry.

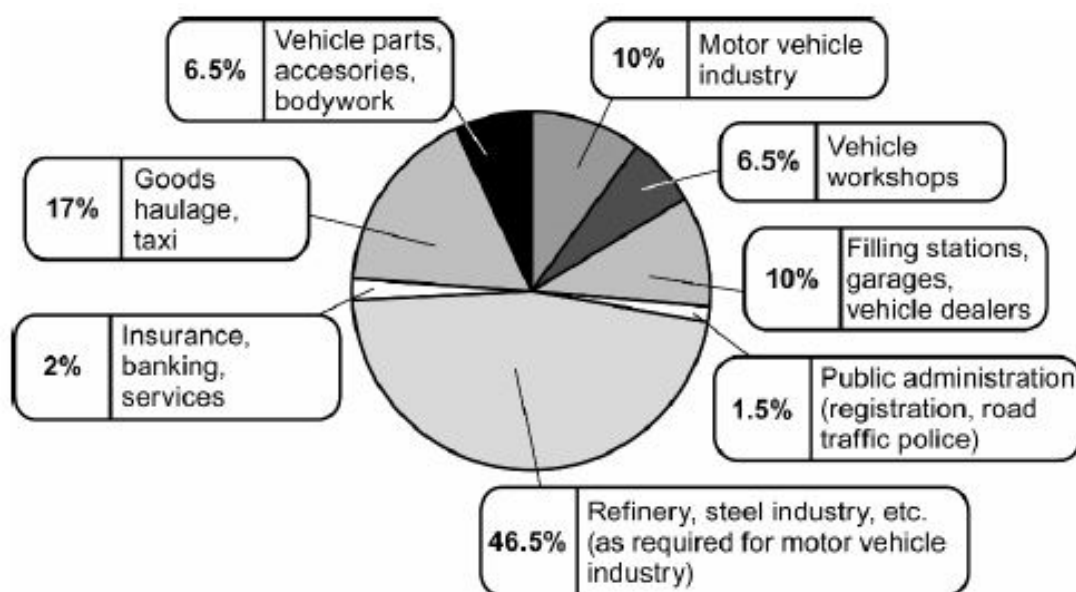


Figure 2 – Breakdown of jobs dependent on the motor transport industry in Germany (Naunheimer *et al.*, 2010)

It is important to underline that on the graph above it is only considered employment since the gasoline station although the chain is longer than what is visible in Figure 3.

Before the gasoline pump, also the fossil fuel reservoir, fuel refinery and road transport create a considerable amount of employment. Although it is desired to get an alternative option to fossil fuels due to the fact of their extinction, new employment will be certainly created.

Motor vehicles have gained great significance for humanity not only because they fulfil the basic human need for mobility but also because they allow door-to-door transport of both people and goods. There are no alternatives to motor vehicles in sight (Naunheimer *et al.*, 2010).

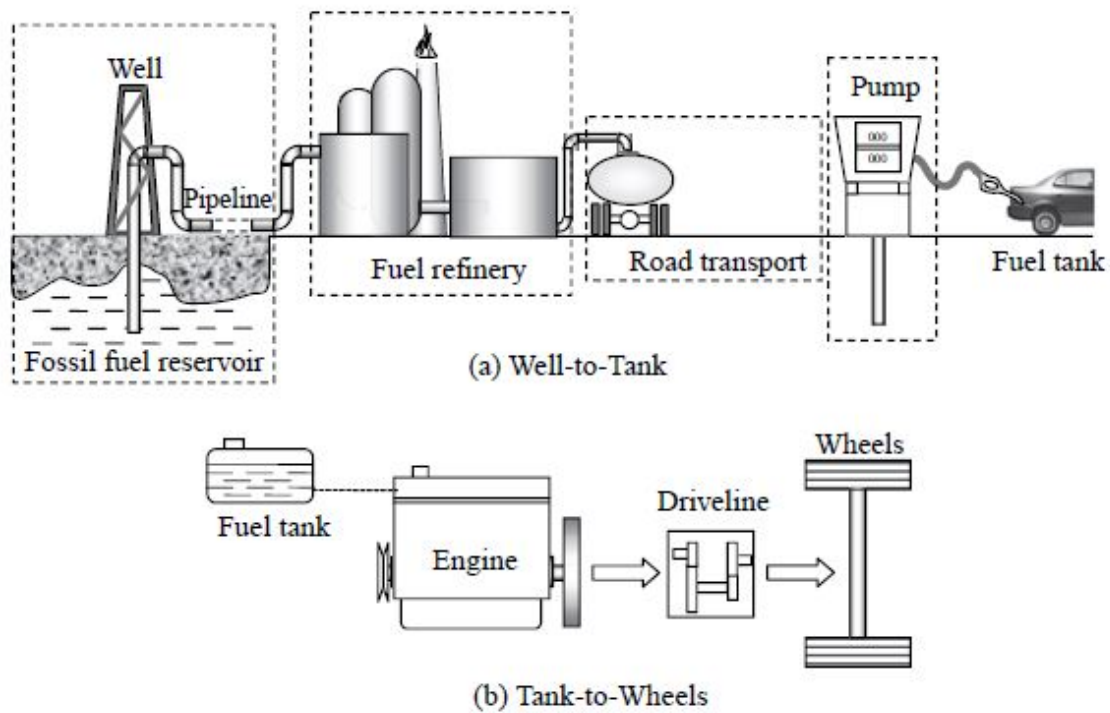


Figure 3 – Overall energy conversion process in vehicle transportation – the Well-to-wheels idea (Mashadi & Crolla, 2012)

In Figure 4, it is shown the various vehicles types and the way road vehicles are divided.

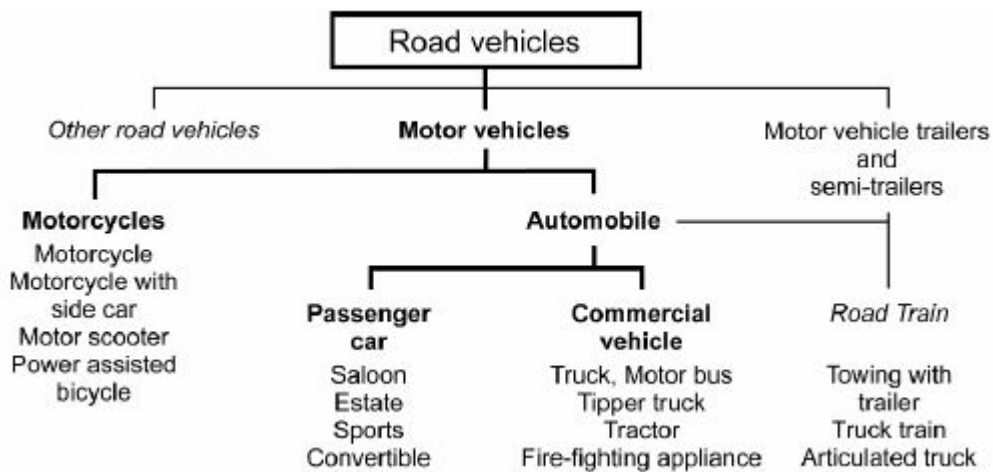


Figure 4 – Classification of road vehicles and motor vehicles to German standard DIN 70010 (Naunheimer *et al.*, 2010)

Statistics refer that motor vehicles are an imperfect trend. Neither environmental negligence or destruction nor the threat of gridlock have alerted people for such dependency and desire of mobility. The number of motor vehicles in the world has risen 10% per year as can be seen on Figure 5, since 1946 (Naunheimer et al., 2010).

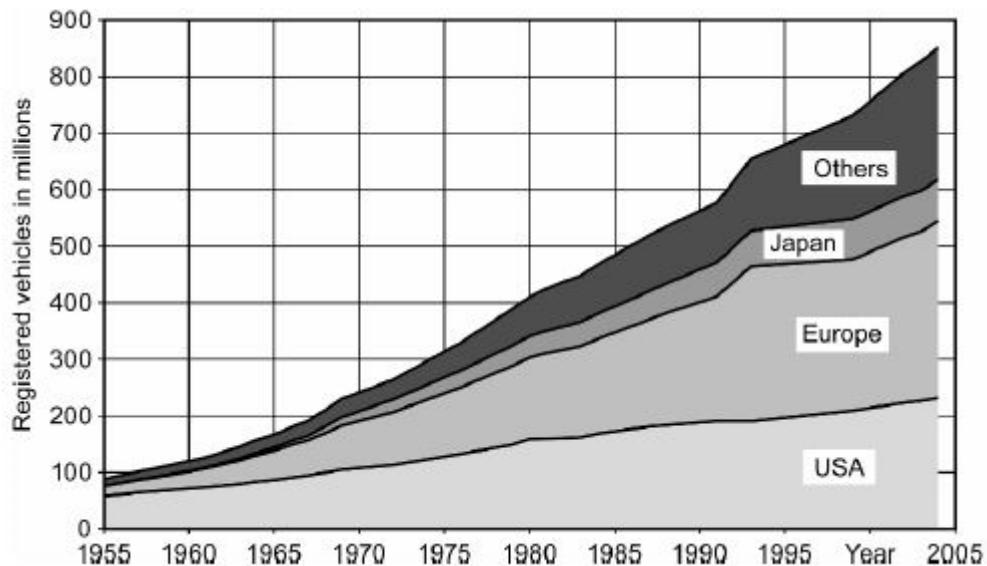


Figure 5 – Increase in the number of motor vehicles worldwide (Naunheimer et al., 2010)

According to the graph of Figure 5, Europe market is the one who presents the highest rise. It is expected the demand for motor vehicles in Eastern Europe and Asia to keep climbing even with the intensive promotion of mass transit. However not so significantly as in industrial countries in the past. It is jussive that the vehicles that are produced and exported in these developing countries be as resource-conserving and efficient as possible to prevent spreading environmental issues to other parts of the globe (Naunheimer et al., 2010). As seen before, automotive industry is a crucial factor in the global economy. In 2004, about 63 million motor vehicles were manufactured: 53 million correspond to passenger cars and 10 million to commercial vehicles (Naunheimer et al., 2010).

There are three centres competing in the development of motor vehicles: Europe, US and Japan/South Korea/China. Europe is the largest manufacturer of passenger cars. In the various European producer countries, the proportion of passenger car production accounted for by small, mid-range and luxury passenger cars varies greatly. In France and Italy are produced mostly small and mid-range passenger cars, in Germany are produced mostly mid-range and luxury passenger cars. In Germany are produced more luxury passenger cars than in the rest of Europe put together (Naunheimer et al., 2010). Each market has specific conditions based on socioeconomic situation of buyers, social values, geographical factors and, not the least, legislation. In order to succeed, motor

vehicles need to fulfil all the requirements of the respective market. Transmission system has a key role as it is the link between the road and the engine. For example, in US over 85% of passenger cars are equipped with automatic transmissions while in Europe it is only 13%. For commercial vehicles over 3.5t gross weight, the gearbox is selected specifically for the particular case of application. There are often different numbers of speeds and different methods of operation (manual or automatic) available for a commercial vehicle gearbox. Assuming that for spare parts, 10% of gearboxes are manufactured in excess, the number of transmission components such as gears and synchronizer packs can be estimated. In 2004, about 240 million gearwheels and 46 million synchronizer packs were produced for passenger car synchromesh gearboxes in Europe. For commercial vehicle constant-mesh and synchromesh transmissions, about 38 million gearwheels and 7.5 million synchronizer packs were manufactured (Naunheimer *et al.*, 2010). In 2009, the total population in the world of cars and light trucks was estimated at around 900 million, with a production in the same year of about 61 million units. More than 99% per cent of these vehicles are equipped with conventional powertrains (Mashadi & Crolla, 2012).

1.1.3 Statistics

1.1.3.1 Vehicles in use

In 2015, 294.2 million motor vehicles were driven in European Union of which 256.1 million were passenger cars. This corresponds to a motorisation rate of 573 units per 1000 inhabitants. Compared to 2005, the amount of passenger cars in use increased 27 million units ((ACEA), 2017).

1.1.3.2 Employment

In terms of employment, automotive industry provides direct and indirect jobs for 12.6 million European citizens where 2.5 million are direct jobs. The number of jobs provided by automotive industry has been rising since 2013 until at least 2015 ((ACEA), 2017).

1.1.3.3 Production

In 2016, Greater China was the most productive continent followed by Europe. In fact, the amount of units produced by Great China has increased significantly, as so South

Asia. For the other continents, it has been oscillating. As a whole, the production of passenger cars around the world has increased on a significant way. This type of vehicle is the most produced in the EU (European Union) (86%) ((ACEA), 2017).

1.1.3.4 Trade

In 2016 North America exported 32.7% of their produced cars, while Asia/Oceania exported 33.1% and EU 20.7. In fact, EU exported 6.3 million vehicles in 2016. Automotive industry generated in 2016 a trade surplus of about 90 billion euros for the EU. Most of the vehicles imported by the EU come from Turkey (about 1 million) and Japan (2nd place) with a significant percentage difference. Annually, about 3 million motor vehicles are imported to the EU. The main market of exportation from the EU is the US (about 1.2 million) ((ACEA), 2017).

1.1.3.5 Environment

The average emission of new cars in the EU is 118.1 gCO₂/km. In 2016, Germany was the country with higher emissions of CO₂. It is evident that there is a higher market demand for more fuel-efficient cars. It has been verified that the number of new cars in the EU with emissions of CO₂ greater than 130 g/km is decreased exponentially. At this moment, most of the cars present emissions in the range of 96-130 g/km and the number of vehicles within this range has increased significantly. The number of new passenger cars in the EU with emissions lower than 95 g/km is increasing slowly ((ACEA), 2017).

1.1.4 Trends in Traffic Engineering

Traffic is “the sum of all processes serving to overcome distance, comprising all relocation of persons, goods and information”. The environment of the automotive transmissions systems is defined by the traffic system. Traffic can be divided in five categories (Naunheimer *et al.*, 2010):

- local traffic: urban traffic;
- regional traffic: traffic within agglomeration areas;
- long-distance traffic: traffic between agglomeration areas;
- continental traffic: long-distance traffic;
- intercontinental traffic.

For passenger and commercial vehicles transmissions, they can only be distinguished between local and long-distance traffic while for buses there is a three-fold distinction between city traffic, local traffic and long-distance traffic (coaches).

The term “transportation performance” defines traffic performance and is related to goods traffic. Since always, the increase of transportation performance has been greater than the population growth.

Modern traffic engineering has a main purpose of solving the following issues (Naunheimer *et al.*, 2010):

- satisfying all transportation requirements;
- increasing the environmental friendliness of the means of transportation;
- reducing primary and secondary energy consumption;
- realising the potentials of electronic communication.

The several means of transport (road, rail, canal and pipeline) can be distinguished into transport systems having in account their purpose or technology (Naunheimer *et al.*, 2010).

1.1.5 Passenger and Goods Transport Systems

Traffic engineering purpose is to develop and provide reliable transport systems. It can be divided in passenger and goods traffic. For passengers, the main transport means are walking, bicycle, motorcycle, private car, taxis, public transport, railways, airplanes and ships. There is a low percentage of people who is carried by public transport and railway as it can be seen in Figure 6.

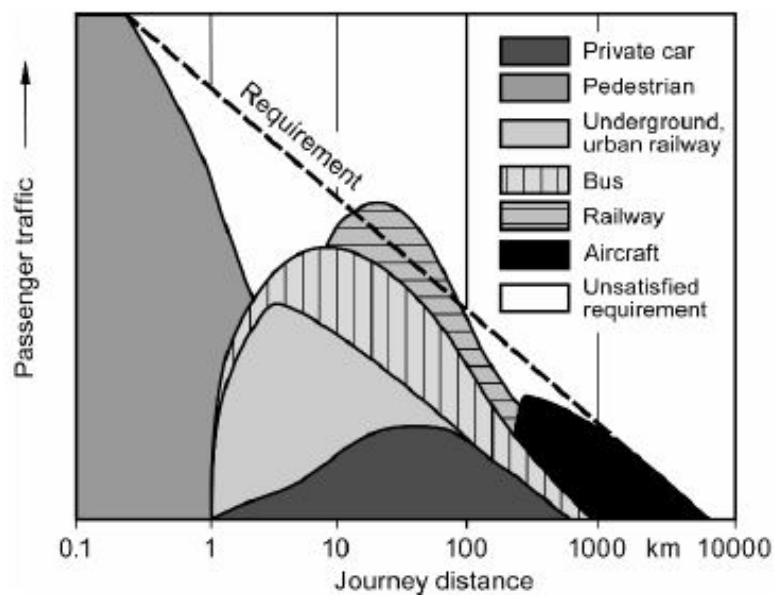


Figure 6 - Comparison of passenger transport supply and demand related to length of journey (Naunheimer *et al.*, 2010)

According to Figure 6, it is conclusive that there is no adequate means of transport in the two distance ranges: between 1 to 10 km and between 100 to 1000 km. Nowadays, passenger cars and taxis are the fastest means of transport for short-distance passenger trips. Because of low average driving speed, city buses are significantly slower than urban high-speed railways given equal idle times. From about 14 km (longer journey distances) high-speed city railways provide shorter travel times than passenger cars and taxis.

There are five means of transport for goods transport:

- Railways;
- Commercial vehicles (road traffic);
- Ships (canals, maritime transport);
- Airplanes (airfreight);
- Pipelines.

Usually some means of transport are joined in order to create a transport chain. Goods traffic on roads needs to be urgently limited. Transport speed, transport flow, space required and transport flow related to the space requirement are the characteristics used to compare the several means of transport of goods. In terms of transport speed and flow, the railway takes the lead followed by trucks. Pipelines, on the other hand, perform unfavourably. Commercial vehicles carry most of the annual volume of goods traffic with rail and barge well behind (Naunheimer *et al.*, 2010).

Similar to passengers' transport, a key feature for goods transport is door-to-door access. This parameter together with speed, economic efficiency and just-in-time delivery to assembly lines are the reasons behind the expressive increase in the number of trucks on roads. Three main points should be considered at the time of choosing a transport system: transportation needs, environmental impact and energy efficiency (Naunheimer *et al.*, 2010).

1.1.6 Classification of vehicles and vehicle use

In a first instance, vehicles are split into passenger cars, commercial vehicles, construction-site vehicles, agricultural tractors and special vehicles.

Passenger cars can be divided into three main groups: under 75 kW engine size, over 75 kW engine size and vans smaller than 3.5t. The commercial vehicle category can be split into buses and trucks. The truck category can then be divided by gross weight. Buses category can be divided according to the stops per kilometre. All this data can be seen of Table 1.

Table 1 – Transmission-oriented classification of motor vehicles by type of vehicle and type of use
(Naunheimer *et al.*, 2010)

Type of vehicle		Type of use			
Passenger cars	Power P < 75 kW	On-road	On-/off-road (building sites)	Off-road	
	Power P > 75 kW				
	Vans < 3.5 t				
Commercial vehicles	Trucks				Light-duty commercial vehicles: GVW < 7.5 t
					Medium-duty commercial vehicles: GVW < 16 t
					Heavy-duty commercial vehicles: GVW > 16 t
	Buses *)				Urban bus
					Interurban bus
					Multi-purpose bus
					Coach
Agricultural tractors					
Construction-site vehicles					
Special vehicles					

1.1.7 Vehicle performance

Since the first road vehicles developed by, for example, Daimler, Benz, Peugeot and Panhard & Levassore in the 1890s and 1900s, people introduced performance features in order to be able to compare different models and vehicles. In the beginning these features were top speed and fuel range. When more powerful engines started to come out, new measures were defined such as acceleration, gradeability and towing performance. Performance could be easily predicted based on the Newton's Second Law of Motion (Mashadi & Crolla, 2012).

According to the pioneering automobile engineer Olley (review paper of 1936), a typical American car of that period weighed about 2 tons (2000 kg) and the engine power was about 100 hp (75 kW). This resulted in an acceleration of approximately 3 m/s^2 , a gradeability of about 11% and a top speed about 140 km/h. The accuracy of these predictions started getting better since 1930 because measurement techniques for engine performance, tyre rolling resistance characteristics and aerodynamic drag effects were significantly improved (Mashadi & Crolla, 2012).

In Figure 7, it is illustrated where all the energy is used in a vehicle longitudinal performance for typical urban and highway conditions (Mashadi & Crolla, 2012).

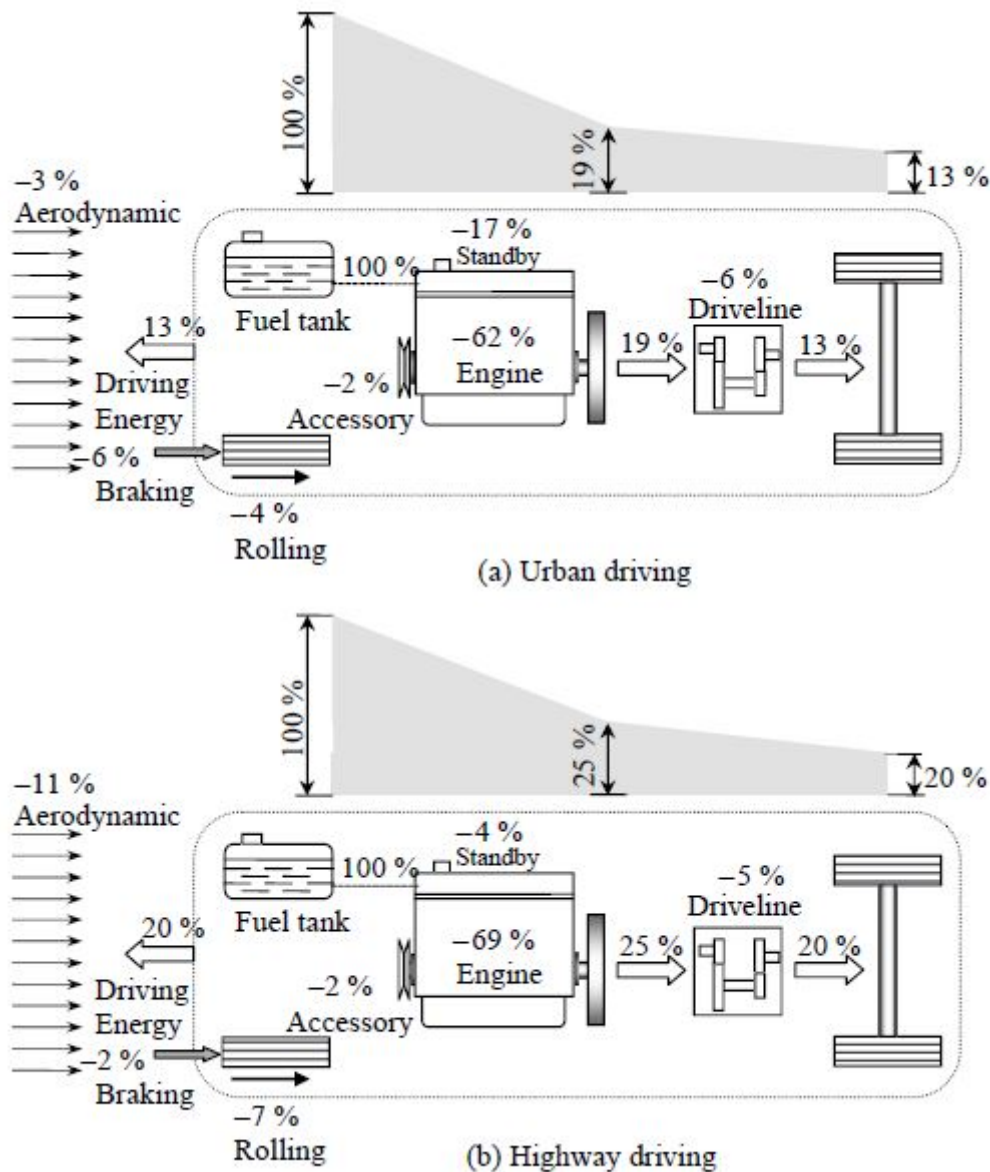


Figure 7 – Example of typical energy flows during urban (a) and highway (b) driving (Mashadi & Crolla, 2012)

In the 1970s, it started to be paid more attention to the fuel economy calculations than vehicle performance. In 1975, as a consequence of the oil crisis of 1973, CAFE (Corporate Average Fuel Economy) by Congress determined federal regulations whose main purpose was to improve the average fuel economy of cars and light trucks sold in the USA (United States of America). This point is the beginning of the interest in both fuel economy and the topic of emissions. Governments became very active in legislating for

the measurement and control of both these aspects of vehicle performance. In recent decades, it started to be required by the consumers data and performance figures to compare different models. Longitudinal performance such as maximum speed, acceleration, hill climbing, towing ability, etc are straight forward to measure. In contrast, comparative data on fuel economy and hence emissions are extremely controversial. The actual method to measure the fuel economy of a vehicle consists on to mount the vehicle on the instrumented dynamometer a drive a standard cycle. The drive cycle consists of a set of data points which specify a speed vs distance travelled profile. Different drive cycles have been developed in order to simulate as close as possible the different types of vehicle operation such as extra-urban, urban, highway and combined urban-highway (Mashadi & Crolla, 2012).

In recent decades, it started to be required by the consumers data and performance figures to compare different models. Longitudinal performance such as maximum speed, acceleration, hill climbing, towing ability, etc are straight forward to measure. In contrast, comparative data on fuel economy and hence emissions are extremely controversial (Mashadi & Crolla, 2012).

The actual method to measure the fuel economy of a vehicle consists on to mount the vehicle on the instrumented dynamometer a drive a standard cycle. The drive cycle consists of a set of data points which specify a speed vs travelled distance profile. Different drive cycles have been defined having in account the different types of vehicle operation such as extra-urban, urban, highway and combined urban-highway. Despite this approach being internationally accepted, there are some differences in the different countries and regions of the world. The current range of standard drive cycles was defined by the world's large three automotive markets: Europe, USA and Asia. The differences between the approaches of each region is based on the driving patterns of the regions. Furthermore, different countries and regions have defined different targets for fuel economy and emissions. This makes difficult for the manufacturers whose vehicles are sold worldwide to fulfil the requirements of all the markets (Mashadi & Crolla, 2012).

Due to these regional differences, it has been a source of considerable controversy and it has been quite difficult to reach a unanimous standard test. In addition, in real-world driving it is very difficult to achieve the ideal results obtained on standard test conditions. This fact comes out with no surprise as the tests and measurements are carried out in laboratory conditions over a repeatable drive cycle. Consumer organizations and popular car publications defend that the data displayed on the windscreen while on sale must be reasonably achievable in real drive conditions. In the EU, the fuel economy of passenger vehicles is measured using two different drive cycles: urban and extra-urban. The urban test-cycle (ECE-15) was introduced in 1999 and simulates a 4 km journey at an average speed of 18.7 km/h and a maximum speed of 50 km/h. The extra urban cycle (EUDC) simulates a mix between urban and highway driving.

The duration of the test is 400 seconds with an average speed of 62.6 km/h and a top speed of 120 km/h. In the USA, procedures are organized by the EPA (Environmental Protection Agency) and were updated in 2008 to include five distinct tests. The information of these tests is then presented on the car sales information. Some say that this information is more objective and reflective of the real-world driving fuel economy performance than the information obtained on the EU tests. In addition to the controversy of tests and measurements between the different markets, there is another issue: fuel economy is quoted in different units around the world. For example, USA and UK (United Kingdom) use apparently the same unit, mpg, although these are not comparable since the US gallon is 0.83 of an imperial gallon. In Europe and Asia, fuel consumption is quoted in l/100 km. A large mpg is comparable to a low l/100 km, which translates itself as : 30 mpg = 9.4 l/100km and 50 mpg = 5.6 l/100 km (Mashadi & Crolla, 2012). In general, drive cycles used in tests are less aggressive than the real-world driving as it is believed in tests are simulated lower values of acceleration and deceleration.

Regarding to emissions, there are two aspects which are commonly referred as “tailpipe emissions”. This name is the consequence of the fact that these emissions emerge from the exhaust pipe as product of the combustion process. The first aspect is the pollutant emissions which includes CO (Carbon monoxide), unburnt HC (Hydro carbons) and NO_x (Oxides of nitrogen). In Europe, in order to reduce the emissions of these pollutants, engine emission standards were defined. This measured lead to significant improvements in harmful emissions from passenger cars. The second aspect is related to CO₂ emission levels of vehicles. A lot of attention have been dedicated to this issue on the early part of the twenty-first century due to global concerns about the environment directly linked to the carbon footprint calculations which are nowadays embedded in all aspects of life. In the UK from 2001, the vehicle taxes were linked to the CO₂ emissions of the vehicle. Vehicles with emissions lower than 100 g/km were free of road taxes. In 2008, some legislation was determined to force European car manufacturers to reduce the average CO₂ emissions of new cars to 130 g/km by 2015 (Mashadi & Crolla, 2012).

1.2 Main goals

Two main goals were proposed for this work. The first one was the contribution to the validation of the DNR subsystem by verifying all the mechanical requirements. The other main goal was to develop a showcase model whose major purpose was to be able to show people all the details of the working principle of a DNR subsystem by allowing interaction with the user and visualization of the inside components of the subsystem.

1.3 Methodology

In order to achieve the proposed goals many other tasks were necessary to be done. For the validation of the DNR subsystem it was necessary to:

- Do intense research about all the mechanisms, components and theoretical topics directly and indirectly related to CVT and DNR subsystems;
- Study the function of the exterior adjacent components of the DNR (for example: torque converter and variator);
- Deeply study the working principle of the proposed DNR subsystem;
- Collect all the necessary information related to the components in study (for example: dimensions and materials);
- Develop numerical models which would lead to results which would be compared to the requirements as also some intermediate calculations to understand the influence of some parameters;
- Do a ramp-down test to compare to the calculated drag torque;

For the development of the showcase model the following tasks were done:

- Research and study of some mechanisms which allow the transmission of motion between the cams and the shifting lever;
- To understand the main requirements and desired function for the showcase;
- To design and calculate the several components of the showcase;
- 3D modelling;
- FEA (Finite elements analysis) to validate the crucial components;
- Technical drawings to ask for quotation;
- Assembly at the supplier.

In this dissertation, it will be explained the working principle of the DNR subsystem which is the base of this thesis. Firstly, it will be studied the DNR schematics in order to understand the principle of a DNR subsystem. Secondly it will be studied the hydraulic circuit to understand the oil flow and by this way to understand which components are pressurized. Having in account the previous point it will be possible to understand the kinematics of the subsystem. During this dissertation, it is also expected to understand the function of some special components. By doing this introduction to the subsystem it will be easier to complete all the tasks previously stated.

1.4 Thesis structure

The thesis structure is divided in seven main chapters. The first one is a contextualization with the topic of study as well as about the work in general. The second chapter is related to the theoretical aspects which imply a deep research of all the mechanisms and components in use in the DNR subsystem. This research it was also useful to

understand the main difference between some transmission and gearboxes' architectures. In the third chapter it is presented all the necessary calculation models and formulas which will be useful in the intermediate calculations for the verification of the mechanical requirements of the DNR. In the fourth chapter it is presented the working principle of the DNR as also all the calculations of the DNR and intermediate tasks. In the same chapter it is also presented the description of the showcase as also all the necessary calculations and FEA for validation. In the conclusion chapter it was done a resume of all the calculations achieved during the thesis development. In the sixth chapter it is presented all the consulted sources of information. Finally, in the last chapter it is presented all the auxiliary information for the thesis development, some extra technical aspects and all the technical drawings of the showcase as also some photos of the final result.

1.5 Welcoming company

Punch Powertrain NV (*Naamloze Vennootschap*), located in Sint-Truiden, Belgium, is an independent full system supplier of powertrains. With over than 40 years of experience in the production of CVT Transmissions, this company also offers electric powertrains for New Energy Vehicles such as Plugin Hybrids (PHs) and Electric Vehicles (EVs), as well as DCT (Dual clutch transmission). Since 2013 Punch Powertrain has achieved an average annual growth of over 50%. This is with no doubt, a challenging and enriching company to develop the master thesis in the field of automotive industry not only because of the enormous amount of knowledge than can be transmitted to student but also because the clear view that can be taken be the student of the complete process of a CVT transmission, from the R&D (Research and Development) to the production line.

LITERATURE REVIEW

2.1 Theoretical aspects

2.2 Calculations' methods and expressions

2 LITERATURE REVIEW

2.1 Theoretical aspects

2.1.1 Power sources

The most implemented source of power so far developed for driving road vehicles is the internal-combustion engine (ICE) with cyclical combustion, working on the spark-ignition or diesel principle. The power developed by the ICE is generated by the burning of fuel inside the engine itself. There are some alternative power units such as the steam engine and the electric motor (Hillier, 2012; Naunheimer et al., 2010). The former requires a boiler to generate the steam, in addition to the engine itself, making the complete installation rather bulky. Furthermore, there are heat losses in the boiler as well as in the engine, therefore it is less efficient than the ICE. During temporary halts, the steam engine continues to consume fuel in order to maintain steam pressure, whereas the internal-combustion engine consumes fuel only when it is actually running (Hillier, 2012).

To operate an electric motor, it is necessary a supply of electrical energy. If this energy is to be carried on the vehicle, it must be in the form of batteries or accumulators. Although, these are both heavy and bulky in relation to the amount of energy they can store, limiting the range, speed and load-carrying capacity of the vehicle. If an external supply of electrical energy is to be used, the vehicle must be connected to a distant power station by some means such as a system of wires, suspended at a safe height above the road, with which contact is made by suitable arms mounted on top of the vehicle and provided with some form of sliding connection to the wires. Such a system has obvious limitations and because of that it cannot be implemented in everyday particular vehicles. It has been invested in the development of more practical electric cars and of hybrid vehicles due to the ecological advantages these types of vehicles offer (Hillier, 2012).

ICE is known for some advantages such as high power-to-weight ratio, relatively good efficiency and relatively compact energy storage. However, there are three fundamental disadvantages (Naunheimer *et al.*, 2010):

- Unlike steam engines or electric motors, the ICE is incapable of producing torque from rest (zero engine speed);
- ICE only produces maximum power at a certain engine speed;
- Fuel consumption is strongly dependent on the operating point in the engine's performance map.

At the late 1800s and early 1900s steam had a long history of development and there wasn't any problem installing enough power to give an adequate performance. On the other hand, fuel economy was poor, the boiler needed firing up prior to a journey and both water storage and usage were problematic. However, electric cars look extremely promising: they are quiet, clean and remarkably easy to operate. The major problem was the range which was limited by the amount of energy that could be stored on the batteries. This problem is even today the subject of many researches. Gasoline cars in that period were less well developed and looked extremely troublesome: they were difficult to start and when running they were noisy, dirty and unreliable. However, they were distinct by a fundamental advantage: energy density of gasoline is about 300 times better than a lead-acid battery. This means that it was worthwhile to invest in the refinement of the gasoline-based powertrain. This refinement has continued to nowadays (Mashadi & Crolla, 2012).

2.1.2 Vehicle powertrain concepts

2.1.2.1 Introduction

The output from the power source dominated by ICE is controlled by a transmission system and driveline to deliver tractive effort to the wheels. All these components, collectively referred to as powertrain system are controlled by the driver. It is the engineering behind powertrain systems which provide the driving force behind the mobility (Mashadi & Crolla, 2012).

Drivers have a range of performance criteria such as: acceleration, top speed, fuel economy, gradeability and towing capacity. These are quantitative features. However, there are also subjective features which have a great impact in the commercial success of a certain model such as driveability, fun to drive, refinement and driving pleasure. Recently, new demands were imposed like emissions (Mashadi & Crolla, 2012).

2.1.2.2 Powertrain subsystems

All the subsystems of a powertrain system are subject to constant research and development in order to improve their performance and the performance of the whole system in terms of efficiency, emissions control, refinement and overall cost.

A powertrain system is composed by four subsystems such as (Mashadi & Crolla, 2012):

- Engine;
- Transmission;
- Vehicle structure;
- Systems operation.

2.1.2.3 *Conventional powertrains*

A conventional powertrain is defined to be composed by an IC engine that the vehicle wheels through a transmission, incorporating a gearbox and a final drive unit. The most common layout of a small passenger car is front-engine, front wheel drive. However, the principles associated with powertrain analysis are applicable to other kinds of layout (Mashadi & Crolla, 2012).

It is known that more than 99% of the 61 million of new cars and light trucks produced in 2009 were equipped with a conventional powertrain. Despite the considerable interest from 2000 onwards in alternative powertrains, described under the general heading of LCVs (Low carbon vehicles), it is known that the principles to analyse and understand conventional powertrain systems will keep being applied for some more decades (Mashadi & Crolla, 2012).

2.1.2.4 *Hybrid powertrains*

During the late 1800s and early 1900s, steam, electric and petrol were the three competing technologies for the powerplant. As expected, each of these had their own advantages and disadvantages which make it unclear to predict which one would be implemented in the future (Mashadi & Crolla, 2012).

The notion of a hybrid vehicle was born between nineteenth and twentieth centuries when it was suggested to combine two powerplants in order to extract the best features of each one. Initially it was not called hybrid. In 1902 Woods gas-electric car realize what is nowadays known as an electric hybrid layout. This vehicle was driven by a motor which doubled as a generator. The vehicle could run on battery power alone at low speeds while the gasoline engine could be used to charge the battery and taking advantage of regenerative breaking (Mashadi & Crolla, 2012).

There are plenty of different powertrain architectures for hybrid vehicles. In 2011, there were three main architectures which were already equipping some vehicle models. These three different architectures are presented in Figure 8. The example of Figure 8

(a) was equipped in Nissan leaf, in (b) it is presented the architecture of Chevrolet Volt and in (c) it is illustrated the architecture of Toyota Prius (Mashadi & Crolla, 2012).

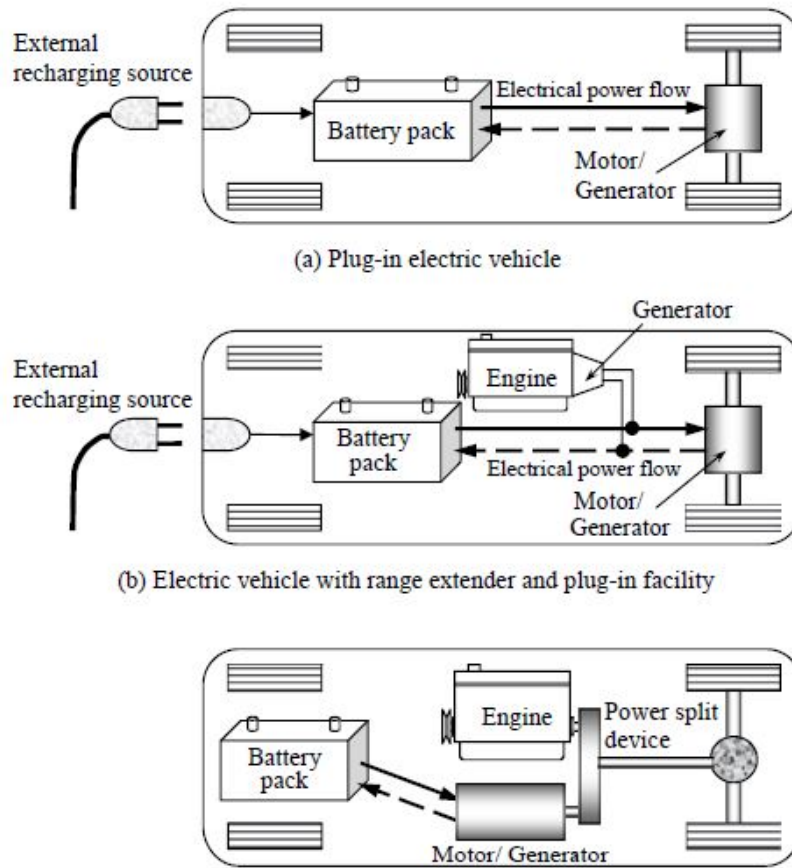


Figure 8 – Three types of typical hybrid/electric vehicle architectures available in 2011 (Mashadi & Crolla, 2012)

2.1.3 Transmissions

2.1.3.1 Introduction

The main goal a vehicle transmission is to transfer engine power to the driving wheels of the vehicle. Shifting which happens inside the transmission allows matching the engine torque and speed with the vehicle's load and speed conditions (Mashadi & Crolla, 2012). The most recent developments have been focused on AMT (Automated manual transmission), DCT and CVT.

In the UK, the transmission system is understood to refer to the complete driveline between the engine and the road wheels. However, in many other countries the term 'transmission' refers to the gearbox unit and applies in particular to the type of arrangement used on a given vehicle (i.e. whether the selection of the gears has to be

carried out manually by the driver or uses automatic means) (Hillier, 2012). On this thesis, “transmission” system is to be understood according to the UK terminology: the complete driveline between the engine and the road wheels. The complete transmission system is composed for different subsystems:

- Gearbox;
- Clutch;
- Propeller shaft;
- Final drive and differential;
- Drive shafts to each driven wheel;
- Universal joints.

The development of a transmission system takes in account the type of vehicle, its power unit and intended use. A classification of vehicles oriented to transmission development can be consulted on chapter 1.1.6 (Naunheimer *et al.*, 2010).

2.1.3.2 Functions of transmissions

Development engineer team should consider the vehicle transmission as a functional whole, in order to guarantee an adequate link between the engine and the drive wheels. The transmission system includes gearbox and moving-off elements, i.e. the entire system to adapt speed and torque including changing gear and starting. The four main functions of a vehicle transmission are to (Naunheimer *et al.*, 2010):

- Enable the vehicle to move-off from rest;
- Adapt power flow;
- Convert output torque T_2 and output speed n_2 ;
- Enable reverse motion;
- Enable permanent power transmission;
- Positive or force locking engine power transmission with minimal loss;
- Control power matching.

In addition to these main requirements there are also other kind of requirements, known as operational requirements, such as (Naunheimer *et al.*, 2010):

- Operational reliability;
- Gearbox costs;
- Ease of repair;
- Ease of operation;
- Power matching;
- Efficiency;
- Installation dimensions and weight;
- Customisability;
- Emissions (noise, oil etc.).

The operational requirements can partially dictate the competitiveness of the transmission.

2.1.3.3 Key factors and Conventions

The key factors of a gearbox are the direction of rotation, the transmission ratio and the torque. In order to be able to compare different transmission architectures, designs and variants, some definitions need to be used as a standard.

2.1.3.3.1 Direction of rotation

The direction of rotation of a powertrain is considered to be positive when the direction of rotation is clockwise in a right-hand Cartesian system of co-ordinates. This is as viewed against the forward direction of movement related to the vehicle, as shown on Figure 9.

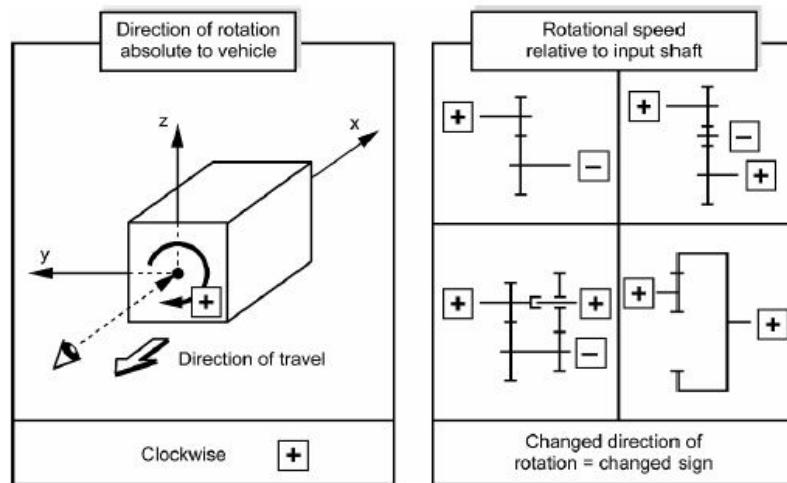


Figure 9 – Definition of direction of rotation in a powertrain (Naunheimer *et al.*, 2010)

2.1.3.3.2 Transmission ratio

The transmission ratio, i_G , is defined as the relationship between the angular velocity ω_1 of the input shaft of a gearbox and ω_2 of the output shaft and can be calculated as follows:

$$i_G = \frac{\omega_1}{\omega_2} = \frac{n_1}{n_2} \quad \{2.1\}$$

The relation between the output speed, n_2 , and the input speed, n_1 in a powertrain component is called the speed conversion. The torque conversion gives the relation between the output torque T_2 and the input torque T_1 of a powertrain component. The following characteristic result for the transmission ratio:

- $i_G > 0$ transmission input and output shaft rotate in the same direction;
- $i_G < 0$ change of direction of rotation in the transmission;
- $|i_G| > 1$ speed reducing ratio;
- $|i_G| < 1$ speed increasing ratio.

In the case of CVT and with transmission combinations:

- $i_G = \infty$ stationary output with rotating input;
- $i_G = 0$ stationary input with rotating output.

The ratios in a gearbox are designated by gear ratio, u . The gear ratio of a gear pair is the ratio of the number of teeth z_2 of the larger wheel to the number of teeth z_1 of the smaller wheel (pinion) (Naunheimer *et al.*, 2010):

$$u = \frac{z_2}{z_1} \text{ with } z_2 \geq z_1 \quad \{2.2\}$$

German standard DIN 3990 states that in the case of spur gears the number of teeth of a wheel with external gearing is positive, and that the number of teeth of a wheel with internal teeth (ring gear) is to be taken as negative (Naunheimer *et al.*, 2010).

2.1.3.3.3 Torque

The torque direction of the transmission input shaft is usually defined as positive. As illustrated in Figure 10, the torque direction changes along a transmission component but not the direction of rotation.

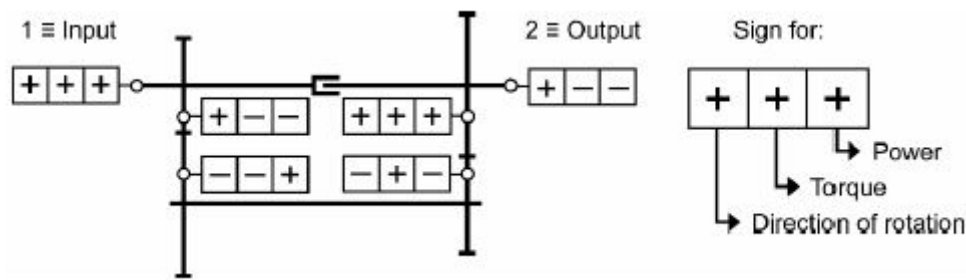


Figure 10 - Sign rules for rotational speed, torque and power (Naunheimer *et al.*, 2010)

The sign of the power P absorbed (positive) and delivered (negative) at a particular point can be determined from the speed of rotation and the torque at that point in the transmission as follows:

$$P = T\omega = 2\pi nT [W] \quad \{2.3\}$$

Müller HW proposed four rules for speeds of rotation, torque values and power values in a transmission:

- All parallel shafts in a transmission rotating in the same direction will have speeds with the same sign;
- In an “input shaft”, the signs for speed of rotation and torque are the same; in an “output shaft”, they are opposite to each other;
- “Input power” is always positive; “output power” is always negative;
- The two equal connecting torque values of a free connecting shaft have opposite signs at the connecting ends.

A transmission consists of three parts. One is the frame. This condition is important because it is necessary to provide a reaction for the difference in force or torque between the input and output side resulting from the conversion of movement. In vehicle transmission, the frame corresponds to the gearbox housing.

In Figure 11, it is illustrated the symbolic representation proposed by Wolf.

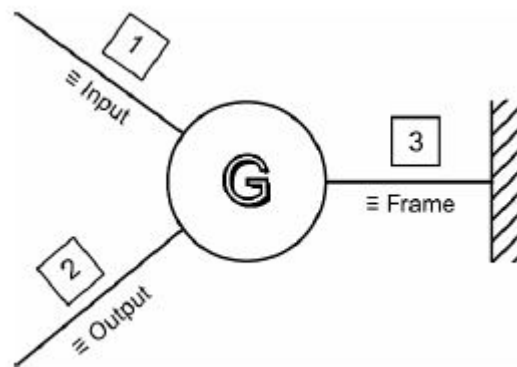


Figure 11 – Wolf transmission symbols (Naunheimer *et al.*, 2010)

2.1.3.4 Latest developments

The most recent developments in transmission systems are (Herdman, 1995; Mashadi & Crolla, 2012):

- Lower-friction lubricants (engine oil, transmission fluid, axle fluid);
- Hydraulic losses in the torque converter;
- Continuously variable transmission (CVT);
- Automated manual gearbox;
- Dual clutch gearbox;
- Increase in the ratio spread between top and first gear.

Systems operation is another subsystem of a powertrain system. This subsystem is also related to the transmission. And also in this subsystem it has been made some effort to improve the performance by (Mashadi & Crolla, 2012):

- Automatically shutting off engine when vehicle is stopped;
- Recapturing wasted energy while braking (regenerative braking);
- Augmenting a downsized engine with an electric drive system and battery (mild hybrid vehicles);
- Improved control of water-based cooling systems so that engines reach their efficient operating temperature sooner.

2.1.3.5 Manual transmissions

Manual transmissions are the oldest type of transmissions and even after new architectures have been developed they are still very popular. Simplicity, low cost and efficiency are the main advantages of this kind of transmissions. Manual transmissions are still the most efficient transmissions although their usage depends on the ability of the driver as frequent shifting is necessary mainly on urban regions (Mashadi & Crolla, 2012). In this type of transmissions the driver of the vehicle is responsible for the shifting (changing from one gear ratio to another) according to the vehicle need (D. A. Crolla, 2009).

2.1.3.5.1 Design and Working Principle

According to Figure 12, transmission is fastened to the engine body by bolts through the bell housing. Inside the bell housing it is assembled the clutch system. The clutch system is bolted to the engine flywheel so by this way the torque and speed of the engine is transmitted to the clutch plate (CP). Consequently, the clutch plate transmits the power from the engine to the input shaft of the transmission. When the clutch is engaged, the

input shaft and the clutch rotate at the same speed as the engine. When clutch is disengaged, the transmission input is disconnected from the engine and shifting can be performed. Shifting is then performed by means of a shifting mechanism (Mashadi & Crolla, 2012).

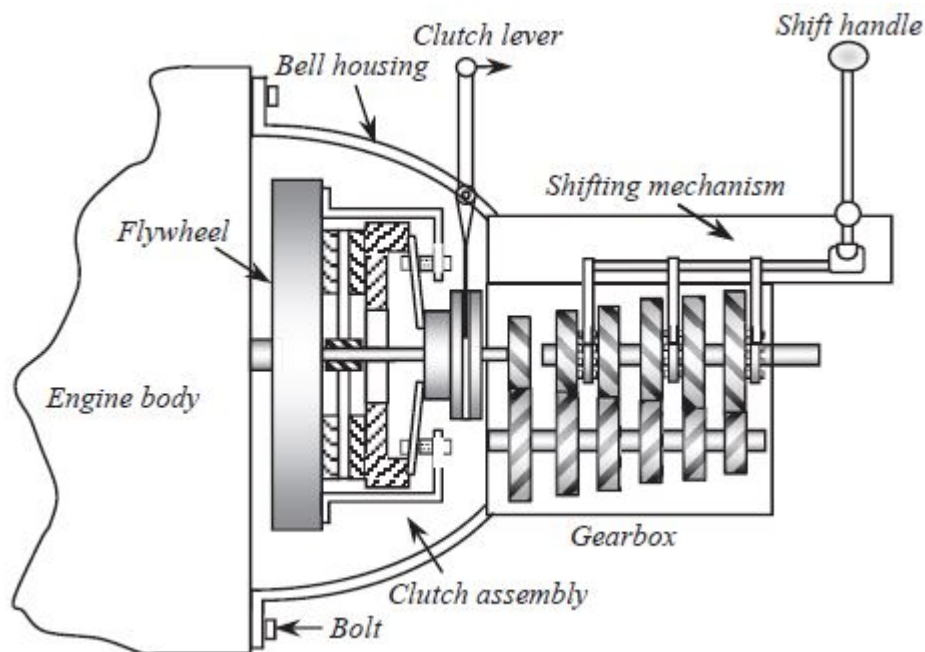


Figure 12 – Manual transmission assembly (Mashadi & Crolla, 2012)

Clutch system is activated when the clutch pedal is released and the gear lever is pulled back, forcing the release bearing to push the diaphragm spring. The seesaw effect on the spring pulls back the pressure plate and consequently the clutch plate is released. The input shaft of the transmission and the clutch plate are directly connected through the splines and both rotate with the same speed (Mashadi & Crolla, 2012).

There are two types of gear change: moving a gear to mesh the opposing gear, or moving a linking member to deliver torque to already meshing gears. The first type is called the “sliding-mesh”. This shifting method is already obsolete. Its concept is illustrated in Figure 14 (a). The second shifting method is called “constant-mesh” and it is used in all modern manual transmissions. Its concept is showed in Figure 14 (b) (Mashadi & Crolla, 2012). In Figure 14, it is illustrated a layshaft type gearbox with in-line input and output shafts.

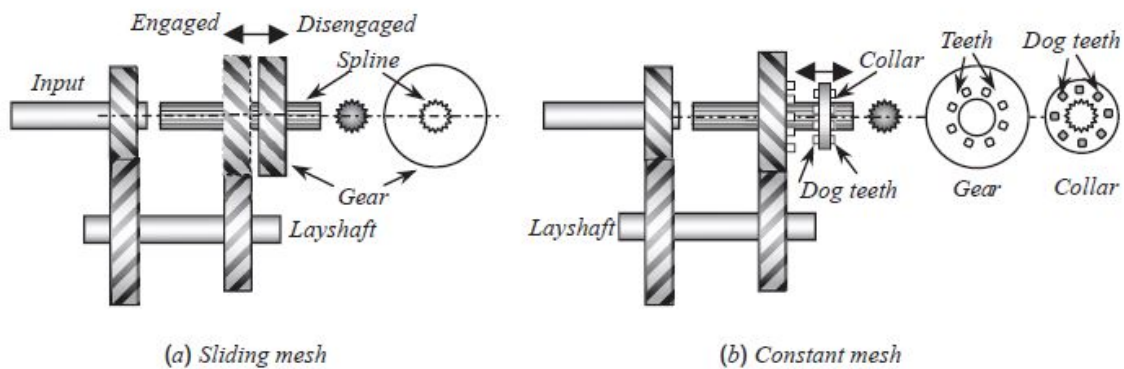


Figure 14 – Sliding mesh and constant-mesh gear meshing methods (Mashadi & Crolla, 2012)

When the clutch is released the power is transferred to the output shaft of the transmission. In the constant mesh gearing method both gears are in constant mesh. However, the gear located on the output shaft has a hole in the centre which means that it is not connected to the output shaft. Not torque is transferred between the gear and the output shaft. The power is transferred by means of a sliding collar with a spline in its core and dog teeth over its sides (Figure 14 (b)). The collar is always turning with the output shaft through the splines and when it is shifted to the side, its dog teeth fit into the matching teeth on the side of the gear and the three (shaft, collar and gear) get connected to each other and power is transferred. Prior to the engagement of the gear and output shaft in constant mesh design, both rotate with different speeds. Due to this fact it is necessary synchronize the speeds of both. The device which is responsible for this action is called synchronizer and it is illustrated in Figure 13 (Mashadi & Crolla, 2012).

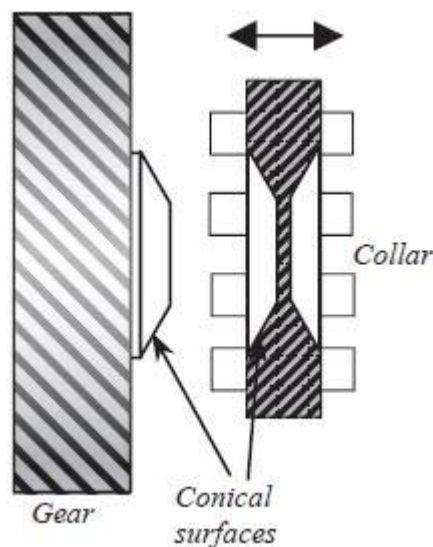


Figure 13 – Synchronizing concept (Mashadi & Crolla, 2012)

In Figure 13, it is presented the concept of two matching conical surfaces which get in contact gradually while the collar slides towards the gear. When, finally, both surfaces get in contact, the speeds gradually synchronize and the teeth subsequently engage (Mashadi & Crolla, 2012).

In Figure 15, it is showed the gear selecting and shifting mechanism in three different views: front, top and left. Usually, for a five-speed transmission there are three forks that control the three collars. Each collar is used to connect two gears when they are moved to the left or right side. The forks are fixed on three rods that can move back and forward. The correct rod is chosen by left-right movement of the shift handle. The choice between two gears is made by moving the shifting handle front or back (Mashadi & Crolla, 2012).

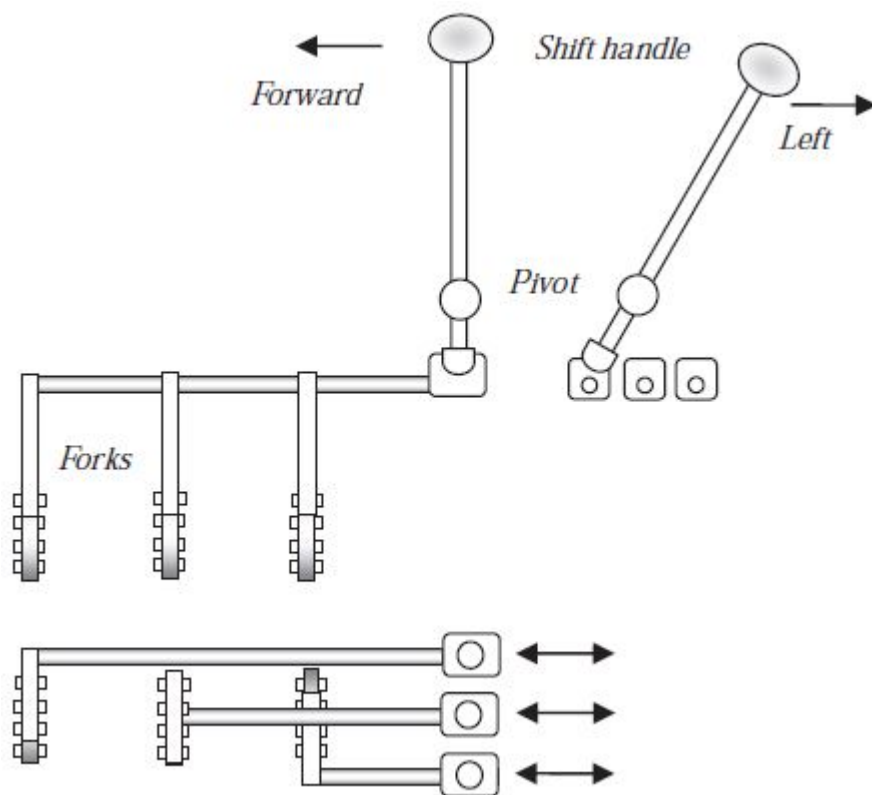


Figure 15 – Shifting mechanism (Mashadi & Crolla, 2012)

2.1.3.6 Automatic transmission

2.1.3.6.1 Introduction

Automatic transmissions make driving an easier task as the driver does not need to worry about shifting. It is only necessary to select a gear when the driver wants to

change from driving forward to backwards and vice-versa or when the driver wants to park (D. A. Crolla, 2009). In automatic transmission, it is required that the driver also choose a gear, but not frequently since there are only four gears: Drive, Neutral, Park and Reverse. While for manual transmissions it is necessary to constantly shift having in account the desired speed, speed limits, road slope and so on, for automatic transmission the driver rarely needs to shift because the transmission itself takes in account the parameters mentioned before. So, the driver is mainly responsible for steering and choose the gear pattern needed to suit the general driving conditions. The automatic gearbox system allows the vehicle to be either moved off or brought to rest in a smooth manner, without the driver having to do anything except press the accelerator or brake pedal. Automatic gearboxes can have a selection of drive methods, ranging from a fluid flywheel or torque converter to conventional gear sets, similar to manual gearboxes but with electronic control of clutch and gear shift (Hillier, 2012).

Early automatic transmissions had limited gear selection such as 'drive' for forwards, 'reverse' for backwards and 'park' and 'neutral' for stationary situations where no drive was transmitted to the road wheels. More modern automatic gearboxes have up to eight forward-moving gears to improve overall performance and fuel efficiency. The conventional reverse, park and neutral complete the automatic transmission assembly (Hillier, 2012).

Despite existing more than one forward moving gears, the shifting can be performed automatically without driver interaction. In Figure 16, it is illustrated an inside view of an automatic gearbox.

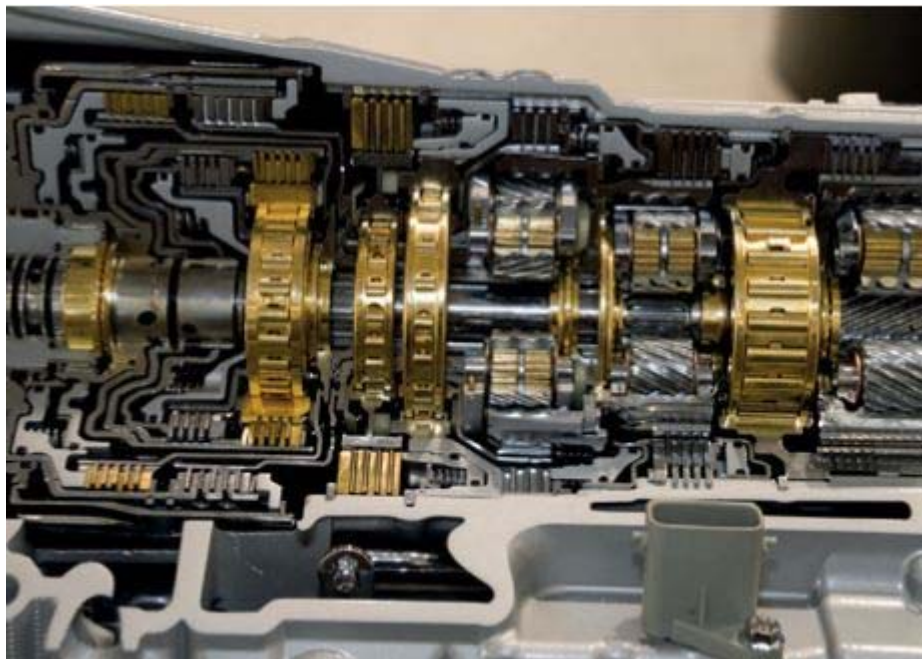


Figure 16 - Automatic gearbox internals (Hillier, 2012)

The inclusion of electronic control systems in modern vehicles gives the driver the option of an automatic gearbox that can also be driven manually. This option can be easily selected (Hillier, 2012).

Due to the recent progress in electronics new types of automatic transmissions have been developed. The first example of this progress is the development of the AMT where well-developed manual transmissions (MT) which are highly efficient, are electronically controlled. Another derivation from manual transmissions is the DCT, known to have a continuous torque flow (Mashadi & Crolla, 2012).

2.1.3.6.2 Conventional automatic transmissions

In a conventional automatic transmission, the clutch system is replaced by a fluid coupling or torque converter in order to eliminate engaging/disengaging movements during gear change. The gearing system is completely different from the other architectures. Instead of conventional gears, planetary or epicyclic gears are used to perform gear ratio. These changes make conventional automatic transmissions totally different from the manual transmissions internally as the external shape is quite similar, as can be seen in Figure 17 (Mashadi & Crolla, 2012).

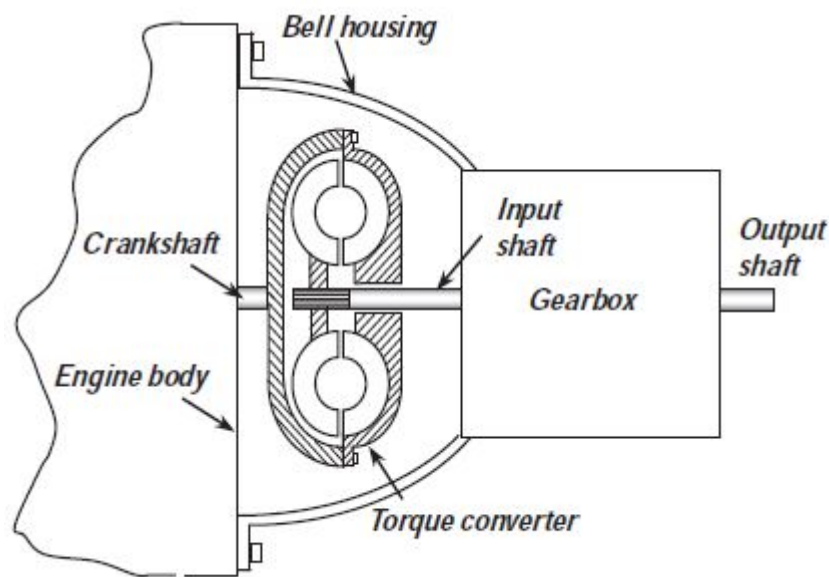


Figure 17 - Overall construction of a conventional automatic (Mashadi & Crolla, 2012)

By using epicyclic gears it is possible to achieve several gear ratios. For multi-speed transmission, several Planetary Gear Sets are necessary. When two sets of PGS are connected, it results in coupled gears (Mashadi & Crolla, 2012).

2.1.3.6.3 AMTs

AMT means “automated manual transmission”. This kind of transmission combines the best characteristics of manual and automatic transmissions: the high efficiency of manual transmissions and the easy operating of automatic transmissions (Mashadi & Crolla, 2012). In addition, there is an economical advantage when compared to MT and AT. AMTs are more efficient than ATs and are programmed to shift more effectively than a regular driver. This transmission architecture is quite linked to Formula 1 (D. A. Crolla, 2009). An AMT is developed from a manual transmission with the addition of electronic controls. The clutch actuation which is the most difficult part of gear changing, is performed automatically while in manual transmission is performed manually. This makes gear shifting more comfortable and driving less exhaustive, so the driver can focus on other operations (Mashadi & Crolla, 2012). This is one more reason why this transmission architecture is being implemented in trucks (D. A. Crolla, 2009).

As gear shifting is performed automatically (shift-by-wire), the mechanical connection between selector lever and transmission is eliminated. When compared to an automatic transmission, AMT presents the following advantages: possibility to use manual transmission manufacturing facilities which results in lower production costs; high efficiency and lower weight. The interruption of flow during shift actuation is presented as the main disadvantage of an AMT (Mashadi & Crolla, 2012).

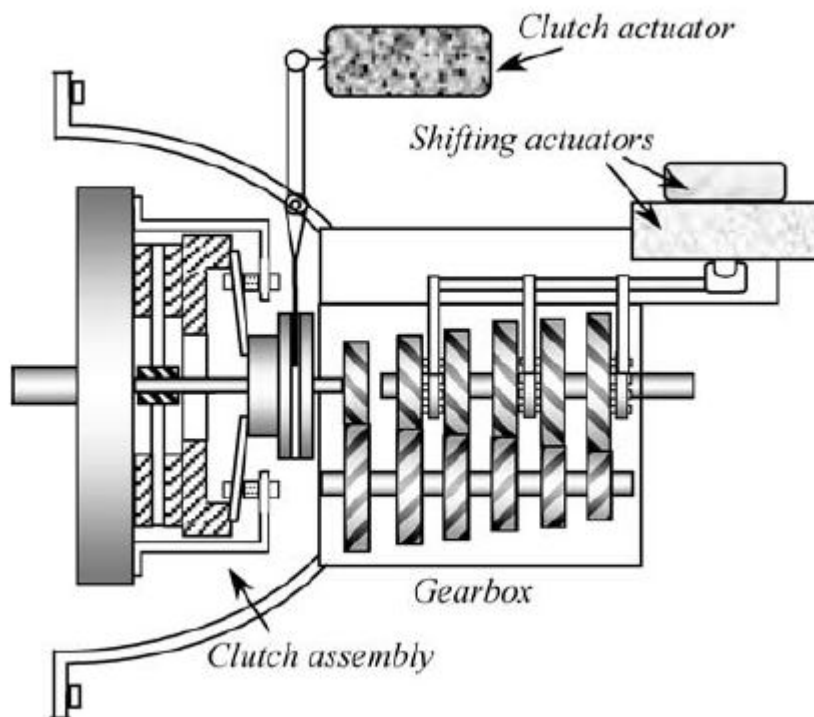


Figure 18 – Actuators for AMT (Mashadi & Crolla, 2012)

It is required two tasks to convert a manual transmission to an AMT (Mashadi & Crolla, 2012):

- Installing three actuator systems for the three actions a driver usually performs: 1 for clutching plus 2 for two movements in gear selection (Figure 18);
- Adding a control unit to make shifting decisions.

2.1.3.6.4 DCTs

DCT are the initials for “dual clutch transmission”. As mentioned before, the main disadvantage of manual transmissions is the torque interruption during gear shifting. According to the working principle of a manual transmission, to perform gear shifting it is necessary to disengage the clutch in order to eliminate transmission of engine’s torque to the gearbox. By this way, gear meshing can be performed by means of a sliding motion. When the next gear is selected, the clutch is engaged (released) to re-establish the torque transmission between the engine and the gearbox. With the concept of sliding motion for gear shifting it is inevitable the torque interruption (Mashadi & Crolla, 2012).

DCT design is based on the concept of manual transmissions. Basically, a DCT is a layshaft-type gearbox with two input shafts. It is necessary a clutch system for each shaft. Clutch systems are controlled hydraulic and electronically. The biggest challenge of DCT is the coordination between engagement of one clutch and disengagement of the other one. Usually, in DCTs odd gears are attached to one shaft and the even gears to the other. By doing this, gear shifting can be performed with no torque interruption. The torque interruption is roughly eliminated because of the engagement of one clutch and disengagement of the other one is performed simultaneously. The results are smooth accelerations comparing to manual and even automatic transmissions (Mashadi & Crolla, 2012).

Contrarily to automatic transmissions in which torque converters transmit the engine torque to the gearbox, in DCTS dry clutches or multi-plate wet clutches are used (Mashadi & Crolla, 2012). This type of transmission presents the following advantages when compared to conventional automatic transmissions (Halderman, 2012):

- Quicker throttle response;
- No drop in engine speed when the driver releases the throttle;
- Instant gear changes;
- Improved fuel economy.

When compared to CVT or conventional AT the disadvantages are the following (Halderman, 2012):

- No torque multiplication advantage of a torque converter;
- Not as fuel efficient as a CVT or transaxle.

Function

As seen before, odd and even gears are placed in distinct shafts. Two input shafts and two power paths are present on DCT. At the end of each input shaft there is a clutch to allow power interruption for gear shifting. In Figure 19, it is illustrated a schematic of a 6-speed DCT.

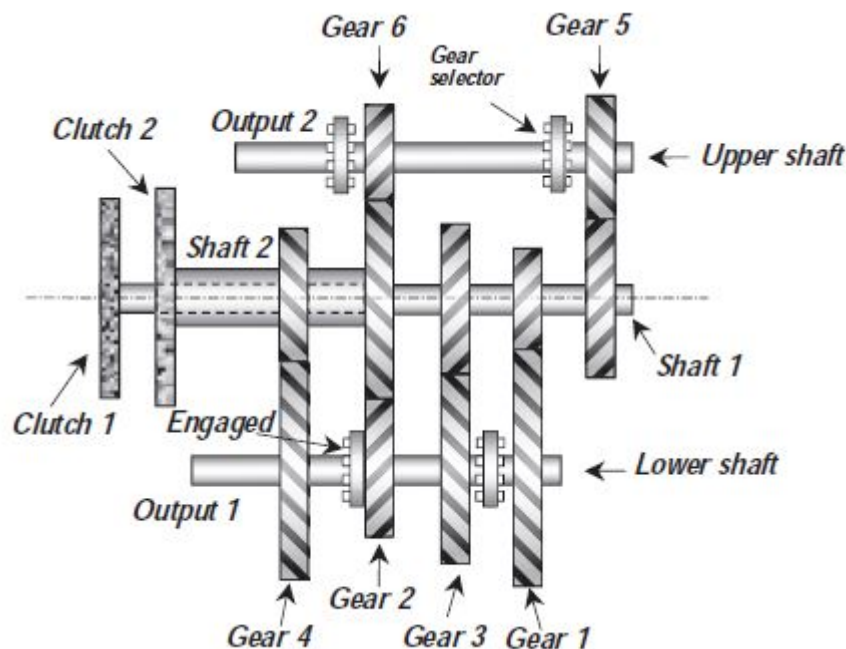


Figure 19 – Schematic of a 6-speed DCT (Mashadi & Crolla, 2012)

According to Figure 19, both input shafts are placed coaxially with one solid shaft (shaft 1). Therefore, shaft 1 is placed inside the shaft 2 (hollow shaft). Each shaft (1 and 2) has a clutch from where the engine power is transmitted. Lower and upper shafts are the output shafts. The power is transferred to each of these shafts by selecting one gear. Like a typical layshaft gearbox, the gears of the output shaft rotate freely until they are connected to the shaft by the gear selector. Only one gear is connected to the output at a time.

Based on Figure 19, if the vehicle is in gear 2, clutch 2 is engaged (released) and shaft 2 is transmitting power to the output shaft 1. To shift to gear 3, the gear selector which is located between gear 1 and 3, will move in direction to gear 3. After that, when output

shaft 1 starts rotating, so does gear 3 and mating gear of input shaft 1. Clutch 1 rotates while it is not engaged. Pre-selecting a gear will force the free clutch to rotate. To shift the gear after pre-selection both clutches need to swap over releasing the clutch which is engaged and engaging the free one. In the example of Figure 19, the engine power is transmitted through shaft 1 and gear 3 to the lower output shaft. The other gear selector is still connected to gear 2 and the input shaft 2 will also rotate freely due to the rotation of gear 2. In a DCT there are two paths for torque flow and during gear shifting torque flow is complex. Torque flow depends on several factors such as: clutch clamp forces, layshaft angular velocity and the type of gear shift (upshift or downshift). Since both clutches engage and disengage simultaneously, if the pressure profiles are not appropriate, high torque variations will be induced which results in undesired longitudinal acceleration inputs to the vehicle and its passengers (Mashadi & Crolla, 2012).

2.1.3.6.5 CVTs

CVT are the initials for “Continuously Variable Transmission”.

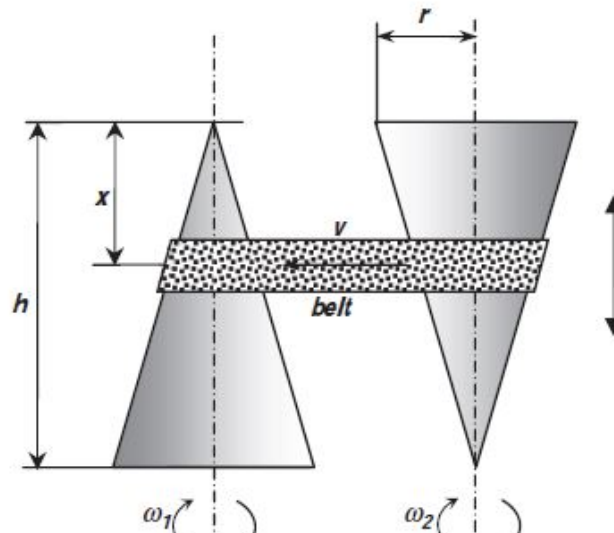


Figure 20 – Conceptual CVT (Mashadi & Crolla, 2012)

CVT concept is based on a principle to transmit power from one rotating shaft to another with continuously variable speeds. The concept is basically two similar cones with a flat wrapped belt around them as showed in Figure 20 (Mashadi & Crolla, 2012).

When the belt moves along the axes of the cones, the value of x varies and for a given speed ω_1 of input shaft, the output angular speed ω_2 can be obtained as follows (Mashadi & Crolla, 2012):

$$\omega_2 = \frac{x}{h-x} \omega_1 [rad / s] \quad \{2.4\}$$

In eq. {2.4}, it is considered zero slip.

For a fixed value of input angular speed, the output speed ranges from low values near zero up to high values. This feature of a CVT is useful in applications where different gear ratios and smooth shifting between ratios are required. The concept illustrated in Figure 20 does not fulfil the requirements of all the applications mainly when high duty cycles and durability requirements are to be fulfilled (Mashadi & Crolla, 2012).

Classification

CVTs can be classified based on different parameters. In Figure 21, it is presented the classification flowchart based on the nature of the production of the output torque.

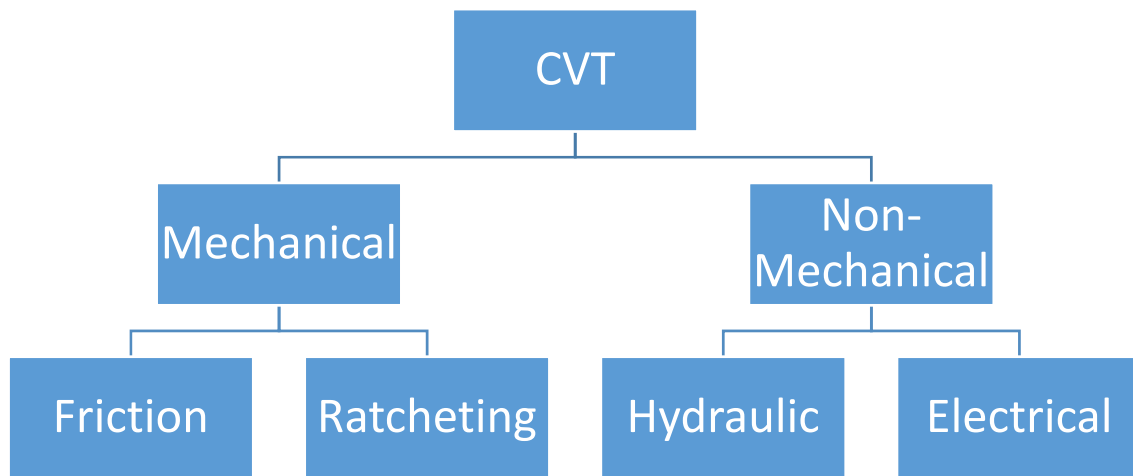


Figure 21 – Automotive CVTs' classification (adapted from (Mashadi & Crolla, 2012))

Friction CVTs

The power provided by an ICE cannot be fully exploited by a traditional gearbox due to the finite number of gears. With an infinite number of gears that is achieved with a CVT the engine can be operated at the ideal operating point in terms of economy or

performance as required (Naunheimer *et al.*, 2010). Friction CVTs produce torque due to the friction between mating surfaces by means of belts or rollers. There are two types of friction CVTs in use in automobiles: belt and toroidal CVTs. A belt CVT system is similar to belt-pulley drive with the exception that the pulleys are not fixed and are able to move apart. In Figure 22 is presented the geometry of a typical belt CVT.

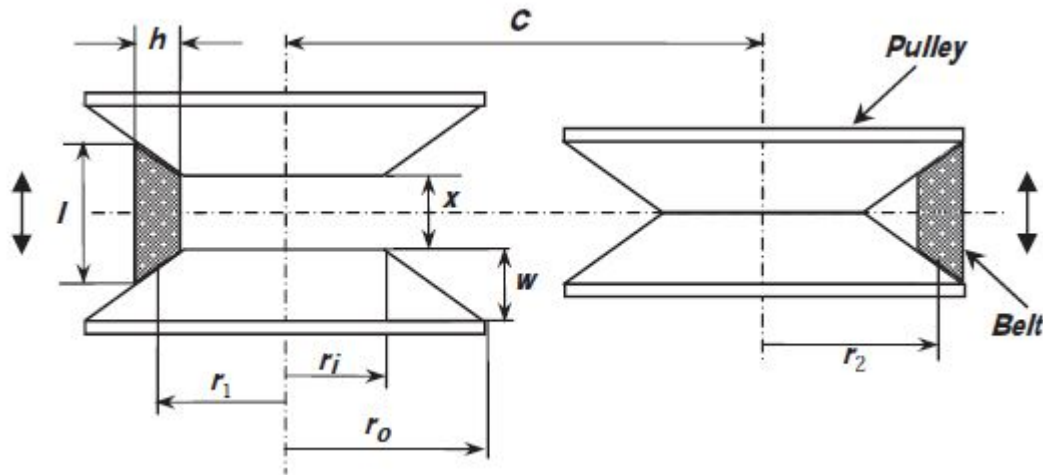


Figure 22 – Geometry of a typical belt CVT (Mashadi & Crolla, 2012)

Pulley transmissions are used in most of the mass-produced passenger cars. The main component of the pulley transmission is the variator. It consists on taper discs and a chain. Power is transmitted by means of friction through the chain. The chain runs between two axially adjustable taper discs (Naunheimer *et al.*, 2010). Both pulleys have fixed axes of rotation at a certain distance from each other. The sides of each pulley are controlled to move apart or together laterally, but the displacements of each pulley are the opposite of the other (Mashadi & Crolla, 2012). This axial adjustment provides variable diameters infinitely varying the ratio (Naunheimer *et al.*, 2010). In Figure 22 it is illustrated an extreme case when the sides of one pulley are fully apart and the sides of the other pulley are fully closed (Mashadi & Crolla, 2012).

One of the crucial issues in friction CVTs is the amount of friction generated at the contacting surfaces or points. Friction is used to create traction forces at the contact region which creates a torque. However, friction is associated to two disadvantages: build-up of heat and sensitivity to wear. Lubricating oil can be a solution but as a consequence it will reduce the friction and torque capacity of the system. Friction force depends not only on coefficient of friction but also on normal load. If the normal load is increased, coefficient of friction can be compensated (Mashadi & Crolla, 2012).

The torque-related pressure is a parameter to take in deep attention since excessive pressure reduces the efficiency of the chain, leading to increased power consumption, power loss by the contact pressure pump and increased stress on the transmission. Chain slipping is undesirable as this can would lead to the destruction of the

transmission. In conclusion, design and reliability of the pressure pump (and its control) is a crucial factor in CVT of this type (Naunheimer *et al.*, 2010).

In the case of chains there are two types: tensional link chains and thrust link chains (Figure 23). The first type is more efficient as less power is required to adapt the chain to the ratio radii and they are better suited to higher torques. For this purpose, short pitch of thrust link chains requires more work. In terms of meshing impact and noise generation, this type of chains play a better role (Naunheimer *et al.*, 2010).

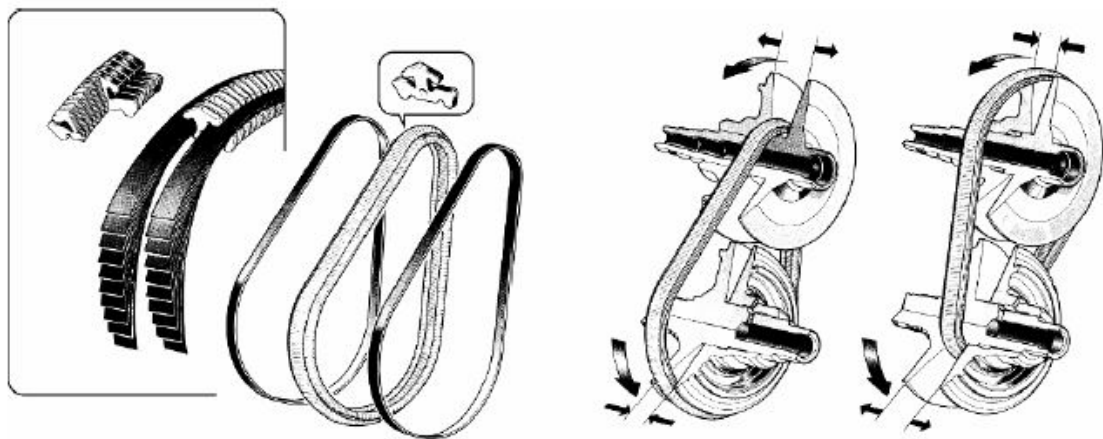


Figure 23 – Elements of a thrust link chain and principle of operation of the variator (Naunheimer *et al.*, 2010)

Ratcheting CVTs

A ratchet is a mechanism that produces rotation in one direction regardless of the direction of the input rotation. The originated output is non-uniform and oscillatory. Ratcheting gearboxes' concept is based on the production of the output rotation from a series of discontinuous rotations summed up at the output. Several mechanisms are built in order to produce a gearbox (Mashadi & Crolla, 2012).

Non-mechanical CVTs

There are two distinct types of non-mechanical CVTs which are based on hydraulic and electrical components. A hydraulic system composed by a pump and a hydraulic motor constitutes a gearbox. The pump is responsible for the production of oil flow and turning the motor. The input mechanical power is transformed to fluid power by the pump and once again to mechanical power by the motor. This system can be converted to a CVT

by selecting a pump and/or a motor of variable displacement type. On the other hand, an electric system consists of a generator and a motor which makes it similar to the hydraulic gearbox. The mechanical input power can rotate the generator and produce electric current that can drive an electric motor and produce mechanical power at the output. The conversion of this system to a CVT is achieved by using electric power circuits that control the motor's voltage or frequency.

2.1.3.7 Gearbox

2.1.3.7.1 Introduction

As seen before, gearbox is one subsystem of a transmission system. The transmission is a subsystem of powertrain.

Modern vehicles equipped with ICE require a gearbox and a differential. Gearboxes provide a range of gear ratios to maintain engine power and speed to allow the vehicle to travel at appropriate road speeds relevant to the driver and environmental requirements. The differential provides a constant torque amplification ratio (final drive) and acts as a power split device for left and right wheels. The role of the gearbox is to provide different torque amplification ratios from the engine to the wheels when it is necessary and at different driving conditions (Hillier, 2012; Mashadi & Crolla, 2012). The number of gears in a vehicle range from three (old cars) to eight (recent models) (Mashadi & Crolla, 2012).

The gearbox can be manual or automatic. The lowest forward gear (first gear) allows the engine to operate at peak power and torque, but produces low road speed. By moving up the gears, the ratio allows the engine revolutions to remain fairly constant within the peak operating efficiency, while increasing road speed. Using a high gear will allow the vehicle to travel at higher road speeds, but lower engine speeds, resulting in increased fuel efficiency and lower emissions (Hillier, 2012).

First gear is generally used to accelerate the vehicle from a standstill position or manoeuvre the vehicle at low speed, such as when parking. That is one more reason why a gearbox is necessary: to help in the acceleration of a vehicle from standstill (Hillier, 2012).

A well-known benchmark for the overall performance of a vehicle is the 0-100km/h time. A reference value can be considered 10 seconds. This implies an acceleration of 2.7 m/s^2 . For this acceleration, for a typical vehicle with a mass of 1000 kg and wheel radius of 30 cm it is required a torque of 710 Nm at the wheel centre. In an electric motor, the maximum acceleration is achieved at start of motion and is up to 2-3 times the average

acceleration. If an acceleration of 2.5 times the average acceleration, the maximum torque at the wheel would be 1775 Nm. In Table 2, it is presented the results comparison between IC engines and EV engines in terms of maximum torques they generate. For this comparison, it was considered engines with an average power of 60 kW (80 hp) (Mashadi & Crolla, 2012).

Table 2 – Gear ratios for different power sources (Mashadi & Crolla, 2012)

Power source (60 kW)	Max torque [Nm]	Overall gear ratio (n)	Gearbox ratio (n_g)
SI Engine (Petrol)	100-120	15-18	4.3-5.1
CI Engine (Diesel)	170-200	9-10	2.6-2.9
Electric motor	250-500	3.5-7	1-2

According to Table 2, the engines (torque sources) have insufficient torque and so they are not suitable to connect directly to the driving wheels (Mashadi & Crolla, 2012).

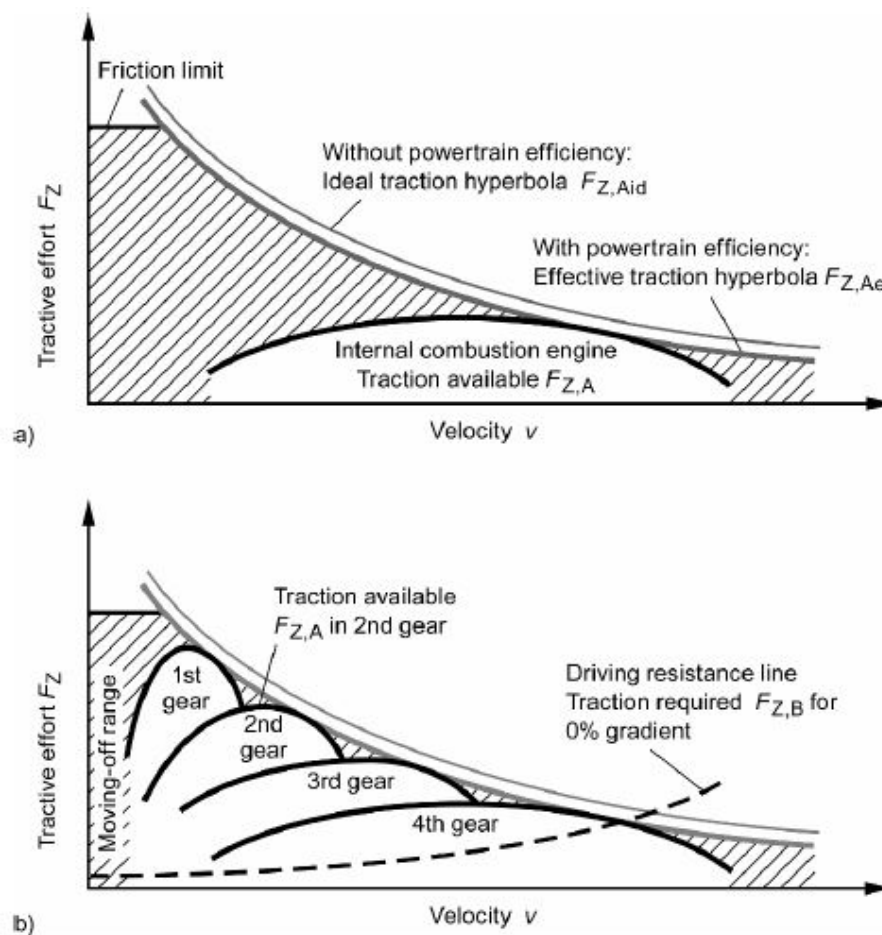


Figure 24 – a) secondary map of an ICE without a gearbox; b) secondary map of an ICE with rear-mounted 4-speed gearbox: traction diagram (Naunheimer *et al.*, 2010)

Another function of the gears is to allow the engine to run at higher speeds when the car is moving slowly, so making use of the vehicle's maximum power and torque for climbing steep hills or accelerating rapidly from low speeds (Hillier, 2012). There is a more technical reason why gearbox use is imperative when the vehicle is equipped with an ICE. If the full-load engine power ($P_{\max}$) was available over the whole speed range, the traction hyperbolas shown in Figure 24 a) would result successfully.

For the ICE, the traction profile would result too. The maximum traction between tyres and road is limited by the friction limit. The whole shaded area of Figure 24 a) cannot be applied without an additional output converter. The output converter must convert the characteristic of the ICE in a way that approximates as closely as possible the traction hyperbola (Figure 24 b)).

There are two possible examples of output converters (Naunheimer *et al.*, 2010):

- Speed converter: mechanical or hydrodynamic clutch;
- Speed-torque converter: geared transmission or continuously variable transmission.

The proportion of impossible driving states (proportion of shaded area) is significantly smaller when an output converter is applied and the power potential of the engine can be better applied.

Based on Figure 24 b), increasing the number of speeds as much as possible gives a correspondingly better approximation to the traction hyperbola. With CVT, the traction hyperbola can correspond to the traction characteristic curve over the range of ratios (Naunheimer *et al.*, 2010).

The second fundamental disadvantage of the ICE presented on chapter 2.1.1 is solved by means of a moving-off element (force locking clutch). The moving-off clutch (master clutch) is generally mounted between the engine and the transmission in the powertrain (Naunheimer *et al.*, 2010).

The purpose of the transmission is to define engine operating points (EOP) that are favourable in terms of efficiency and performance. Low gears are used at low speeds, and should provide high tractions for accelerating and hill climbing. On the other hand, high gears, are used at higher speeds and must allow proper matching between engine torque-speed characteristics with vehicle speed and acceleration (Mashadi & Crolla, 2012).

2.1.3.7.2 Manual gearbox

The manual gearbox consists of sets of gears that amplify the output given by the engine to enable the driving force at the road wheels to be increased sufficiently to overcome the resistance to movement of the vehicle. The gearbox enables the engine speed to be

kept within its working limits irrespective of the speed of the vehicle. It also allows a 'neutral' position where the engine can run without moving the vehicle.

Early manual gearboxes had four forward-moving gears and a reverse gear to drive the vehicle backwards. Modern gearboxes now consist of up to six forward-moving gears in manual gearboxes (up to eight in automatic gearboxes). This enhances the vehicle's overall performance and fuel efficiency. By having more gear ratios, the engine's power can be kept within the peak power band to improve acceleration and performance (Hillier, 2012).

2.1.4 Planetary gear set

Planetary gear sets or Epicyclic gear sets are responsible for the gear ratios in automatic transmissions. Figure 25 shows a typical PGS composed by a sun gear S , a ring gear (or annulus gear) R and a number of planets P that have their rotation axle on a common carrier known as planet carrier C . The example of PGS presented on Figure 25 is similar to the one studied in this thesis. The PGS has three sets of concentric gears meshing at two different diameters. The R has internal teeth and the other gears have external teeth. Most of the times in ATs the PGS are operated with one of the shafts fixed to the gearbox housing making the remaining shafts the input and output (D. A. Crolla, 2009).

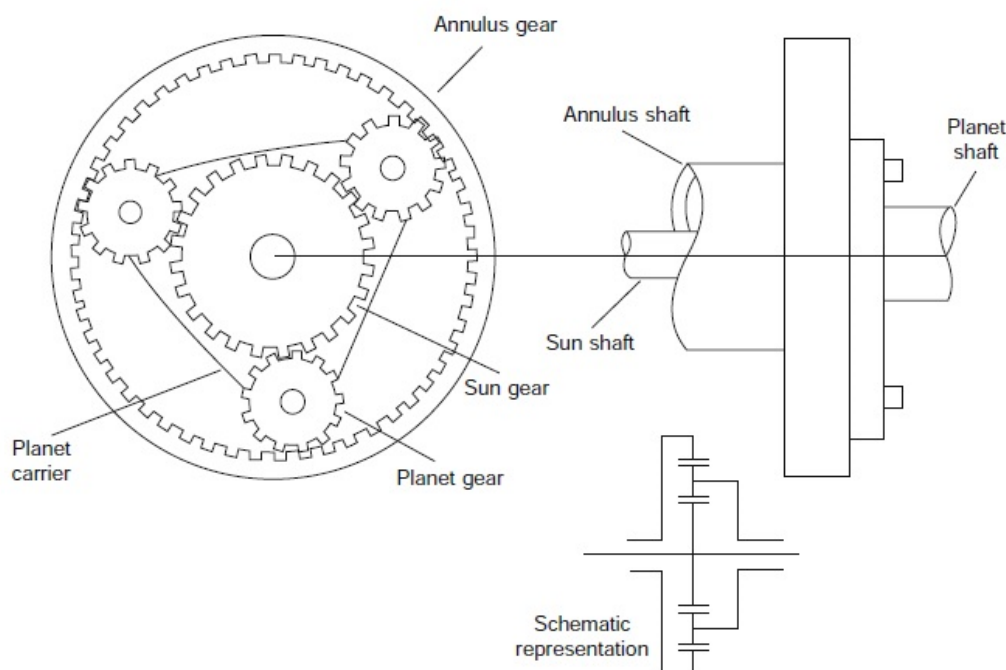


Figure 25 – Planetary gear set (D. A. Crolla, 2009)

In addition to the presented information in this chapter, more topics were studied due to its importance in the understanding of the system in study. The extra topics are presented on Appendix (Extra theoretical aspects).

2.2 Calculations' methods and expressions

2.2.1 Springs

2.2.1.1 Introduction

Springs are elastic components which exert reaction forces, or torques, and absorb energy, which is usually stored and later released, infrequently or over continuous cycles. Usually, springs are made of metal however they can also be made of plastic for application with light loads (Juvinall & Marshek, 2012).

In engineering, springs are commonly used to control movement, apply (reaction) forces (to external loads), limit impact forces, reduce vibration and force measurement (Carvil, 1993). Springs have been intensively studied. They are mass-produced and consequently low cost, and different configurations have been found for a variety of applications (Shigley, Mischke & Budynas, 2002).

In the design development, both the level of stress and the amount of deflection are important, usually at the same time (T. H. Brown, 2005).

There are several types of mechanical springs, such as:

- Spiral springs;
- Leaf springs;
- Helical springs;
- Spring washers;
- Moulded springs.

Helical springs can be organized in three distinct categories: compression, tension and torsion springs. Diverse types of spring washers can be seen in Figure 26.



Figure 26 – Types of spring washers (Juvinall & Marshek, 2012)

In this report, only Belleville washers, Wave springs and compression springs will be studied as only these were used in the design of the Drive-Neutral-Reverse subsystem. Below it can be found specific information related to these three types of springs.

2.2.1.2 Diaphragm springs

This type of springs is mainly used in dry clutches to generate force for the generation of friction on the clutch plate. These springs have a conical shape with a number of slots cut out in the upper part as can be seen in Figure 27. The tips of the fingers come in contact with the release bearing when clutch pedal is depressed. At the engaging stage the release bearing is held away from the diaphragm spring's fingers (Mashadi & Crolla, 2012).



Figure 27 – Simple form of a diaphragm spring
(Mashadi & Crolla, 2012)

2.2.1.3 Compression springs

When the word spring is used, its image is always associated to the image of a helical spring. Typically, they have circular cross sections and their coils are cylindrical, though other shapes are possible. Although other cross sections such as square are also common (T. H. Brown, 2005).

A compression spring is an open-coil helical spring that reacts to a compressive load applied axially. As expected, when a compression coil spring is loaded, making it shorter, it pushes back against the load and tries to get back to the original length.

2.2.1.3.1 Applications

Compression springs can be applied on automotive engines and stamping presses as well as in medical devices, cell phones, electronics and sensitive instrumentation

devices. Cone shape metal springs are generally used in applications where low solid height and increased resistance to surging are required (Spring, 2011).

2.2.1.3.2 Direction of wind

Similar to a screw type thread, coil spring can be wound in either left or right-hand direction. For applications where a spring operates inside another one, it is necessary to coil the springs in opposite directions. If a spring is screwed onto a thread, the direction of the helix must match the one of the thread, i.e., a left hand wound spring will spiral in the same direction as a left hand threaded screw (Spring, 2011).

2.2.1.3.3 End types

In Figure 28 it is presented the four types of ends which are usually applied with compression springs.

A spring with plain ends has a noninterrupted helicoid. The ends are the same as if a long spring has been cut into sections. Springs with plain ends which are squared or closed are obtained by deforming the ends to a zero-degree helix angle.

For important applications, where a better transfer of load is required, springs should be both squared and ground (Shigley *et al.*, 2002).

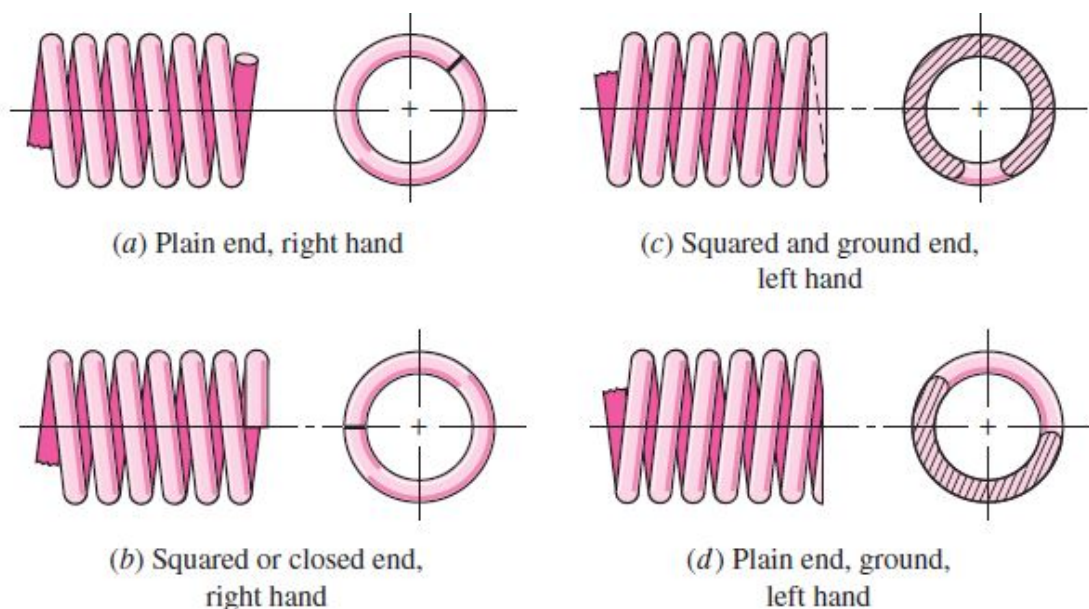


Figure 28 – Types of ends for compression springs (Shigley *et al.*, 2002)

2.2.1.3.4 Counting coils

The total number of coils is counted from consecutive tips according to Figure 29.



Figure 29 – Counting coils for compression springs (Spring, 2011)

The number of coils and the spring length are directly affected by the type of end used. Consequently, solid height and the pitch will be affected, too. These relations can be visualized in Figure 30.

Term	Type of Spring Ends			
	Plain	Plain and Ground	Squared or Closed	Squared and Ground
End coils, N_e	0	1	2	2
Total coils, N_t	N_a	$N_a + 1$	$N_a + 2$	$N_a + 2$
Free length, L_0	$pN_a + d$	$p(N_a + 1)$	$pN_a + 3d$	$pN_a + 2d$
Solid length, L_s	$d(N_t + 1)$	dN_t	$d(N_t + 1)$	dN_t
Pitch, p	$(L_0 - d)/N_a$	$L_0/(N_a + 1)$	$(L_0 - 3d)/N_a$	$(L_0 - 2d)/N_a$

Figure 30 – Influence of the types of ends of compression springs (Shigley *et al.*, 2002)

2.2.1.3.5 Induced residual stresses

Set removal or presetting is a process applied during the manufacturing process of the compression springs in order to induce useful residual stresses. This process consists in producing the spring longer than needed and afterwards compressing it to its solid height. By doing this, the required final free length is set. Furthermore, since the yield

strength of the springs' material has been exceeded, residual stresses are induced opposite in direction to those induced in service.

During design development, it is recommended to consider an excess of 10 to 30 percent of the initial free length, for springs to be preset.

If the stress at the solid height is greater than 1.3 times the torsional yield strength, distortion may occur. If this stress is much less than 1.1 times, it is difficult to control the resulting free length.

Set removal is especially useful when the spring is used for energy-storage applications, as this process increases the strength of the spring. For applications where the springs are submitted to fatigue, set removal should not be applied (Shigley *et al.*, 2002).

2.2.1.3.6 Spring materials

Springs can be manufactured either by hot or cold forming. The choice of the manufacturing process depends on the size of the material, spring index and desired properties.

Winding of the spring induces residual stresses through bending on the mechanical spring, although these are normal to the direction of the torsional working stresses in a coil spring. It is quite frequent in spring manufacture to relieve the springs after winding by mean of a thermal treatment. There is a vast variety of materials suitable for the manufacturing of springs, such as:

- Plain carbon steels;
- Alloy steels;
- Corrosion-resisting steels;
- Phosphor bronze;
- Spring brass;
- Beryllium copper;
- Nickel alloys.

The main and more common property for comparison of the different materials is the tensile strength. However, to know the tensile strength of a certain material, the diameter of the wire must be known because the tensile strength varies significantly with the wire diameter.

The processing of the material influences the tensile strength too. Although the torsional strength is a necessary property to the spring design and its performance's analysis, spring materials are most of the times tested only for tensile strength. A reasonable explanation for this is the low cost of this kind of tests. Posteriorly a rough deduction of

the torsional yield strength may be obtained by assuming that the tensile yield strength is between 60 and 90 per cent of the tensile strength (Shigley *et al.*, 2002).

2.2.1.3.7 General calculations

In this chapter, it is presented a calculation process for compression springs and some formulas. There are several ways to calculate the different parameters of a compression spring. Not always, this method was followed, however the formulas were based on the ones presented above. This is because sometimes some parameters were already defined. The process presented here was useful to understand the calculation process of a compression spring design.

First of all, to design a spring is necessary to define the appropriate material taking in account the working environment and kind of application. Shear modulus and tensile strength depends on the material and these values will be crucial on the spring design.

The next step is about calculating the mean diameter and the inner or outer diameter of the spring. At this time, it is necessary to do an assumption of the value of the coil diameter. To find out the optimum value of the coil diameter some iterations should be done. It is important to compare the ID (Inner diameter) of the spring to any work over rod requirements. Furthermore, lowest OD (Outer diameter) (or ID) dimension (having in account the tolerances) should be incorporated when examining the work over rod requirements (Spring, 2011).

Mean diameter and ID can be calculated by means of the following equation (Spring, 2011):

$$D_m = OD - d_c [m] \quad \{2.5\}$$

$$ID = OD - 2d_c [m] \quad \{2.6\}$$

Where OD represents the outer diameter of the spring and is expressed in meters [m]; d_c represents the coil diameter and is expressed in meters [m].

As expected, the diameter of a compression spring increases when compressed. The value of this increment depends on the pitch. At this step, OD expansion is calculated and then this value is compared to any work in hole requirements. It is necessary to consider the great side of the OD tolerance when examining the work in hole requirements (Spring, 2011).

$$OD_{\text{expansion}} = \left[\sqrt{D_m^2 + \frac{p^2 - d_c^2}{\pi^2}} + d_c \right] - OD[m] \quad \{2.7\}$$

In which D_m represents the mean diameter of the spring and is expressed in meters [m]; p corresponds to the pitch and is a nondimensionalise variable; and π corresponds to the mathematical constant.

In the following step, pitch is calculated and consequently coils per mm, and spring index. It must be verified that pitch of the spring is not greater than the OD, otherwise coiling difficulties will appear during the manufacturing. As previously mentioned, some factors depend on the type of end of the spring so as a consequence spring index depends on the type of end, too (T. H. Brown, 2005; Spring, 2011). Spring index formulas as a function of the type of end are presented on Table 3.

Table 3 – Spring index of compression springs (adapted from (Spring, 2011))

Type of end			
Open	Open/Ground	Closed	Closed/Ground
$\frac{FL - d_c}{N_A}$	$\frac{FL}{N_T}$	$\frac{FL - 3d_c}{N_A}$	$\frac{FL - 2d_c}{N_A}$

$$\text{Coils per inch} = \frac{1}{p} \left[1 / m \right] \quad \{2.8\}$$

$$\text{Spring index} = C = \frac{D_m}{d_c} \quad \{2.9\}$$

It is crucial to analyse the spring index value. If it is not between 4 and 10, further examination and considerations are required, such as (Spring, 2011):

- Special tooling (pencil arbors for tight index springs);
- Special packaging (for flimsy or slinky type-high index springs);
- Stress relieving (would either not be applicable for springs with indexes over 25 or would require special handling to prevent distortion);
- Tangling issues;
- No grinding (because grinding high index springs will actually worsen squareness);
- No additional operations, such as plating or tumbling (because of tangling issues);

So, it is recommended to do some iterations changing some initial estimated values in order to avoid manufacturing difficulties and price rise.

Next step is about choosing the design criteria because each criteria has its own calculation method (Spring, 2011):

- Design based on physical dimensions;
- Design based on spring rate;
- Design based on two loads;
- Design based on one load and spring rate;
- Design based on one load and free length.

After defining the spring rate and the number of active coils, the number of total coils can be calculated. This does not apply to designs that are based on physical dimensions (Spring, 2011).

The formulas to calculate the N_T are presented on the table of Figure 30. To calculate SH and verify that all the requirements are fulfilled and that the all the load heights are above solid height. It is necessary to consider a variation of 3% to the nominal solid height to calculate the maximum solid height. Formulas to calculate the SH is presented on the table of Figure 30 but in this table “solid length” was used instead of “solid height”.

As most of the times, there are load requirements. The stress at the load heights must be calculated and compared to the tensile strength of the spring’s material. If the percent stress (formula presented below) at any load height is greater than 40% (and the spring will not be compressed more than this particular height) then a set operation or allow for set should be considered. If the percent stress is greater than 60% a re-design must be considered (T. H. Brown, 2005; Spring, 2011).

$$\sigma = \frac{8F \cdot D_m}{\pi d_c^3} [Pa] \quad \{2.10\}$$

$$\text{Corrected stress} = \frac{8F \cdot D_m}{\pi d_c^3} K [Pa] \quad \{2.11\}$$

In which F corresponds to the load and is expressed in Newtons [N], where (T. H. Brown, 2005):

$$K = \frac{4C - 1}{4C - 4} + \frac{0.615}{C} [] \quad \{2.12\}$$

Where C corresponds to the spring index and is a nondimensionalized variable.

The percent stress can be calculated as follows (Spring, 2011):

$$\text{Percent stress} = \frac{\sigma}{R_m} \cdot 100 [\%] \quad \{2.13\}$$

Where σ corresponds to the stress and is expressed in Pascal [Pa]; R_m corresponds to the material tensile strength and is expressed in Pascal [Pa].

Unless the working range is specifically known, the stress at solid height must be examined. If the percent stress at solid height is greater than 40% then a set operation or allow for set should be considered. If the percent stress is greater than 60%, a re-design must be done. If an overstressed situation is presented, it must be known how the spring will be used in order to see if it is possible to avoid the overstress. Overstress load and respective stress can be calculates as follows (Spring, 2011):

$$F_s = SR(FL - SH) [N] \quad \{2.14\}$$

$$\sigma_s = \frac{8F_s \cdot D_m}{\pi \cdot d_c^3} [Pa] \quad \{2.15\}$$

$$\text{Percent stress at solid height} = 100 \cdot \frac{\sigma_s}{R_m} [\%] \quad \{2.16\}$$

Where FL (Free length) corresponds to the free length of the spring and is expressed in meters [m]; SR corresponds to the spring stiffness and is expressed in Newton per meter [N/m]; and SH corresponds to the solid height of the spring and is expressed in meters [m].

Design to physical dimensions

In order to calculate the main properties of a spring based on the physical dimensions the following information is required:

- Wire diameter;
- Outside diameter or hole diameter or inside diameter or rod size;
- Free length;
- Number of coils (active/inactive)
- Ends configuration;
- Solid height.

In addition to the general considerations presented above, spring rate must be calculated (T. H. Brown, 2005):

$$SR = \frac{G \cdot d_c^4}{8N_A \cdot D_m^3} [N / mm] \quad \{2.17\}$$

Where G corresponds to the material's shear modulus and is expressed in Pascal [Pa]; N_A corresponds to the number of active coils and is a nondimensionalize variable.

Spring rate is generally defined between 20 and 80% of the available deflection where it is linear (Spring, 2011).

Design to spring rate

With the purpose to calculate the number of active coils of a compression spring the following information is required:

- Wire diameter;
- Outside diameter or hole diameter or inside diameter or rod size;
- Free length;
- Spring rate;
- Number of inactive coils;
- Ends configuration;
- Solid height;

In addition to the general considerations presented above, number of active coils must be calculated (T. H. Brown, 2005).

$$N_A = \frac{G \cdot d_c^4}{8 \cdot SR \cdot D_m^3} [] \quad \{2.18\}$$

Design to two loads

The following information is necessary to design a compression spring for two loads:

- Wire diameter;
- Outside diameter or hole diameter or inside diameter or rod size;
- Load at height 1;
- Load at height 2;
- Ends configurations;
- Solid height;
- Number of inactive coils.

In addition to the general considerations presented above, the following parameters must be calculated:

- Spring rate:

$$SR = \frac{\Delta F}{\Delta h} = \frac{F_2 - F_1}{h_1 - h_2} [N / m] \quad \{2.19\}$$

Where F_2 corresponds to the load at working height 2 and is expressed in Newton [N]; F_1 corresponds to the load at working height 1 and is expressed in Newton [N]; h_1 corresponds to the working height 1 and is expressed in meters [m]; and h_2 corresponds to the working height 2 and is expressed in meters [m].

- Number of active coils is calculated according to equation {2.18}.
- Free length:

$$FL = h_1 + \frac{F_1}{SR} [m] \quad \{2.20\}$$

Design to one load/rate

When is the intention of the designer to calculate a spring for one load or based on its rate the following information is necessary:

- Wire diameter;
- Outside diameter or hole diameter or inside diameter or rod size;
- Load at height 1;
- Spring rate;
- Ends configurations;
- Solid height;
- Number of inactive coils.

In addition to the general considerations presented above, the following parameters must be calculated:

- Number of active coils can be calculated according to equation {2.18};
- Free length can be calculated based on equation {2.20}.

Design to one load/free length

The following information is required to calculate the compression spring for one load and with the free length as a requirement:

- Wire diameter;
- Outside diameter or hole diameter or inside diameter or rod size;
- Load at height 1;
- Free length;
- Ends configurations;
- Solid height;
- Number of inactive coils.

In addition to the general considerations presented above, the following parameters must be calculated:

- Spring rate:

$$SR = \frac{\Delta F}{\Delta h} = \frac{F_1}{FL - h_1} [N / m] \quad \{2.21\}$$

- Number of active coils according to equation {2.18}

2.2.1.3.8 Fatigue

Springs are almost always subject to fatigue loading. In fact, all the springs applied on the DNR subsystem are under dynamic loading conditions. In case of an expected life of millions of cycles without failure, spring must be designed for an infinite life. This is the case of the valve springs of a combustion engine (Shigley *et al.*, 2002).

Shot peening can be applied on springs dynamically loaded in order to improve fatigue strength. It may increase the torsional strength at least twenty per cent. Shot size is about 1/64 inches, so spring coil wire diameter and pitch must allow for complete coverage of the spring surface (Shigley *et al.*, 2002). The highest values of torsional endurance limits of springs steels are those reported by Zimmerli. He discovered that

size, material and tensile strength haven't any effect on the endurance limits (infinite life only) of spring steels in sizes under 10 mm. Endurance limits tend to level out at high tensile strengths however the justification for this phenomenon is still unknown (Shigley *et al.*, 2002). Unpeened springs were tested from a minimum torsional stress of approximately 138 MPa to a maximum of 621 MPa. Peened springs were tested from a range of approximately 138 MPa to 931 MPa (Shigley *et al.*, 2002). The following endurance strength components for infinite life were found to be:

- Unpeened:

$S_{sa} = 241 \text{ MPa}$ and $S_{sm} = 379 \text{ MPa}$

- Peened:

$S_{sa} = 398 \text{ MPa}$ and $S_{sm} = 534 \text{ MPa}$

The torsional modulus of rupture S_{su} is necessary for the designer's torsional fatigue diagram and may be calculated as follows (Shigley *et al.*, 2002):

$$S_{su} = 0.67 R_m [Pa] \quad \{2.22\}$$

For shafts and other kind of machine elements it is quite common fatigue loading in the form of completely reversed stresses. Although, compression springs are never used as both compression and extension springs. Furthermore, compression springs are usually installed with a preload so that the working load is additional.

Amplitude force must be calculated as follows (Shigley *et al.*, 2002):

$$F_a = \frac{F_{\max} - F_{\min}}{2} [N] \quad \{2.23\}$$

Where $F_{\max}$ corresponds to the maximum load applied on the spring and is expressed in Newton [N] and $F_{\min}$ corresponds to the minimum load applied on the spring and is expressed in Newton [N].

Midrange force may be obtained as follows (Shigley *et al.*, 2002):

$$F_m = \frac{F_{\max} + F_{\min}}{2} [N] \quad \{2.24\}$$

Shear stress amplitude is (Shigley *et al.*, 2002):

$$\tau_a = K_B \frac{8F_a D_m}{\pi d^3} [Pa] \quad \{2.25\}$$

In which F_a corresponds to the amplitude force and is expressed in Newtons [N], where K_B is obtained as follows (Shigley *et al.*, 2002):

$$K_B = \frac{4C + 2}{4C - 3} \quad \{2.26\}$$

K (eq. {2.12}) can be used instead of K_B .

Midrange shear stress is given by (Shigley *et al.*, 2002):

$$\tau_m = K_B \frac{8F_m D_m}{\pi d_c^3} [Pa] \quad \{2.27\}$$

Slope line is given by ("Formulas for Compression Spring Fatigue Design," n.d.):

$$r = \frac{\tau_a}{\tau_m} [] \quad \{2.28\}$$

Where τ_a corresponds to the shear stress amplitude and is expressed in Pascal [Pa] and τ_m corresponds to the midrange shear stress and is expressed in Pascal [Pa].

Shear endurance limit according to Gerber is given by (Shigley *et al.*, 2002):

$$S_{se}^{Gerber} = \frac{S_{sa}}{1 - \left(\frac{S_{sm}}{S_{su}} \right)^2} [Pa] \quad \{2.29\}$$

Where S_{sa} corresponds to the shear stress amplitude limit and is expressed in Pascal [Pa]; S_{sm} corresponds to the maximum endurance strength component for infinite life and is expressed in Pascal [Pa] and S_{su} corresponds to the torsional modulus rupture and is expressed in Pascal [Pa].

Shear endurance limit according to Goodman is given by ("Formulas for Compression Spring Fatigue Design," n.d.):

$$S_{se}^{Goodman} = \frac{S_{sa}}{1 - \left(\frac{S_{sm}}{S_{su}} \right)} [Pa] \quad \{2.30\}$$

Shear stress amplitude limit according to Gerber failure criteria can be calculated as follows ("Formulas for Compression Spring Fatigue Design," n.d.):

$$S_{sa}^{Gerber} = \frac{r^2 S_{su}^2}{2 S_{se}} \left(-1 + \sqrt{1 + \left(\frac{2 S_{se}}{r \cdot S_{su}} \right)^2} \right) [Pa] \quad \{2.31\}$$

In which r corresponds to the slope line and is a nondimensionalized variable.

Shear stress amplitude limit according to Goodman failure criteria can be obtained as follows ("Formulas for Compression Spring Fatigue Design," n.d.):

$$S_{sa}^{Goodman} = \frac{r S_{se} S_{su}}{r S_{su} + S_{se}} [Pa] \quad \{2.32\}$$

The value of S_{sa} according to Sines is equal to S_{sa} according to Zimmerli.

Factor of safety against fatigue is given by ("Formulas for Compression Spring Fatigue Design," n.d.):

$$SF_f = \frac{S_{sa}}{\tau_a} \quad \{2.33\}$$

2.2.1.4 Wave springs

A wave spring is like a compression spring made of coiled flat wire with waves along the coils to give it a spring effect. This kind of springs can replace conventional round wire compression springs in applications where tight load deflection in space critical environments is required. Generally, for the same load, wave springs occupy between 30 and 50% of the compressed height space compared to round wire springs. This provides not only space savings but also smaller assemblies which use less material and consequently lower production costs.

Wave springs are manufactured from a single filament of flat wire formed in continuous and precise coils with uniform diameter and waves. It must be taken in account the material for high temperatures or corrosive operating environments.

Wave springs can be found in a wide variety of applications where structural considerations restrict the installation height, where uniform force distribution is needed and/or uniform rotational pressure is required over 360°. This kind of spring washers is suitable for both static and slightly dynamic load applications. However, fatigue is critical for dynamic load applications. For higher life cycles is it necessary a stronger spring. Special solutions must be found for extremely dynamic applications, since considerable stress on the sine wave curves can cause material abrasion at the contact point. Typical applications include connectors, clutches, valves, pumps, seals, medical equipment, etc (Spring, 2011).

In Figure 31 is presented the terminology of a wave spring that will be applied in the following calculations.

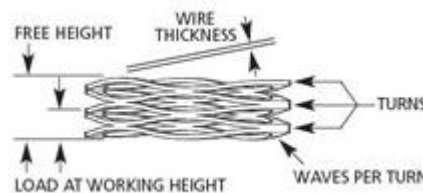


Figure 31 – Terminology of a wave spring
(Spring, 2011)

There are four critical factors to be taken in account during design development of a wavy washer (Spring, 2011):

- Constraints of the application: Bore/shaft, ID/OD, etc;
- Load;
- Working height;
- Material.

2.2.1.4.1 Design general considerations

If a spring is designed to a static application, the percentage of stress at working height must be less than 100%. If this value is greater than 100%, spring will take a set. In case of dynamic application, the percentage of stress at working height must be less than 80%. If this value is greater than 80% spring will take a set. If the work height per turn is less than two times wire thickness, the spring will operate in a nonlinear range and actual loads may be higher than calculated.

Number of turns must be between 2 and 20. Number of waves per turn must be in half increment. Minimum radial wall must be equal to three times wire thickness. Maximum radial wall must be equal to ten times wire thickness.

It is not recommended to completely compress a wave spring (solid height) (Spring, 2011).

2.2.1.4.2 General calculations

Deflection, can be calculated as follows:

$$d = \frac{F \cdot K_w \cdot D_m^3 \cdot Z_w}{E \cdot b_w \cdot t_w^3 \cdot N_w^4} \cdot \frac{ID}{OD} [m] \quad \{2.34\}$$

Where F corresponds to the load and is expressed in Newton [N]; K_w corresponds to the multiple wave factor (Table 4) and is a nondimensionalized variable; D_m corresponds to the mean diameter and is expressed in meters [m]; Z_w corresponds to the number of turns and is a nondimensionalized variable; E corresponds to the materials' Young's Modulus and is expressed in Pascal [Pa]; b_w corresponds to the radial wall (width) of the wavy washer and is expressed in meters [m]; t_w corresponds to the wire thickness and

is expressed in meters [m]; N_w corresponds to the number of waver per turn and is a nondimensionalized variable; ID corresponds to the inner diameter of the wavy washer and is expressed in meters [m] and OD corresponds to the outer diameter and is expressed in meters [m].

Spring rate can be calculated according to the following formula:

$$SR = \frac{F}{d} = \frac{E \cdot b_w \cdot t_w^3 \cdot N_w^4}{K_w \cdot D_m^3 \cdot Z} \cdot \frac{OD}{ID} [N / m] \quad \{2.35\}$$

Where d corresponds to the deformation of the spring when compressed and is expressed in meters [m].

The value of K_w can be found via correspondence according to Table 4.

Table 4 – Multiple wave factor (Spring, 2011)

N_w	K_w
2.0-4.0	3.88
4.5-6.5	2.9
7.0-9.5	2.3
10 and over	2.13

OD expansion is a crucial factor mainly for wavy springs as this kind of springs is most of the times assembled in a housing (hole) (that is the case of the studied wavy spring).

$$OD_{expansion} = (0.02222 R_w \cdot N_w \cdot t_w) + b_w [m] \quad \{2.36\}$$

In which R_w corresponds to the effective wave radius and is expressed in meters [m].

Where:

$$R_w = \frac{(4W_H^2 + W_L^2)}{8W_H} [m] \quad \{2.37\}$$

In which W_H corresponds to the half wave free height and is a nondimensionalized variable; and W_L corresponds to the half wave length and is a nondimensionalized variable.

$$\theta_w = \arcsin\left(\frac{W_L}{2R_w}\right) \quad \{2.38\}$$

$$W_L = \frac{\pi D_m}{2N_w} \quad \{2.39\}$$

$$W_H = \left(\frac{\frac{FL}{Z} - t_w}{2} \right) \quad \{2.40\}$$

2.2.1.4.3 Fatigue

Operating stress can be calculated as follows:

$$\sigma = \frac{3\pi F \cdot D_m}{4b_w \cdot t_w^2 \cdot N_w^2} [Pa] \quad \{2.41\}$$

In case of a static application, need of presetting can be verified according to Table 5.

Table 5 – Wave springs design stresses for static/low cycles (Spring, 2011)

Static/Low cycles	
No preset	100% R_m
Preset springs	150% R_m

In case of dynamic applications, cycle life can be estimated according to Table 6.

Table 6 – Wave springs design stresses for dynamic service applications (Spring, 2011)

Dynamic service	
10 ⁴ cycle life	80% R _m
10 ⁵ cycle life	53% R _m
10 ⁶ cycle life	50% R _m
10 ⁷ cycle life	48% R _m

Fatigue stress ratio can be calculated based on the following formula:

$$X = \frac{\sigma - \sigma_1}{\sigma - \sigma_2} \quad \{2.42\}$$

Where σ_1 corresponds to the stress at the working height 1 and is expressed in Pascal [Pa]; σ_2 corresponds to the stress at the working height 2 and is expressed in Pascal [Pa]; and σ corresponds to the operating stress and is expressed in Pascal [Pa].

The value of σ_1 must be less than σ .

The cycle life can be estimated based on Table 7.

Table 7 - Wavy spring estimated cycle life (Spring, 2011)

X	Estimated life cycle (nr. of cycles)
<0.40	Under 30000
0.40-0.49	30000-50000
0.50-0.55	50000-75000
0.56-0.60	75000-100000
0.61-0.67	100000-200000
0.68-0.7	200000-1000000
> 0.7	Over 1000000

2.2.1.5 Belleville washers

Belleville washers are also called by Belleville springs or Disc springs. The main advantage of these type of springs is that it occupies only a small space in height (A. A. D. Brown, 1981).

Belleville washer is an annular dished steel ring which deflects axially under load (Carvil, 1993). The slight conical shape gives the washer a spring property. When compared to

helical springs, Belleville washers' conical shape provides them the ability to support high loads with relatively small deflections and solid heights (Spring, 2011).

Several springs may be used in series or parallel arrangements to give lower or higher stiffness, respectively (Carvil, 1993).

For a given deflection, a higher reaction load may be obtained by nesting, that is, to stack the springs in parallel. On the other hand, stacking the springs in series provides a larger deflection for the same load. When stacked in series, there is danger of instability (Shigley *et al.*, 2002). In Figure 32 it is shown the representation of the two stacking configurations for Belleville washers.

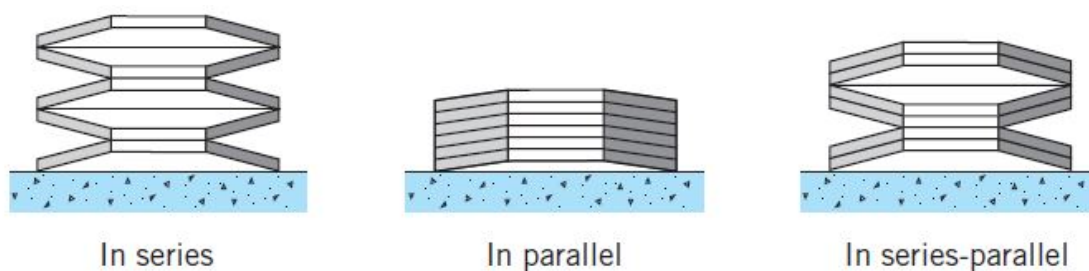


Figure 32 – Stacked Belleville washers (Juvinall & Marshek, 2012)

This kind of spring washers can also be applied as locking devices, but only in applications with low dynamic loads (Spring, 2011).

In this chapter, all the necessary data related to the disc springs can be found.

In Figure 33, it is illustrated a cross section of a Belleville washer and the nomenclature adopted for the following calculations.

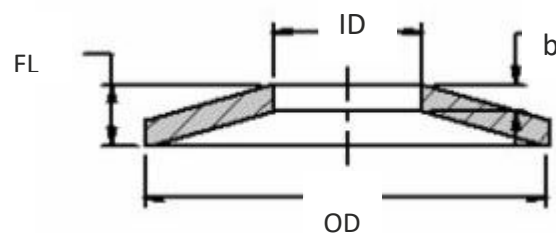


Figure 33 – Main dimensions for Belleville washers (Spring, 2011)

2.2.1.5.1 General calculations

The diameters' ratio can be calculated according to the following equation (Spring, 2011):

$$\delta = \frac{OD}{ID} \quad \{2.43\}$$

Where OD corresponds to the outer diameter of the disc spring and is expressed in meters [m]; ID corresponds to the inner diameter and is expressed in meters [m].

The load at maximum working height can be calculated as follows (Spring, 2011):

$$F_{h_{\max}} = \frac{4E \cdot b^3 \cdot d_{\max}}{(1-\nu^2) \cdot \gamma_1 \cdot OD^2} [N] \quad \{2.44\}$$

Where E represents the material's Young's modulus and is expressed in Pascal [Pa]; b corresponds to the thickness and is expressed in meters [m]; $d_{\max}$ represents the maximum allowed deflection and is expressed in meters [m]; ν corresponds to the material's Young's Modulus and is expressed in Pascal [Pa]; γ_1 represents the calculation coefficient and is a nondimensionalized variable.

The load at any working height can be obtained as follows (Spring, 2011):

$$F = \frac{4E \cdot b^4}{(1-\nu^2) \cdot \gamma_1 \cdot OD^2} \cdot \frac{d}{b} \cdot \left[\left(\frac{h_B}{b} - \frac{d}{b} \right) \cdot \left(\frac{h_B}{b} - \frac{d}{2b} \right) + 1 \right] [N] \quad \{2.45\}$$

In which d represents the spring's deflection and is expressed in meters [m]; h_B represents the unloaded height of truncated cone of free spring and is expressed in meters [m]. Where:

$$h_B = FL - b[m] \quad \{2.46\}$$

In which FL corresponds to the unloaded spring height and is expressed in meters [m].

The stress at a certain point can be calculated as follows (Spring, 2011):

$$\sigma = \frac{4E \cdot b \cdot d}{(1-\nu^2) \cdot \gamma_1 \cdot OD^2} \cdot \left[\gamma_2 \left(\frac{h_B}{b} - \frac{d}{2b} \right) + \gamma_3 \right] [Pa] \quad \{2.47\}$$

In which γ_2 and γ_3 represent calculation coefficients and are nondimensionalized variables. Where (Spring, 2011):

$$\gamma_1 = \frac{1}{\pi} \cdot \frac{\left(\frac{\delta-1}{\delta} \right)^2}{\frac{\delta+1}{\delta-1} - \frac{2}{\ln \delta}} \quad \{2.48\}$$

$$\gamma_2 = \frac{1}{\pi} \cdot \frac{6}{\ln \delta} \left(\frac{\delta-1}{\ln \delta} - 1 \right) \quad \{2.49\}$$

$$\gamma_3 = \frac{\delta-1}{\pi} \cdot \frac{3}{\ln \delta} \quad \{2.50\}$$

In which δ represents the diameter's ratio and is a nondimensionalized variable.

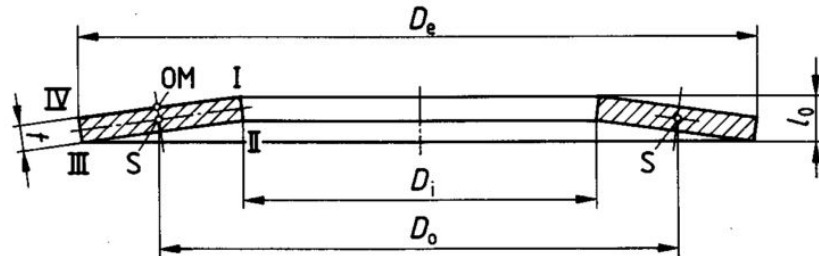
2.2.1.5.2 Fatigue

Static stress

In the case of disc springs made of spring steel (DIN EN 10 132-4), stress σ_{OM} in a flattened condition should not exceed the tensile strength of approximately 1600 MPa of the material (*Disc springs - theory and practice*, n.d.). In the case of springs made from DIN 17 221 or DIN 17 222 steel that are submitted to static loading or to moderate fatigue conditions, the stress at the point designated OM (Figure 34) shall be approximately equal to the yield strength of the material used (i.e. 1400 to 1600 N/mm²) (DIN 2093, 1992).

When submitted to higher stresses, it is likely that the springs will suffer from creep or relaxation (DIN 2093, 1992). In case of higher levels of stress, high deflection may be expected with a minor degree of direct resetting. Up to around 5000 load cycles, it is still considered a static application (*Disc springs - theory and practice*, n.d.).

Conical disc spring of group 1 or 2



Conical disc spring of group 3

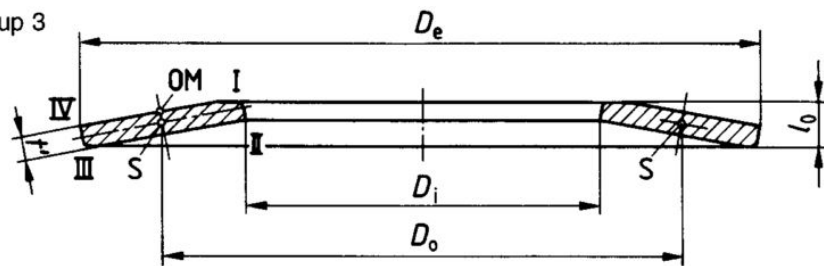


Figure 34 – Cross section of a single disc spring, including the relevant points of loading (“DIN 2093,” 1992)

Dynamic stress

Minimum pre-stress to avoid cracking

Springs that are subject to fatigue loading must be designed and installed in such a way that the initial deflection is from about $0.15h_0$ to $0.20h_0$ in order to avoid cracking at the upper inner edge (point I of Figure 34) as a result of residual stresses from the setting processes (DIN 2093, 1992).

It is possible to counteract the influence of residual tensile strength by providing enough pre-stress in the spring. Depending on the stress level of the disc springs, a greater pretension deflection may be necessary, or a smaller deflection may be enough (*Disc springs - theory and practice*, n.d.).

Stress in the work range

When submitted to vibrating stress, the response of a disc spring is determined by the tensile stress occurring on the underneath of the spring. The number of load cycles to fracture results from the minimum stress limit σ_u assigned to the maximum spring deflection. Whether the stress levels occurring at cross section point II or III are decisive depends on the dimensioning of the disc spring. (*Disc springs - theory and practice*, n.d.)

Endurance and fatigue strength diagrams

The graphs from Figure 35 to Figure 36 illustrate the endurance life of conical disc springs submitted to fatigue loading that have not been shot peened.

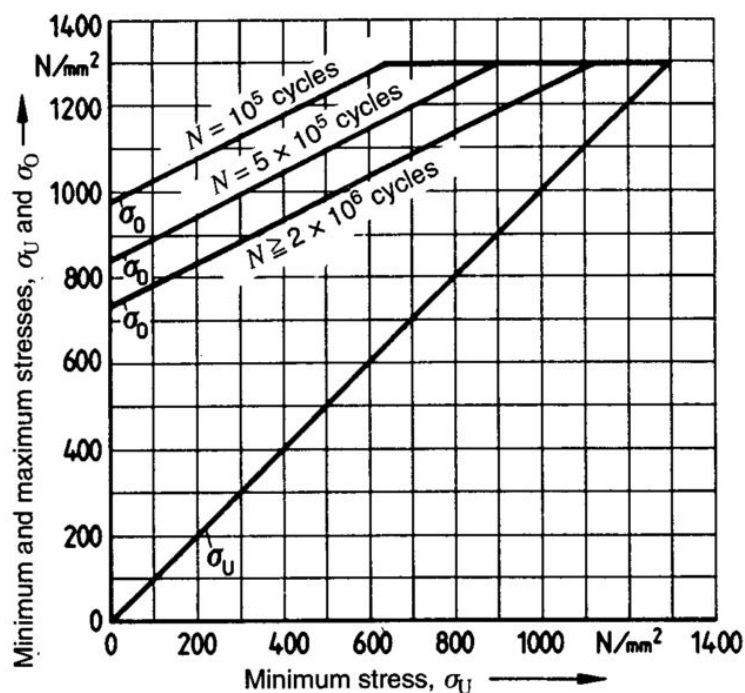


Figure 35 – Graphical representation of endurance life of springs where thickness is less than 1,25 mm (DIN 2093, 1992)

The information provided on these graphs represents the results of laboratory testing using fatigue testing equipment capable of producing sinusoidal loading cycles and the statistical results obtained for a 99% probability of endurance life. The test pieces were ten single discs with hardened surfaces, stacked in series, designed for use at ambient temperature, provided with an internal or external centring element with a smooth finish, having a minimal initial deflection from about $0.15h_0$ to $0.20h_0$.

These graphs specify guideline values for the range of stress, σ_H , as a function of the minimum stress, σ_U , at three different numbers of stress cycles, N_C , namely where N_C is less than or equal to $2 \cdot 10^6$, equal to 10^5 , and equal to $5 \cdot 10^5$. Intermediate values for other values of stress cycles may be estimated based on this information.

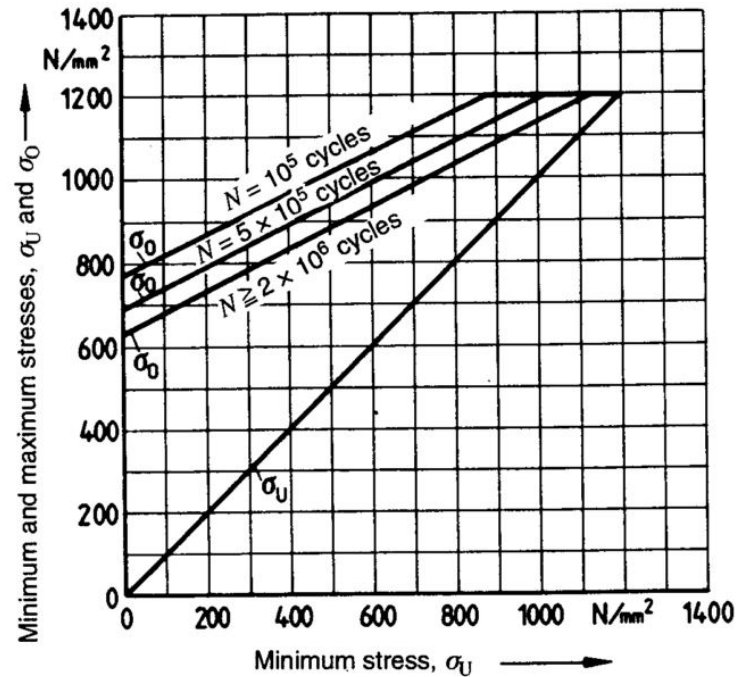


Figure 37 – Graphical representation of endurance life of springs where $6 \text{ mm} < \text{thickness} < 14 \text{ mm}$ (DIN 2093, 1992)

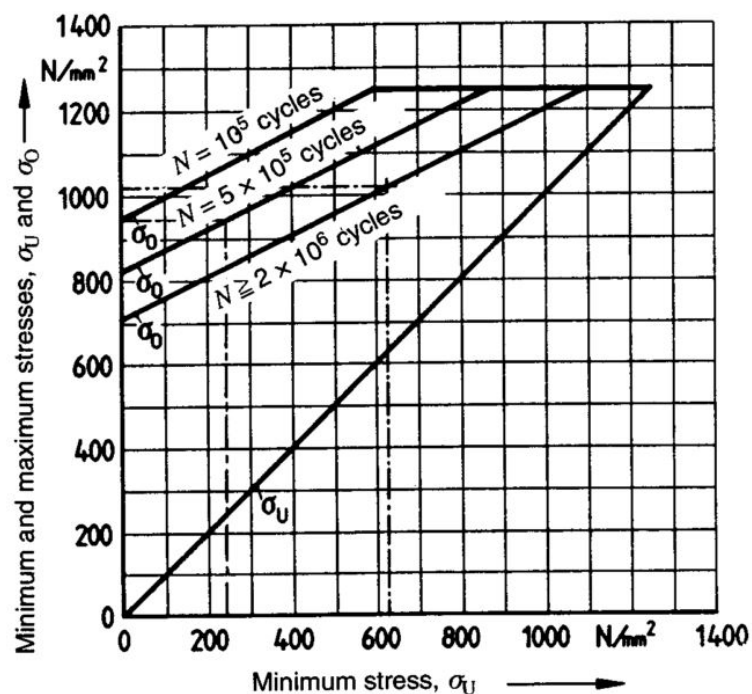


Figure 36 – Graphical representation of endurance life of springs where $1,25 \text{ mm} \leq \text{thickness} \leq 6 \text{ mm}$ (DIN 2093, 1992)

Calculations

On this chapter are presented all the necessary equations to study the fatigue of disc springs. First of all, it is necessary to calculate all the four coefficients, as follows (*Disc springs - theory and practice*, n.d.):

$$K_1 = \frac{1}{\pi} \cdot \frac{\left(\frac{\delta-1}{\delta}\right)^2}{\frac{\delta+1}{\delta-1} - \frac{2}{\ln \delta}} \quad \{2.51\}$$

$$K_2 = \frac{6}{\pi} \cdot \frac{\frac{\delta-1}{\ln \delta} - 1}{\ln \delta} \quad \{2.52\}$$

$$K_3 = \frac{3}{\pi} \cdot \frac{\delta-1}{\ln \delta} \quad \{2.53\}$$

In which δ represents the diameter's ratio and is a nondimensionalized variable.

The value of K_4 is not calculated directly because this value depends if there are contact surfaces between disc springs and other components. As the disc spring used on the DNR subsystem has no contact surfaces with other components, the value of K_4 can be considered as follows (*Disc springs - theory and practice*, n.d.):

$$K_4 = 1 \quad \{2.54\}$$

After that, stresses at different points may be calculated as follows (*Disc springs - theory and practice*, n.d.):

$$\sigma_{OM} = -\frac{4E}{1-\nu^2} \cdot \frac{b^2}{K_1 \cdot OD^2} \cdot K_4 \cdot \frac{d}{b} \cdot \frac{3}{\pi} [Pa] \quad \{2.55\}$$

$$\sigma_I = -\frac{4E}{1-\nu^2} \cdot \frac{b^2}{K_1 \cdot OD^2} \cdot K_4 \cdot \frac{d}{b} \cdot \left[K_4 \cdot K_2 \left(\frac{h_0}{b} - \frac{d}{2b} \right) + K_3 \right] [Pa]$$

{2.56}

$$\sigma_{II} = -\frac{4E}{1-\nu^2} \cdot \frac{b^2}{K_1 \cdot OD^2} \cdot K_4 \cdot \frac{d}{b} \cdot \left[K_4 \cdot K_2 \left(\frac{h_0}{b} - \frac{d}{2b} \right) - K_3 \right] [Pa]$$

{2.57}

$$\sigma_{III} = -\frac{4E}{1-\nu^2} \cdot \frac{b^2}{K_1 \cdot OD^2} \cdot K_4 \cdot \frac{1}{\delta} \cdot \frac{d}{b} \left[K_4 (K_2 - 2K_3) \left(\frac{h_0}{b} - \frac{d}{2b} \right) - K_3 \right] [Pa]$$

{2.58}

$$\sigma_{IV} = -\frac{4E}{1-\nu^2} \cdot \frac{b^2}{K_1 \cdot OD^2} \cdot K_4 \cdot \frac{1}{\delta} \cdot \frac{d}{b} \left[K_4 (K_2 - 2K_3) \left(\frac{h_0}{b} - \frac{d}{2b} \right) + K_3 \right] [Pa]$$

{2.59}

Where K_1 , K_2 , K_3 and K_4 represent calculation coefficients and are nondimensionalized variables; h_0 corresponds to the initial cone height of disc springs without ground ends.

Positive stress values correspond to tensile stress and negative stress values correspond to compressive stresses.

Where the value of h_0 is given by (DIN 2093, 1992):

$$h_0 = FL - b [m] \quad \{2.60\}$$

2.2.2 Centrifugal force in rotating cylinder

Centrifugal force in rotating cylinder is the centrifugal force exerted by a fluid in a piston, if the cylinder rotates about its symmetry axis ("Centrifugal force in rotating hydraulic cylinders - MATLAB," n.d.). This kind of cylinders are used in control mechanism, friction clutches, brakes, etc.

Some basic assumptions and limitations need to be taken in account:

- No inertial effects are considered in the model which means that the angular velocity of the cylinder is constant or changing at very low speed;
- Fluid inertia is neglected;
- Fluid compressibility is neglected.

Centrifugal force in a cylinder can be calculated as follows ("Centrifugal force in rotating hydraulic cylinders - MATLAB," n.d.):

$$F_c = \frac{\pi \rho \omega^2}{4} \left(OR^4 - IR^4 - 2r_p^2 (OR^2 - IR^2) \right) \quad \{2.61\}$$

Where π represents the mathematical constant; ρ represents the density of the fluid and is expressed in kilogram per cubic meter [kg/m^3]; ω represents the rotating speed and is expressed in radians per second [rad/s]; OR represents the outer hydraulic radius and is expressed in meters [m]; IR represents the inner hydraulic radius and is expressed in meters [m]; r_p represents the fluid entry radius and is expressed in meters [m].

In terms of design, the following relation must be achieved ("Centrifugal force in rotating hydraulic cylinders - MATLAB," n.d.):

$$r_p < \sqrt{\frac{(OR^2 + IR^2)}{2}} \quad \{2.62\}$$

In Figure 38, it is illustrated the effect of a rotating cylinder in a fluid (White, 2010).

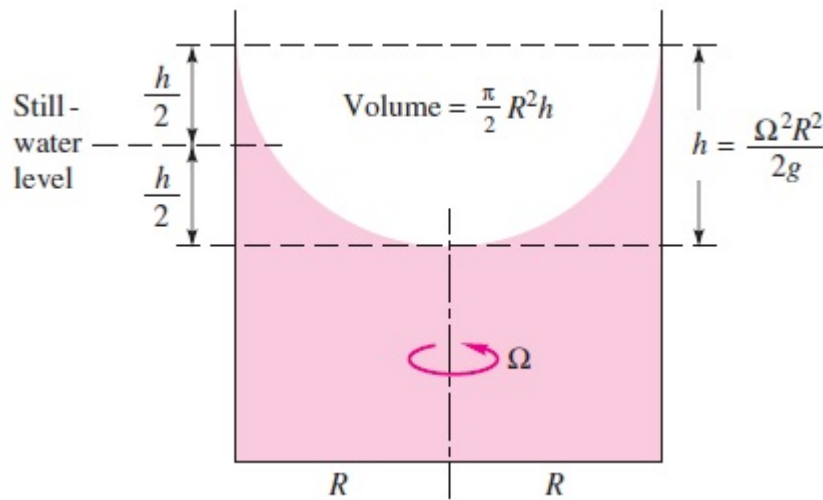


Figure 38 - Effect of centrifugal force on a fluid (White, 2010)

2.2.3 Transmissible torque

The transmissible torque of a wet clutch in multi-plate design can be obtained with the axial force, mean friction radius, friction coefficient and the number of friction surfaces, as follows (Naunheimer *et al.*, 2010):

$$T_C = F_{ax} \cdot R_F \cdot \zeta \cdot N_s [N.m] \quad \{2.63\}$$

In which F_{ax} represents the axial force applied on the clutch plates and is expressed in Newtons [N]; R_F represents the mean friction radius and is expressed in meters [m]; ζ represents the friction coefficient between clutch plates and is a nondimensionalized variable and N_s corresponds to the number of frictional surfaces and is a nondimensionalized variable. Where the mean friction radius can be obtained, as follows (Naunheimer *et al.*, 2010):

$$R_F = \frac{2}{3} \cdot \frac{(OR_F^3 - IR_F^3)}{(OR_F^2 - IR_F^2)} [m] \quad \{2.64\}$$

Where OR_F and IR_F represent the outer and inner friction surface radius, respectively and are expressed in meters [m].

2.2.4 Drag torque of wet clutches

A lot of research has already been made in this field of study. Despite not being completely dominated the mathematical models are feasible and are getting closer to the real values. The actual models are enough accurate for an approximate prediction of the drag torque while for validation of the system is advisable to use real tests.

Reduction of drag torque of wet clutches is a significant factor to improve transmission efficiency. Reduction of drag torque in disengaged wet clutches is the focus of transmission research because it is one of the potentials to improve transmission efficiency (Shihua & Zengxiong, 2010).

There was already a model for drag losses before research in this topic has been made by (Shihua & Zengxiong, 2010). This model is called by "Traditional model". Traditional model is not accurate to predict the decrease after oil film shrinking due to the difficulty

in modelling the flow pattern between two plates. In this model flow pattern was considered as laminar flow and full oil film in the gap between two plates.

In conclusion, traditional model is not accurate at high speed region (Shihua & Zengxiong, 2010). The equation of the tradition model can be seen below (Shihua & Zengxiong, 2010):

$$T_D = \frac{N_s \cdot \mu \cdot \pi (R_2^2 - R_1^2) \cdot \omega \cdot r_m^2}{h_p} [N.m] \quad \{2.65\}$$

Where N_s represents the number of friction surfaces and is a nondimensionalized variable; μ represents the fluid's dynamic viscosity and is expressed in Pascal second [Pa.s]; π represents the mathematical constant; R_2 and R_1 represent the outer and inner radius of the friction plates, respectively and is expressed in meters [m]; ω represents the rotating speed and is expressed in radians per second [rad/s]; r_m represents the mean radius and is expressed in meters [m]; and h_p represents the clearance between consecutive clutch plates and is expressed in meters [m].

This equation can only represent the rising portion of a typical drag torque curve at the low speed region when the clearance between clutch plates is full of oil film (Shihua & Zengxiong, 2010). Due to poor accuracy of the traditional model it was necessary to reveal the flow pattern through the clearance between the rotating plate and the stationary one and build a precise mathematical model to predict the characteristics of the drag torque. A lot of research on this topic was done by (Shihua & Zengxiong, 2010). It was developed a mathematical model using experimental verification. A test rig was set up to test the drag torque of disengaged wet clutch consisting of single plate and separate plate. In order to study the flow pattern, it was installed a high-speed camera. By this way it was possible to study the characteristics of oil film shrinking.

Visual test reveal that the oil film begins to shrink from outer to inner radius at the stationary plate and only flows along the rotating plate after shrinking. Meanwhile, drag torque decreases sharply due to little contact area between stationary plate and oil. According to the model analysis, the acceleration of the flow in radial direction caused by centrifugal force is the main reason for the shrinking at the constant feeding flow rate. It is important to underline the fact that the results of the model agree well with test data for non-grooved wet clutch. The wet clutch plates in study are grooved (Shihua & Zengxiong, 2010).

In Figure 39 it is presented the behaviour of oil film at different rotating speeds between the rotating plate and the stationary plate.

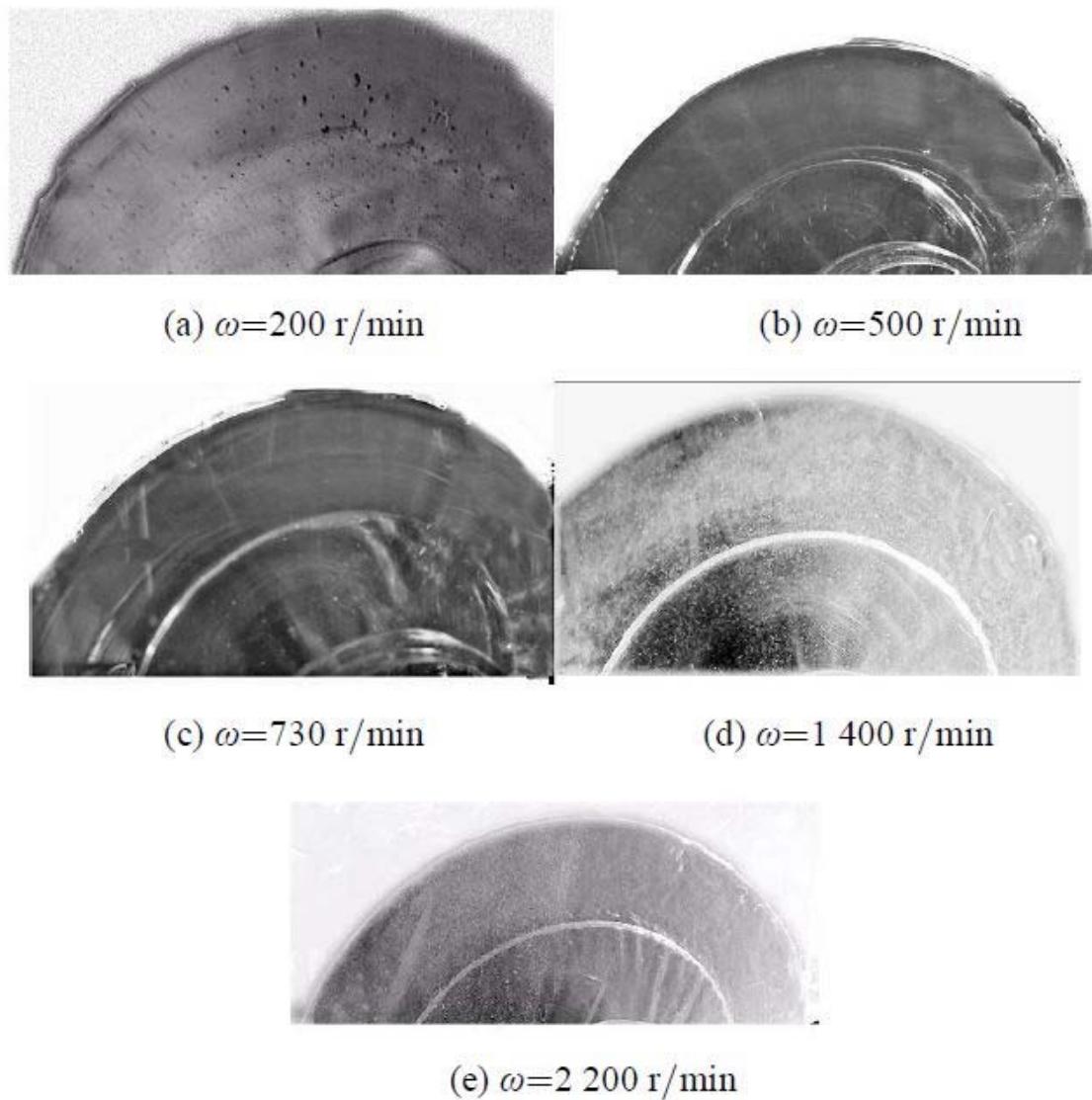


Figure 39 – Plate-view flow pattern in the gap at different rotating speed (Shihua & Zengxiong, 2010)

At low rotating speed, screwed flow in the clearance appears, especially in the inner radius region. It is also possible to state the clearance is full of oil. Increasing the speed Figure 39 (b) and (d), oil film in outer region starts to shrink and aeration emerges in the outer region due to centrifugal acceleration and continuity equation. In Figure 39 (d) it is possible to observe that screwed flow gradually disappears instead of radial flow. There is no oil attached to stationary plate and the oil flows radially only along rotating plate, whereas the fluid flowing near the rotating plate wall is more likely similar to the boundary layer flow over a single rotating plate. Cavity appears and flow pressure reduces caused by the radial acceleration of the fluid. In Figure 39(e), the fluid flows radially due to strong centrifugal force and becomes oil fog outside the clearance. From above visual observation, rotating speed is the most important factor on flow pattern.

On the other hand, flow pattern affects the drag torque through integral area calculation (Shihua & Zengxiong, 2010).

In Figure 40 is presented a scheme of what was explained above regarding to the flow pattern at different rotating speeds.

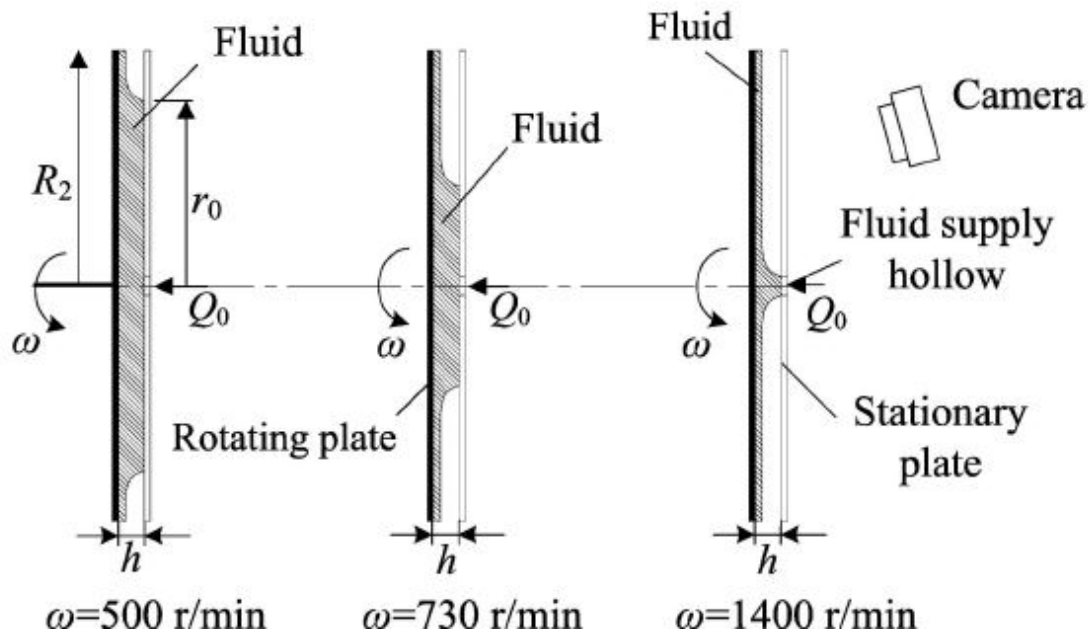


Figure 40 – Flow pattern in the clearance at different rotating speed (Shihua & Zengxiong, 2010)

Based on (Jibin & Zengxiong, 2009), drag torque increases linearly before oil film shrinking due to full oil film in the clearance.

In Figure 41 it is presented the comparison between the results of mathematical models and the experimental tests. It was applied and considered a constant feeding flow rate of 3L/min. It is conclusive that the traditional model is a straight line and is only similar to the other models until approximately 500 rpm. After that the values of drag torque are completely different. Analysing the experimental results, it is conclusive that from 1250 rpm drag torque is almost zero. This means that there is no shear stress on the plates. It is easy to conclude that no oil film leads to little shear stress (Shihua & Zengxiong, 2010). On the other hand, for circumferential degree mathematical model, the values are decreasing but do not reach zero. This means that this mathematical model is not accurate to predict the reduction of drag torque.

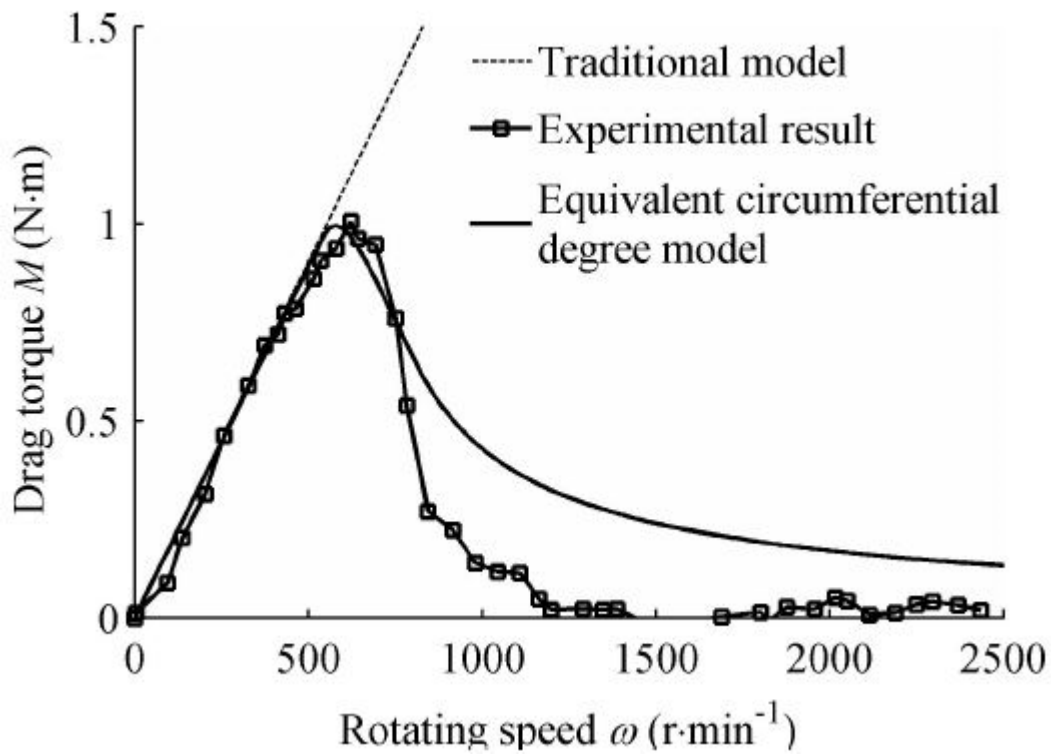


Figure 41 – Comparison between different mathematical models and experimental tests results (Shihua & Zengxiong, 2010)

Hereafter, it will be presented a mathematical model which is based on the following assumptions:

- Oil is incompressible and steady state;
- Flow in the wet clutch clearance is laminar and symmetrical;
- Gravity is neglected;
- Wet clutch plate in non-grooved;
- Friction plate and counter plate don't have any glide with its surface layer oil.

The oil motion in the radial direction between the plates is mainly driven by the centrifugal force, whereas the viscous tend to resist the motion. The relative rotating motion causes the drag torque due to oil viscosity (Heyan *et al*, 2013).

Based on the assumptions presented above and on Navier-Stokes equations, boundary conditions equations can be written as (Shihua & Zengxiong, 2010):

$$\begin{cases} v_r(r, 0) = 0, v_r(r, h_p) = 0 \\ v_\theta(r, 0) = 0, v_\theta(r, h_p) = \omega R \\ p_2 = p_0, p_1 = 0 \end{cases} \quad \{2.66\}$$

Where p_1 is the existing pressure at outer radius, p_2 is the flow pressure at inner radius, p_0 is the feeding pressure and h_p is the axial gap or clearance between consecutive plates.

In Figure 42, it can be found an auxiliary schematic of disengaged wet clutch with all the terminology that will be used on this mathematical model.

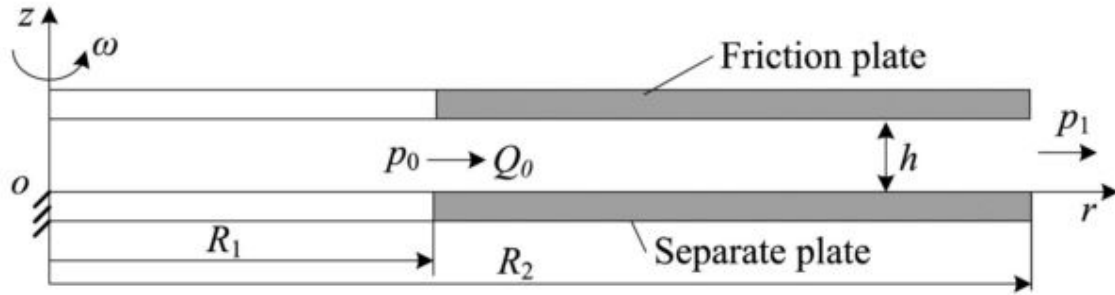


Figure 42 – Schematic of disengaged wet clutch (Yuan Shihua, Peng Zengxiong, 2010)

Radial speed distribution can be divided in three distinct stages. The first stage is caused by steady flow between two fixed plates. In this stage flow speed becomes lower when radius increases. The second stage is a consequence of centrifugal force. Its speed increases with the rise of rotating speed and radius. Finally, third stage is caused by flow inertia. It cannot be neglected when flow rate is very large. Therefore, centrifugal inertia and flow inertia accelerate the radial speed of oil in the clearance. As rotating speed increases, centrifugal speed distribution becomes dominant so that radial speed is strongly accelerated as it can be seen in Figure 39 (e) (Shihua & Zengxiong, 2010).

According to (Shihua & Zengxiong, 2010), radial oil pressure distribution along the radial direction can be calculated based on eq. {2.67}.

$$p(R) = \frac{6\mu Q}{\pi h_p^3} \ln \frac{R_1}{R} - \frac{3\rho\omega^2}{20} (R_1^2 - R^2) - \frac{27\rho Q^2}{140\pi^2 h_p^2} \left(\frac{1}{R^2} - \frac{1}{R_1^2} \right) [Pa]$$

{2.67}

Where Q represents the needing feed flow rate and is expressed in cubic meters per second [m^3/s]; R corresponds to an arbitrary radius and is expressed meters [m].

According to eq. {2.67}, it is possible to conclude that the rotating speed and flow rate affect pressure distribution of the fluid, especially when Q and ω are very high. This

equation is a junction of four pressure distributions. The first is caused by feeding pressure, the second one is caused by the steady flow between two fixed plates, the third is caused by centrifugal inertia and the fourth is caused by fluid flow inertia. Feeding pressure can be rewritten as:

$$p_0 = \frac{6\mu Q}{\pi h_p^3} \ln \frac{R_1}{R_2} - \frac{3\rho\omega^2}{20} (R_1^2 - R_2^2) - \frac{27\rho Q^2}{140\pi^2 h_p^2} \left(\frac{1}{R_2^2} - \frac{1}{R_1^2} \right) [Pa] \quad \{2.68\}$$

There is a strong phenomenon of suction in the inner radius region as rotating speed is very high, which is running as a centrifugal pump (Shihua & Zengxiong, 2010).

In some papers, such as (Shihua & Zengxiong, 2010), import and export pressure of friction pair are considered the same. However, in the real clutch they are not the same. Pressure difference between import and export of friction pair can be considered as follows (Heyan *et al.*, 2013):

$$p(R_1) - p(R_2) = \Delta p [MPa] \quad \{2.69\}$$

Where $p(R_1)$ and $p(R_2)$ represent the pressure on the outer and inner radius, respectively and are expressed in Pascal [Pa].

According to (Heyan *et al.*, 2013), the following equation can be used to calculate the ideal flow rate for full oil film in the clearance. This equation was based on the Navier-Stokes equations considering the centrifugal effect.

$$Q = \frac{6\mu \cdot \ln \frac{R_1}{R_2}}{\frac{27\rho}{70\pi^2 \cdot h_p^2} (R_2^{-2} - R_1^{-2})} + \sqrt{\left(\frac{6\mu}{\pi \cdot h_p^3} \ln \frac{R_1}{R_2} \right)^2 - \frac{81\rho^2 \omega^2 (R_2^{-2} - R_1^{-2}) (R_1^2 - R_2^2) - 540\rho (R_2^{-2} - R_1^{-2}) \Delta p}{700\pi^2 h_p^2}} + \frac{\frac{27\rho}{70\pi^2 h_p^2} (R_2^{-2} - R_1^{-2})}{\left[m^3 / s \right]} \quad \{2.70\}$$

Where Δp represents the pressure difference between import and export friction pair and is expressed in Pascal [Pa].

Ideal flow rate for full oil film means the needing feed flow rate to maintain full oil film in the clearance between the plates. It is conclusive that there is a relation between the needing flow rate and rotating speed: the larger ω is, the larger Q is necessary to maintain single-phase flow in the clearance (Shihua & Zengxiong, 2010) and (Heyan *et al.*, 2013). In order to visualize the relation between needing feed flow rate and rotating speed, a simulation was done. The goal of this simulation was to obtain a trend chart of needing feed flow rate. The trend chart can be found in Figure 43 and the simulation conditions can be found in Table 8.

Table 8 – Conditions of clutch simulation (Heyan *et al.*, 2013)

R_1 [mm]	R_2 [mm]	h_p [mm]	Q_i [L/min]	μ [Pa.s]	ρ [kg/m ³]
86	125	0.6	1/1.5	0.086	875

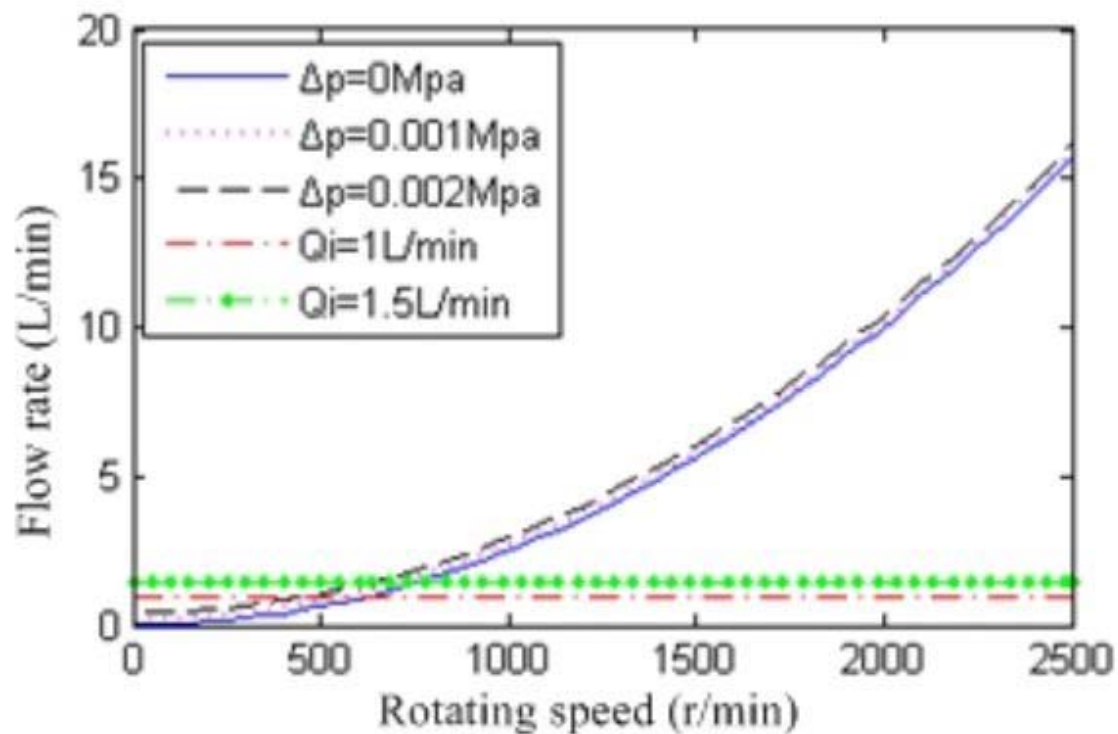


Figure 43 – Needing feed flow rate curve for full oil film (Heyan *et al.*, 2013)

By observing the trend chart (Figure 43), the needing feed flow rate for full oil film increases with the rise of the rotating speed of the friction plate. But the actual feed

flow is constant regardless of the rising of rotating speed. This means that the oil film will shrink when the needing feed flow rate exceed the actual feed flow rate in high speed. This phenomenon can be seen in Figure 44 (Heyan *et al.*, 2013).

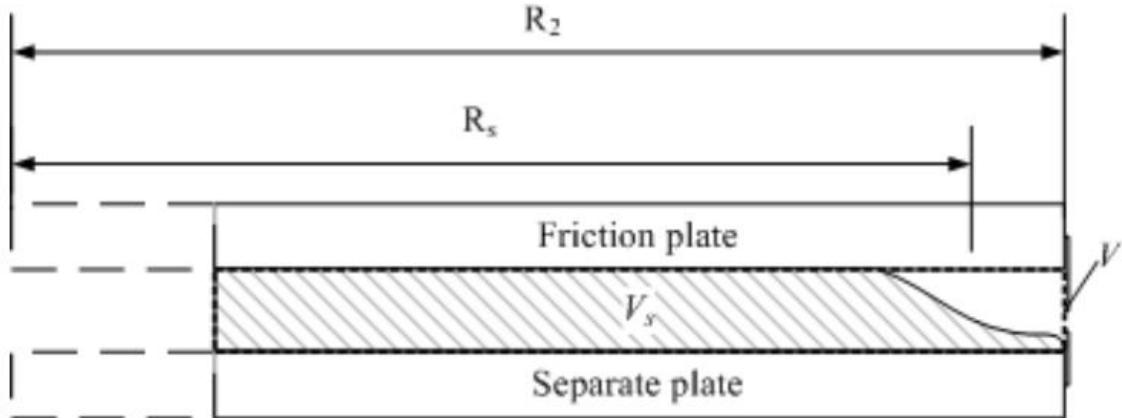


Figure 44 – Schematic diagram of partial oil filming in the clearance (Heyan *et al.*, 2013)

The principle for oil film shrinking can be explained as that the clutch clearance cannot be full of oil when actual feed flow rate is lower than the needing feed flow rate (Heyan *et al.*, 2013). When,

$$Q_i \geq Q, R_s = R_2 \quad \{2.71\}$$

When,

$$Q_i < Q \quad \{2.72\}$$

Where R_s represents the equivalent radius of oil film and is expressed in meters [m] and Q_i represents the input flow rate and is expressed in cubic meters per second [m³/s].

Based on the relationship of oil flow rate and volume between import and export, equivalent radius of oil film can be defined as:

$$R_s = \sqrt{\frac{Q_i}{Q} R_2^2 + R_1^2 \left(1 - \frac{Q_i}{Q}\right)} [m] \quad \{2.73\}$$

A simulation was done by (Heyan *et al.*, 2013) in order to find out what is the relation between equivalent radius and rotating speed for different Δp and Q_i . Simulation conditions are presented on Table 8. The resulting plot can be shown in Figure 45.

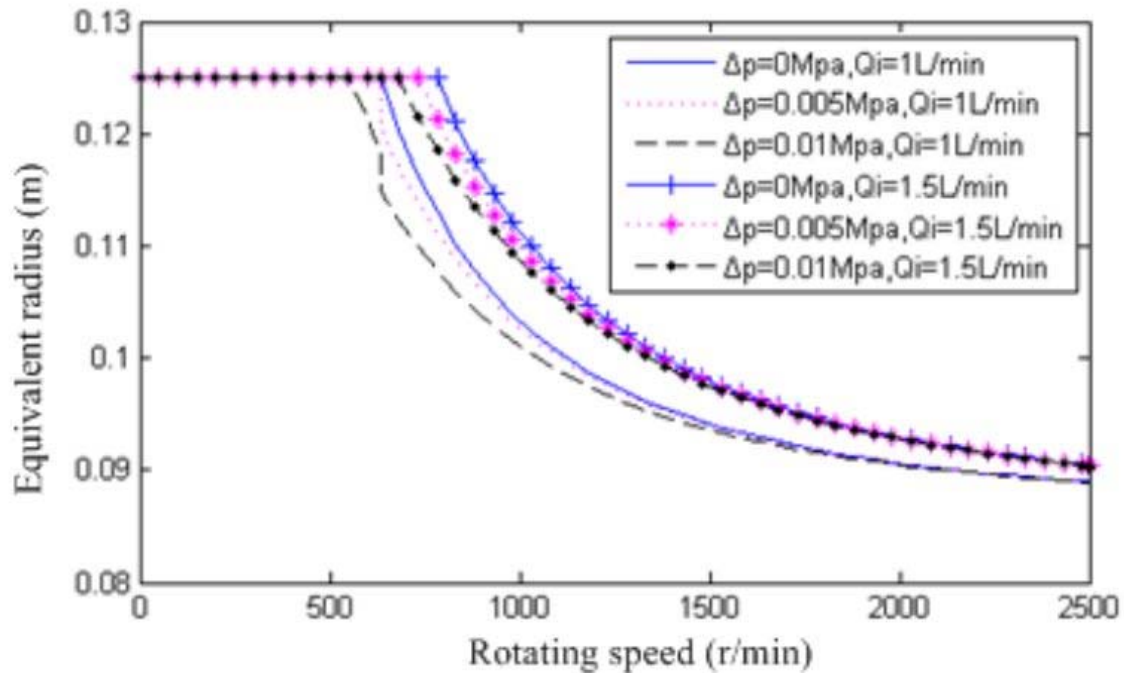


Figure 45 – Equivalent radius curve with rotating speed (Heyan *et al.*, 2013)

By studying the graph above it is possible to conclude that at the beginning none of the factors (Δp and Q_i) have an effect on the value of R_s . There is an explanation for this. When the rotating speed is low, there is a full oil film and the equivalent radius equals the outer radius (eq. {2.71}). When rotating speed becomes higher, the equivalent radius starts to drop (Heyan *et al.*, 2013).

According to Newton internal friction theorem, drag torque of each of the rotating plate can be expressed as:

$$T_{D_i} = \frac{\pi \mu \omega}{2h_p} (R_s^4 - R_l^4) [N.m] \quad \{2.74\}$$

In Figure 46 a trend chart of drag torque is plotted based on the simulation conditions of Table 8.

It is expected that the resulting graphs of the actual clutch in study are similar to the shape of this: a linear rise of the drag torque with the rise of the rotating speed for low speed; when the speed increased to a certain point, the drag torque began to decrease. In high rotating speed, the drag torque decreased with the rise of the pressure difference and increased with the rise of oil flow rate.

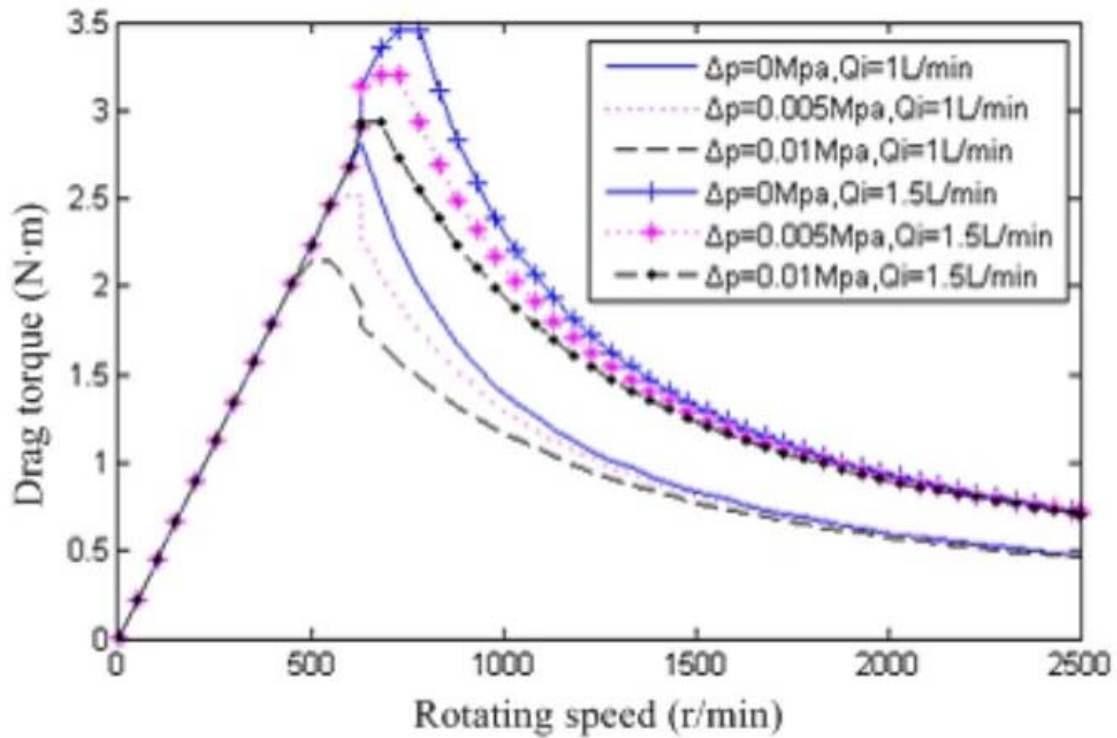


Figure 46 – Drag torque of single friction pair (Heyan *et al.*, 2013)

The resulting value of the drag torque for all the clutch plates can be obtained as follows,

$$T_D = \frac{N_s \pi \mu \omega}{2h_p} (R_s^4 - R_l^4) [N.m] \quad \{2.75\}$$

2.2.5 Planetary Gear Set ratio

PGS are complex mechanisms which great characteristics. Geared transmissions can be divided into:

- Gear sets with fixed axles;
- Epicyclic gears.

Gear sets with fixed axles refer to gearing where the positions of the axes of all the gear wheels are fixed relative to the gearbox housing. In the case of epicyclic gears or PGS, a revolving bar, which is known as the carrier or spider, carries the axes of the planetary gears. The differences between both types is explicit in Figure 47. Planetary transmissions have at least three planetary gears on a carrier or spider as shown in Figure 47. The purpose is to ensure a uniform and lower stress. The transmission ratio does not depend on the number of planetary gears or the number of teeth of each one.

Planetary gears main purpose is to reverse the direction of rotation (Naunheimer *et al.*, 2010).

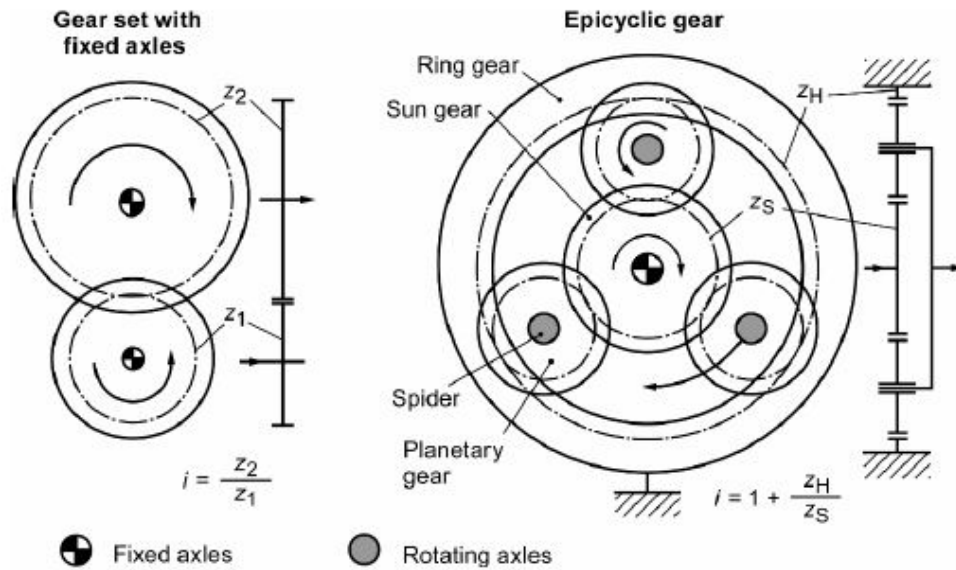
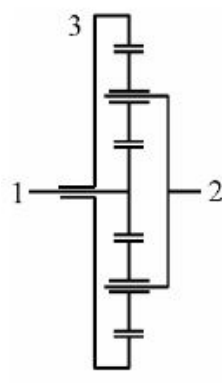


Figure 47 – Gear set with fixed axes and epicyclic gear (Naunheimer *et al.*, 2010)

When the carrier is fixed to the housing, the PGS becomes a gear set with fixed axes. With only one PGS it is possible to obtain nine different combinations. This can be achieved by fixing one by one the ring gear, sun gear and carrier acting as a frame. The remaining gear are the input and output (Naunheimer *et al.*, 2010).

The ratios of each combination or state of motion are defined by the number of teeth of the gears. The frame refers to the gear that is fixed to the housing. The different ratios can be calculated based on Table 9.

Table 9 – States of motion and ratios of a simple PGS (Naunheimer *et al.*, 2010)



State of motion	Type of gearbox	Input	Output	Frame	Planetary step ratio
a)	Gear set with fixed axes	1	3	2	$i_P = \frac{n_1}{n_3} = i_S = -\frac{z_3}{z_1}$
b)		3	1		$i_P = \frac{n_3}{n_1} = \frac{1}{i_S} = -\frac{z_1}{z_3}$
c)	Planetary transmission single drive	1	2	3	$i_P = \frac{n_1}{n_3} = 1 - i_S = 1 + \frac{z_3}{z_1}$
d)		2	1		$i_P = \frac{n_2}{n_1} = \frac{1}{1 - i_S} = \frac{1}{1 + \frac{z_3}{z_1}}$
e)		2	3	1	$i_P = \frac{n_2}{n_3} = \frac{1}{1 - \frac{1}{i_S}} = \frac{1}{1 + \frac{z_1}{z_3}}$
f)		3	2		$i_P = \frac{n_3}{n_2} = 1 - \frac{1}{i_S} = 1 + \frac{z_1}{z_3}$

1 ... Sun gear
2 ... Spider
3 ... Ring gear

2.2.6 PGS Kinematics

PGS is a complex gearing system and to calculate the speeds (angular or linear) of each gear of this system requires to deeply understand the working principle of a PGS and particularly what happens to the PGS in the different modes of the DNR.

The generalized equation of Willis can be derived to the following equation (Müller, 1982):

$$\omega_b - i_{bc}n_c - (1 - i_{bc})n_a = 0 \quad \{2.76\}$$

Where n_c and n_a represent the rotating speed of the gear c and a , respectively and are expressed in radians per second [rad/s]; i_{bc} represents the ratio between gear b and c and is a nondimensionalized variable. This generalized equation can be adapted as follows (Liu, 2017):

$$\omega_s + i \cdot \omega_R - (1 + i)\omega_C = 0 \quad \{2.77\}$$

In which ω_R and ω_s represent the rotating speed of the ring and sun gear, respectively and are expressed in rad/s. Where the value of i can be obtained by two different ways (Liu, 2017):

$$\left\{ \begin{array}{l} i = \frac{Z_R}{Z_S} \\ i = \frac{r_R}{r_S} \end{array} \right. \quad \{2.78\}$$

Where Z_R and Z_S represent the number of teeth of the ring gear and sun gear, respectively; r_R and r_S represent the radius of the ring and sun gear, respectively and are expressed in meters [m].

2.2.7 Cams

A cam is one of the several types of clamping elements. Figure 48 shows a cam being actuated by a force F . The force is applied on a handle of length L measured from point

C (centre of the pivot). This force is exerting a clamping force P and a frictional force $\zeta_1 P$ at the point of contact D (on the interface between the cam and the clamped component). These forces create a reaction force R with a frictional force $\zeta_2 R$ on the pivot. The radius vector r and the position angle θ (measured from the low point A) define any point on the contour of the cam. The variable e is the initial eccentricity and together with the height H define the location of D relative to C. Height H depends on the cam geometry and θ (Henriksen, 1973).

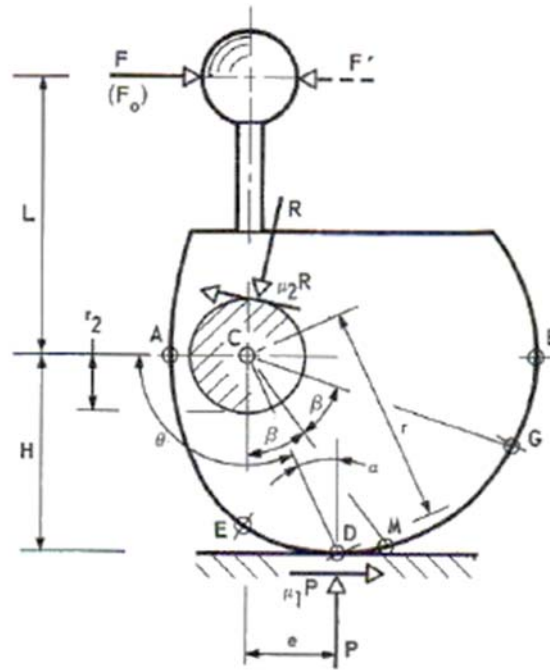


Figure 48 – The mechanics of a cam of arbitrary contour
(Henriksen, 1973)

Table 10 – Coefficients of friction μ for wedges, cams, pivots and bearings made of hardened steel (Henriksen, 1973)

Wedge or Cam Acting on	Coefficient of Friction μ_1	
	Clamping	Release
Hardened steel	0.19 to 0.20	0.19 to 0.20
Machine steel	0.17 to 0.19	0.20
Cast iron	0.15 to 0.17	0.17 to 0.19
Aluminum alloy	0.17 to 0.18	0.18 to 0.20
Laminated plastic	0.12 to 0.16	0.15 to 0.18
Pivots and bearings	Coefficient of Friction μ_2	
	Good Condition	Neglected
	0.03 to 0.06	0.10 to 0.15

The surface and lubrication conditions at point D and on the pivot, are different that is why it was used two different coefficients of friction. It was considered $\zeta_1 > \zeta_2$. The values of the coefficients of friction can be consulted on Table 10 as a reference (Henriksen, 1973). These values depend on the supplier and on other factors, so it is desirable to having in account the coefficient of frictions from the datasheets of the components.

Friction is always present on this kind of applications. In fact, the friction makes possible the self-locking of the system. The equilibrium equation for the clamping phase is (Henriksen, 1973):

$$P \cdot e + \zeta_1 \cdot P \cdot H + \zeta_2 \cdot R \cdot r_2 = F \cdot L \quad \{2.79\}$$

Where P represents the load at the cam and is expressed in Newton [N]; e represents the horizontal distance between point D and point C and is expressed in meters [m]; ζ_1 represents the friction coefficient at point D and is a nondimensionalized variable; H represents the vertical distance between point C and point D and is expressed in meters [m]; ζ_2 represents the frictional coefficient in the pivot and is a nondimensionalized variable; R represents the reaction force and is expressed in Newton [N]; r_2 represents the pivot's radius and is expressed in meters [m]; F corresponds to the load applied in the handle and is expressed in Newton [N]; and L represents the distance between point C and the handle and is expressed in meters [m].

Cams are designed in such a way to ensure that P is greater than F (10 to 20 times). As good approximation the value of R can be considered as follows (Henriksen, 1973):

$$R \cong 1,03P[N] \quad \{2.80\}$$

Then (Henriksen, 1973):

$$\begin{aligned} P(e + \zeta_1 H + 1.03\zeta_2 r_2) &= FL \Leftrightarrow \\ \Leftrightarrow \frac{P}{F} &= \frac{L}{e + \zeta_1 H + 1.03\zeta_2 r_2} \end{aligned} \quad \{2.81\}$$

Assuming that the cam at this point is clamped in a self-locking position with the force P it is necessary to find out what is the necessary force F' applied on the handle to release the cam. This values can be calculated based on the following equation (Henriksen, 1973):

$$\begin{aligned}
 Pe - \zeta_1 PH - \zeta_2 Rr_2 &= -F' L \Leftrightarrow \\
 \Leftrightarrow \frac{F'}{P} &= \frac{-e + \zeta_1 H + 1.03\zeta_2 r_2}{L}
 \end{aligned} \tag{2.82}$$

If the returned values of F' is positive, it means that the cam was self-locked. The theoretical limit for self-locking is when $F' = 0$ and consequently $R = P$. The theoretical condition for self-locking is given by (Henriksen, 1973):

$$\begin{aligned}
 -e + \zeta_1 H + \zeta_2 r_2 &= 0 \Leftrightarrow \\
 \Leftrightarrow \frac{\zeta_1 H + \zeta_2 r_2}{e} &= 1
 \end{aligned} \tag{2.83}$$

The safety factor is obtained as follows (Henriksen, 1973):

$$\frac{\zeta_1 H + \zeta_2 r_2}{e} = SF \tag{2.84}$$

The efficiency of the cam, which is dictated by the friction losses, is given by (Henriksen, 1973):

$$\frac{F_0}{F} = \frac{e}{e + \zeta_1 H + 1.03\zeta_2 r_2} \tag{2.85}$$

2.2.7.1 *Spiral cam*

Cams used for fixture clamping can be of two types: Archimedes spirals or circular eccentrics. An example of an Archimedes spiral is illustrated in Figure 49. Figure 49 is an auxiliary to the following calculations relative to the Archimedes spiral cam.

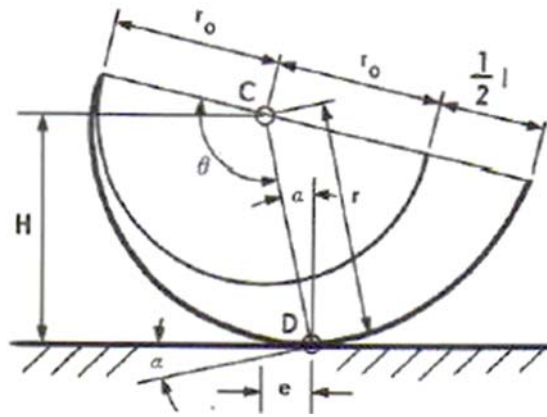


Figure 49 – The mechanics of an Archimedes spiral cam
(Henriksen, 1973)

The fundamental equation for an Archimedes spiral is given by (Henriksen, 1973):

$$r = \frac{l}{2\pi} \theta [m] \quad \{2.86\}$$

In which θ represents an angle and is expressed in radians [rad]. Where the angle θ is in radians and l is the increase in radius vector r for one revolution.

The rise is the increase in r along a given length of contour. The rise over the entire operating range is called the “throw”.

The slope of the tangent and the normal to the curve is determined by (Henriksen, 1973):

$$\tan \alpha = \frac{l}{2\pi r} \quad \{2.87\}$$

Where r represents the radius of the exterior cam spline and is expressed in m. The angle α is equivalent to a wedge angle. While l is constant, r and α vary with θ ; α increases with l and decreases with the increasing r . For fixture clamping cams the variation of the angle α over the useful length of the cam is small, usually around 1 degree.

At an arbitrary point D:

$$\begin{cases} e = r \sin \alpha [m] \\ H = r \cos \alpha [m] \end{cases} \quad \{2.88\}$$

$$\phi = \theta - \psi - 90 [\text{rad}] \quad \{2.90\}$$

Then (Henriksen, 1973):

$$H = R - E \cos(\theta - \psi) [m] \quad \{2.91\}$$

$$e = E \cos \phi = E \sin(\theta - \psi) [m] \quad \{2.92\}$$

The variable e is independent of R and depends only on the fixed eccentricity E and the angle. $\theta - \psi$ corresponds to the cam rotation angle (Henriksen, 1973). The maximum value of ψ is determined by (Henriksen, 1973):

$$\sin \psi = \frac{E}{R} \quad \{2.93\}$$

THESIS DEVELOPMENT

- 3.1 CVT transmission
- 3.2 External components
- 3.3 DNR working principle
- 3.4 Calculations data
- 3.5 Analytical Calculations
- 3.6 Results comparison between FWD and REV clutch
- 3.7 Ramp-down test
- 3.8 Comparison between tests and analytical calculations
- 3.9 Requirements description

3.10 Design validation

3.11 Development of the showcase model

3.12 Preliminary assembly feasibility study

3.13 Showcase FEA

3.14 Drawings

3.15 Poster

3 THESIS DEVELOPMENT

3.1 CVT transmission

The Drive-Neutral-Reverse subsystem in study is part of a CVT (Continuously Variable Transmission). A cross-section of the CVT is shown in Figure 51.

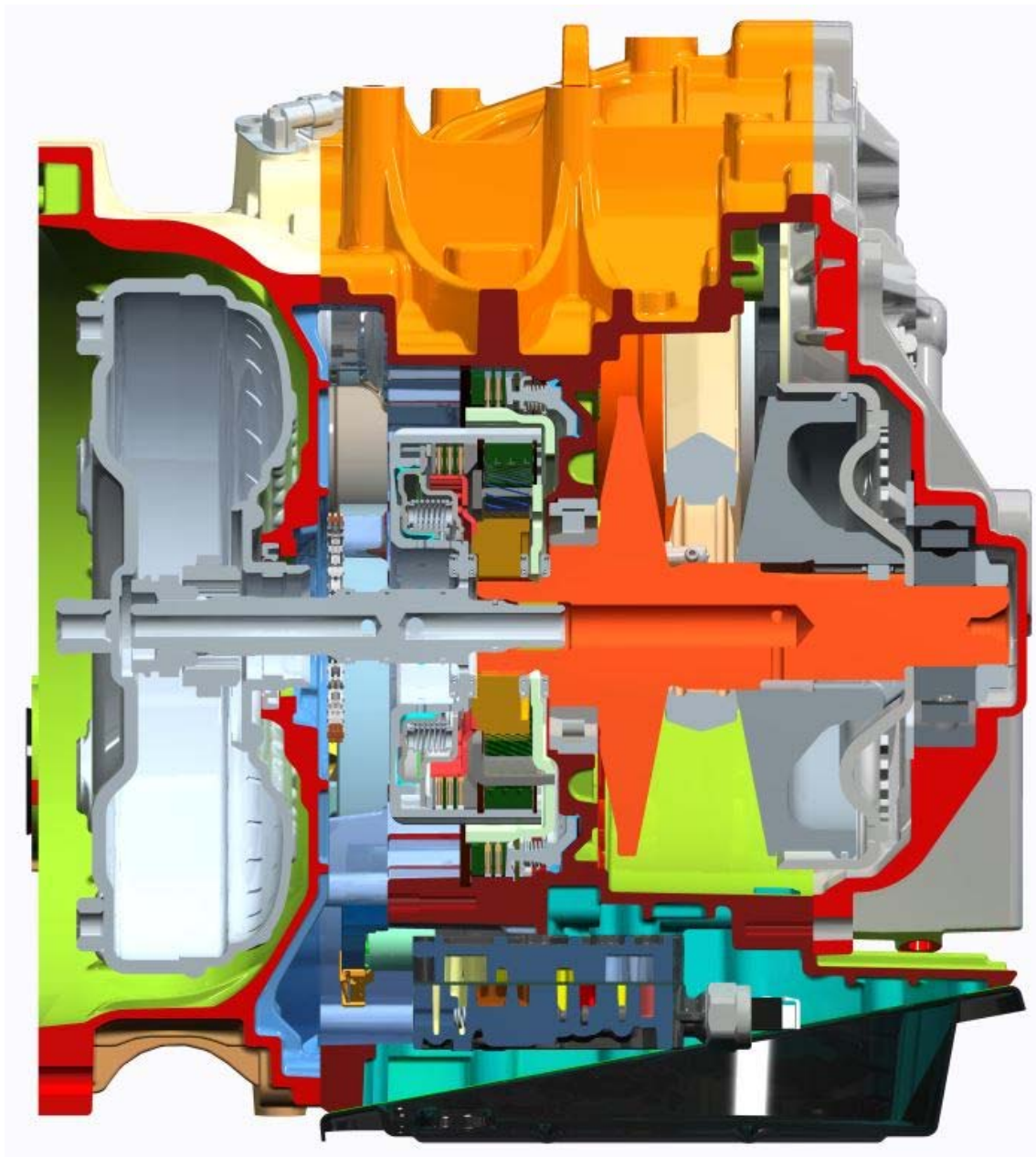


Figure 51 - Cross section of the CVT (left side) (courtesy of Punch Powertrain NV)

In Figure 52 it is underlined the DNR subsystem.

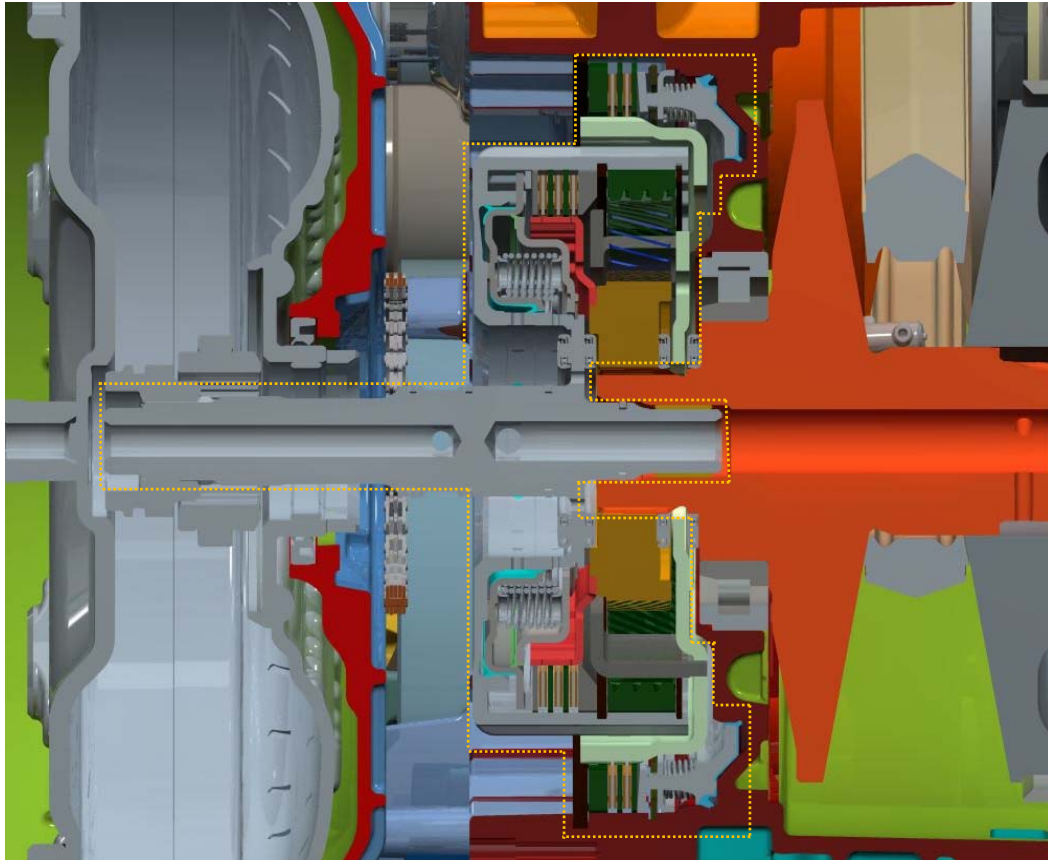


Figure 52 – DNR location inside the CVT (courtesy of Punch Powertrain NV)

3.2 External components

In Figure 53 it is underlined the different external components that will be referred during this thesis:

1. Torque converter (yellow line);
2. Stator plate (red line);
3. Pulley (blue line);
4. Primary shaft (black line);

The location of the sprocket will be stated forward due to the visual difficulty to see it. The transmission housing (TRH) corresponds to the part whose crossed surfaces are red.

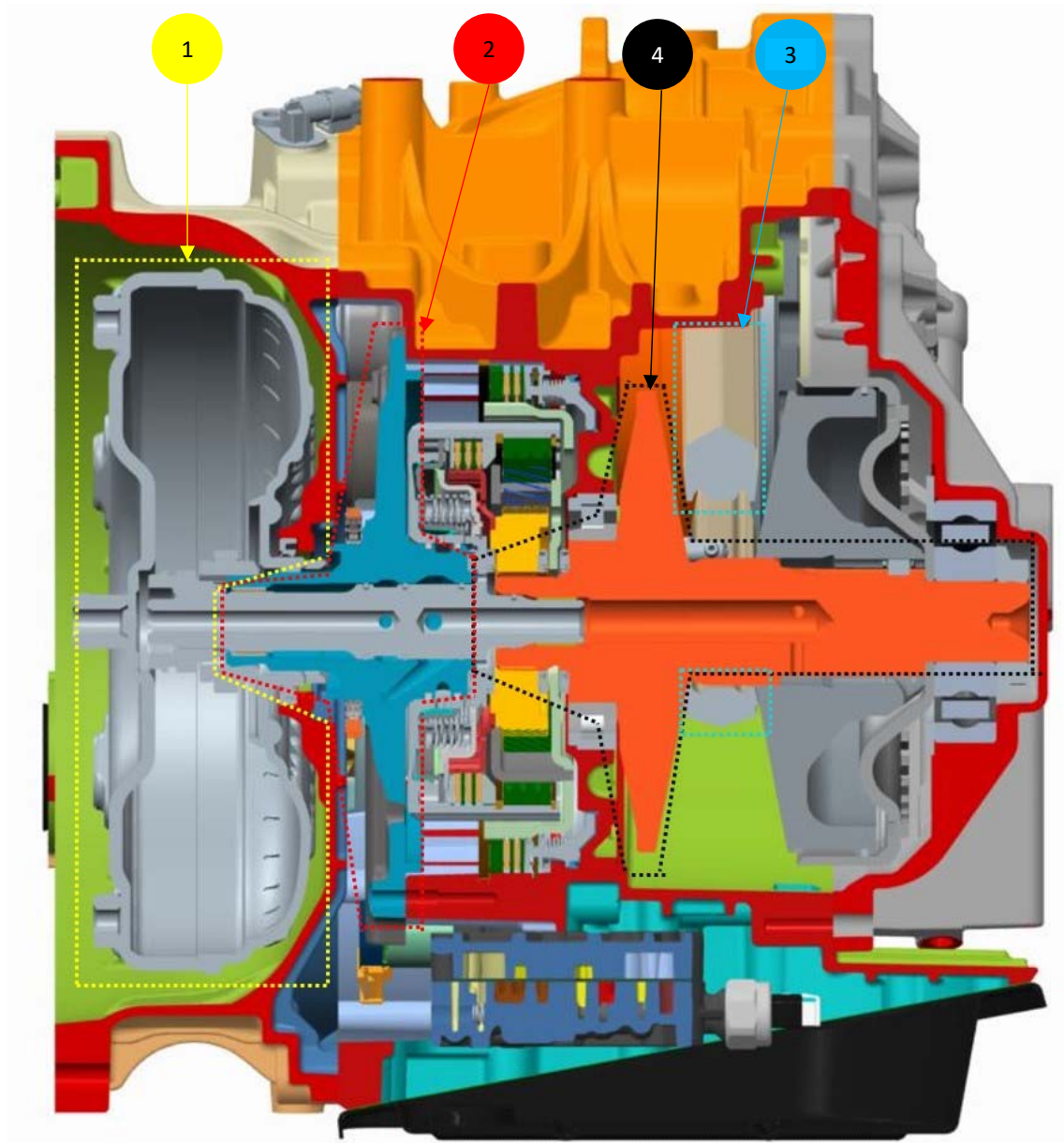


Figure 53 – Location of external components in the transmission (courtesy of Punch Powertrain)

3.3.1 DNR schematics

As previously mentioned, a DNR subsystem allows the shifting between Drive gear (D) and Reverse gear (R) passing through the Park and Neutral positions.

Figure 54 illustrates the DNR schematics and will be an auxiliary for the explanation on this chapter.

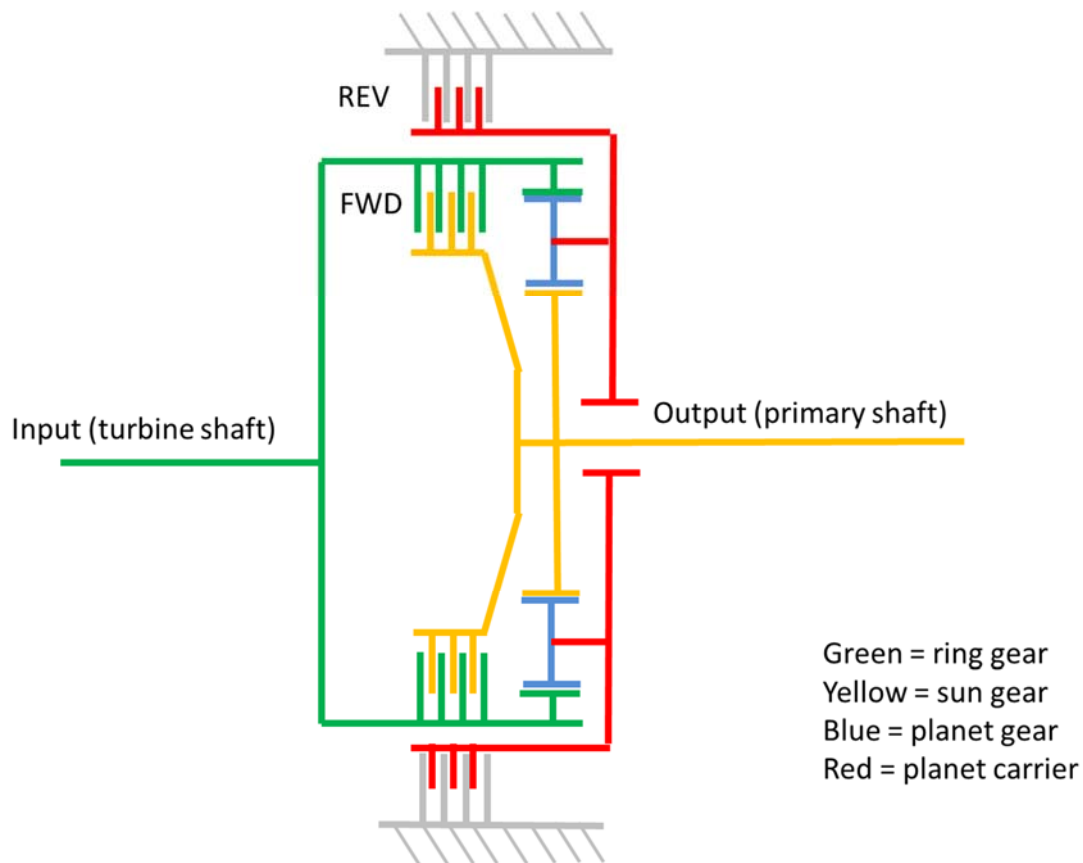


Figure 54 - DNR schematics (courtesy of Punch Powertrain NV)

DNR subsystem can be divided in two distinct groups. The first group is Forward (FWD) and the second one is Reverse (REV). As suggested by the names FWD group transmits the torque for Drive gear and the REV group transmits the torque for Reverse gear. Output shaft direction is different for FWD and REV. The change of direction of the rotation of the wheels of the vehicle is done inside the DNR by the planetary gear set (which will be presented forward). In addition to output direction, torque and transmission ratio are different too. Such factors make it ideal to do a separation between FWD and REV.

DNR subsystem is fixed inside the CVT transmission housing. The transmission housing is steady (grey part of Figure 54). Coupled to the primary shaft there is the sun gear (yellow part of Figure 54) which is part from the PGS. This means that the sun gear rotates solidary with the shaft.

Engine torque/rotational motion are transmitted to the DNR subsystem by the turbine shaft (green part of Figure 54) which is the input shaft of the DNR. The output of the DNR is the primary shaft (yellow shaft of Figure 54) which will transmit the torque to the belt.

In FWD, torque transmission from the input to the output shaft is done by a clutch set. The number of clutches varies according to the value of torque which is necessary to transmit, clutches' material friction coefficient and dimensions. In Figure 54 it is visible that the turbine shaft is connected to the PGS by the ring gear. When FWD clutch set is engaged, turbine shaft becomes solidary to the primary shaft. According to the working principle of a PGS when sun gear and ring gear are rotating, planet gear rotates too (Figure 55) although REV group does not rotate because the REV clutch set is open. The motion of the planet gears is a consequence of the rotation of both sun and ring gears.

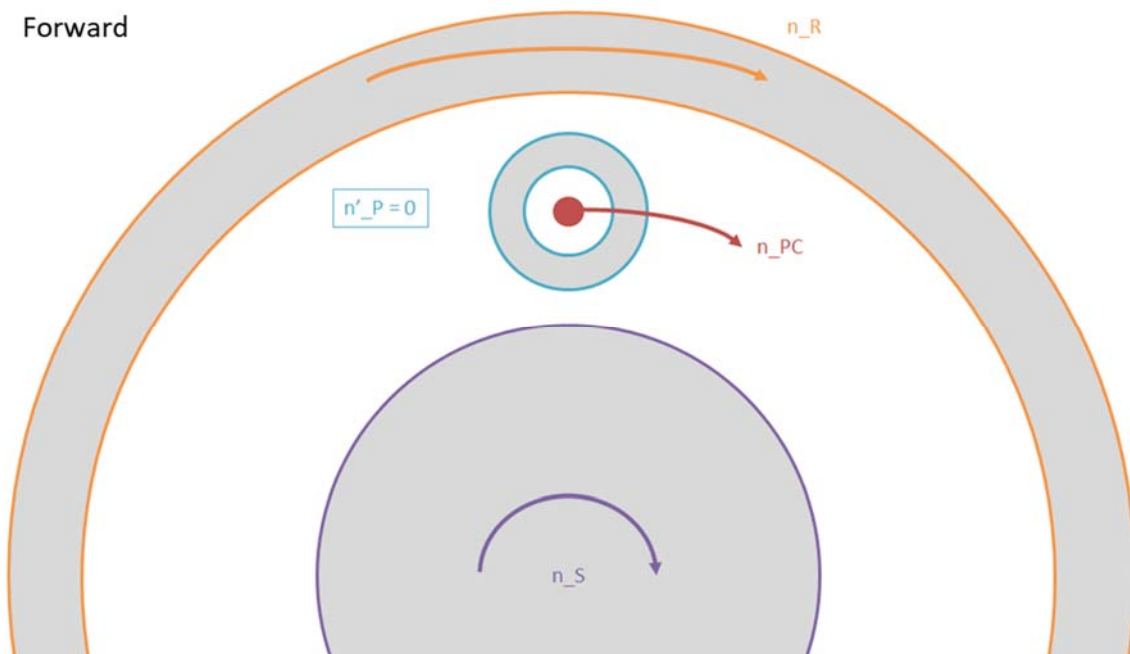


Figure 55 – PGS scheme for drive gear

According to Figure 54 reverse group is connected to the planet gears though the planet carrier (red part of Figure 54) and ring gear. When reverse clutch set is engaged, planet carrier will not rotate because clutch set is pressed against the housing which is steady. Having this in account when reverse clutch is activated, planet carrier is steady, turbine shaft is rotating so sun gear will rotate in a different direction (Figure 56).

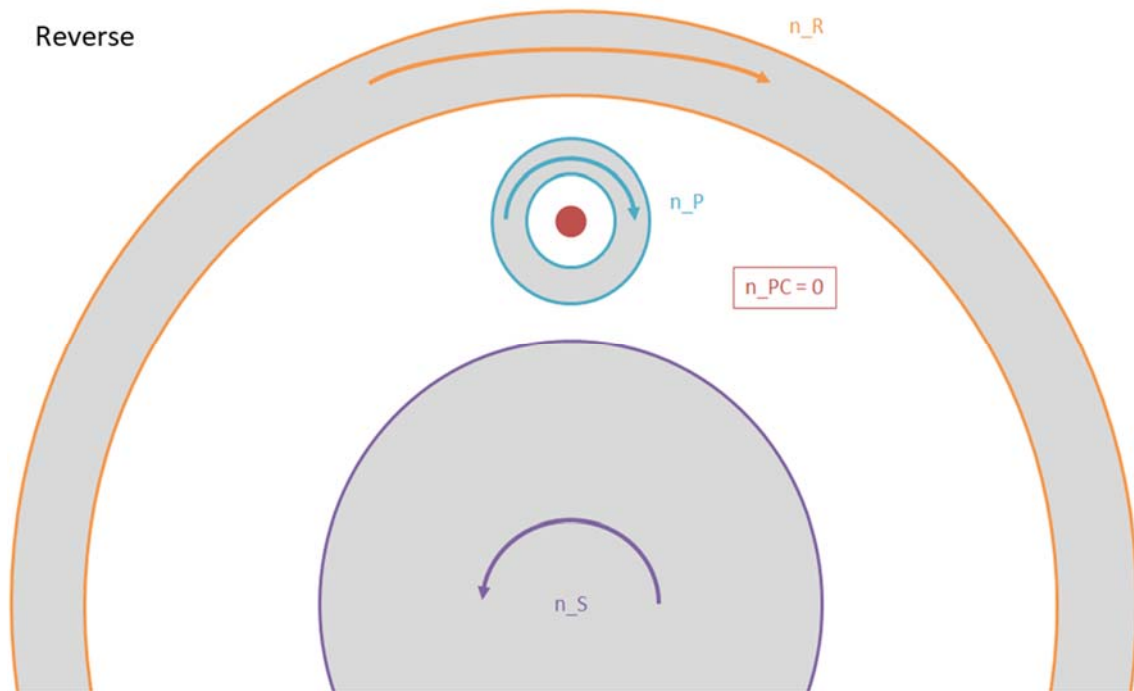


Figure 56 – PGS scheme for reverse gear

Finally, in Neutral, turbine shaft is rotating because the engine does not stop so the ring gear is rotating too; reverse clutch set is not engaged so planet carrier can rotate freely and as a consequence sun gear keeps steady (Figure 57). FWD clutch is disengaged too.

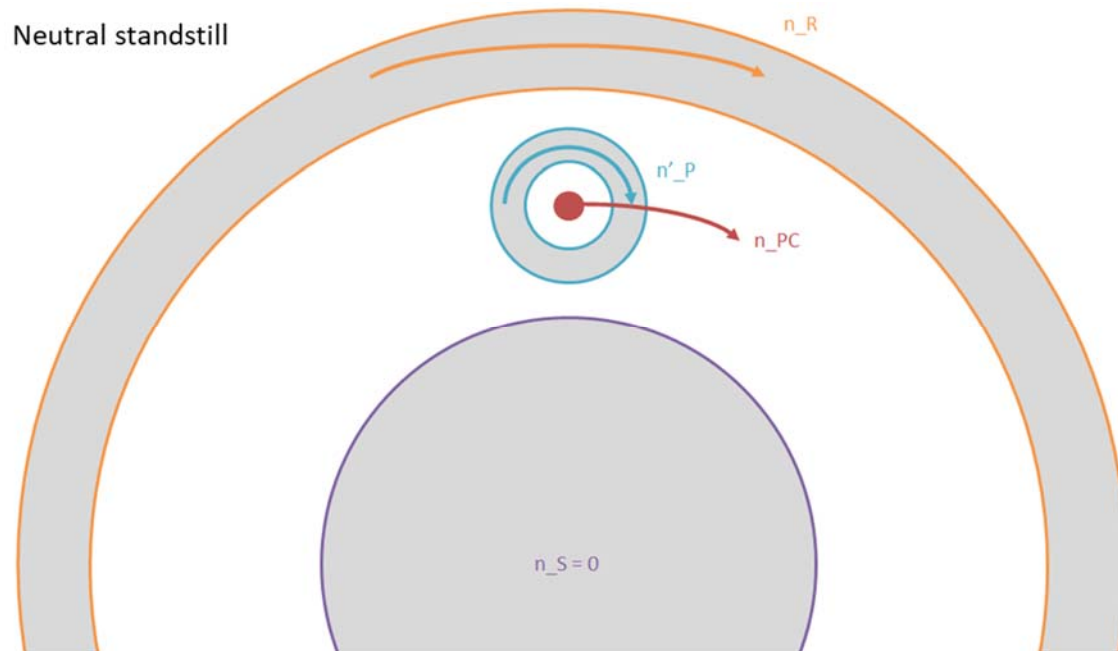


Figure 57 - PGS scheme for neutral gear

3.3.2 List of components

In this chapter, it will be presented a list of all the components which make part of the DNR subsystem as an auxiliary to the following chapters. It is important to have an idea of the components which will be approached during this dissertation in order to better understand it.

In the turbine shaft assy (assembly) are assembled all the parts regarding to the FWD set (Table 11) and Figure 58.

Table 11 – Turbine shaft assy BOM

Components	Number	Qty.
Clutch carrier welded assy	1	1
Piston FWD clutch	2	1
FWD spring pack	3	1
Compensation chamber FWD clutch	4	1
Snap ring compensation chamber FWD clutch	5	1
Wavy washer FWD clutch	6	1
FWD separator plate	7	3
Friction plate FWD clutch	8	3
Endplate FWD clutch	9	1
Snap ring FWD clutch	10	1
Ring gear	11	1
Snap ring FWD clutch narrow	12	1
Seal TC assy dia. 28.4	13	3
Seal TC assy dia. 20	14	1

PGS is the responsible for the transmission ratio in the different gears and all the components are expressed on Table 12 and illustrated on Figure 59.

Table 12 - PGS assy BOM (Bill of materials)

Component	Number	Qty.
Planet carrier welded assy	1	1
Planet thrust washer	2	6
Needle bearing	3	6
Planet gear	4	3
Planetary pin	5	3



Figure 58 – Exploded view of the turbine shaft assy (courtesy of Punch Powertrain NV)

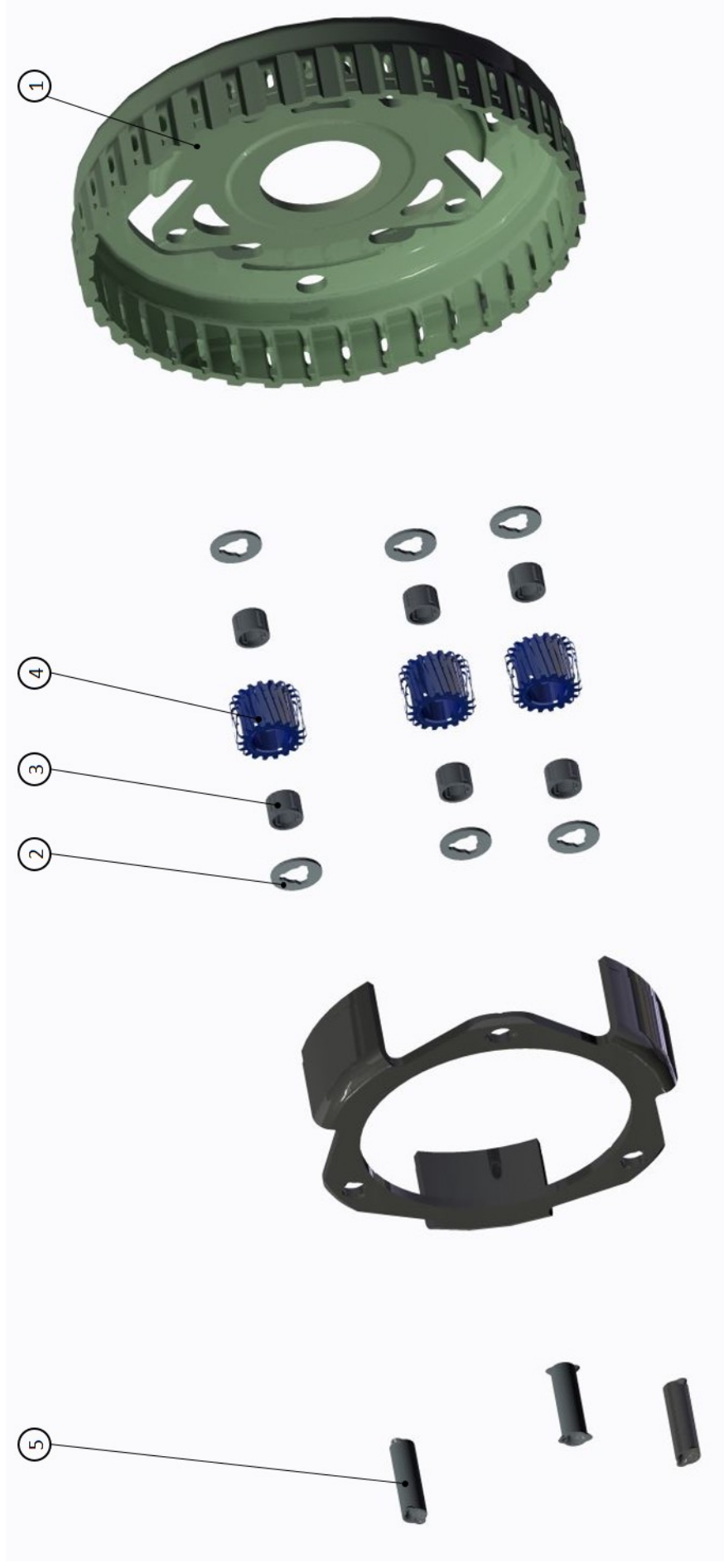


Figure 59 – Exploded view of the PGS (courtesy of Punch Powertrain NV)

The list of Table 13 involves all the component referent to the REV clutch set plus the other pre-assembled groups that will be assembled together in the transmission housing (Figure 60).

Table 13 - BOM to assemble in the transmission housing (complete DNR)

Component	Number	Qty.
REV piston	1	1
Spring pack REV clutch	2	1
Snap ring REV piston	3	1
Disc spring REV piston	4	1
REV separator plate	5	2
Friction plate REV clutch	6	2
Endplate REV clutch	7	1
Snap ring REV clutch	8	1
Needle roller thrust bearing	9	1
PGS assy	10	1
Sun gear assy	11	1
Turbine shaft assy	12	1

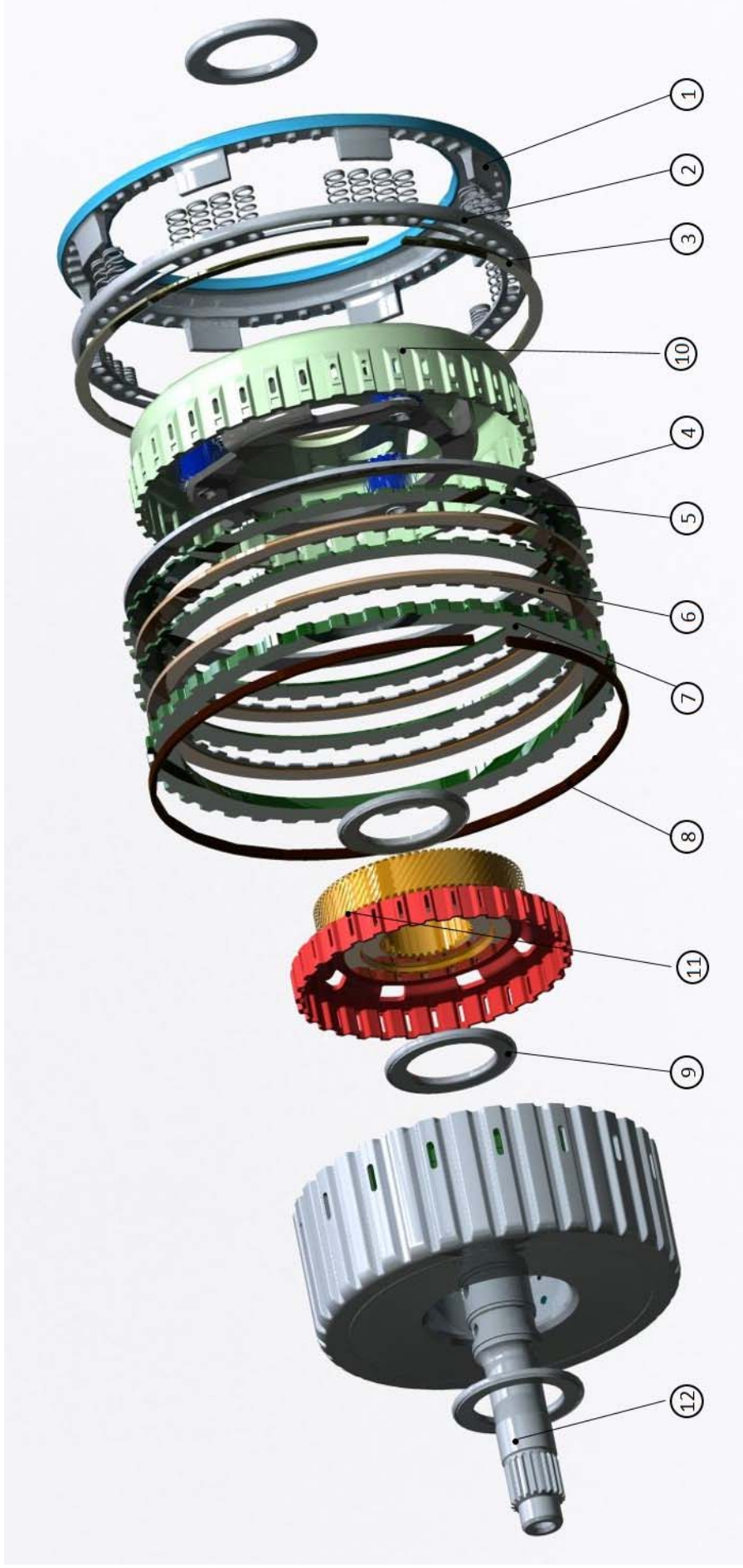


Figure 60 – BOM of the DNR components to be assembled in the CVT (courtesy of Punch Powertrain NV)

3.3.3 Rotational kinematics of components

The main goal of this chapter is to analyse which components can have rotational motion. This is important to study the oil flow of the DNR subsystem and to better understand the axial kinematics of the mechanism. This study is so important to the oil flow analysis due to centrifugal force.

According to Figure 61, turbine shaft and torque converter are connected which means that the turbine shaft rotates when the torque converter does too. In the same picture, it is also possible to see that the turbine shaft is not fixed to the transmission housing so there is nothing blocking its rotation.

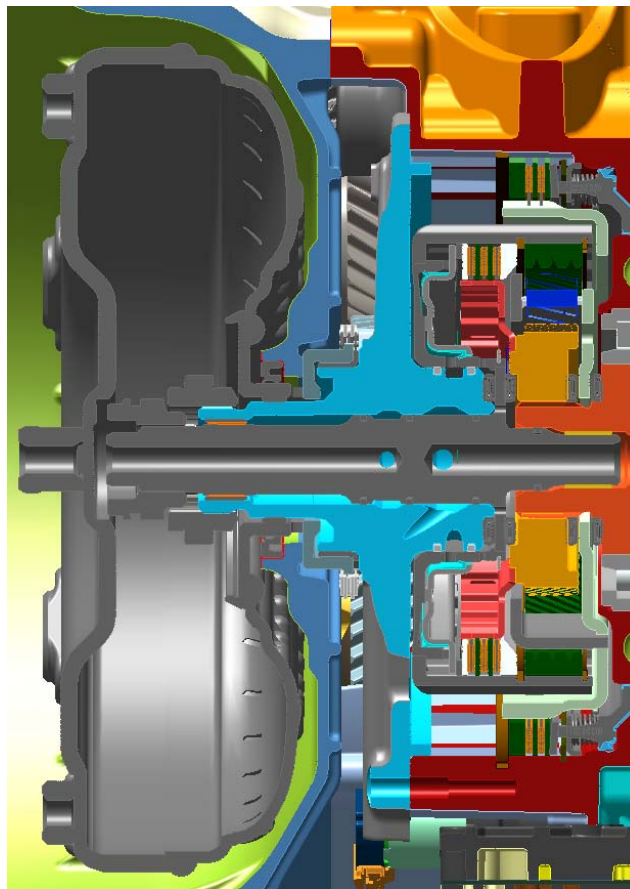


Figure 61 – Torque converter and DNR inside the CVT (courtesy of Punch Powertrain NV)

FWD piston is assembled inside the CC (clutch carrier) so it is susceptible to rotation. The mould injected elastomer material in the piston creates a friction contact between both components (Figure 62). However, this is not the main purpose of the rubber. The main purpose of the elastomer material is to work as a seal in order to prevent leaks and by

this way to create a chamber. This friction contact will transmit the rotational movement of the CC to the FWD piston. However, the main mean of transmission is the centripetal force.

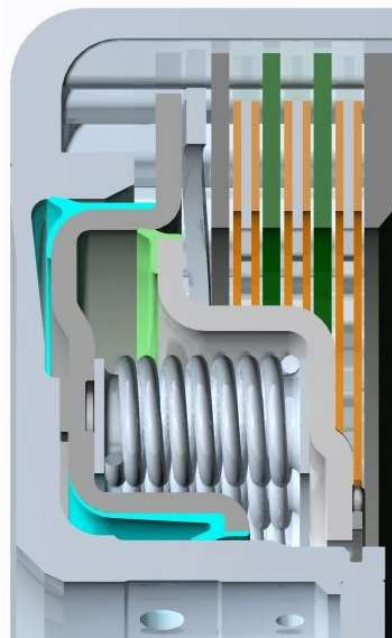


Figure 62 - Piston assembled in the CC (courtesy of Punch Powertrain NV)



Figure 63 - Splines matching between the FWD separator plate (white) and the CC (black) (courtesy of Punch Powertrain NV)

The wavy washer will rotate, too, because of the centripetal force. However, there may be some sliding between the wavy washer and the turbine shaft cage mostly during accelerations.

On the right side of the compensation chamber there is a snap ring. Between the forward piston and compensation chamber it is assembled the spring pack (Figure 62). Having this in account and that the spring pack is not assembled in free length of the springs (so it is loaded) it can be concluded that the compensation chamber and the spring pack are also rotating solidary with the forward piston.

Regarding to the forward clutch separator plate it rotates solidary with the clutch carrier due to its spline which matches with the clutch carrier's spline (Figure 63). The matching between forward separator plate and clutch carrier can be seen in Figure 63.

Based on Figure 64 it is obvious that the clutch plate doesn't have an outer spline but an inner one. This means that it doesn't rotate solidary with the clutch carrier.

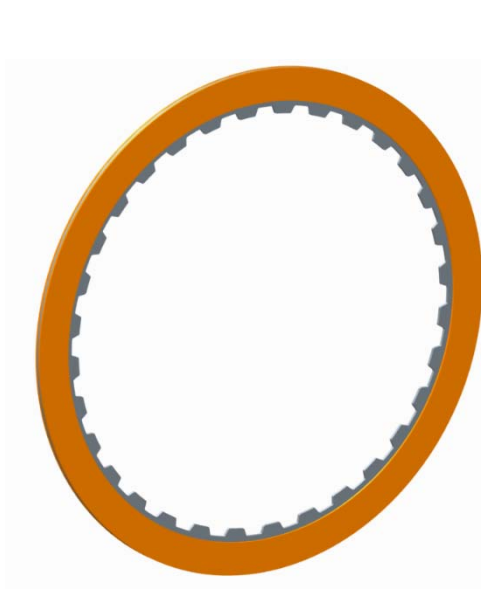


Figure 64 – FWD clutch plate (courtesy of Punch Powertrain NV)



Figure 65 – Matching between sun gear (red) and clutch plates (courtesy of Punch Powertrain NV)

Further ahead it will be explained the function of the clutch plates and why they don't have an outer spline. Inner spline of the clutch plates matches the outer spline of one of the gears of sun gear assembly. This means that the sun gear is solidary with the clutch plates. In Figure 65 it can be seen the matching between clutch plates and sun gear. In Figure 66 it can be seen sun gear. The gear which matches with the clutch plates is the red one. It is important to state that this assembly is welded which means that both gears rotate solidary.

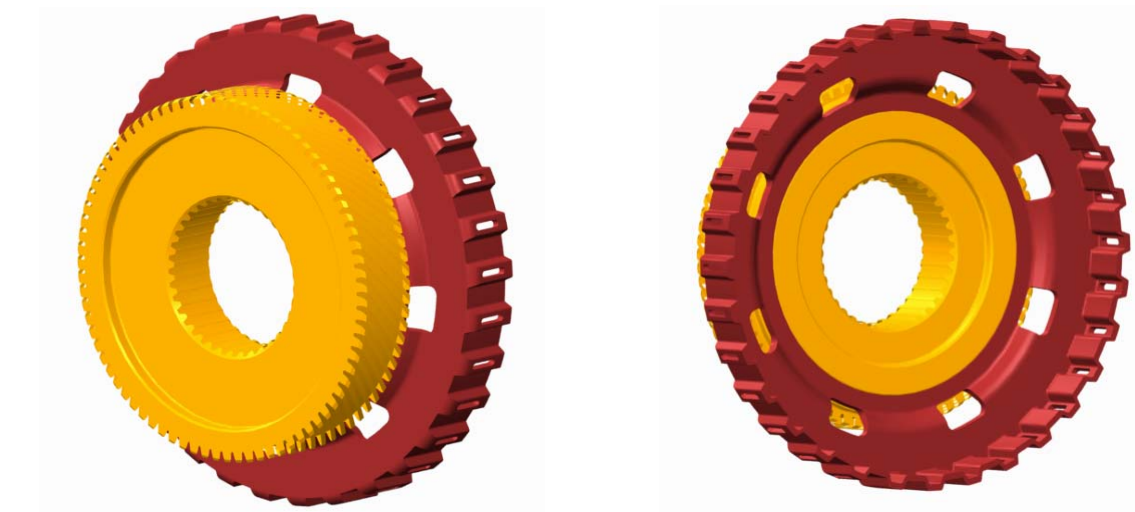


Figure 66 – Sun gear (courtesy of Punch Powertrain NV)

On the right side of the last clutch plate there is an endplate (endplate forward clutch). This component presents a spline in the outer diameter which matches with the clutch carrier as it is presented in Figure 67.

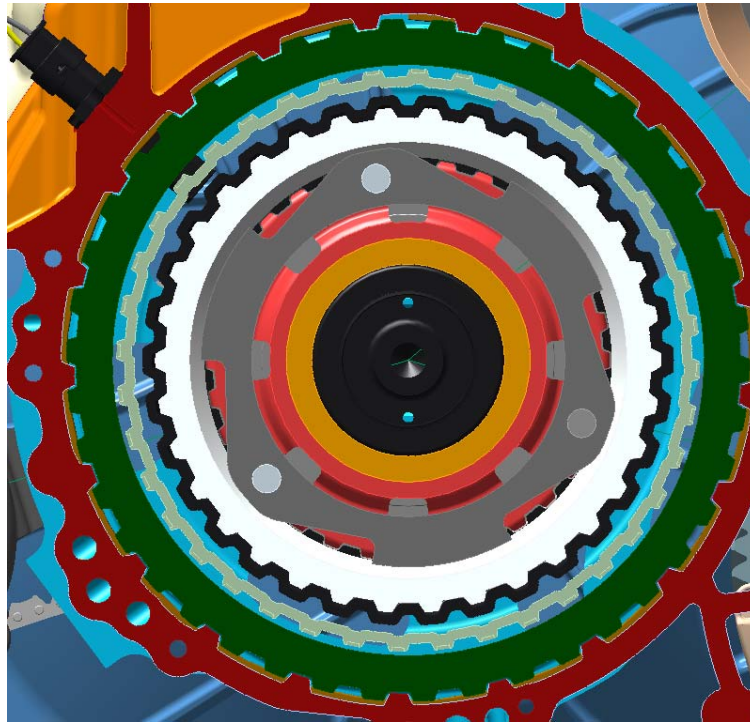


Figure 67 – Matching between forward endplate and clutch plates (courtesy of Punch Powertrain NV)

The other gear of the sun gear assembly (yellow part) will transmit the rotary motion to the planet gears of the PGS as it is presented in Figure 68. In the same picture, it is possible to see that the ring gear outer spline matches with the CC.

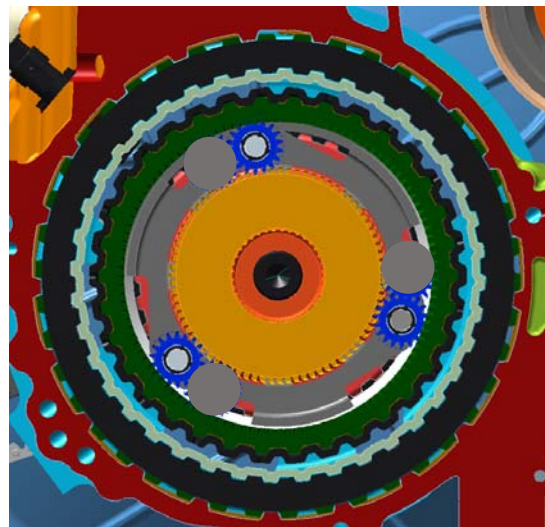


Figure 68 – PGS: cross section (side and front view) (courtesy of Punch Powertrain NV)

In Figure 58 it can be seen the outer and inner spline of the ring gear.

Regarding to the reverse piston as previously stated, it is placed in the transmission housing which is steady (Figure 69). This means that the reverse piston does not have rotational motion.

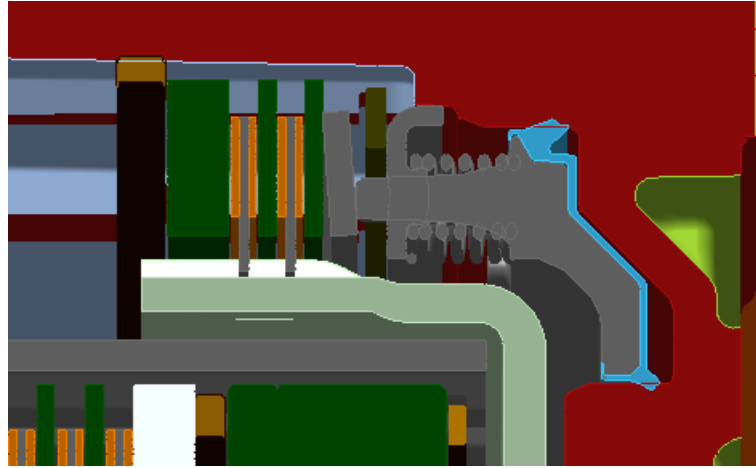


Figure 69 – REV piston location (courtesy of Punch Powertrain NV)

Reverse separator plate has a spline (Figure 60) which matches with the transmission housing, so it is conclusive that these components do not have rotational motion too (Figure 70).

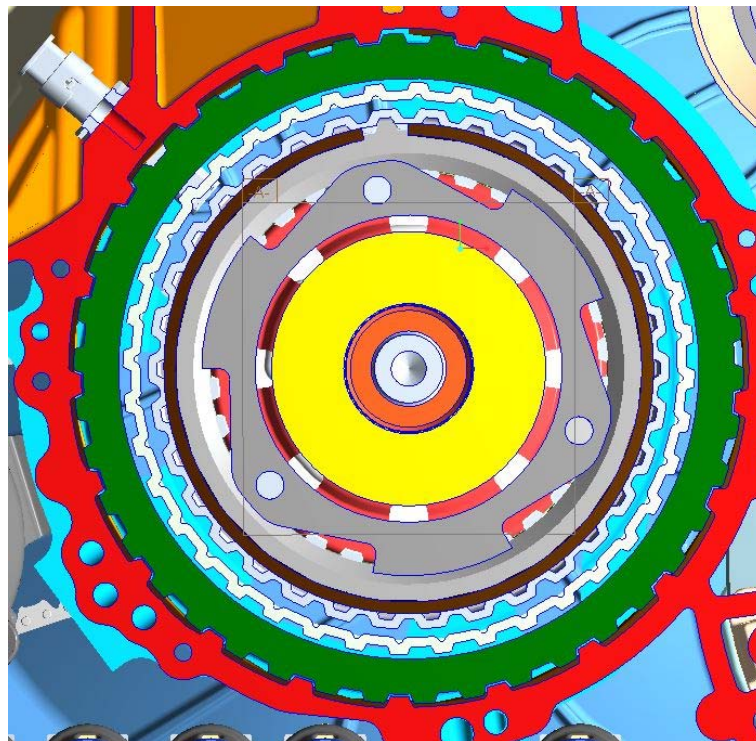


Figure 70 – Spline matching between transmission housing and REV separator plate (courtesy of Punch Powertrain NV)

Therefore, the following components keep steady too:

- Spring pack reverse clutch;
- Snap ring reverse piston;
- Disc spring reverse clutch.

Reverse clutch plates' spline matches with PGS carrier's spline (Figure 71) which means that both components are able to rotate.

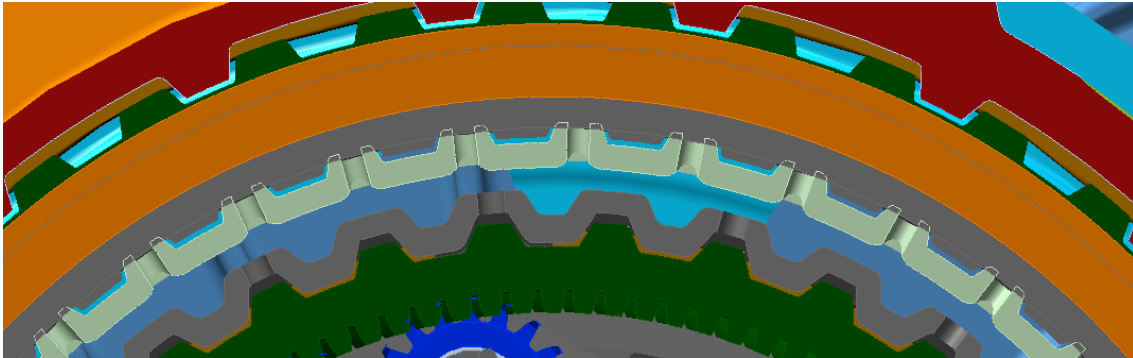


Figure 71 – Spline matching between REV clutch plates and PGS carrier (courtesy of Punch Powertrain NV)

Finally, endplate reverse clutch keeps steady too, because of the matching between its spline and the housing (Figure 72).



Figure 72 - Spline matching between endplate reverse clutch and transmission housing (courtesy of Punch Powertrain NV)

3.3.4 Hydraulic circuit analysis

3.3.4.1 Overview

According to the hydraulic diagram (Figure 73) it is possible to realize that there are three outputs which are directly related to the DNR subsystem. These outputs are the followings:

- FWD clutch;
- Reverse clutch;
- Lubrication DNR.

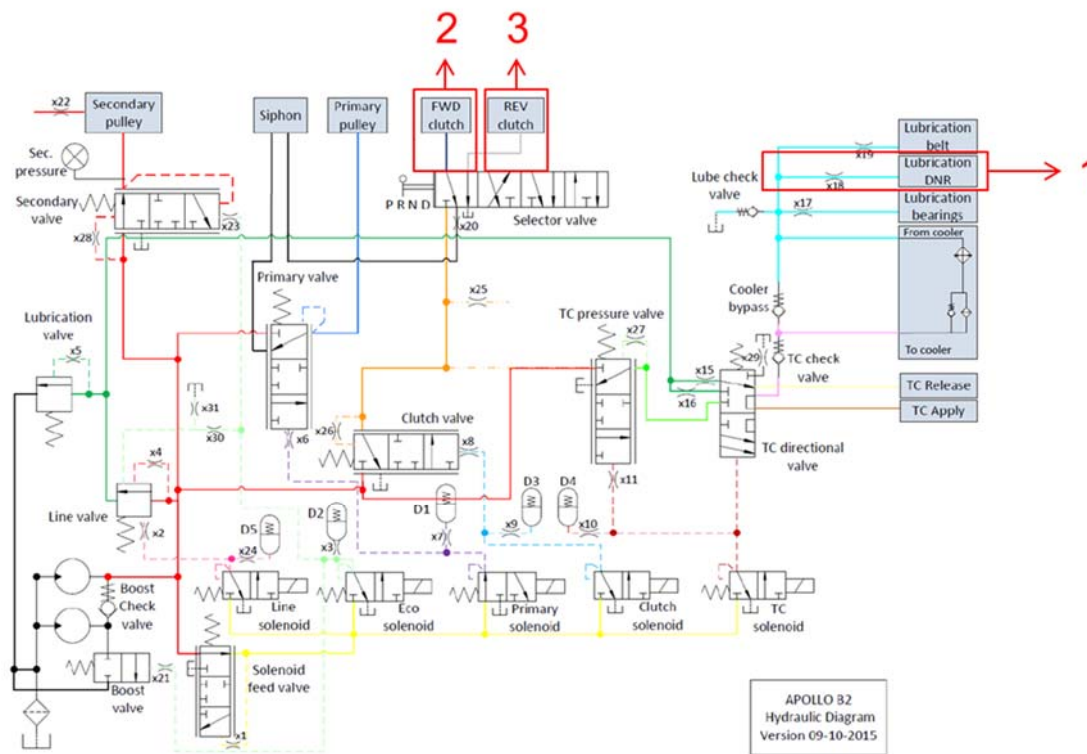


Figure 73 – Hydraulic circuit scheme (courtesy of Punch Powertrain NV)

According to the diagram it is possible to conclude that FWD and reverse clutches cannot be pressurized at the same time. This statement is guaranteed by the hydraulic valve whose position is defined by the position of the gear selector which is controlled by the vehicle driver. This means that FWD and REV clutches will be or not pressurized according to this valve. With this valve it is possible, for example to guarantee that when the driver moves the gear selector to Neutral mode all the clutches will be released or even when the driver moves the gear selector to Drive mode, REV clutch will be released and vice-versa.

This diagram is also conclusive regarding to the functions of the fluid: lubrication of the subsystem and activation of clutches.

Lubrication is an important factor on this subsystem. Due to the rotating motion of the gears a considerable amount of heat will be generated so it must be cooled down. In addition to cooling, lubrication itself is really important to eliminate the friction between surfaces.

Regarding to the properties of the fluid it must be ensured that this does not see its properties changed when running under different temperatures.

The starting point for the analysis of the circuit will be the stator plate as this is the part which distributes the oil before it reaches the DNR subsystem. Looking attentively to this component it is possible to see that there are two channels which are dedicated to the DNR subsystem.

In Figure 74 it is highlighted the channels which are dedicated to the DNR subsystem. It is also possible to see that one channel is for the lubrication circuit and the other belongs to the FWD clutch system.

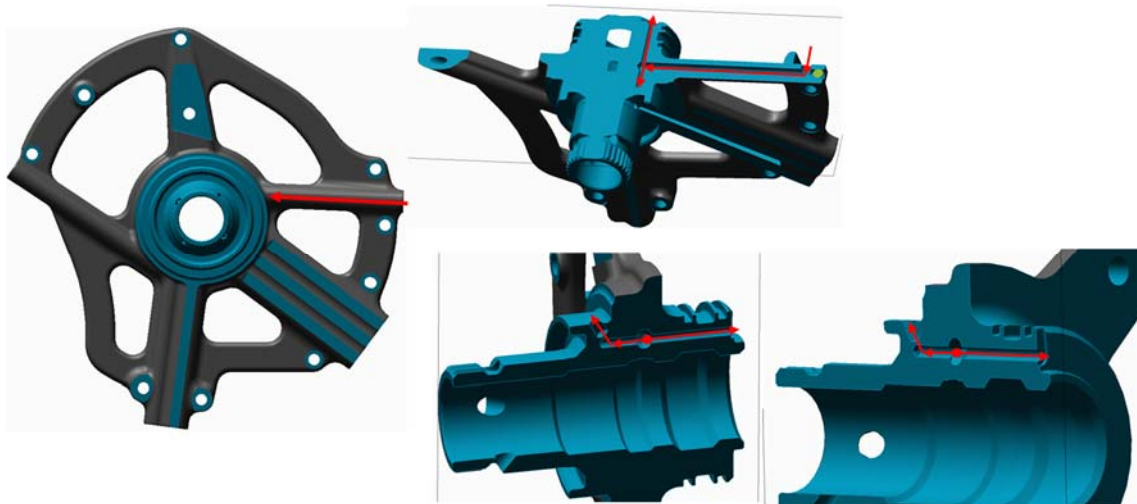


Figure 74 – Channels dedicated to DNR oil flow of the stator plate (courtesy of Punch Powertrain NV)

3.3.4.2 *Lubrication flow circuit*

In Figure 76 (cutting section A) it is presented the lubricant inlets on the stator plate. One is dedicated to the lubricant circuit and the other one is dedicated to the FWD clutch activation.

In Figure 75 it is presented the location of the cutting section of Figure 76.

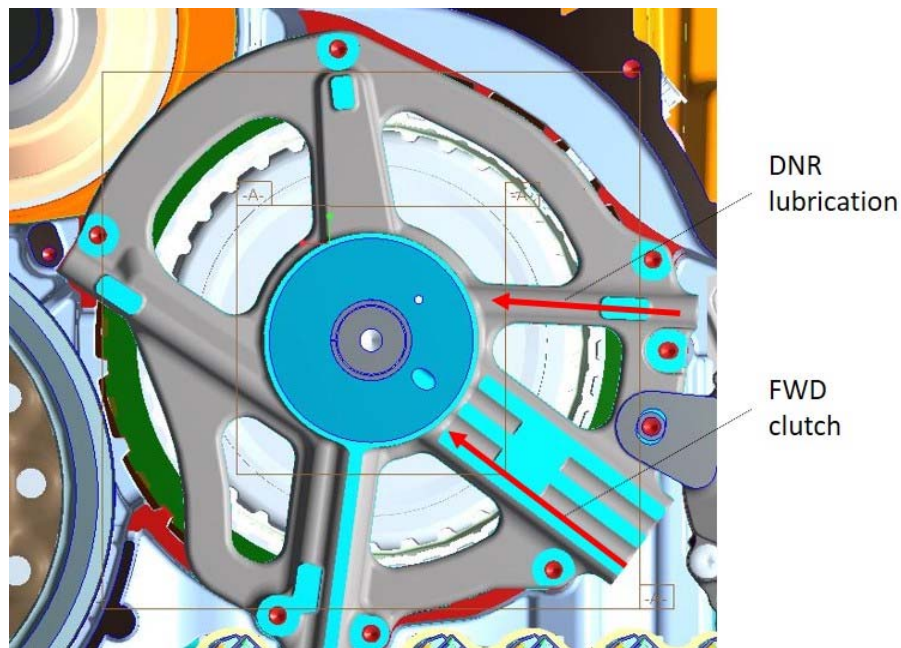


Figure 76 – Cross section A (courtesy of Punch Powertrain NV)

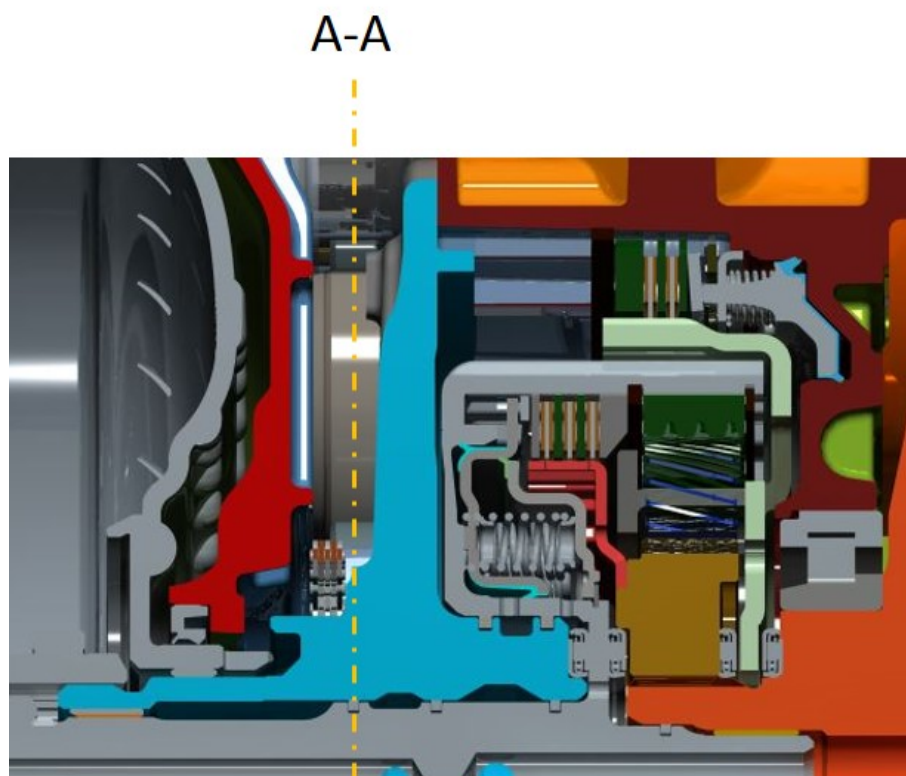


Figure 75 – Location of cross section A (courtesy of Punch Powertrain NV)

In Figure 78 it is possible to see the channels through where the oil will flow.

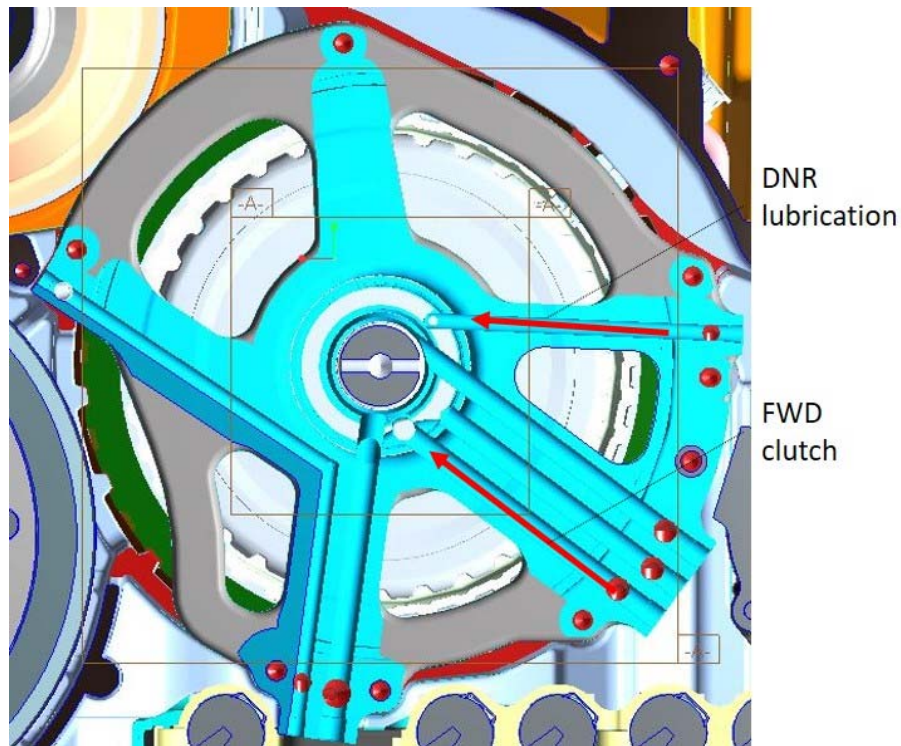


Figure 78 – Cross section B (courtesy of Punch Powertrain NV)

Figure 77 shows the location of the cutting section B.

B-B

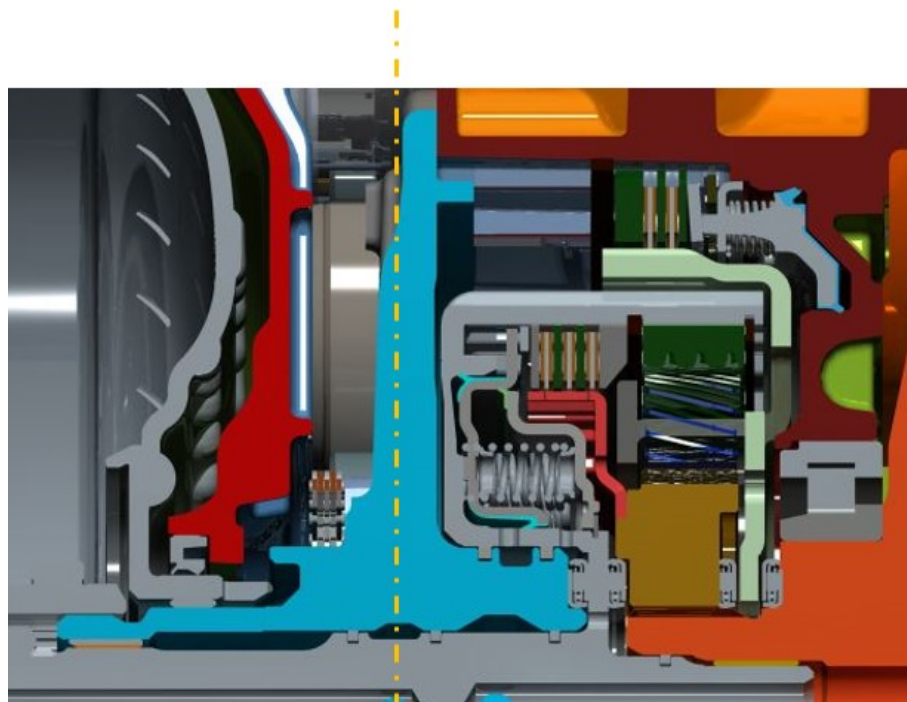


Figure 77 – Location of cross section B (courtesy of Punch Powertrain NV)

On the side view (Figure 77) it is possible to see from where the oil flows in.

By analysing the stator plate deeply (Figure 79) it can be seen that inside the previously identified lubrication channel there are two possible paths that the fluid can take.

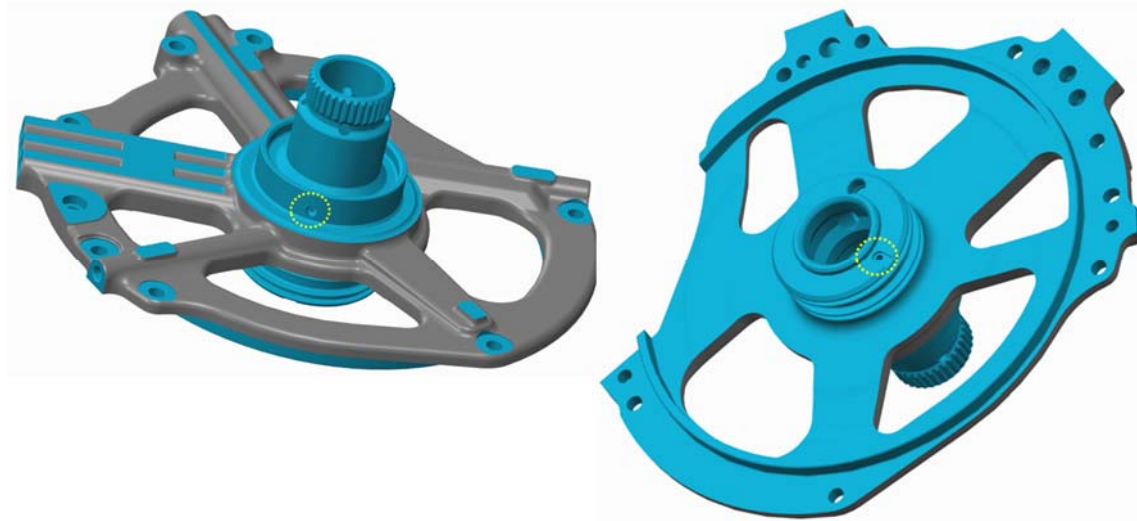


Figure 79 – Stator plate's lubrication outputs (courtesy of Punch Powertrain NV)

In Figure 80 it is possible to see the location of two outputs of the channel in question.

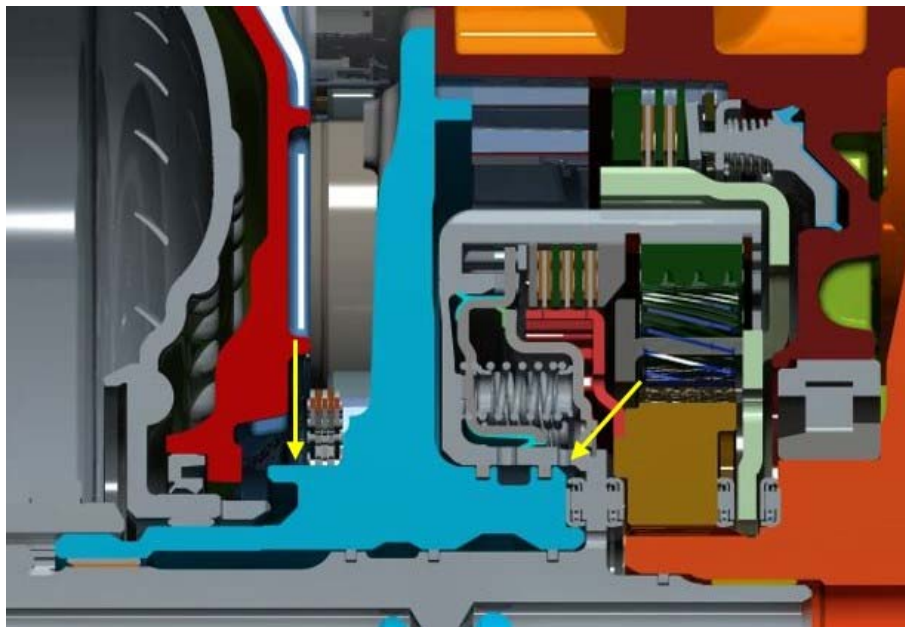


Figure 80 - Location of the stator plate's oil outputs (courtesy of Punch Powertrain NV)

Having in account Figure 81 (cutting section G), contrarily to previously stated, the path of the stator plate in study is not dedicated to the DNR subsystem but shared with

another subsystem. According to Figure 81 it is possible to conclude that one path ends in the sprocket.

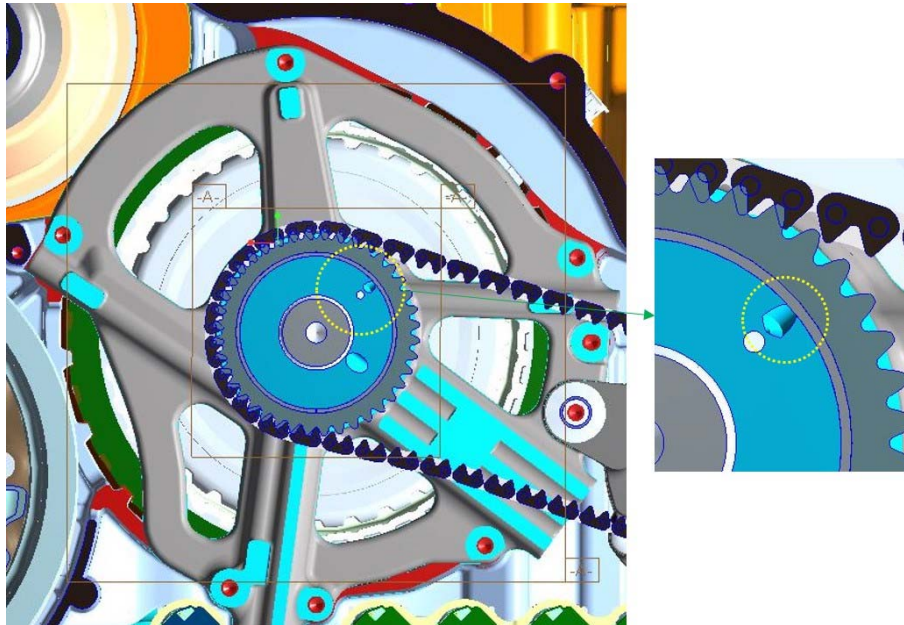


Figure 81 – Cross section G (courtesy of Punch Powertrain NV)

In Figure 82 it can be seen the location of the cutting section of cutting section G.

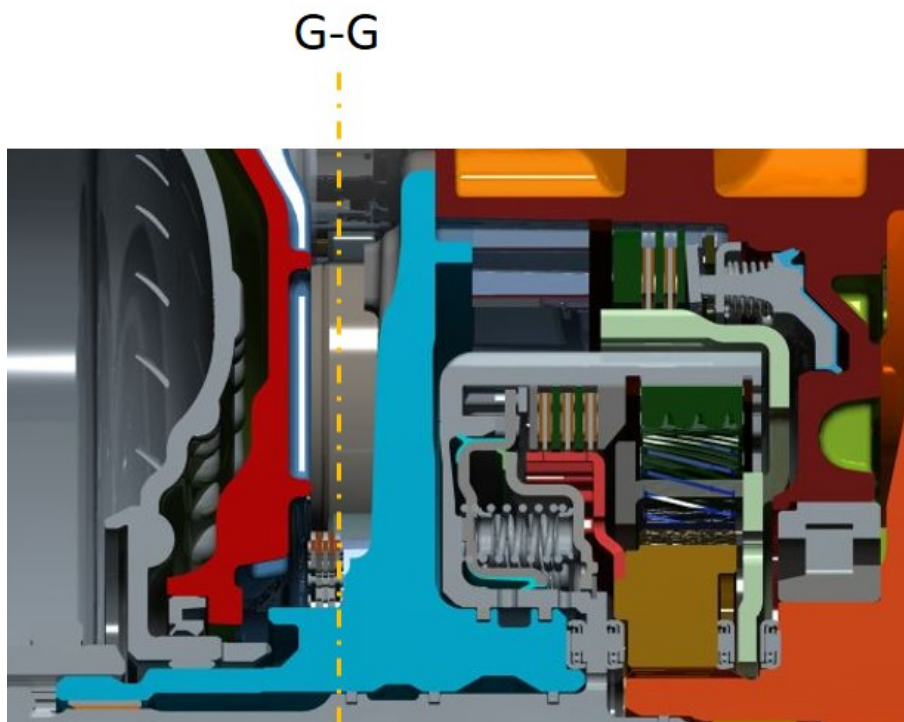


Figure 82 – Location of cutting section G (courtesy of Punch Powertrain NV)

Forward we will see that the other one continues to inside the DNR subsystem.

The output in this area has the function of directing the flow to the sprocket assembly and in this way the use of a bearing is avoided. In Figure 79 (left picture) it can be seen the hole which directs the flow to the sprocket.

Component 1 and component 2 (Figure 83) are welded. The chain rotates the sprocket (component 1) and as a consequence the component 2 rotates too. The oil film between the stator plate (blue part) and component 2 originates the effect of a bearing so it is not necessary to insert one there because the oil film is good enough to lubricate it.

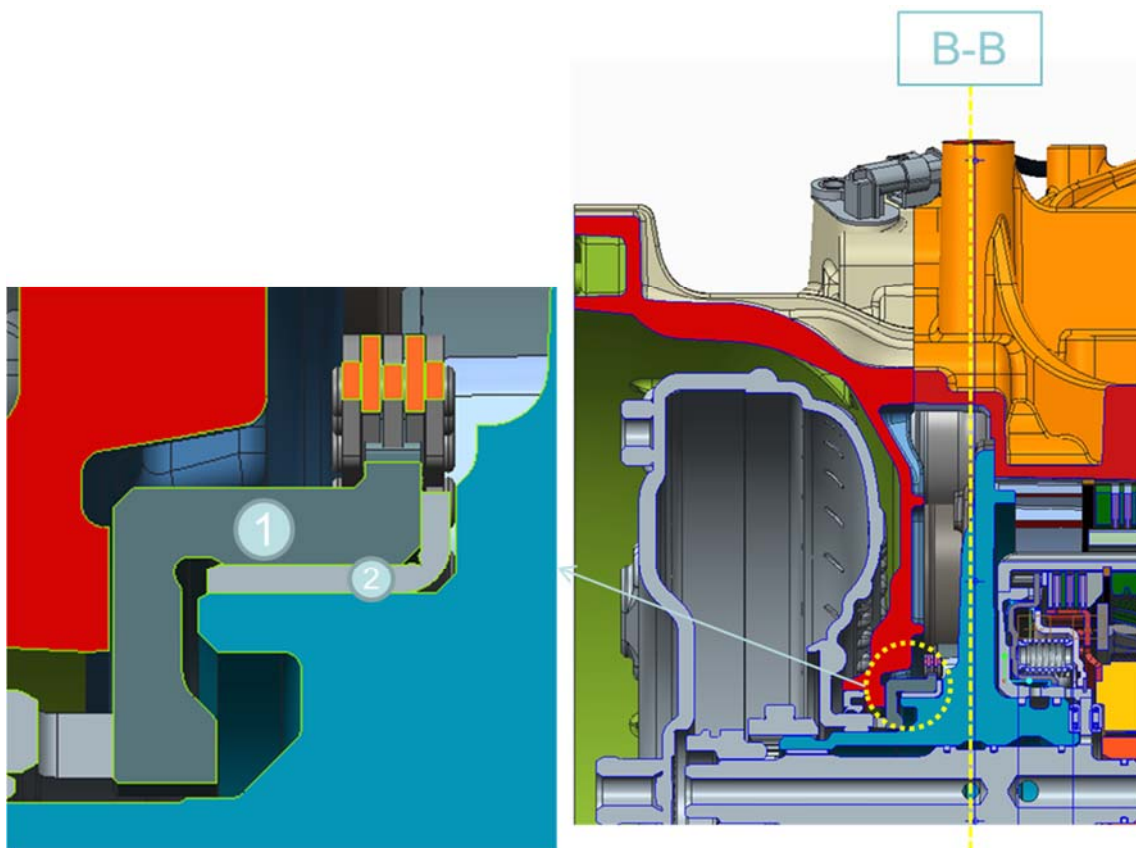


Figure 83 – Detail of the sprocket (courtesy of Punch Powertrain NV)

The other output presented on Figure 79 is the output to lubricate the DNR subsystem.

In Figure 80 it is possible to see where the oil flow output of the stator plate is located. The one on the left has just been presented. The other one on the right is the output which will send the flow to the DNR subsystem.

In Figure 85 (cutting section H) it is possible to see the other output of lubrication channel of the compensation chamber.

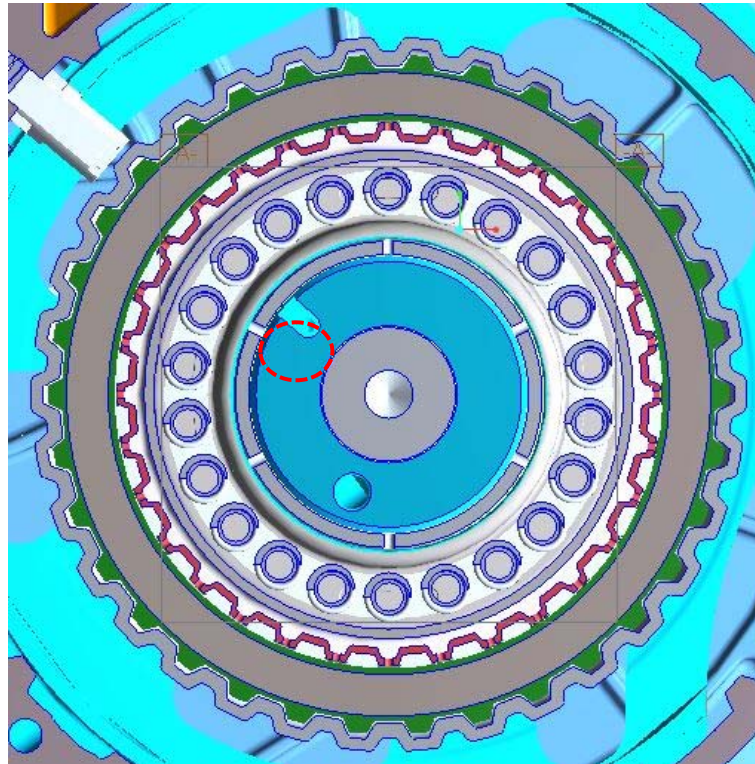


Figure 85 – Cross section H (courtesy of Punch Powertrain NV)

In Figure 84 it is presented the location of the cutting section H.

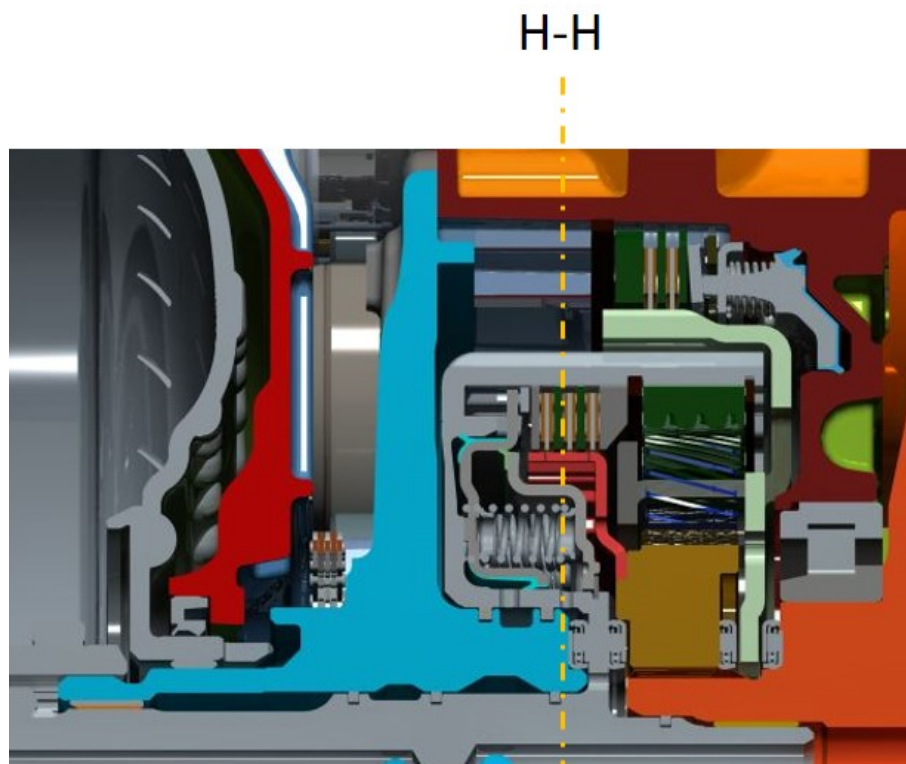


Figure 84 - Location of cutting section H (courtesy of Punch Powertrain NV)

It is now explicit that the lubrication path of the DNR starts on the FWD clutch. The fluid flows out of the stator plate and inside the forward springs pack through the holes on the turbine shaft assy (Figure 86).

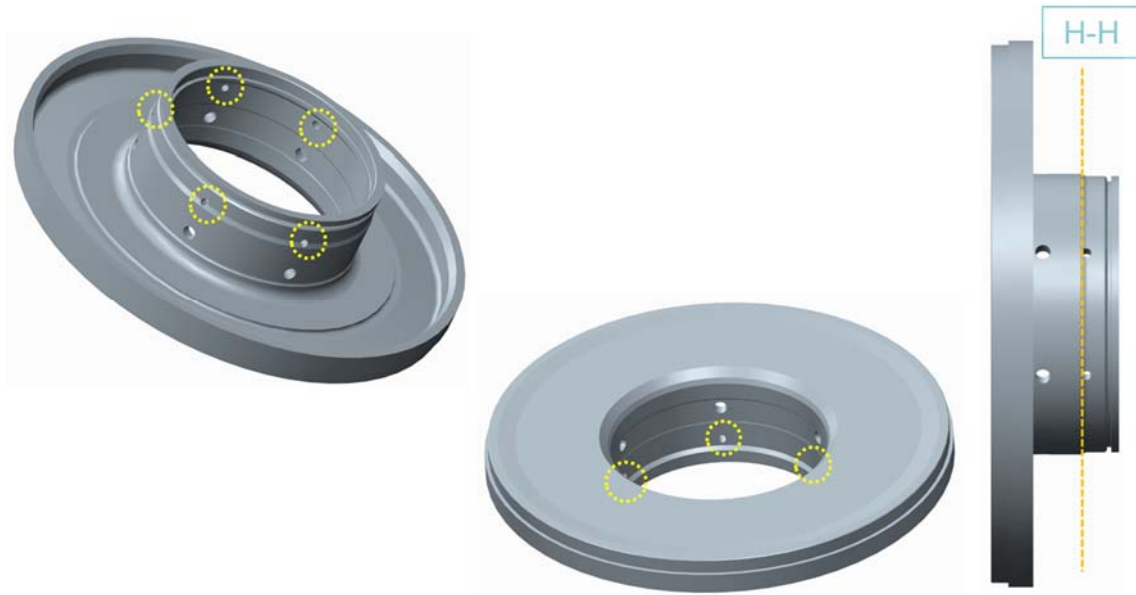


Figure 86 – Turbine shaft assy (part of) (courtesy of Punch Powertrain NV)

In Figure 87 it is presented the oil flow. It is expected that the oil flows all over the circumferential perimeter of the components due to the fact that the clutch carrier is rotating (centrifugal force).

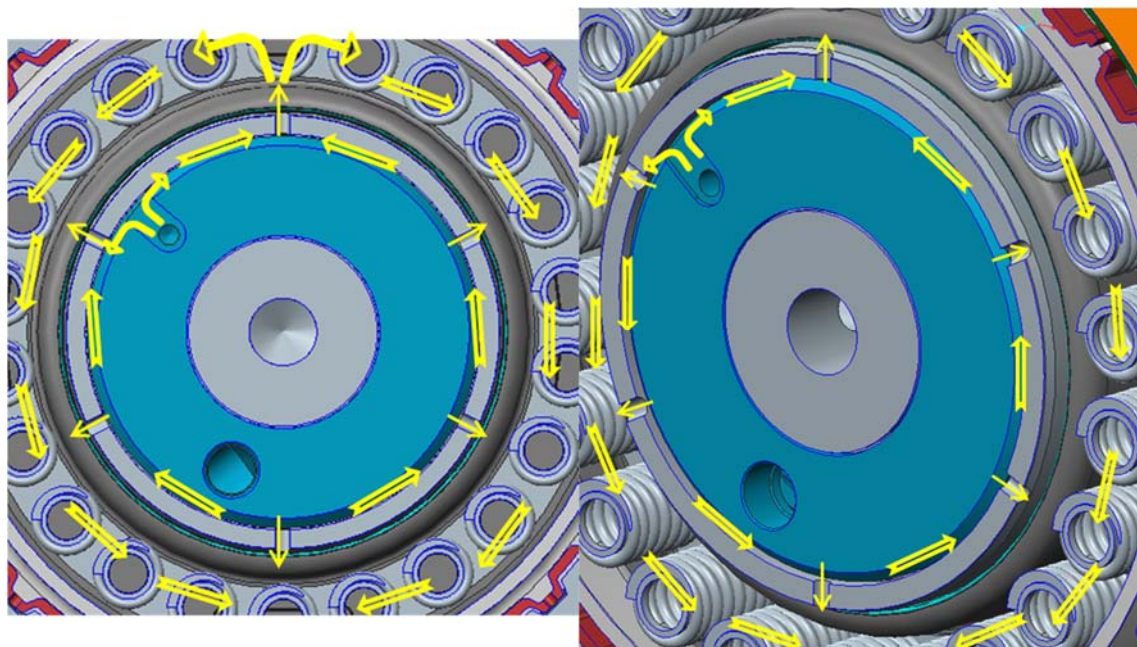


Figure 87 - Oil flow on cutting section H (front and perspective view respectively) (courtesy of Punch Powertrain NV)

In Figure 84 it is possible to see where the hole of the clutch carrier is located inside the DNR subsystem.

After reaching the springs pack, when the chamber is not full, the oil will go to the outer diameter of the balance dam according to Figure 89 due to the centrifugal force as long as the clutch carrier is rotating.

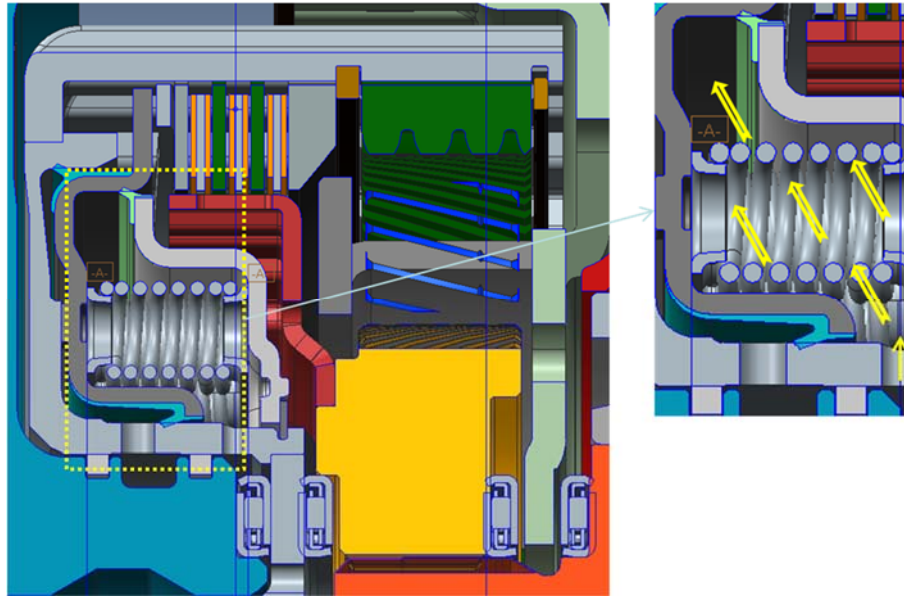


Figure 89 - Oil flow inside compensation chamber (courtesy of Punch Powertrain NV)

Once the chamber is full, some oil will start going to the right direction (reverse mechanism direction) in order to lubricate the other components.

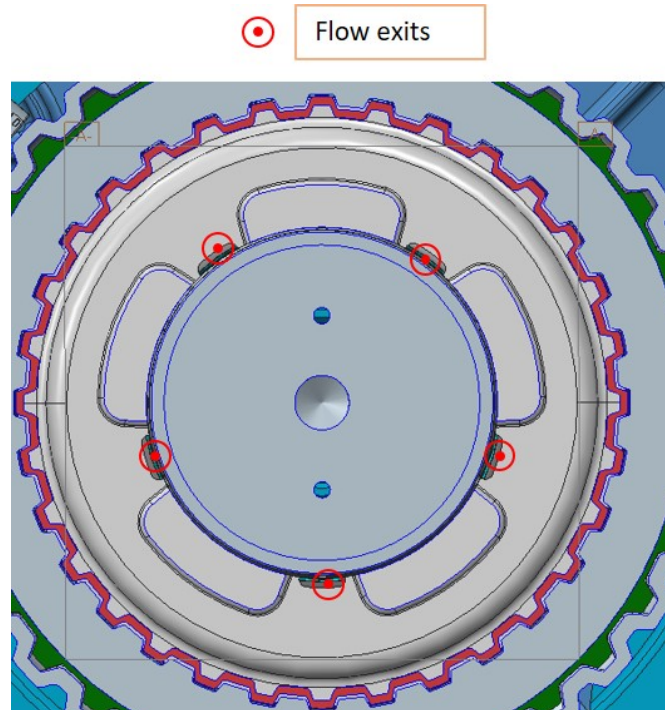


Figure 88 – Flow exits present on cross section I (courtesy of Punch Powertrain NV)

This fact can be assumed because there are some holes on the compensation chamber which allow the flow of the oil to go through. The lubricant will reach the other components flowing through the compensation piston. Cross section I (Figure 88) is an auxiliary to understand through where the oil flows. In Figure 90 it is possible to see the compensation piston and its holes to the oil flow.

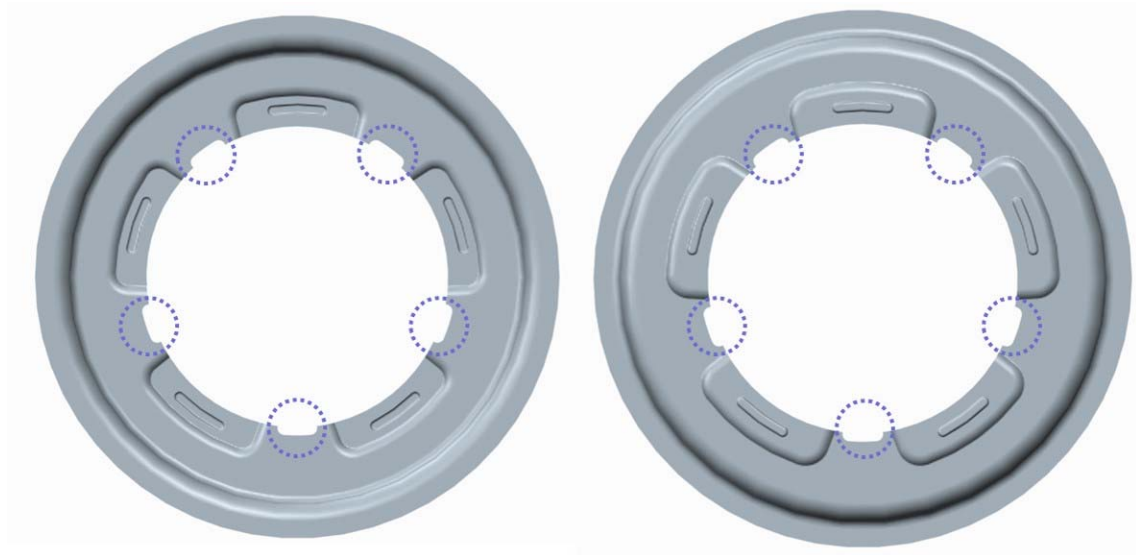


Figure 90 – Compensation chamber flow holes (courtesy of Punch Powertrain NV)

In Figure 91 it is visible the normal view of the cutting section I where the location of the compensation piston's holes can be seen.

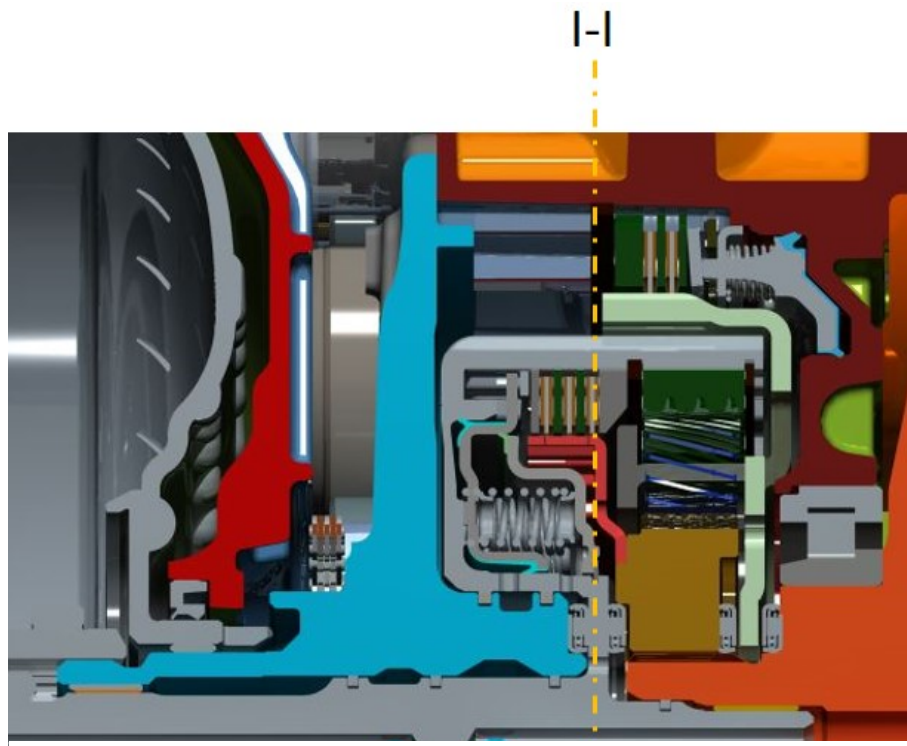


Figure 91 – Location of cross section I (courtesy of Punch Powertrain NV)

When the entire volume (between forward piston and compensation piston) is fulfilled with oil, it follows two directions. The first one is the outer diameter of the balance dam as previously explained but as long as all the area is fulfilled with oil, inevitably part of it is directed to the right side of the DNR subsystem. That flow can be seen in Figure 92.

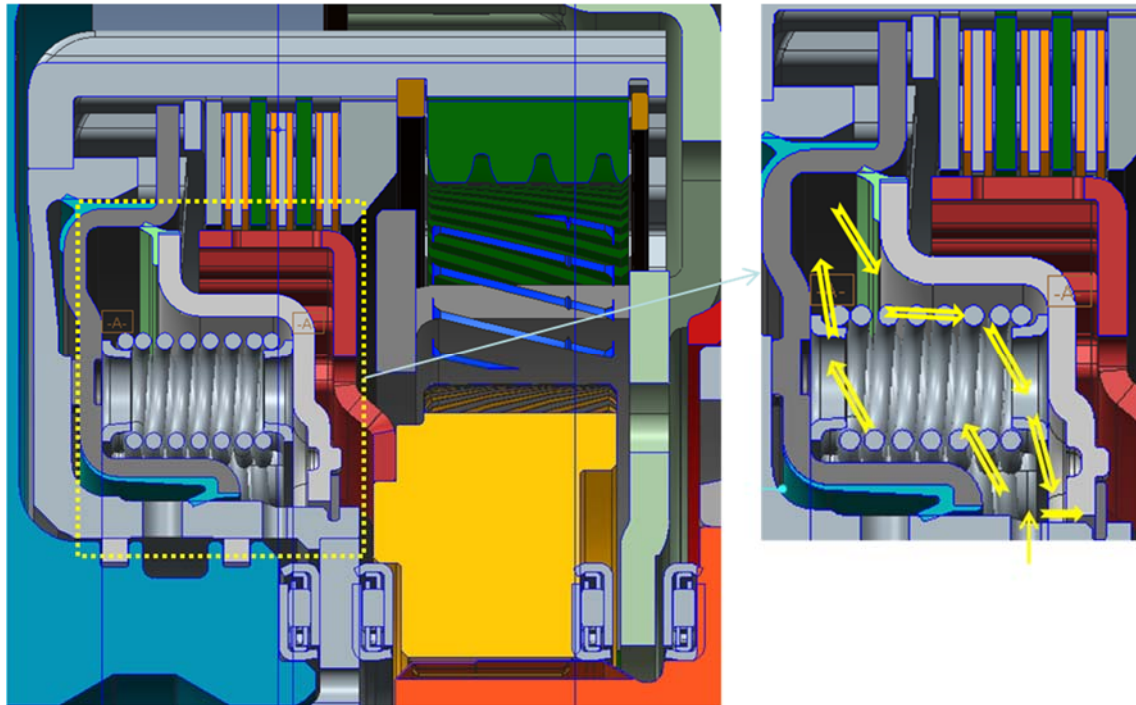


Figure 92 – Oil flow inside compensation chamber (courtesy of Punch Powertrain NV)

As we go forward on the section of the transmission, at first sight in the cross section J (Figure 93) the compensation piston's holes seem to be obstructed by the snap ring of the compensation chamber.

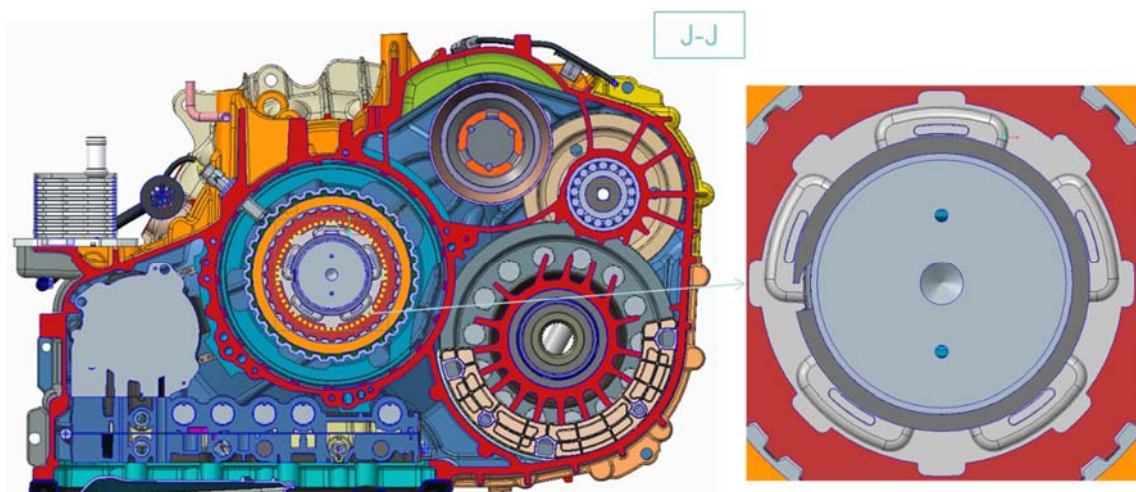


Figure 93 – Cutting section J

In a perspective view it is possible to conclude that the holes are not obstructed (Figure 94) due to existence of ledges on the compensation piston between holes. The main purpose of these ledges is to create an opening for the lubricant. Otherwise the oil flow circuit would finish here due to the presence of the snap ring.

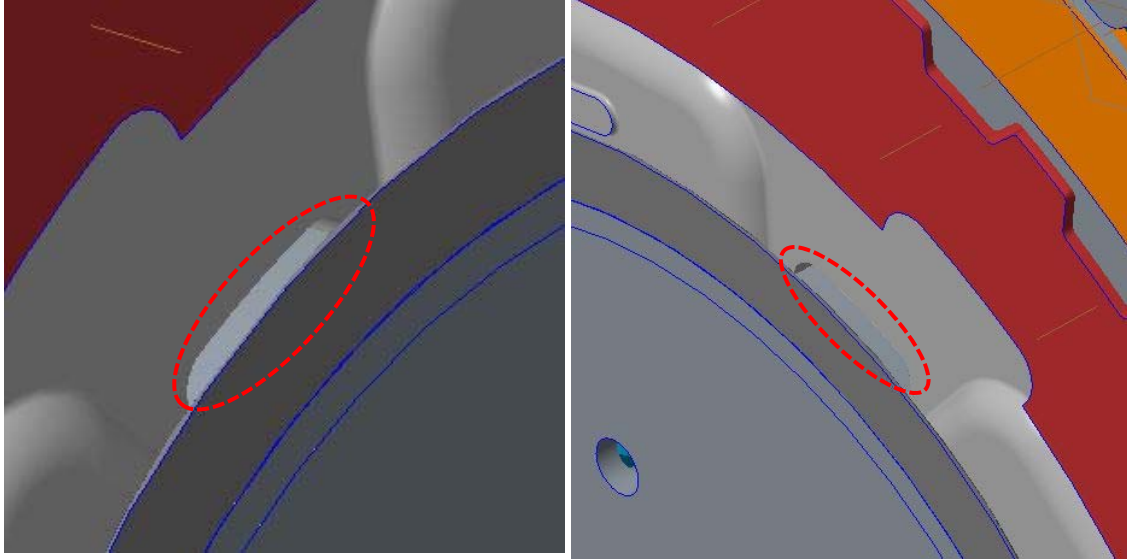


Figure 94 - Detail of perspective view of cutting section J (courtesy of Punch Powertrain NV)

Due to the centrifugal force and the presence of the snap ring the oil flows according to the Figure 95.

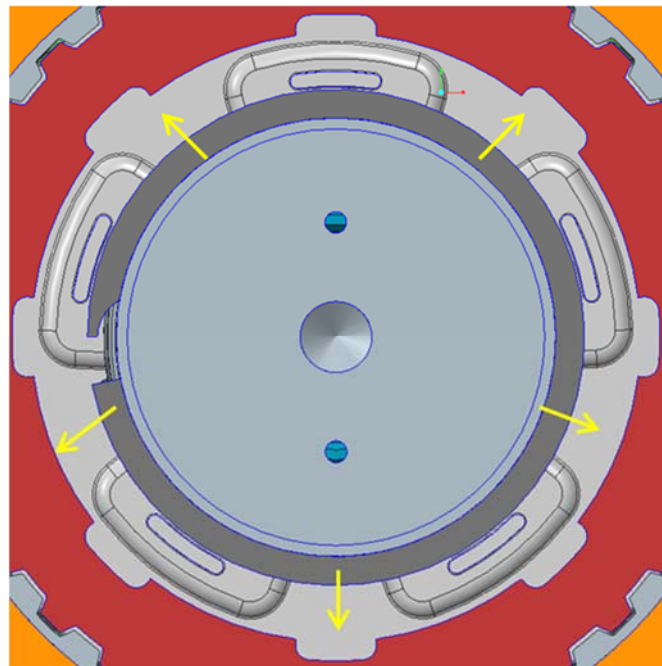


Figure 95 - Oil flow on cutting section J (courtesy of Punch Powertrain NV)

In a different view (Figure 97) it is possible to see the vertical gap between the ledges of the compensation piston and the snap ring and through which the lubricant flows.

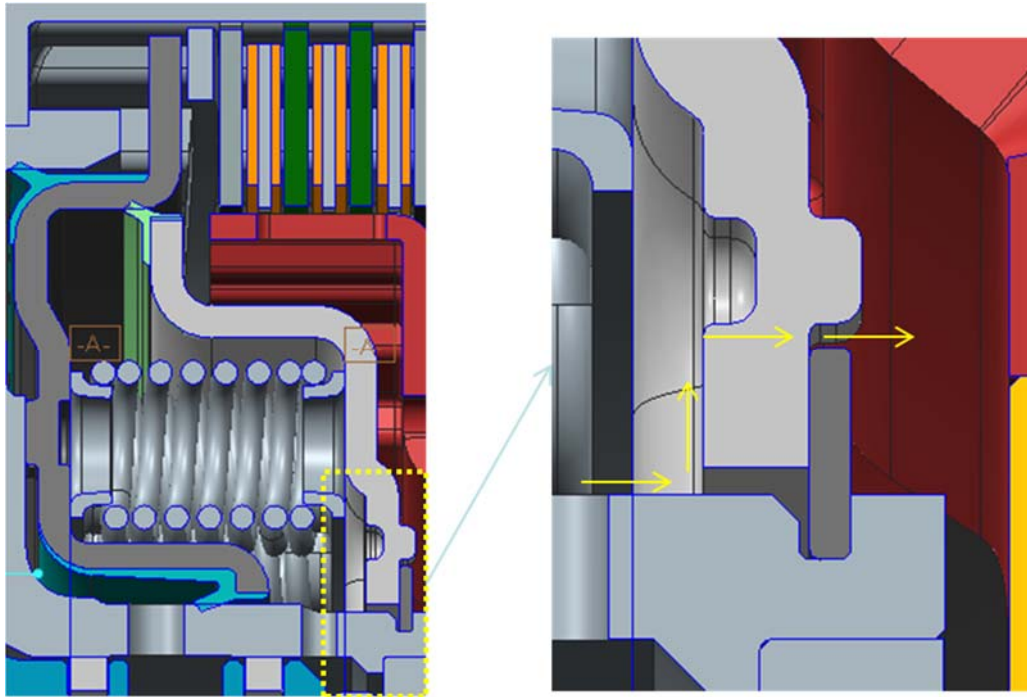


Figure 97 - Gap between compensation piston and snap ring (courtesy of Punch Powertrain NV)

In Figure 96 is presented the location of the cutting section J.

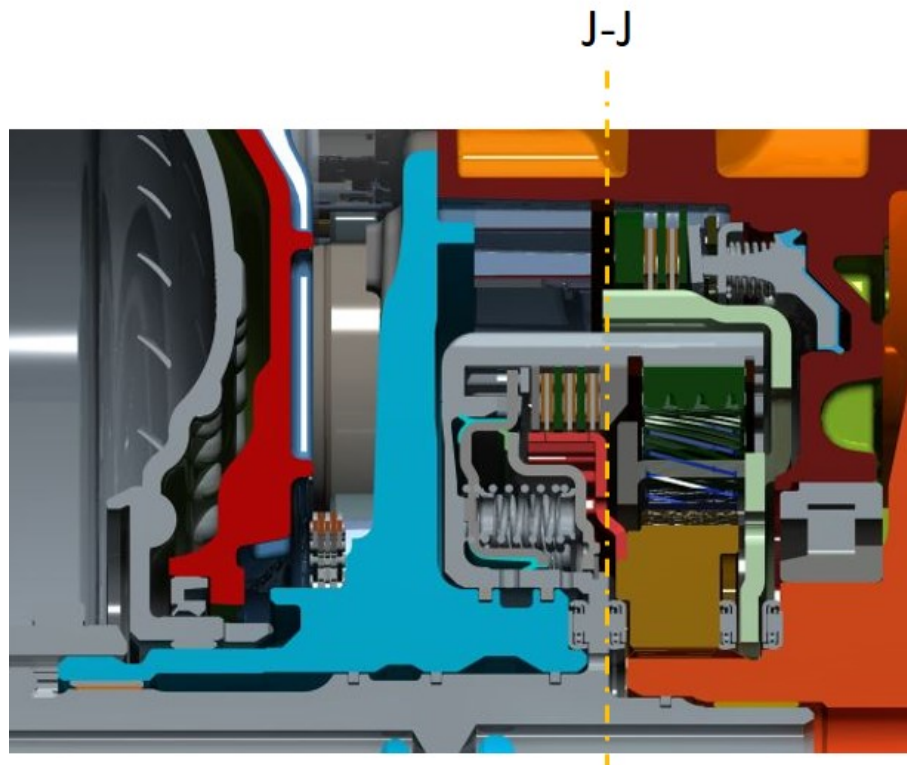


Figure 96 – Location of cross section J (courtesy of Punch Powertrain NV)

After flowing through the gap between the compensation chamber and the snap ring the fluid will get into a chamber which is created by the volume between the compensation piston and the sun gear. When the fluid arrives at this location it can follow 2 different directions. The first direction is the vertical one, once again because of the centrifugal force the fluid will be thrown to the outer diameter in direction to the forward clutch. More than 80% percent of the flow goes to the forward clutch plates as desired. A minimum fraction of oil (the remaniscant) goes to the right direction to the gears and bearings. It is presumable that the flow takes the right direction (direction of the reverse clutch) due to the existence of some holes on the sun gear.

The path of the lubricant fluid can be seen in Figure 98.

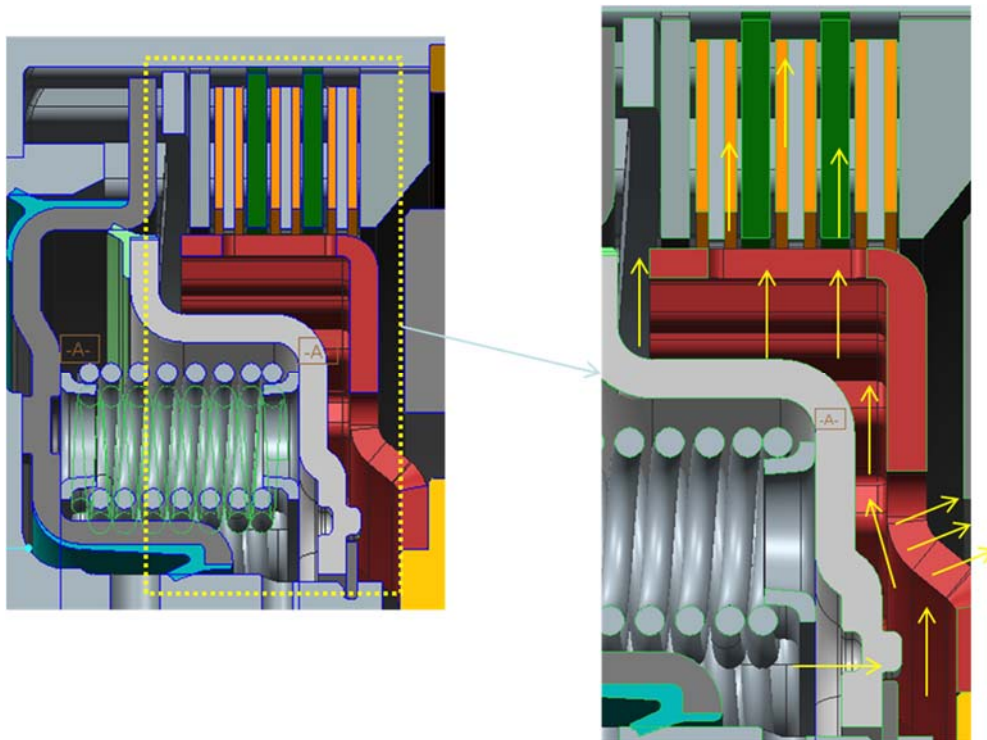


Figure 98 – Oil flowing through the sun gear (courtesy of Punch Powertrain NV)

In Figure 99 it is possible to see the holes of the sun gear which allow the fluid to go in two different directions: clutch plates and right direction of the DNR. The holes which provide an axial oil flow are presented on the Figure 99 (a). The holes which provide radial oil flow are presented on Figure 99 (b). As mentioned above it is desired that 80% of the oil flow reaches the clutch plates so it is necessary that the sum of the areas of the radial holes is larger than the summing of the areas of the axial flow. It is not demanding that the summing area of the radial holes is 80% of the total area of holes (radial plus axial) due to the fact that the centrifugal force is higher than the force which pushes the oil in the axial direction so more oil will be sent to the radial direction.

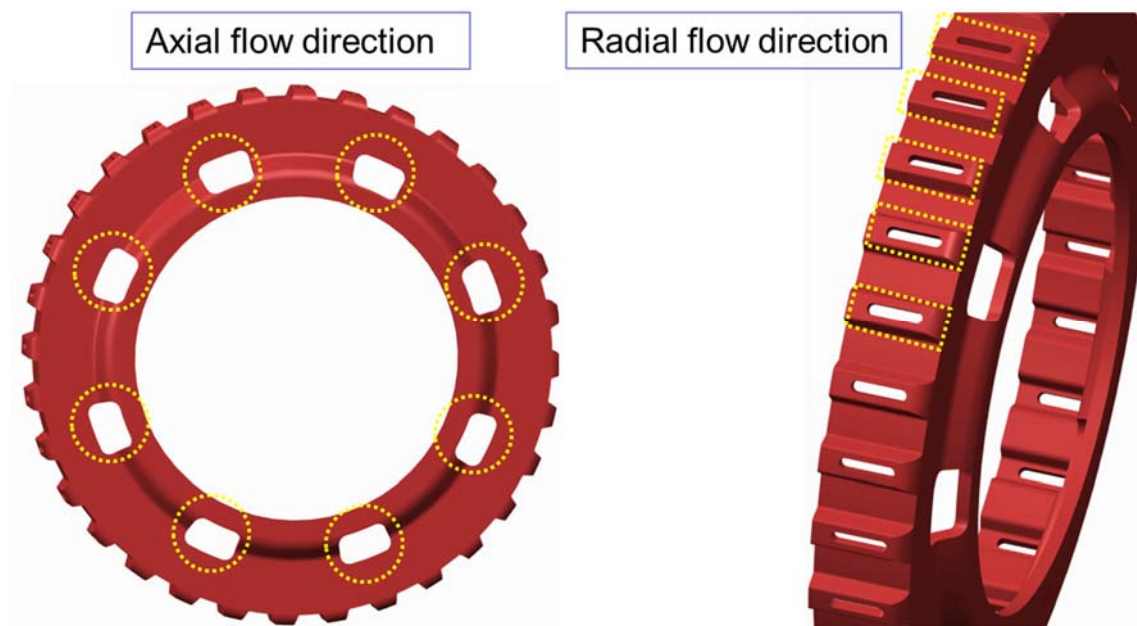


Figure 99 - Sun gear oil outputs: axial flow direction outputs (a) and radial flow direction outputs (b) (courtesy of Punch Powertrain NV)

In cutting section K (Figure 100) it is possible to see the axial holes of the sun gear. It is also possible to see that there isn't any component blocking the oil so the oil will keep flowing to the next components of the DNR subsystem. In Figure 100 it is presented the point through where the oil comes.

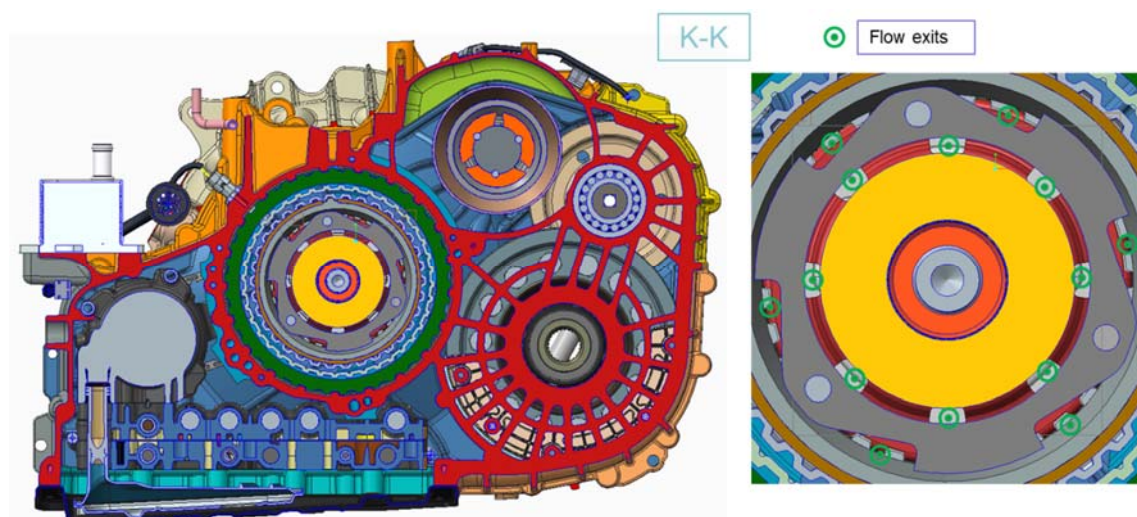


Figure 100 – Cross section K (courtesy of Punch Powertrain NV)

In Figure 101 it can be seen a perspective view of the cutting section K.

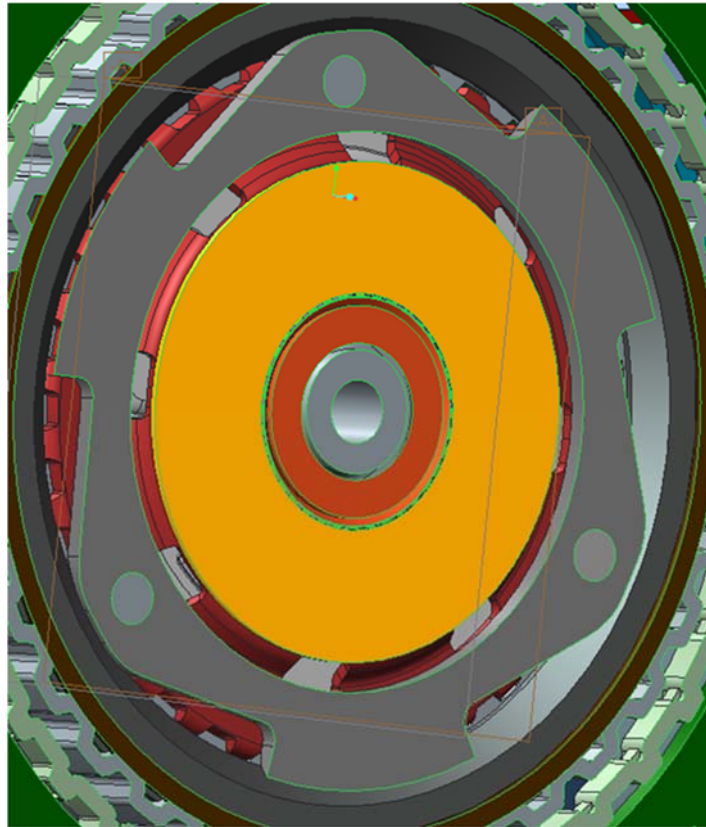


Figure 101 – Perspective view of cutting section K (courtesy of Punch Powertrain NV)

In Figure 102 it is presented the location of the cutting section K.

K-K

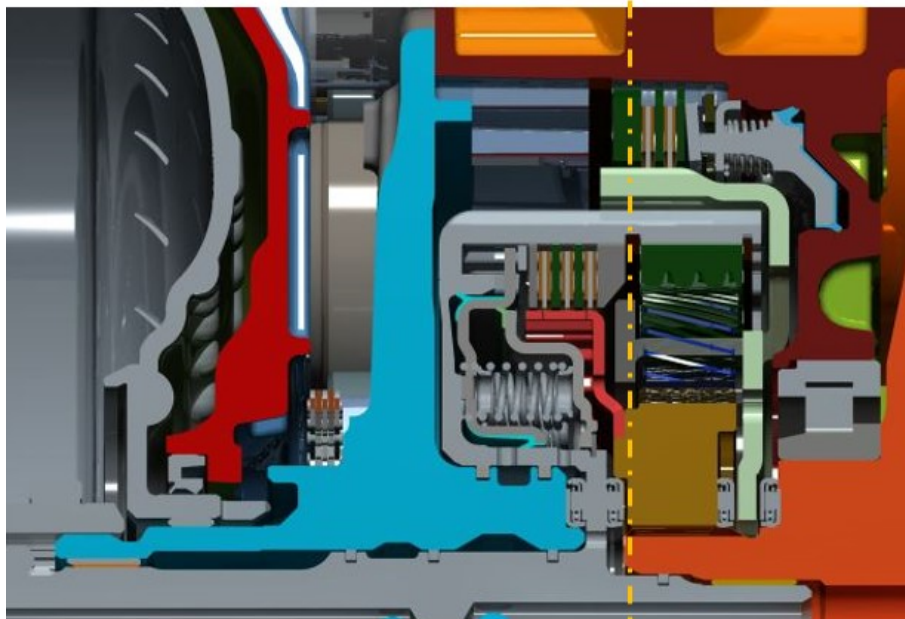


Figure 102 – Location of cutting section K (courtesy of Punch Powertrain NV)

According to Figure 103 the oil can follow two different paths after leaving the sun gear. One path is horizontal (axial direction) and the other one is vertical (radial direction). The oil leaves the sun gear through the axial direction so now it keeps that path. The lubricant also goes to the radial direction due to the centrifugal force.

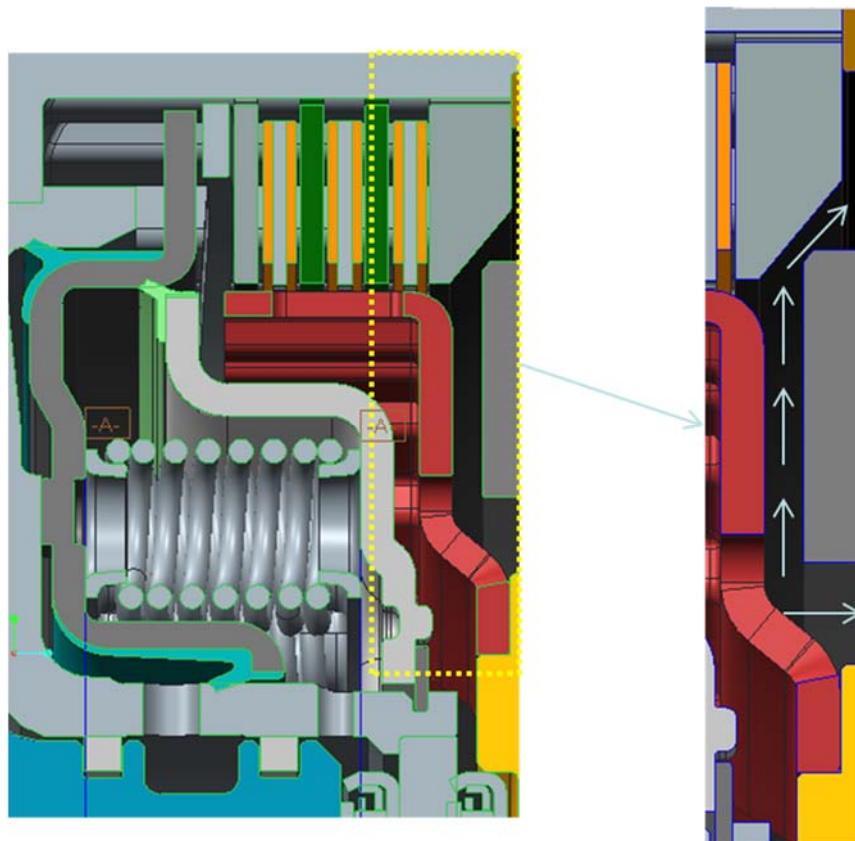


Figure 103 - Oil flow immediately after leaving sun gear (courtesy of Punch Powertrain NV)

After that the oil will reach the PGS (Planetary gear set). As previously mentioned at this assembly there are several components which need to be cooled and lubricated. Some of the components which need to be lubricated are: ring gear, sun gear, planets and the bearings inside the planets.

In cutting section L (Figure 104) it is possible to see that the lubricant reaches the contact areas between the planetary gears and the sun gear and the contact areas between the planetary gears and the ring gear. So, at this point cooling and lubrication are ensured for the contact sets between gears. In Figure 104 it is also possible to see where the oil comes from.

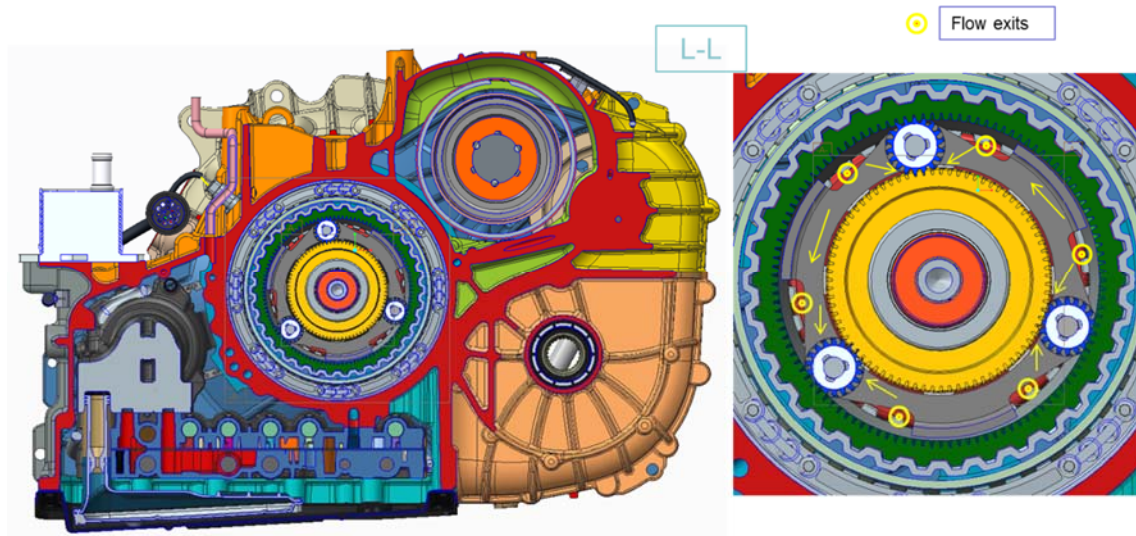


Figure 104 – Cutting section L (courtesy of Punch Powertrain NV)

To lubricate the bearings inside the planets it was created three holes on the washers. It is important to underline the fact that the quantity of oil that goes to these bearings is just the necessary to lubricate them but not to cool them. The gears don't need a significant quantity of oil because they just need an oil film to work properly. Although it is important to ensure that they are correctly cooled.

In Figure 105 it can be seen a perspective view of the cutting section L and in Figure 106 the location of the cutting section.



Figure 105 – Perspective view of cross section L (courtesy of Punch Powertrain NV)

When the oil leaves the PGS it can flow in two different directions. Once again, radial and axial direction.

At a first sight it could be assumed that the oil would flow according to Figure 107 (a) close to the reverse piston. However, the centrifugal force makes the oil flow up immediately after leaving the PGS according to Figure 107 (b). Then the oil flows close to the PGS cage until the radial surface (spline) of the turbine shaft assembly. There is a spline in the radial surface of this component in order to spread the oil in the radial direction due to centrifugal force and by this way ensure the oil will reach the radial surface of the PGS cage through several holes. Once again, the oil will be spread due to centrifugal force to the clutches.

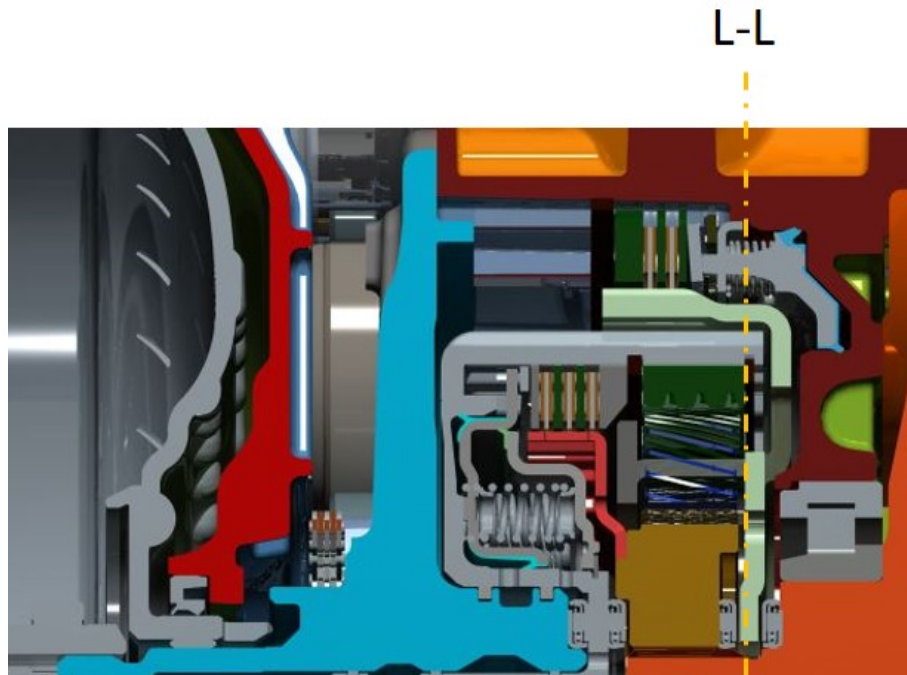


Figure 106 – Location of cutting section L (courtesy of Punch Powertrain NV)

When reverse clutch is pressed, the planetary gear set is steady. This means that the planetary gear set cage would not create a centrifugal force which would push the oil to the clutch plates. For this reason, the flow circuit of the first picture (Figure 107 (a)) is wrong. It is impossible a significant and necessary quantity of oil goes to the right direction as indicated in the first picture, but it is in a low quantity. As the cage of the forward system is always rotating this means that this component will always create a centrifugal force. This force is desired to send the oil to the reverse clutch plates. On both forward cage and planetary gear set cage there are holes (similar to the red part of the sun gear assembly) to increase the quantity of oil which goes to the reverse clutch plates.

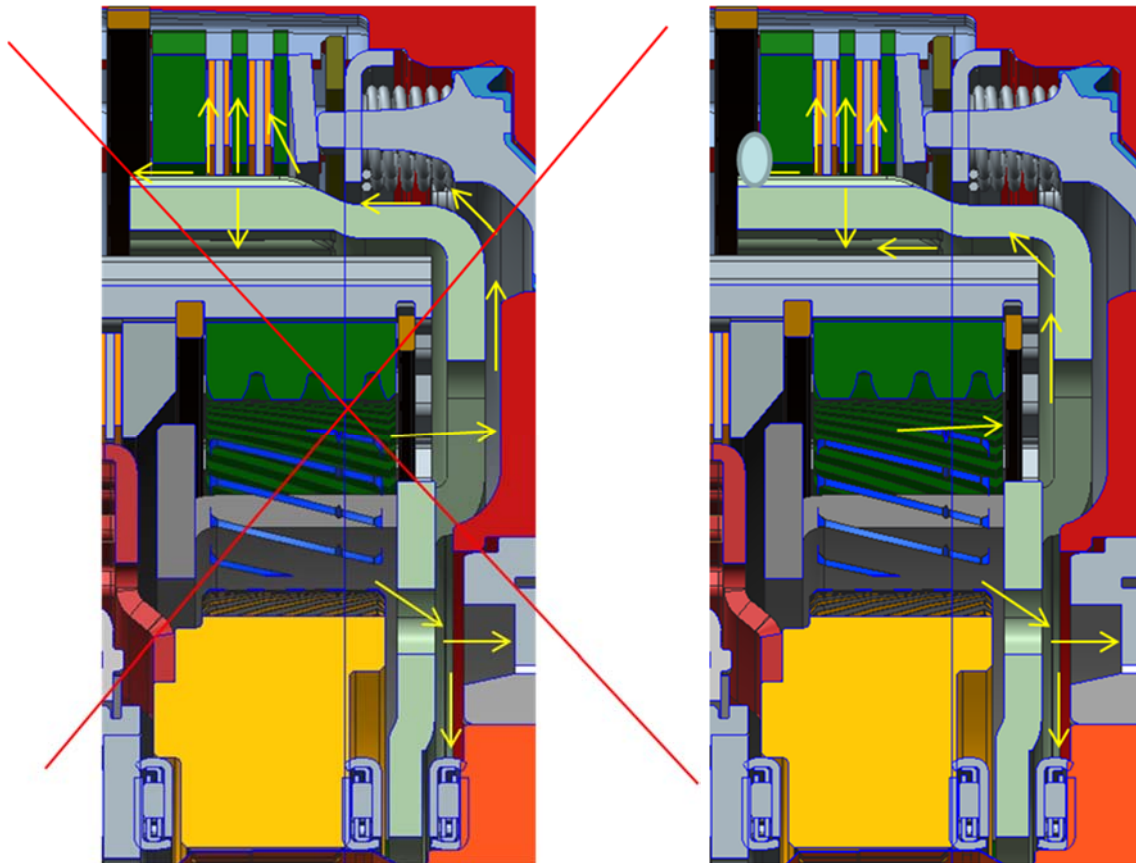


Figure 107 - Oil flow when leaving PGS: incorrect oil flow (a) and correct oil flow (b) courtesy of Punch Powertrain NV)

On the signed area of Figure 107 (b) (blue circle) it is located a wall which was created to avoid the lubricant flows to the left direction as it would be wasted. When the components which have rotational motion are not rotating at high speed or at least at a speed not high enough to generate centrifugal force able to direct the oil flow radially some oil will go down. As it can be seen in Figure 108 some of that oil will lubricate and cool the thrust bearings and the slider between the secondary shaft and the sun gear.

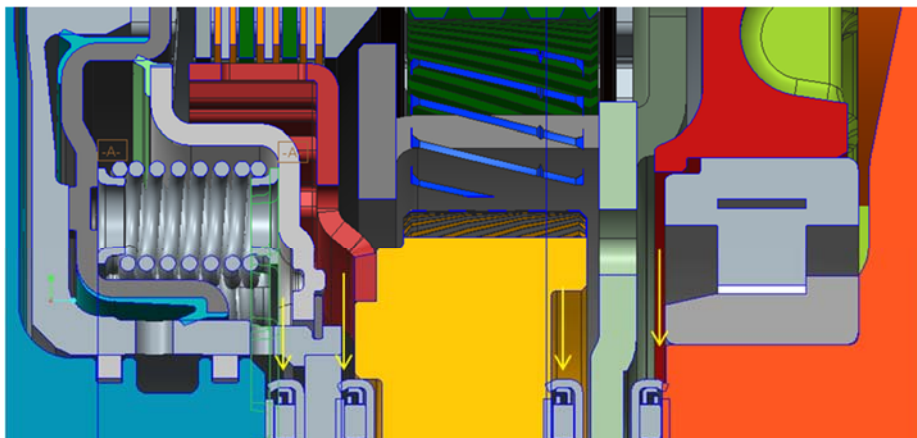


Figure 108 – Oil flow on thrust bearings (courtesy of Punch Powertrain NV)

The pointed-out component on Figure 109 whose main purpose is to spray the belt with lubricant will also spray some oil to the DNR subsystem, mainly for the bearing. The path of the oil is presented on the same figure.

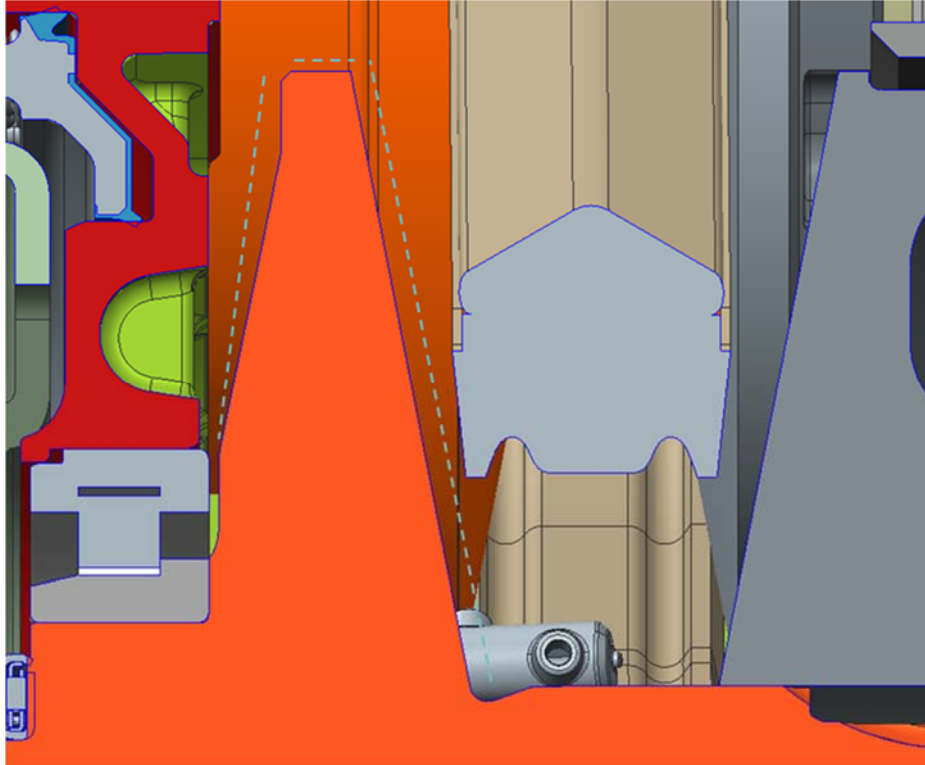


Figure 109 - Oil path which comes from the primary (output) shaft (courtesy of Punch Powertrain NV)

In Figure 110 is presented a chamber. In this chamber, high pressure is built which will push the oil stocked in the chamber to the DNR subsystem. By this way the thrust washer will be lubricated. This is only possible because at high pressure the seals will deform, and the deformation creates a gap great enough to let some oil flow through. In Figure 110 is also signed the path of the oil from the high-pressure chamber to the thrust bearing (yellow lines).

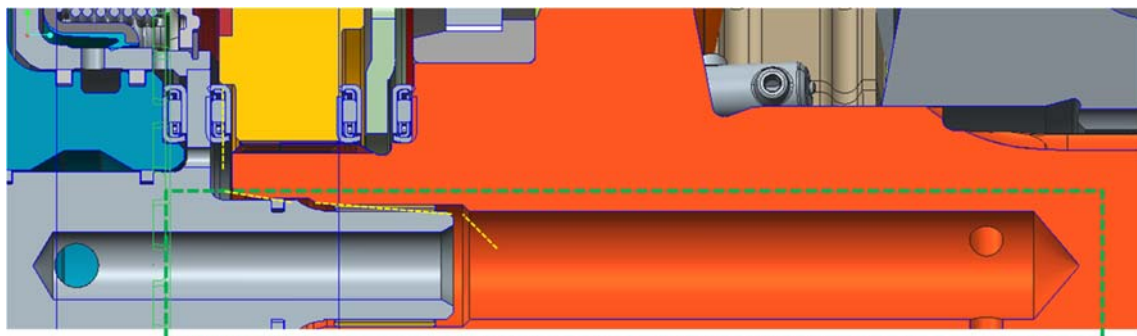


Figure 110 - High pressure chamber between turbine shaft and primary shaft (courtesy of Punch Powertrain NV)

In Figure 111 it is illustrated a resume of the lubricant path.

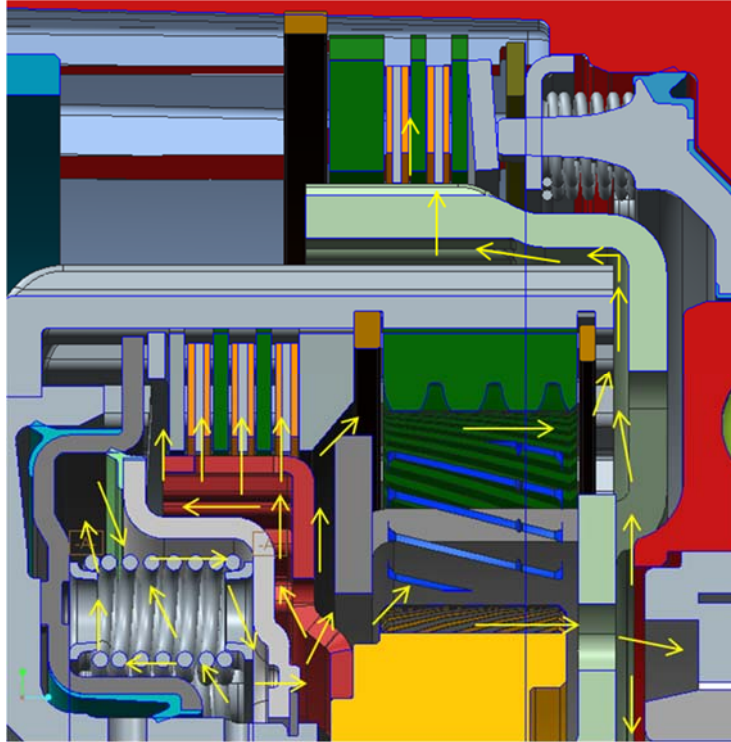


Figure 111 – Resume of the lubricant flow (courtesy of Punch Powertrain NV)

3.3.4.3 FWD clutch flow circuit

In Figure 112 it is possible to see once again the stator plate. Previously they were presented the holes for lubrication circuit. Now they are going to be presented the holes of the forward clutch circuit. In Figure 112 are pointed out some spheres. These balls main goal is to ensure that the oil flows through the right path. The paths that the balls are blocking are not used for flow. These paths are a consequence of the manufacturing process. As this part is machined (drilling process) new holes had to be made to allow doing the necessary and desired paths. In order to “close” the unwanted holes / paths it was applied these balls. To only way to avoid the use of these balls would be to change the manufacturing process and choose one other such as injection.

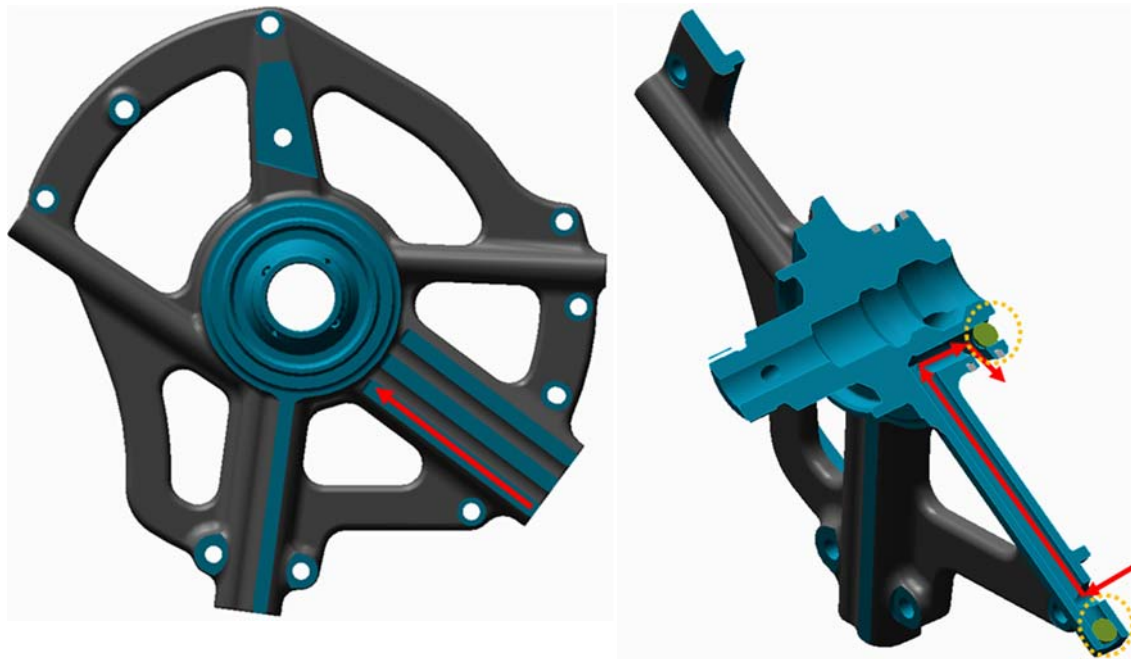


Figure 112 - Stator plate's channels for forward clutch oil circuit (courtesy of Punch Powertrain NV)

The holes illustrated in Figure 113 are a consequence of the manufacturing process as previously mentioned.



Figure 113 – Stator plate's channels for forward clutch oil circuit (courtesy of Punch Powertrain NV)

In Figure 114 it can be seen cross section D where it is presented the output hole of the stator plate. This is the hole from where the oil goes into the DNR subsystem.

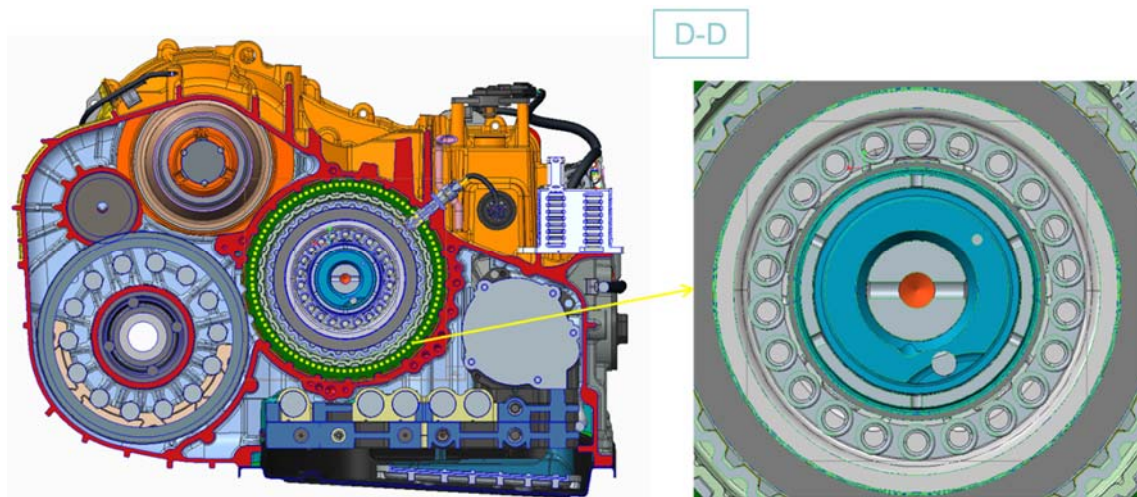


Figure 114 – Cross section D (courtesy of Punch Powertrain NV)

In Figure 115 it is presented the location of the cutting section D.

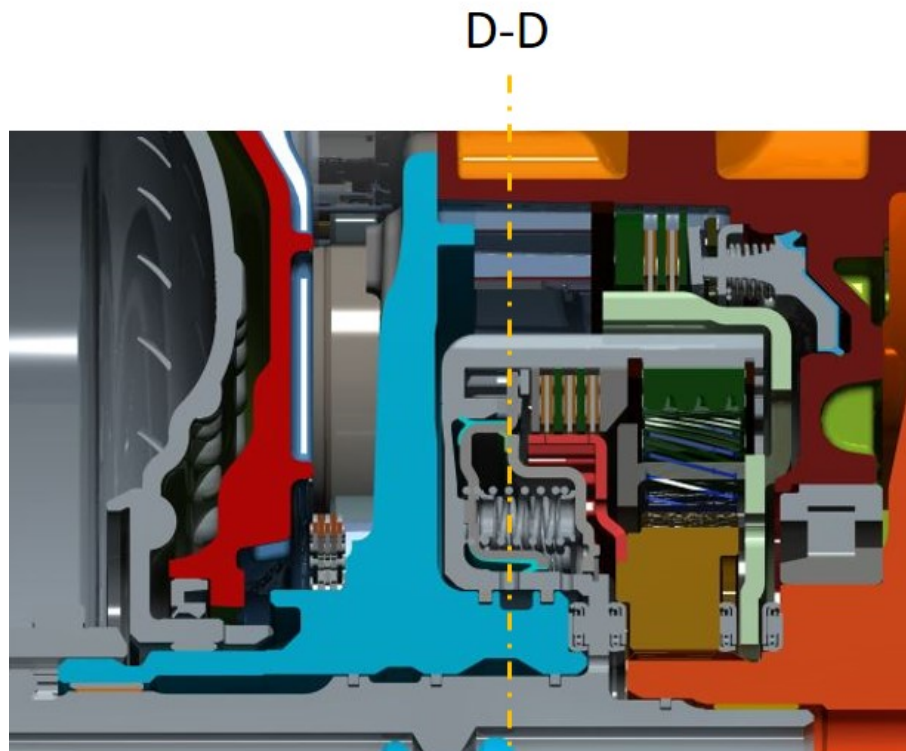


Figure 115 – Location of cross section D (courtesy of Punch Powertrain NV)

When the oil leaves the hole of the stator plate it starts a circular path due to the fact that the turbine shaft is rotating. After that the oil follows a radial path due to the centrifugal force through the presented holes (Figure 116).

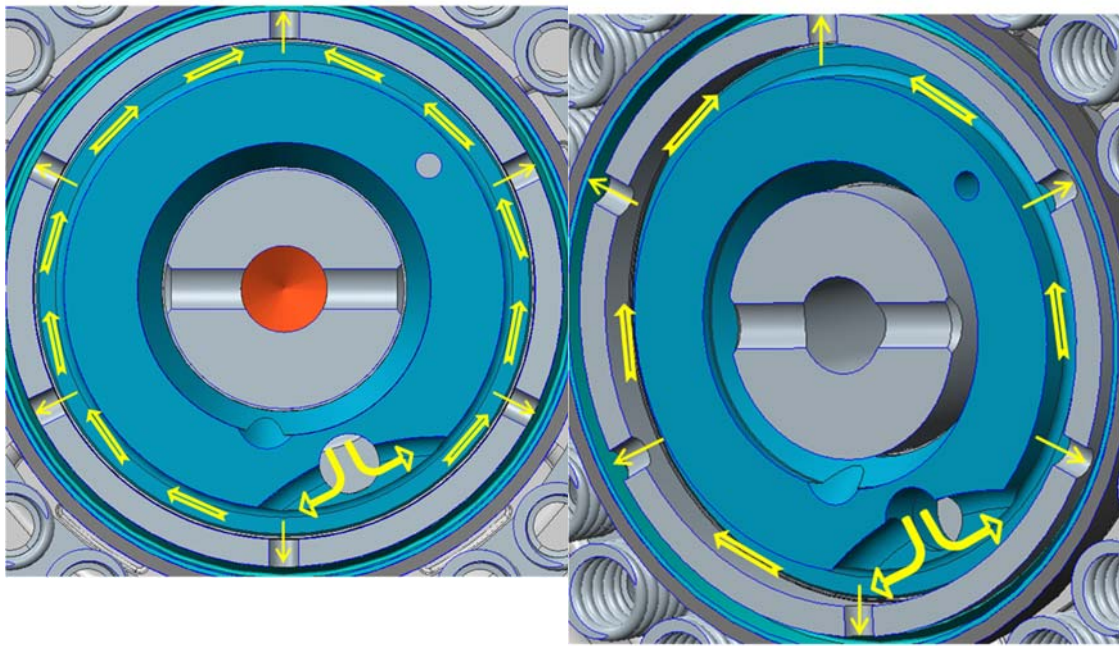


Figure 116 – Oil flow on cross section D (courtesy of Punch Powertrain NV)

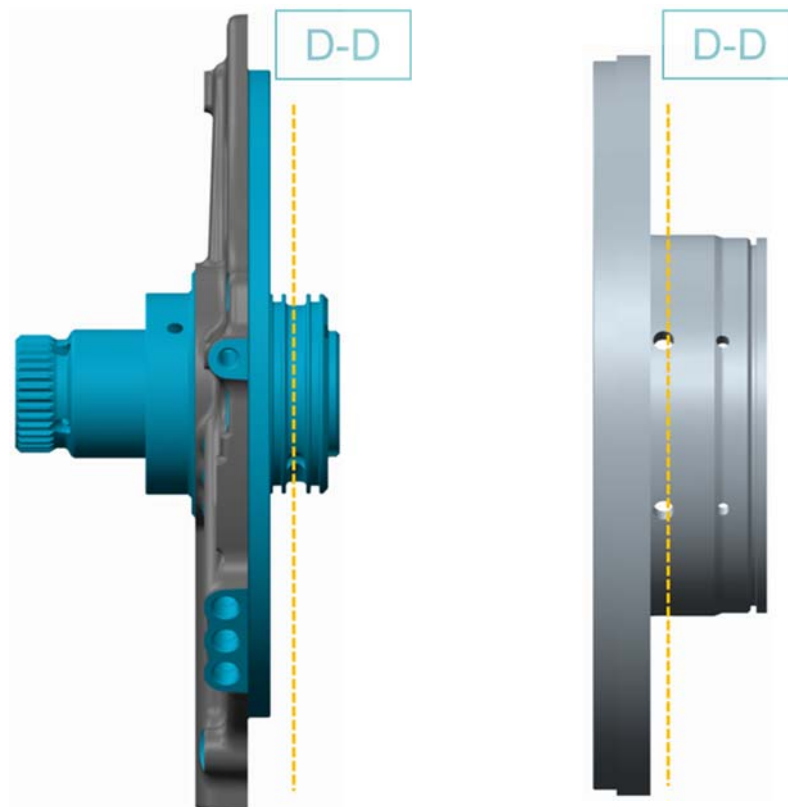


Figure 117 - Holes for oil flow in stator plate (a) and turbine shaft assembly (b) (courtesy of Punch Powertrain NV)

In Figure 117 it is illustrated the holes from where the oil flows through in the turbine shaft and stator plate. Regarding to the turbine shaft it was built six holes because there wasn't enough room to have only one hole with the necessary area to guarantee the necessary flow rate. So, it was used six holes with a total area equal to the necessary area to guarantee the flow rate. In addition, if it was used only one hole it would originate a lot of turbulence and unbalancing.

It is important to underline the fact that the principle of the holes' area is applied to all the components because if the total area of the holes is not greater than the necessary or at least equal to the area which ensures the right flow rate some components cannot be cooled or lubricated correctly. Furthermore, as the oil flows through more than one part it could create turbulence in the previous part which is not desired at all as it could create air bubbles and the system would become unstable.

In Figure 118 they are presented perspective views of the components.



Figure 118 - Perspective view of turbine shaft assy and stator plate (courtesy of Punch Powertrain NV)

The oil will reach a chamber between the turbine shaft assembly and the forward piston. The pressurized oil will pull the forward piston (Figure 119).

When the engine is steady, air appears in the chamber between the turbine shaft assembly and the forward piston. The air isn't desired at all because it will create turbulence. When the engine starts running the chamber is fulfilled with air. When the engine starts running and the oil starts coming into the chamber due to the centrifugal force and as it is heavier than the air, it will be thrown to the outer diameter and the air will go down. Then the air will go out through the seals which are housed in the blue part. When the engine stops again the oil tends to flow away again. This is bad when the car is equipped with START/STOP technology. So, for short periods of time it has to be

ensured that the flow rate which goes away is as low as possible. Otherwise if the driver presses full throttle there will not be enough oil in the chamber, so the clutches will not be pressed and the car will not move or at least not with the required torque.

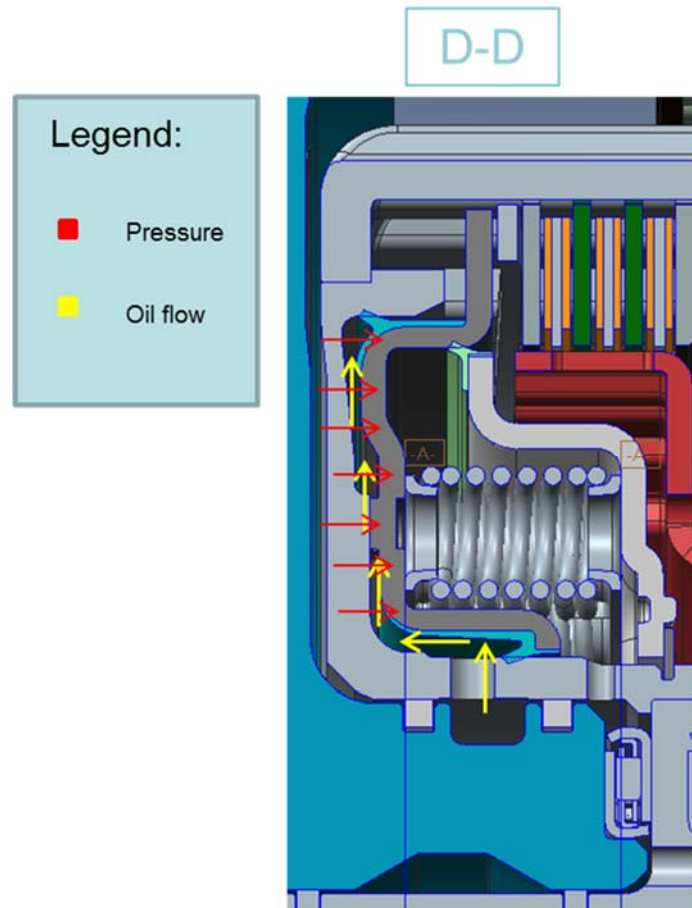


Figure 119 - Chamber between turbine shaft assembly and forward piston (courtesy of Punch Powertrain NV)

3.3.4.4 *REV clutch flow circuit*

In Figure 120 it is possible to see the CVT transmission housing. In the same picture it is illustrated two holes. The upper one is only for prototype purposes while the other one on the bottom is dedicated to the oil circuit. The oil gets in and out through the same hole.

Transmission housing and reverse piston will create a chamber. When the oil reaches this chamber, it will create pressure and by this way the piston will move and press the clutch plates.

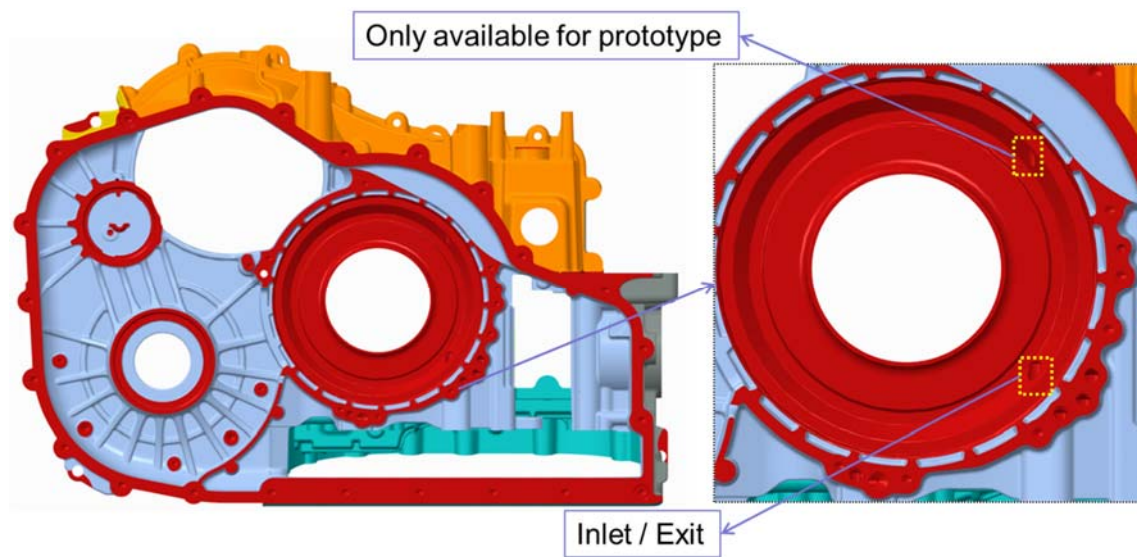


Figure 120 – Reverse piston housing (courtesy of Punch Powertrain NV)

In Figure 121 it is explicit that the hole comes from the bottom area of the transmission. On this area, it is located the hydraulic control system, so it is conclusive that the oil which reaches the reverse system is sent directly from the hydraulic unit.

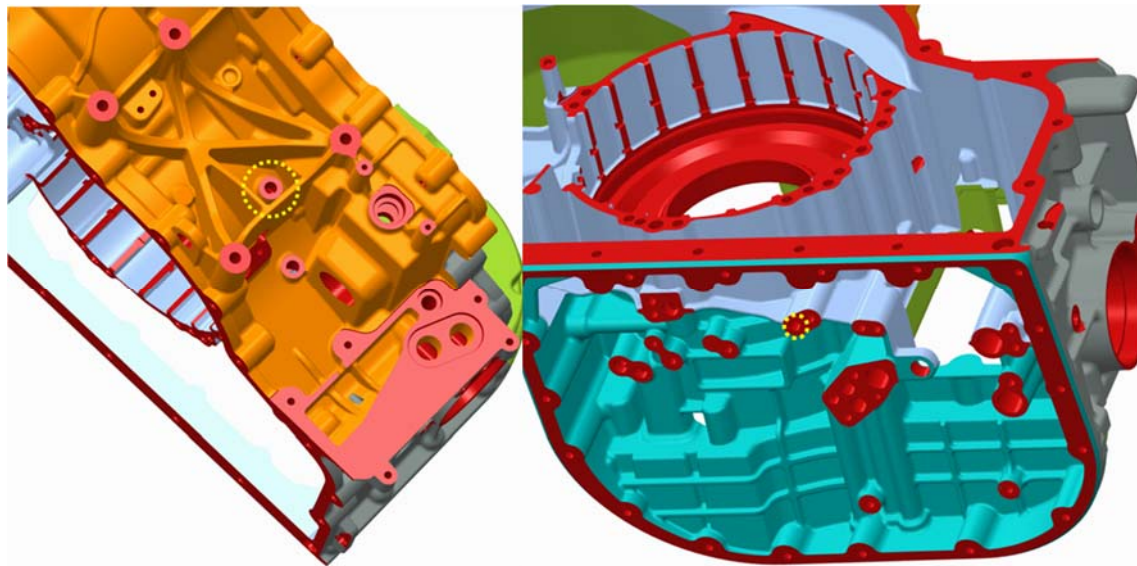


Figure 121 - Perspective view of the inlets/exits of transmission housing: channel available only for prototype (a) and serial channel (b) (courtesy of Punch Powertrain NV)

After reaching the volume created between reverse piston and the transmission housing the hole spreads all over this chamber according to Figure 122.



Figure 122 – Oil path inside reverse chamber
(courtesy of Punch Powertrain NV)

When the chamber is full of oil the pressure starts to be created. This pressure will be applied in reverse piston (Figure 123) which will force it to move axially.

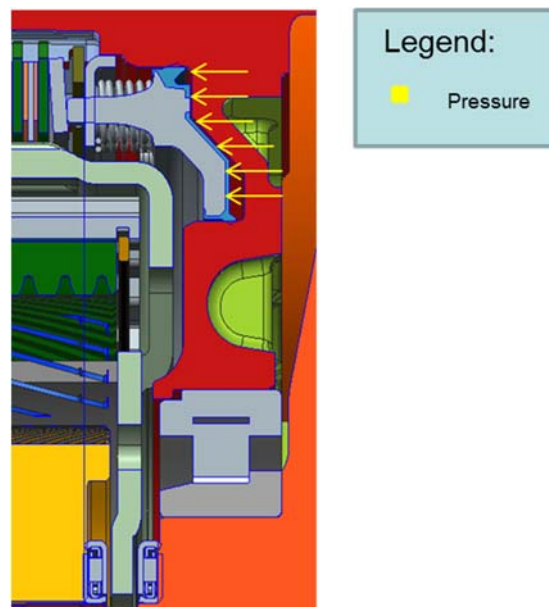


Figure 123 – Illustration of the pressure inside reverse chamber (courtesy of Punch Powertrain NV)

In the reverse clutch system, there is always air inside the chamber. As previously explained contrarily to the forward system in reverse the housing is steady (CVT transmission housing) so the same principle (used for forward) cannot be used to push

the air out. This means that there is always some air present in the reverse chamber which will originate some instability. A solution for this problem would be to use the exit for prototype measuring to push the air out. Another solution would be to create an inlet for oil on the upper part of the chamber which will force the air to exit through a channel on the lower section of the chamber.

3.3.5 Kinematics of clutches

At this moment, it was already approached the oil flow in forward and reverse clutch groups and the rotary kinematics of all the components. We are now in conditions to study the clutches' kinematics.

3.3.5.1 FWD clutch

When the pressure in the chamber between the forward piston and the clutch carrier produces a force greater than the springs pack load plus the wavy washer load, the piston starts to move axially. After that starts, a chain: the wavy washer will press separator plate which will, press one clutch plate and so on until the last clutch plate is pressed against the endplate of the forward clutch. When the clutch plates are pressed they get steady relative to the clutch carrier. This means that the clutch plates are rotating with the clutch carrier. This rotational motion will be passed to the sun gear. What comes next was already explained in the paragraph DNR schematics.

3.3.5.2 REV clutch

The working procedure of the reverse clutch is similar to the forward although there isn't rotational motion. When the oil reaches the chamber between the reverse piston and the transmission housing a pressure is built. This pressure will press the reverse piston in order to move it axially. Like in the forward clutch, the pressure on reverse has to translate itself in a force greater than the springs load. In the reverse system instead of a wavy washer there is a Belleville washer. This one has to be compressed too otherwise there will be no contact in clutch plates. When the reverse piston fingers touch the Belleville washer and applies force on it, deforms it. This will create an area of contact between the washer and the reverse separator plate. The separator plate presses the clutch plate. The last clutch plate is compressed against the endplate. When all the components are pressed against each other, all of them are steady. In conclusion when the clutch plates are pressed the PGS cage keeps steady and is not able to rotate.

3.3.6 Gears

In this sub-chapter, it will be presented a resume of what happens when the different gears are selected mainly on the point of view of the clutches and PGS.

3.3.6.1 Drive gear

When D gear is selected, FWD clutch is engaged. This means that the FWD clutch plates get closer, touching each other. A chamber between the CC and the FWD piston is created and the FWD spring pack and wavy washer are compressed. Due to the coefficient of friction, torque will be transmitted between clutch plates and then to the sun gear through the teeth. Clutch plates, sun gear and CC rotate solidary. As seen before, at this point REV clutch is disengaged. Sun gear is the input of the PGS as so the ring gear. The carrier rotated freely. The sun gear is connected to the primary shaft. Figure 124 is an auxiliary to this explanation.

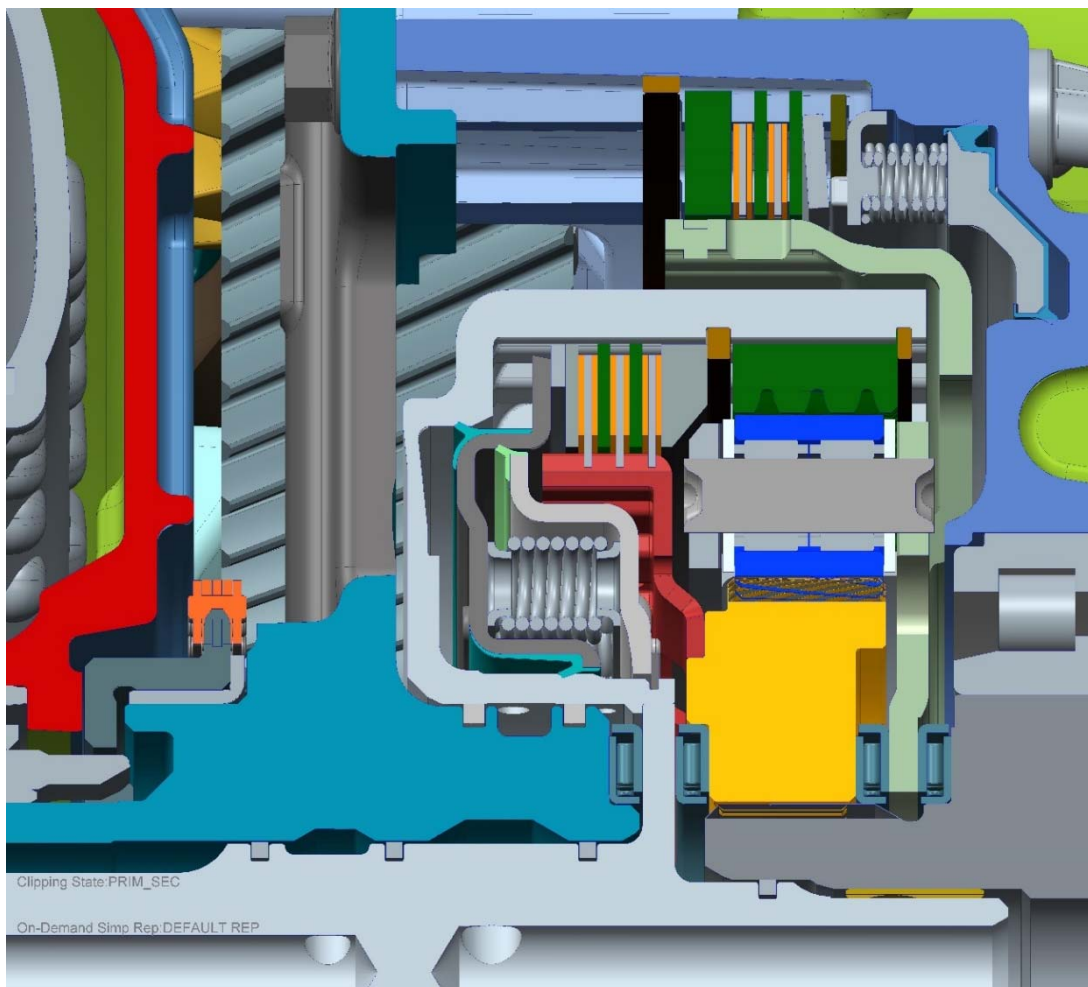


Figure 124 – Drive gear (courtesy of Punch Powertrain NV)

3.3.6.2 *Neutral gear*

When N gear is selected, FWD and REV clutch are disengaged. This means that there is no transmissible torque. Ring gear is the input of the PGS since it is connected to the CC. Sun gear can also work as an input because if the car starts moving the variator and consequently the primary shaft will transmit that motion to the sun gear. In N the engine is rotating and the possible motion from the variator cannot change the motion of the engine which means that the motion which comes from the variator is absorbed by the PGS. When the car is in N mode usually, the vehicle is steady. However, it is possible that when the transmission is N, the car moves forwards or backwards for small operations. Figure 125 is an auxiliary to this explanation.

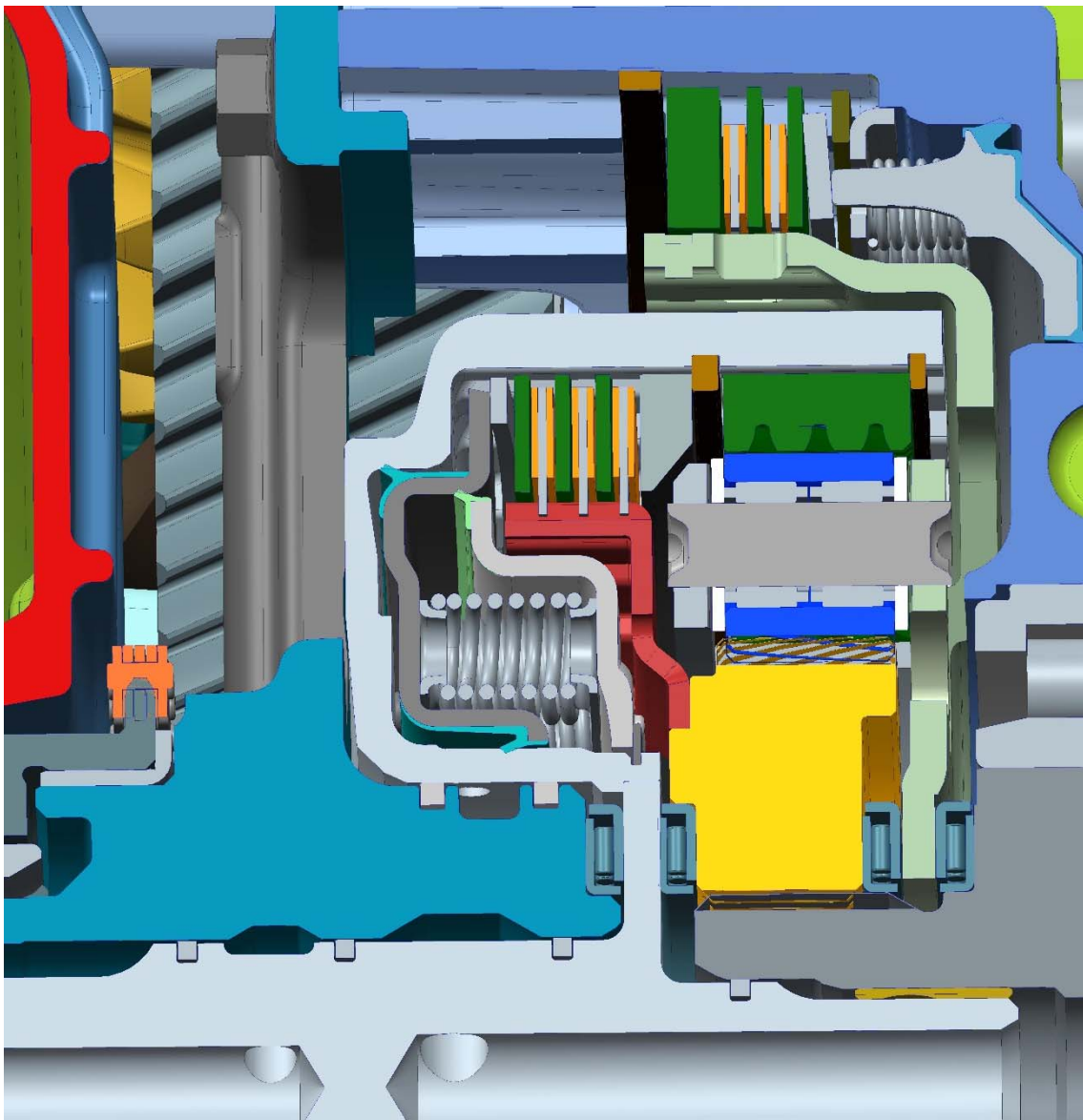


Figure 125 – Neutral gear (courtesy of Punch Powertrain NV)

3.3.6.3 *REV gear*

When R gear is selected, REV clutch is engaged. This means that the REV clutch plates get closer, touching each other. A chamber between the transmission housing and the REV piston is created and the REV spring pack and disc spring are compressed. Due to the coefficient of friction, torque will be transmitted between clutch plates and then to the PGS carrier through the teeth. As REV clutch endplates are connected to the transmission housing, when REV clutch is engaged there will be no motion which means that the carrier will be steady. The carrier is steady, ring gear receives the motion from the CC and the sun gear transmits it to the variator. The sun gear is connected to the primary shaft. Figure 126 is an auxiliary to this explanation.

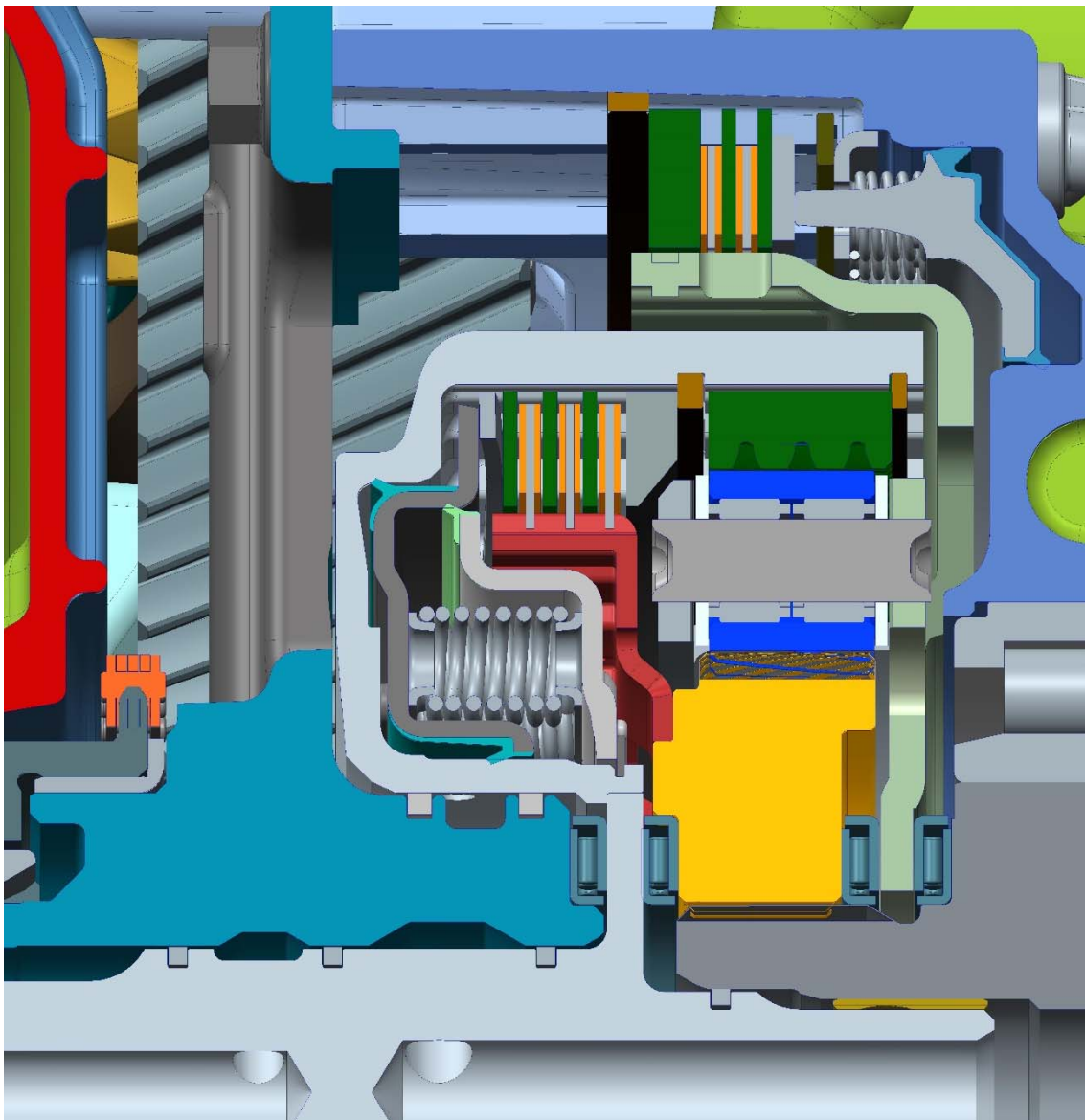


Figure 126 – Reverse gear (courtesy of Punch Powertrain NV)

3.3.7 Thrust bearings orientation

Thrust bearings or Axial needle roller bearings play an important role in the DNR subsystem. There are four thrust bearings in the DNR subsystem. As illustrated in Figure 127, similar to other types of bearings there are an inner and an outer ring. This means that the face of the left side corresponds to the inner ring and the face of the right side corresponds to the outer ring. Both rings can rotate independently (and that is the aim of a bearing).

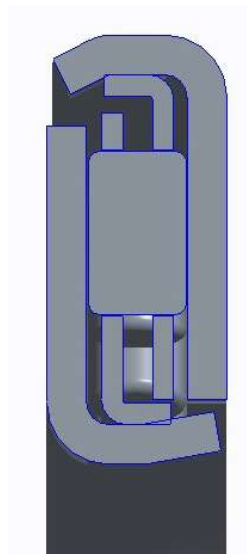


Figure 127 – Thrust bearings' cross section

In Figure 128, it is presented a cross section of the CVT transmissions where it is possible to see the location and the orientation of the bearings. In some cases, the orientation of the bearings is arbitrary, but not in this case. The incorrect positioning of the bearing leads to the destruction of it. All the bearings are numbered to make easier to identify them (Figure 128).

Regarding to the bearing number 1, it is visible that it is assembled between the stator plate (left blue part) and the turbine shaft assy. (right grey part). The stator plate is steady but not the turbine shaft. This means that the same ring of the bearing cannot touch both components at the same time. One face and correspondent ring need to face only one component. The correct positioning is the inner ring supported on the stator plate and the outer ring face touching the turbine shaft. If the bearing was inversely assembled (as for example bearing nr. 4) the inner ring flange would touch at the same time the stator plate (ring) and the turbine shaft (face). This would lead to the scratching and friction of the flange.

Bearing number 2 is assembled exactly the same way bearing number 1. On the left of the bearing is the turbine shaft (grey part) and on the right, is the sun gear (yellow part). Both components rotate but they can rotate in opposite directions and at different speeds which means that one ring of the bearing is for the turbine shaft and the other to the sun gear. The only way to guarantee this, is to orient the flange of the inner ring to the left and consequently the flange of the outer ring to the outer, according to Figure 128. If the bearing was assembled on a different orientation (as bearing number 4) the outer ring flange would touch at the same time the sun gear and the turbine shaft.

Once again, bearing number 3 orientations is assembled with the same orientation of bearing number 1. On the left of the bearing is the sun gear (yellow part) and on the right, is the PGS carrier (green part). Inside the inner ring of the bearing is the primary shaft (orange part). Sun gear and PGS carrier are able to rotate but can rotate at different speeds and opposite directions. Sun gear and primary shaft rotate solidary. The correct position of the bearing is the inner ring flange to the left and the outer ring flange to the right. If it was assembled on the contrary orientation (as bearing number 4) the inner ring flange would touch at the same time the primary shaft and the PGS carrier that are allowed to rotate at different speeds and directions.

As already realized, bearing number 4 is the only one which is oriented on a different way. In this case the outer flange is to the left side and the inner ring flange to the right side. On the left is the PGS carrier (green part) and on the right, is the primary shaft (orange part). If the bearing was assembled on a different orientation (as the other bearings) the inner ring flange with touch at the same the time the carrier and the primary shaft.

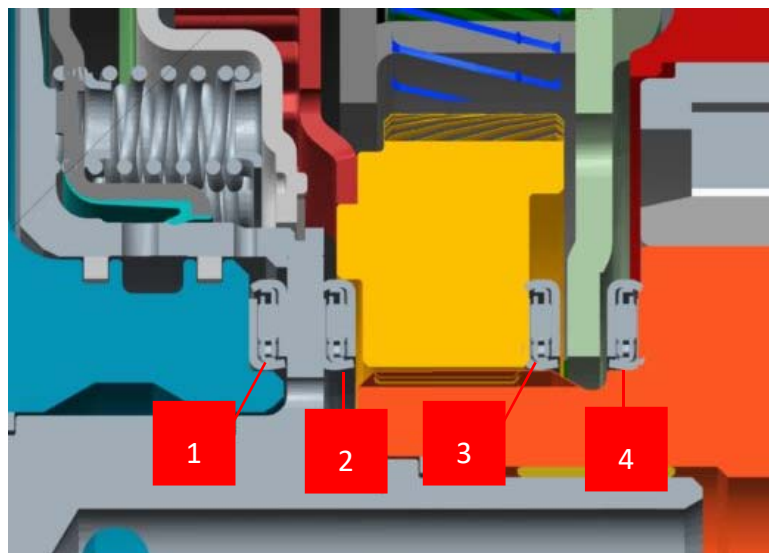


Figure 128 - Thrust bearings correct orientation

3.4 Calculations data

All the data that will be used on the calculations and any other explanations regarding to the DNR components and materials' properties are presented on chapter 6.33.

3.5 Analytical Calculations

3.5.1 Calculation scenarios

3.5.1.1 FWD clutch set

In the calculations related to the FWD clutch set, it was considered four different scenarios:

Table 14 - Calculation scenarios for FWD clutch set

Scenario nr.	Description
1	Nominal scenario
2	Worst scenario in terms of transmissible torque (dimensions' tolerances)
3	Best scenario in terms of transmissible torque (all parameters)
4	Worst scenario in terms of transmissible torque (all parameters)

The second scenario takes in account the tolerances of the dimensions. This means that the final dimensions have a negative effect on the obtained results of the transmissible torque. By this way it is possible to verify what is the influence of the dimensions' tolerances. The third and fourth scenarios have in account all the factors that influence positive or negatively the transmissible torque. The aim of the simulation of all these distinct scenarios is also to verify that even on the worst scenario all the requirements are fulfilled.

3.5.1.2 REV clutch set

In the calculations related to the REV clutch set, it was considered three different scenarios:

Table 15 – Calculations' scenarios for REV clutch set

Scenario nr.	Description
1	Nominal scenario
3	Best scenario in terms of transmissible torque (all parameters)
4	Worst scenario in terms of transmissible torque (all parameters)

The third and fourth scenarios have in account all the tolerances that influence positive or negatively the transmissible torque, not only in terms of dimensions but also in terms of loads and so on. As expected, in the nominal scenario, nominal values are considered.

3.5.2 Springs

3.5.2.1 FWD spring pack

In this chapter, all the calculation related to the FWD spring pack will be presented and discussed.

3.5.2.1.1 General calculations

It is known that the stiffness of the spring pack and all the other springs will be necessary for several calculations. So, the first step to be done is the calculation of the stiffness of all the springs. If this value is already provided by the supplier, it will be verified according to the springs properties.

Firstly, it is calculated the average load at maximum and minimum working height. The values of $F_{\max(h_{\max})}$ and $F_{\min(h_{\max})}$ (Table 138) will be used to obtain the value of the average load at maximum working height. The values of $F_{\max(h_{\min})}$ and $F_{\min(h_{\min})}$ (Table 138) will be used to obtain the value of the average load at minimum working height.

Then, it is necessary to calculate the value of the maximum compression. This is the maximum displacement that can be induced on the springs in a free state. Maximum compression can be calculated, as follows:

$$d_{\max} = h_{\max} - h_{\min} [mm] \quad \{3.1\}$$

It is important to see that the value of $h_{\max}$ is equal to the value of the max. installed height.

Now, all the necessary values to calculate the spring pack stiffness have been obtained.

The stiffness must be calculated according to eq. {2.19}. The value of spring index is an important parameter as it will use forward on other calculations, and can be calculated according to eq. {2.9}. The value of the mean diameter is given by eq. {2.5}. All the values calculated on this chapter regarding to the nominal scenario are expressed on Table 16.

Table 16 - Results of general calculations for FWD spring pack

Variable	Unit	Value
$F_{h\max}$	N	929.200
$F_{h\min}$	N	1660.050
$d_{\max}$	mm	3.340
SR	N/mm	218.817
C	---	6.265
D_m	mm	7.718

Scenarios

In this chapter, it was also calculated the spring rate according to the third and fourth scenarios. These values will be crucial in the calculation of the transmissible torque for FWD clutch set. The results are expressed on Table 17 and Table 18.

In order to calculate the spring rate for the 3rd scenario it was considered the minimum loads at both maximum and minimum height from Table 138.

Table 17 – Results of general calculations for FWD spring pack (3rd scenario)

Variable	Unit	Value
$F_{h\max}$	N	808.8
$F_{h\min}$	N	1504.6
SR	N/mm	208.323

On the other hand, to calculate the spring rate for the 4th scenario it was considered the maximum loads at maximum and minimum working heights from Table 138.

Table 18 - Results of general calculations for FWD spring pack (4th scenario)

Variable	Unit	Value
F_{hmax}	N	1049.6
F_{hmin}	N	1815.5
SR	N/mm	229.311

3.5.2.1.2 Suppliers verification

The value of the spring rate of the compression springs of the FWD spring pack was provided by the supplier. At this stage, this value is being verified, as follows:

$$SR = \frac{SR^{FWD\ SP}}{N} [N / mm] \quad \{3.2\}$$

To obtain the value of SR for each compression spring, it must be used the values of the stiffness of the FWD spring pack (Table 16) and the number of compression springs of the spring pack (Table 138).

The obtained value was 8.416 N/mm and the value provided by the supplier was 8.42 N/mm (Table 138).

The value of the coils stiffness can also be calculated based on eq. {2.17}. The error was calculated for both methods, comparing always to the theoretical value (provided by the supplier). The number of active coils was calculated according to eq. {2.18} as also the respective error. All the results are expressed on Table 19.

Table 19 – FWD spring pack supplier's verification

Variable	Unit	Value
SR	N/mm	8.416
Error	%	0.047
SR (acc. To eq. {2.17})	N/mm	8.493
Error	%	0.871
N_A	---	5.951
Error	%	0.871

Based on the value of the calculated errors, which are acceptable because they are low, it is conclusive that the suppliers' reference values can be validated.

3.5.2.1.3 Pre-load conditions

Pre-load conditions are the necessary conditions to start compressing the spring pack.

On the following calculations, it was considered a rotating speed of 6500 rpm. Firstly, it is necessary to calculate the resulting centrifugal force corresponding to the chosen rotating speed as calculated on section 0. After that, maximum allowable hydraulic force applied on the springs is calculated:

$$F_H^{\max} = F_{\min(h \max)} - F_C [N] \quad \{3.3\}$$

It should be considered the minimum load at maximum working height of the spring pack (808.800 N) in order to simulate the worst scenario due to the fact that pre-load is quite an important point to have in account in terms of safety. For example, if the maximum load at maximum working height had been considered instead of the minimum load, for a spring with lower spring rate or lower load at maximum working height, the vehicle would be able to start moving even in Neutral gear. This could be originated by the driver pressing the accelerator, which would increase the centrifugal force and would force the spring pack to be compressed.

The following step is about calculating the pressure which corresponds to the maximum allowable hydraulic force considering the hydraulic area of the piston.

The results for nominal and second scenario are presented on the following sections.

On Figure 129, it is illustrated the relation between rotating speed and necessary pressure to start compressing the spring pack. In the same graph, it is plotted the relation for nominal and second scenario.

It is possible to see that there is not any significant difference between both scenarios. As expected, when the value of rotating speed rises, the value of pressure decreases. This was expected because of the centrifugal force which increases its value when the rotating speed rises.

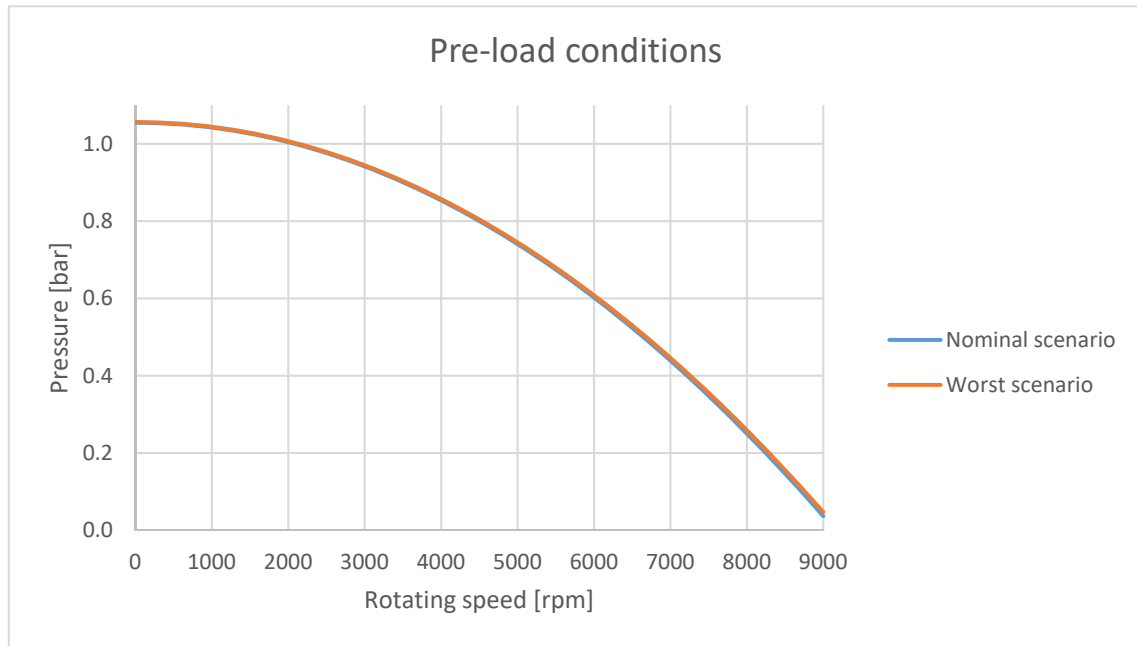


Figure 129 – Graph of pre-load conditions for FWD clutch

Nominal scenario

The results obtained for the nominal scenario are presented on the table below.

Table 20 – Results of pre-load conditions calculations for nominal scenario of FWD clutch

Variable	Unit	Value
F_c	N	407.063
$F_h^{\max}$	N	401.737
$p_{\max}$	bar	0.524

Second scenario

The results obtained for the second scenario are presented on the table below.

Table 21 – Results of pre-load conditions calculations for second scenario of FWD clutch

Variable	Unit	Value
F_c	N	402.906
$F_h^{\max}$	N	405.894
$p_{\max}$	bar	0.530

Third scenario

The results obtained for the third scenario are presented on the table below.

Table 22 - Results of pre-load conditions calculations for third scenario of FWD clutch

Variable	Unit	Value
F_{hmax}	N	808.800
F_c	N	422.743
F_h^{max}	N	386.057
p_{max}	bar	0.502

Fourth scenario

The results obtained for the fourth scenario are presented on the table below.

Table 23 - Results of pre-load conditions calculations for fourth scenario of FWD clutch

Variable	Unit	Value
F_{hmax}	N	1049.600
F_c	N	389.786
F_h^{max}	N	659.814
p_{max}	bar	0.862

3.5.2.1.4 Fatigue

Regarding to the fatigue calculations, firstly, it is calculated the value of F_a and F_m according to eq. {2.23} and {2.24}, respectively.

F_{max} and F_{min} were considered to be the average load at h_{min} and h_{max} , respectively.

Amplitude force per spring can be obtained as follows:

$$F_a^{perspring} = \frac{F_a}{N} [N] \quad \{3.4\}$$

Midrange force can be calculated as follows:

$$F_m^{perspring} = \frac{F_m}{N} [N] \quad \{3.5\}$$

The values of τ_a and τ_m may be calculated according to eq. {2.25} and {2.27}, respectively. The values of K and K_B can be obtained based on eq. {2.12} and {2.26}, respectively. The values of D_m and d_c can be found on Table 16 and Table 136, respectively. S_{su} value is given by eq. {2.22}. After that, it is necessary to calculate the value of the slope line according to eq. {2.28}. The following step is about to calculate S_{se} according to Gerber and Goodman and respective S_{sa} values. The equations for these calculations are from {2.29} to {2.32}.

It is important to underline the fact that the values of S_{sa} according to Zimmerli considered for these calculations are for unpeened springs. This is because the springs applied on the DNR subsystem are not peened.

Finally, it is calculated the value of the safety factor according to the three different criteria based on eq. {2.33}.

The results for fatigue's calculations are expressed on Table 24.

Table 24 - Results of fatigue calculations of FWD spring pack

Variable	Unit	Value
F_a	N	365.425
$F_a^{per spring}$	N	14.055
F_m	N	1294.625
$F_m^{per spring}$	N	49.793
ID	mm	6.486
K	---	1.241
K_B	---	1.227
τ_a	MPa	183.265
τ_m	MPa	649.271
S_{su}	MPa	1132.300
r	---	0.282
S_{se}^{Gerber}	MPa	271.407
S_{sa}^{Gerber}	MPa	182.710
$S_{se}^{Goodman}$	MPa	362.252
$S_{sa}^{Goodman}$	MPa	169.798
S_{sa}^{Sines}	MPa	241.000

SF_f^{Gerber}	---	0.997
SF_f^{Goodman}	---	0.927
SF_f^{Sines}	---	1.315

It is desirable to obtain a value of SF (safety factor) greater than 1. This is not verified for the SF according to Gerber and Goodman. However at least in the case of Gerber the SF is much close to 1. Both criteria are more conservative that Sines. According to Sines, the SF is considerably greater than one. Having all this is account, the FWD SP (spring pack) in terms of fatigue can be validated.

3.5.2.2 Wavy washer

In this chapter, it will be presented all the results and calculations regarding to the wavy washer. The main purpose of the wavy washer in this particular application is to smoothly transmit the torque to clutch plates.

3.5.2.2.1 General calculations

As seen before in order to calculate the spring rate it is necessary to know the deflection and the loads applied. The maximum deflection must be calculated based on the following equation:

$$d_{\max} = FL - b[mm] \quad \{3.6\}$$

The value of mean diameter will be crucial for further calculations and should be calculated as follows:

$$D_m = \frac{OD + ID}{2}[mm] \quad \{3.7\}$$

The radial wall can be calculated as follows:

$$b_w = \frac{OD - ID}{2}[mm] \quad \{3.8\}$$

The input data for the equations presented above can be found on Table 139.

The results are expressed on Table 25.

Table 25 - Results of general calculations of wavy washer

Variable	Unit	Value
$d_{\max}$	mm	0.8
D_m	mm	131.0
b_w	mm	4.4

3.5.2.2.2 Spring stiffness

Nominal scenario

The value of spring stiffness was not provided by the supplier. So, this value will not be verified but calculated. The value of SR must be calculated according to eq. {2.19}. The value of $F_{h\max}$ is expressed on Table 139 and $d_{\max}$ was previously calculated. The obtained values are expressed on Table 26.

Table 26 – Wavy washer supplier's verification

Variable	Unit	Value
SR	N/mm	1000.0

Other Scenarios

In this chapter, it was also calculated the spring rate according to the third and fourth scenarios. These values will be crucial in the calculation of the transmissible torque for FWD clutch set. The main differences between both scenarios is the load at maximum working height. In the case of the fourth scenario it is considerably higher comparing to the 3rd scenario. All the other parameters presented on Table 27 and

Table 28 are not so distinct.

Table 27 - Results of general calculations of wavy washer (3rd scenario)

Variable	Value	Unit
FL	mm	2.45
b	mm	1.6
d _{max}	mm	0.85
F _{hmax}	N	510
SR	N/mm	728.571

Table 28 - Results of general calculations of wavy washer (4th scenario)

Variable	Value	Unit
FL	mm	2.25
b	mm	1.50
d _{max}	mm	0.75
F _{hmax}	N	690.00
SR	N/mm	1380.00

It is evident that the difference in terms of SR between all the scenarios is quite significant.

3.5.2.2.3 Fatigue

On this chapter, it will be presented the results of fatigue analysis of the wavy washer.

First of all, it was calculated the load at lower and upper working height. It is known that the load at upper working height is zero because it corresponds to the free length of the spring. The lower working height was considered to be 1.75 mm. After that, operating stresses are calculated for both working heights. Finally, it is calculated the fatigue stress ratio.

All the results are presented on Table 29.

Table 29 - Results of fatigue calculations of the wavy spring

Variable	Unit	Value
F ₁	N	600.000
F ₂	N	0
σ ₁	MPa	486.648
σ ₂	MPa	0
X	---	0.676

Based on Table 7 and on the results of Table 29, fatigue stress ratio translates itself as an estimated life cycle between 100000 to 200000 cycles for a static load.

For a dynamic service, the highest value of stress corresponds to approximately 32% of the tensile strength of the material. This value translates itself in more than 10^7 life cycle.

Due to the fact that the wavy washer will work under dynamic conditions, the respective values should be considered for the validation of the wavy washer. Furthermore, it is applied a heat treatment to the wavy springs which will improve the behaviour of the spring as also its life cycle.

3.5.2.3 REV spring pack

In this chapter, it will be discussed and presented all the results and calculations regarding to the REV spring pack.

3.5.2.3.1 General calculations

The calculation method used for REV spring pack is the same used for FWD spring pack.

The necessary data to calculate the average load at maximum and minimum working height is expressed in Table 141. In the same table, it is expressed the values of maximum and minimum load at maximum and minimum working height. The results are presented in Table 30.

Table 30 – Results of general calculations for REV spring pack (nominal scenario)

Variable	Unit	Value
$F_{h \max}$	N	528.250
$F_{h \min}$	N	854.500
$d_{\max}$	mm	2.800
SR	N/mm	116.518
C	---	8.843
D_m	mm	7.861

The results for second and third scenarios are presented on Table 31 and Table 32.

Table 31 - Results of general calculations for REV spring pack (second scenario)

Variable	Unit	Value
$F_{h \max}$	N	467.900
$F_{h \min}$	N	774.600
$d_{\max}$	mm	2.800
SR	N/mm	109.536

Table 32 - Results of general calculations for REV spring pack (third scenario)

Variable	Unit	Value
$F_{h \max}$	N	588.600
$F_{h \min}$	N	934.400
$d_{\max}$	mm	2.800
SR	N/mm	123.500

3.5.2.3.2 Suppliers verification

Similar to FWD spring pack, the spring rate of the compression springs was provided by the supplier. This value was verified. The values of stiffness of the REV spring pack and the number of compression springs are expressed in Table 30.

The results for the different scenarios are presented on Table 33, Table 34 and

Table 35.

The reference value of the supplier is 3.640 N/mm.

Table 33 – REV SP suppliers' verification (nominal scenario)

Variable	Unit	Value
SR	N/mm	3.641
Error	%	0.03

Table 34 - REV SP suppliers' verification (second scenario)

Variable	Unit	Value
SR	N/mm	3.423
Error	%	5.960

Table 35 - REV SP suppliers' verification (third scenario)

Variable	Unit	Value
SR	N/mm	3.859
Error	%	6.030

This means that the value of the supplier can be validated because the error results are relatively low.

3.5.2.3.3 Preload conditions

The method to calculate the pre-load conditions regarding to the REV SP is different from the one used for FWD spring pack. In this case centrifugal force is not taken in account since it is null in the REV clutch set because there isn't any kind of rotational motion.

In Figure 130, it is presented a graph illustrating the evolution of the hydraulic force with the increase of the hydraulic pressure. The horizontal lines correspond to the necessary load to be applied on the SP to start deforming it for the different scenarios. By using this graph, it is very easy to know what hydraulic force corresponds to a certain value of pressure and also at which pressure, will the SP start being compressed. For the calculation of the hydraulic force of each scenario it was considered the following values of hydraulic area:

Table 36 – Hydraulic area of each scenario for pre-load calculations of REV clutch set

Variable	Unit	Value
A_H (nominal scenario)	mm^2	10687.698
A_H (second scenario)	mm^2	10723.324
A_H (third scenario)	mm^2	10652.073

For the nominal scenario it was considered the nominal dimensions of the transmission housing while for the other scenarios it was considered the tolerances.

By analysing the graph of Figure 130, it is conclusive that there isn't any significant difference between the lines of the distinct scenarios.

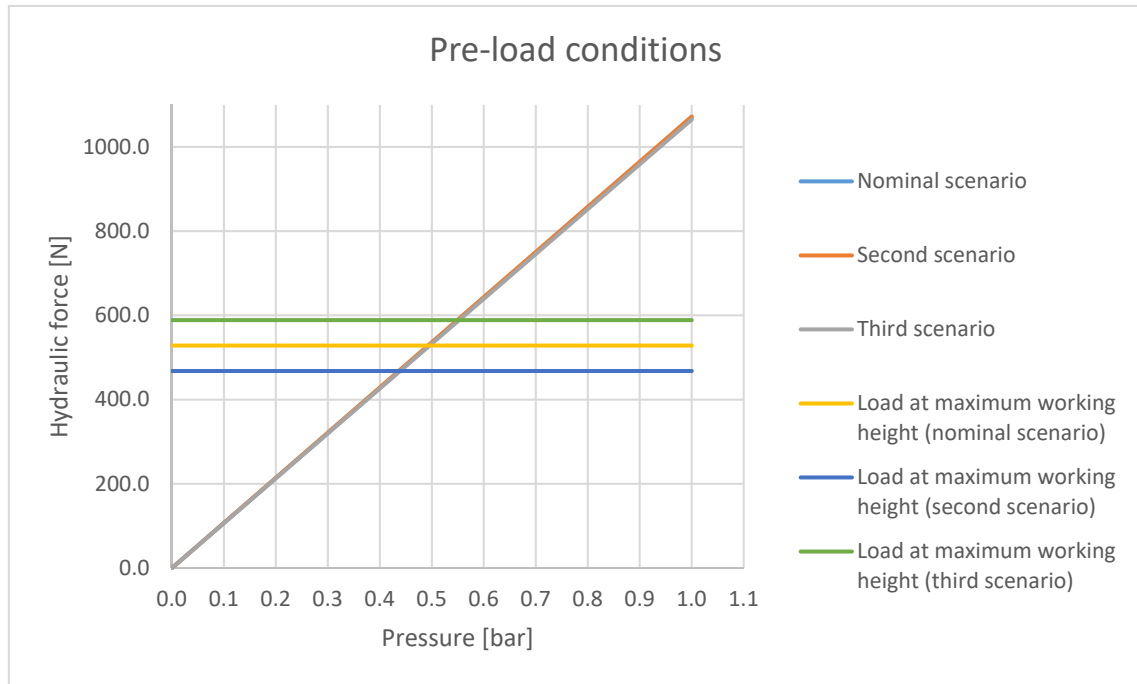


Figure 130 – Graph of pre-load conditions of REV clutch set

Nominal scenario

Table 37 - Results of pre-load conditions calculations for nominal scenario of REV clutch

Variable	Unit	Value
$F_h^{\max}$	N	528.25
$p_{\max}$	bar	0.494

Second scenario

Table 38 - Results of pre-load conditions calculations for second scenario of REV clutch

Variable	Unit	Value
$F_h^{\max}$	N	467.9
$p_{\max}$	bar	0.436

Third scenario

Table 39 - Results of pre-load conditions calculations for third scenario of REV clutch

Variable	Unit	Value
$F_{h\max}$	N	588.6
$p_{\max}$	bar	0.553

3.5.2.3.4 Fatigue

The calculations' method used for REV spring pack in terms of fatigue is exactly the same one used for the FWD spring pack calculations on chapter 0.

Results for fatigue calculations of REV spring pack are presented on the following table.

Table 40 - Results of fatigue calculations of REV spring pack

Variable	Unit	Value
F_a	N	163.125
$F_{a\text{ per spring}}$	N	5.098
F_m	N	691.375
$F_{m\text{ per spring}}$	N	21.605
ID	mm	6.972
K	---	1.165
K_B	---	1.154
τ_a	MPa	169.230
τ_m	MPa	717.250
S_{su}	MPa	1132.300
r	---	0.236
S_{se}^{Gerber}	MPa	271.407
S_{sa}^{Gerber}	MPa	166.275
S_{se}^{Goodman}	MPa	362.252
S_{sa}^{Goodman}	MPa	153.761
S_{sa}^{Sines}	MPa	241.000
SF_f^{Gerber}	---	0.983
SF_f^{Goodman}	---	0.909
SF_f^{Sines}	---	1.424

The values of SF must be greater than 1. Although only the SF according to Sines is according to this besides the values are close to 1. It is important to have in mind that these springs are submitted to thermal treatments (heat treatments) which improve the properties of the metal. This improvement is not taken in account in the calculations. Furthermore, the theorems of Gerber and Goodman are significantly more conservative than Sines'. In conclusion, in terms of fatigue, REV spring pack can be validated.

3.5.2.4 Belleville washer

Some factors of the Belleville washer or as also known, disc spring, will be presented on this chapter. The main purpose of the Belleville washer in this particular application is to smoothly transmit the torque to clutch plates.

In these calculations three different scenarios was considered. The first scenario takes in account all the nominal values of the different parameters. The main difference between second and third scenario is related to the load at working height of 1.75 mm. For the calculations of the second scenario it was considered the minimum load while for the third scenario it was considered the maximum one.

3.5.2.4.1 General calculations

Firstly, it is necessary to calculate the maximum compression according to eq. {3.6}.

The necessary values for this equation are expressed on Table 142.

The value obtained is expressed on the table below.

Table 41 – General calculations' results of Belleville washer

Variable	Unit	Value
$d_{\max}$	mm	0.750

The value of the spring stiffness was not provided by the supplier. So, it needs to be calculated (eq. {2.19}). All the necessary values to calculate the spring stiffness are present on Table 142. On Table 42 it is expressed the values of load assumed for the calculation of the spring rate for 2nd and 3rd scenarios.

Table 42 – Input data for the calculations of the spring stiffness of the disc spring (2nd and 3rd scenarios)

Variable	Unit	Value
F (2 nd scenario)	N	630.080
F (3 rd scenario)	N	801.920

The final results of the spring stiffness for the three scenarios can be consulted on Table 43.

Table 43 – General calculations' results of the disc spring

Variable	Unit	Value
SR (nominal case)	N/mm	1432.000
SR (2 nd scenario)	N/mm	1260.160
SR (3 rd scenario)	N/mm	1603.840

3.5.2.4.2 Fatigue

All the necessary equations and graphs for the calculation of fatigue of the disc springs are available on chapter 2.2.1.5.2.

Firstly, it was calculated the value of δ with the data from Table 142. Then, all the coefficients were calculated. Finally, all the stresses were calculated.

All the results are presented on Table 44.

Table 44 - Results of fatigue calculations of disc spring

Variable	Unit	Value
δ	---	1.124
h_0	mm	0.750
K_1	---	0.199
K_2	---	0.993
K_3	---	1.013
K_4	---	1.000
σ_{OM}	MPa	-207.344
σ_I	MPa	-257.592
σ_{II}	MPa	182.355
σ_{III}	MPa	230.457
σ_{IV}	MPa	-160.870

The values of σ_o and σ_u correspond to maximum and minimum fatigue stress, respectively (DIN 2093, 1992). This means that points II and III are decisive to the determination of the cycle life of the spring (*Disc springs - theory and practice*, n.d.).

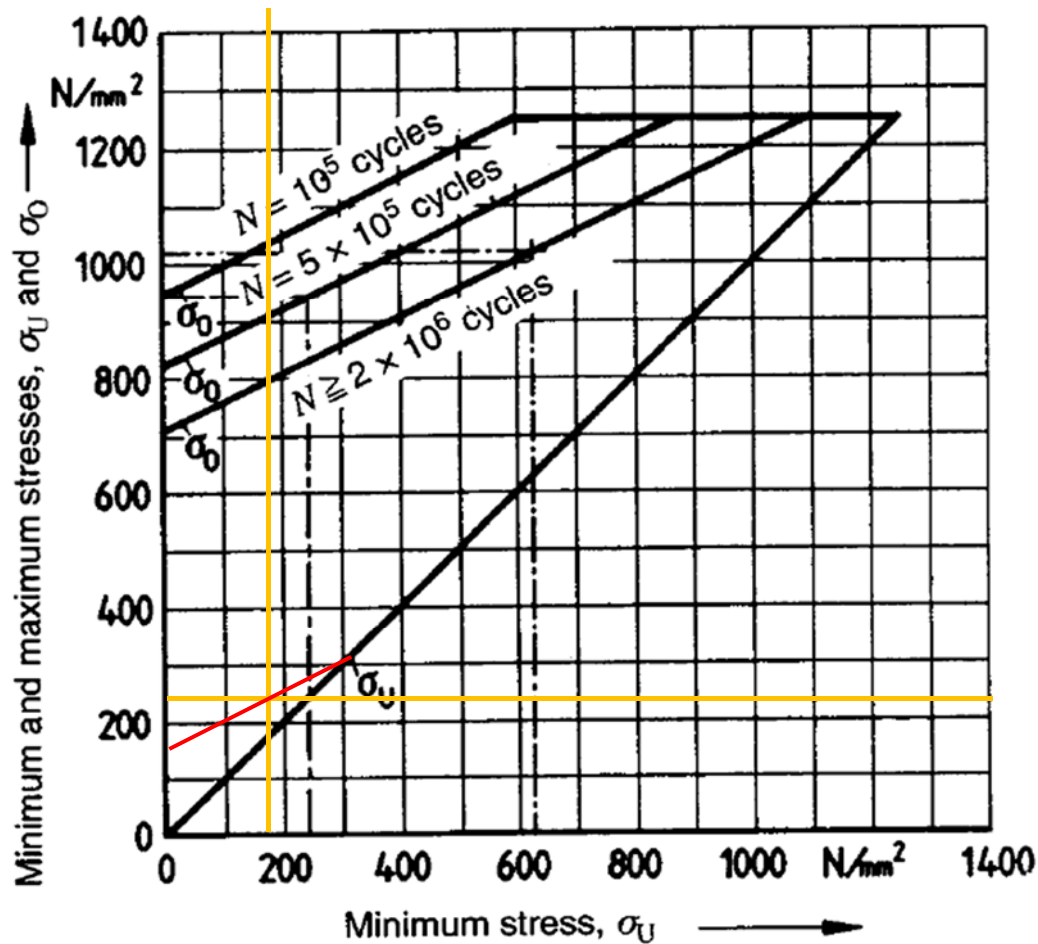


Figure 131 – Resulting graph for fatigue cycle life study of the disc spring (adapted from ("DIN 2093," 1992))

Based on the graph of Figure 131, it is expected an infinite life for the disc spring due to the location of the resulting line (red). The lower the line is located the greater the life cycle is expected to be which makes sense since the fatigue stresses applied on the disc spring are relatively low.

3.5.3 Transmission oil

On the datasheet provided by the supplier it was not expressed the value of dynamic viscosity of the oil. However, this value can be easily calculated:

$$\mu = \lambda \cdot \rho [cP] \quad \{3.9\}$$

This calculation was done for all the temperature values presented on Table 135. It was obtained the following table:

Table 45 - Dynamic viscosity of transmission oil

T [°C]	μ [cP]
-40	9900.800
-30	2432.060
-20	810.960
-10	334.276
0	161.680
10	87.962
20	52.322
30	33.512
40	22.823
50	16.268
60	12.030
70	9.232
80	7.250
90	5.844
100	4.810
110	4.023
120	3.419
130	2.948
140	2.566
150	2.263

After getting all the necessary values, it was important to understand the relation between all the oil properties and its temperature because the transmission oil will operate in a wide range of temperatures. By knowing the behaviour of the different properties at different temperatures it is possible to predict how the temperature will affect oil properties and all the other factors that depend on the oil properties. The ideal way to express the relation between the different properties and the temperature is with graphs.

In Figure 132, it is presented the relation between oil density and temperature based on Table 135.

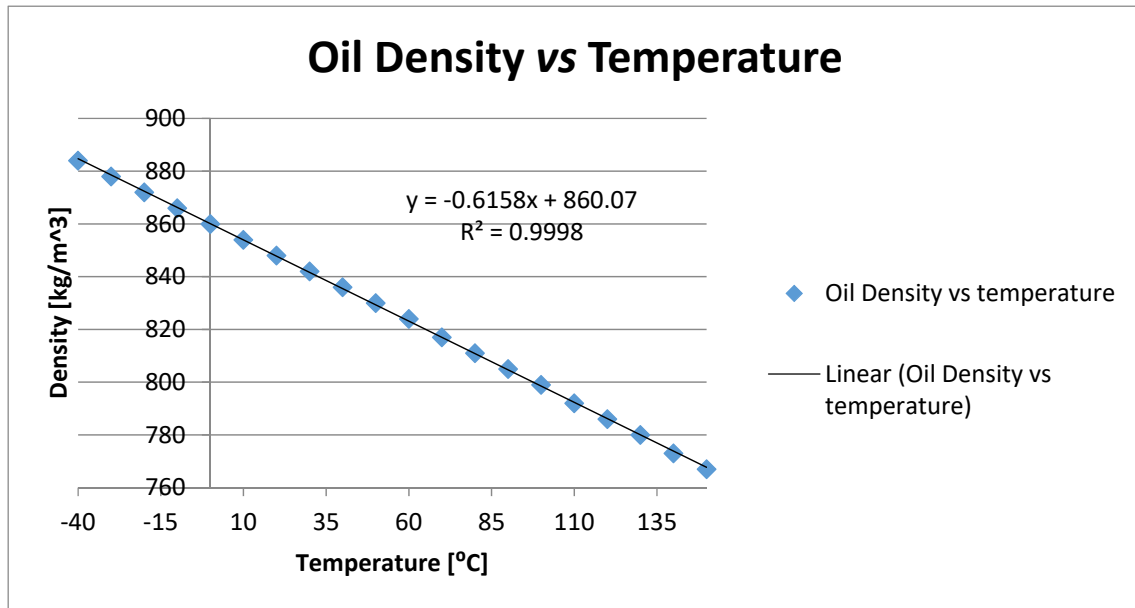


Figure 132 – Graph oil density vs temperature

Based on the above graph it is possible to conclude that the relation between density and temperature is linear. When temperature rises, the value of density decreases. It is presented on the graph the tendency line with a value of R^2 much close to 1.

In Figure 133, it is illustrated a graph relating oil kinematic viscosity and temperature.

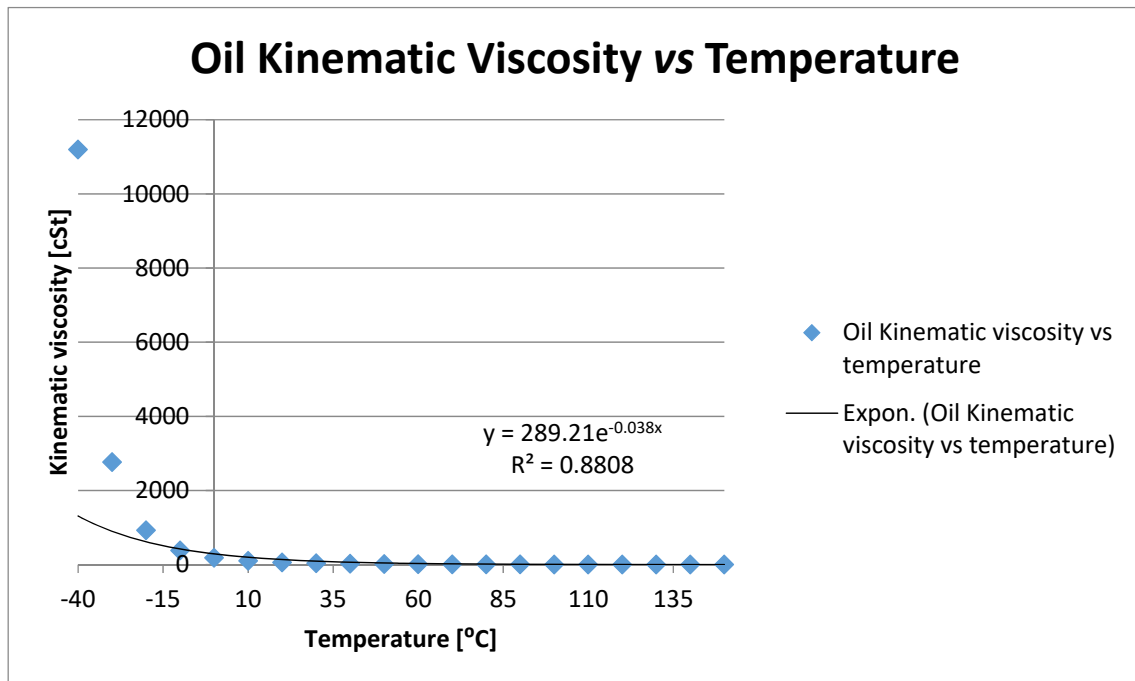


Figure 133 - Graph oil kinematic viscosity vs temperature

According to the graph of Figure 133, there is an exponential relation between oil kinematic viscosity and temperature. At low temperatures, from -40°C to approximately 0 °C, the relation is not well defined by the tendency line. Although, from 0 °C and up the tendency line defines well the relation. Due to the low accuracy at low temperatures, the value of R^2 for the tendency line is about 0.9 which is not so close to 1 as the previous tendency line. Despite this, it is possible to conclude that this is the best tendency line for this relation. This is not problematic because most of the times, transmission oil will operate on the positive temperature range.

Based on the graph of Figure 133 it is conclusive that the exponential trendline is not the most suitable one as long as it is not accurate at lower temperatures (approximately lower than -15 °C). In order to obtain a curve that better describes the trendline mainly in the range between 40 and 100 °C it was consulted the standard ASTM D341-03. According to this standard the best equation is the following ((ASTM International, 2003)):

$$\log \log(Z) = A - B \log(T) \quad \{3.10\}$$

Where A and B represent constants and are nondimensionalized; T corresponds to the temperature in Kelvin; $\log$ correspond to the logarithm to base 10 and Z is a constant and is calculated as follows (ASTM International, 2003):

$$Z = \nu + 0.7 \quad \{3.11\}$$

In which ν corresponds to the kinematic viscosity in cSt.

By substituting eq. {3.11} on eq. {3.10}, the following eq. is obtained:

$$Z = 10^{10(A - B \log(T)) - C} \quad \{3.12\}$$

The error is calculated as follows:

$$Error = |\log(\nu_C) - \log(\nu_D)| \quad \{3.13\}$$

In which ν_c corresponds to the calculated value of the kinematic viscosity (according to eq. {3.12}) in cSt and ν_D corresponds to the value of the kinematic viscosity provided on the datasheet of the oil in cSt.

The error was calculated for each temperature value. It is also calculated the sum of all the errors. By using the solver function of Excel®, an interaction process was done. The set objective is to minimize the value of the sum of the errors by changing the value of the constants A , B and C . It was selected the GRG nonlinear solving method.

It was obtained the following values for the constants A and B for limits at 40 and 100°C:

$$\begin{cases} A = 8.133 \\ B = 3.194 \\ C = 0.700 \end{cases} \quad \{3.14\}$$

The obtained value for the sum of the errors was 0.006 which is considerably little. The resulting equation is the following:

$$Z = 10^{10(8.133 - 3.194 \log(T)) - 0.700} \quad \{3.15\}$$

For all the calculations in which the value of the kinematic viscosity was required it was used the trendline presented on the graph of Figure 133 however the best approach would be to use eq. {3.15}.

The table with all the values is presented in the appendix on 6.33.

In Figure 134, it is presented a graph with the relation between oil dynamic viscosity and oil temperature.

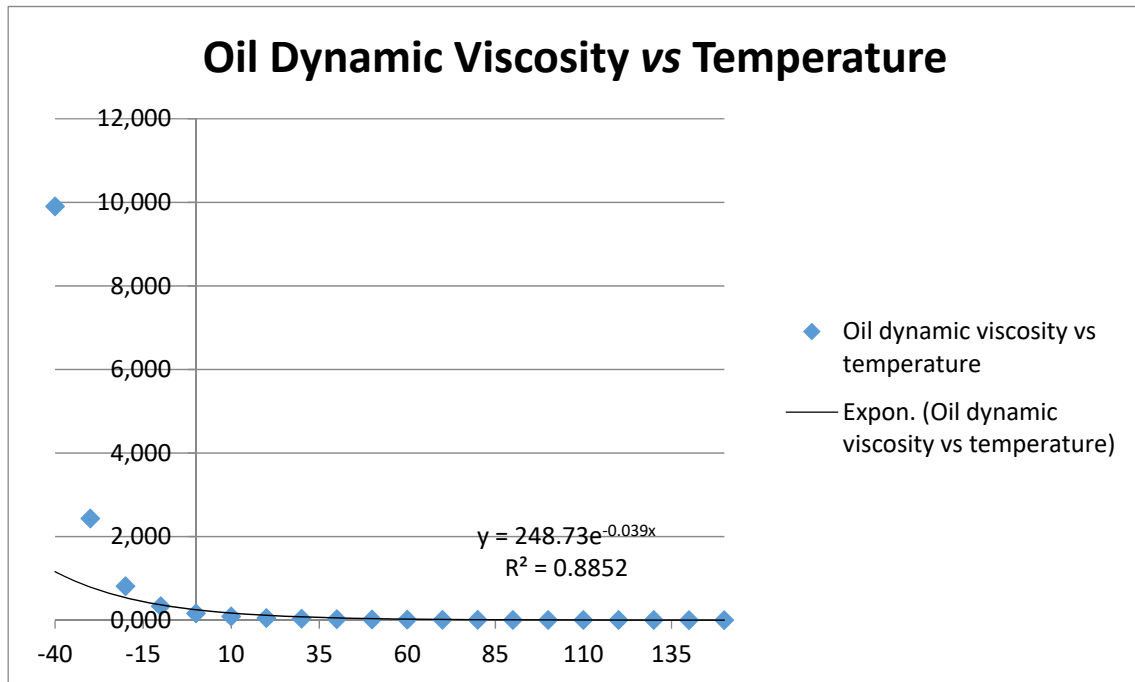


Figure 134 - Oil dynamic viscosity vs temperature

The behaviour of the dynamic viscosity is similar to the behaviour of the kinematic viscosity. Once again, at low temperatures the tendency line is not so accurate as desired but for the positive range temperature it is accurate enough. As happens to kinematic viscosity, at low temperatures there is a high variation on its values while on the positive temperature range the curve (tendency line) gets stable.

3.5.4 Clutch slipping

Clutch slipping phenomenon can be verified in three distinct occasions:

1. When the vehicle is steady (for example at a traffic light), the driver engages the clutch and then accelerates the vehicle;
2. When the driver presses full throttle on the accelerator when the vehicle is already moving;
3. When creeping effect is verified. This is when the vehicle is steady, the driver takes of the feet from the brake pedal and the car starts moving slowly. The engine is rotating at ± 800 rpm and the car is moving at ± 5 km/h. The creeping phenomenon can take until 20 seconds. After that clutches will be open in order to cool themselves with the lubricant oil. As it is an extended period (20 seconds)

of sleeping the temperature rises significantly that is why it is necessary cooling after a certain period of time.

Normally, the slipping speed of the clutches is at maximum 200 rpm (most of the times 100 rpm).

Slipping effect creates wear on the clutch plates (friction material) which decrease the efficiency of the clutch due to the lower coefficient of friction. As it is known, the wear of the clutches increases along the time. Friction ratio decreases along the time (it is already a known graph of the wear along the time). To compensate the decrease of the friction ratio the controls, after a certain amount of km it will be applied more force on the clutches to compensate the friction ratio decrease.

3.5.5 FWD clutch set

In Figure 135, it is illustrated the convention of signs (positive) for the calculations regarding to the FWD clutch set.

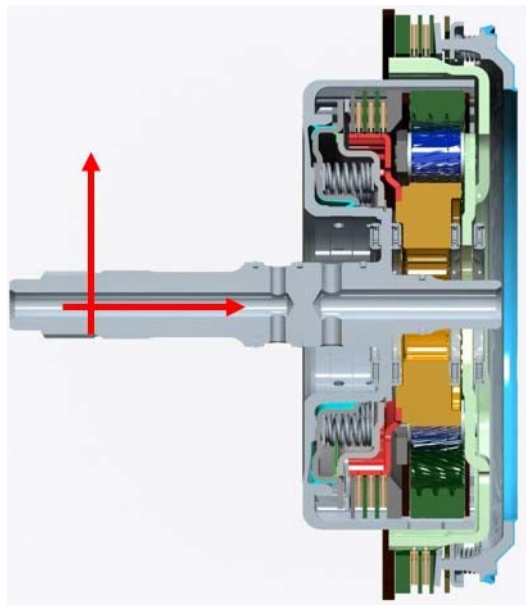


Figure 135 – Signs convention for the calculations related to FWD clutch set

3.5.5.1 General calculations

Firstly, it will be calculated the hydraulic area of the FWD piston, as follows:

$$A_H = \frac{\pi}{4} (OD_H^2 - ID_H^2) [mm^2] \quad \{3.16\}$$

For the following calculations, it was considered a nominal pressure of 15 bar and a rotating speed of 6500 rpm.

After that, hydraulic force that is exerted on FWD piston is calculated. This force is exerted by the fluid that is between FWD piston and the balance dam. The chamber built between FWD piston and balance dam is pressurized with 15 bar.

$$F_H = p \cdot A_H [N] \quad \{3.17\}$$

Now, it is necessary to calculate the value of the centrifugal force on the FWD piston and balance dam. It is certain that there is centrifugal force on both components because they rotate solidary with the clutch carrier. As mentioned before, centrifugal force is built in components which are rotating and have a fluid in it. That is the case of these components. To calculate the centrifugal force, eq. {2.61} must be used. It is important to verify the value of r_p for both components.

It is necessary to obtain the value of the resulting centrifugal force. It is known that the concavity of both components is oriented in opposite sides. This means that the centrifugal force has different orientations horizontally.

Centrifugal force of the balance dam has a negative orientation while centrifugal force of the FWD piston has a positive orientation. On the calculations, the negative sign for the centrifugal force of the BD was already introduced, so eq. {2.61} was adapted by adding the minus sign on the beginning of the equation.

The resulting centrifugal force is calculated as follows:

$$F_C = F_C^{FWDpiston} + F_C^{BD} [N] \quad \{3.18\}$$

The results of centrifugal force of the BD, FWD piston and total related to the rotating speed were calculated for some points in order to obtain graphs. All the calculated values are presented on Table 98 (Appendix).

Hydraulic force that is applied in FWD piston has a positive direction as so resulting centrifugal force. Total force on FWD piston can be obtained as:

$$F^{FWDpiston} = F_H + F_C [N] \quad \{3.19\}$$

In Figure 136, it is illustrated a schematic of FWD springs association.

It is known that there is a play between FWD clutch set plates. The all play is presented on chapter 3.5.8.

The springs association was divided in two stages. On the first stage, SP is compressed absorbing the play between plates. At this point, wavy washer wasn't compressed yet. When there is no more play between plates the wavy washer starts to be compressed (2nd stage).

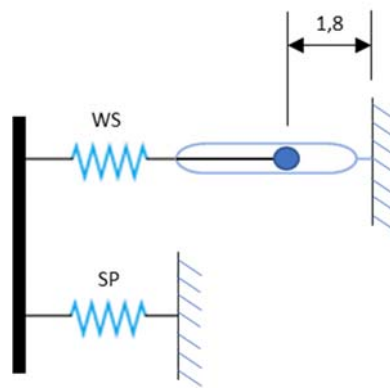


Figure 136 - FWD springs association schematic

According to the tolerances stack-up, the first stage compression is 1.8 mm.

First of all, it is necessary to calculate the reaction force of the spring pack at the end of 1st stage.

For nominal and second scenario, RF_{x1} is calculated on different ways. On the next equations' system, the first equation corresponds to the nominal case and the other one to the second scenario.

$$\begin{cases} RF_{x1} = SR^{SP} \cdot \Delta x1 + F_{hmax}^{SP} \\ RF_{x1} = SR^{SP} \cdot \Delta x1 + F_{max_{hmax}}^{SP} \end{cases} [N] \quad \{3.20\}$$

The value of SR and F_{hmax} and $F_{max_{hmax}}$ can be found in Table 16.

After that, it is calculated the force applied on the 2nd stage:

$$F_{x2} = RF_{x1} - F^{FWDpiston} [N] \quad \{3.21\}$$

The next step is about calculating the stiffness of the springs' parallel association:

$$SR_{\parallel} = SR^{SP} + SR^{WS} [N / mm] \quad \{3.22\}$$

The value of SR^{SP} can be obtained on Table 16 and the value of SR^{WS} can be obtained on Table 25.

The result of $SR_{\parallel}$ is presented on Table 46.

On the following step, it is necessary to find out how much the springs will compress on the 2nd stage. There are two possible compression deflection scenarios for the second stage. The first one is the complete compression of the spring pack and the other one is the complete compression of the wavy washer. To find out what is the correct scenario it is necessary to calculate the maximum deflection that can be travelled by the springs after travelling 1.8 mm on the 1st stage.

$$\begin{cases} x_2^{SP} = d_{\max}^{SP} - \Delta_{x1} \\ x_2^{WS} = FL^{WS} - b^{WS} \end{cases} [mm] \quad \{3.23\}$$

The value of the maximum travelling for the spring pack can be found on Table 16 and the values for the second equation can be found on Table 139.

The results are expressed on Table 46.

As it can be seen, the lowest value is 0.8 mm, that corresponds to the complete compression of the wavy washer. The springs association cannot be compressed more than 0.8 mm on the 2nd stage, however the spring pack could still be compressed 1.540 mm but not the wavy washer.

Now, it can be calculated the reaction force at the end of the second stage:

$$RF_{x2} = SR_{\parallel} \cdot \Delta_{x2} [N] \quad \{3.24\}$$

On the following step, it will be calculated the value of the force that is transmitted to the clutch discs:

$$F^{CP} = F_{x2} - RF_{x2} [N] \quad \{3.25\}$$

In order to better understand the springs association is was created a graph where springs total travelling and springs' stiffness are related: Figure 137.

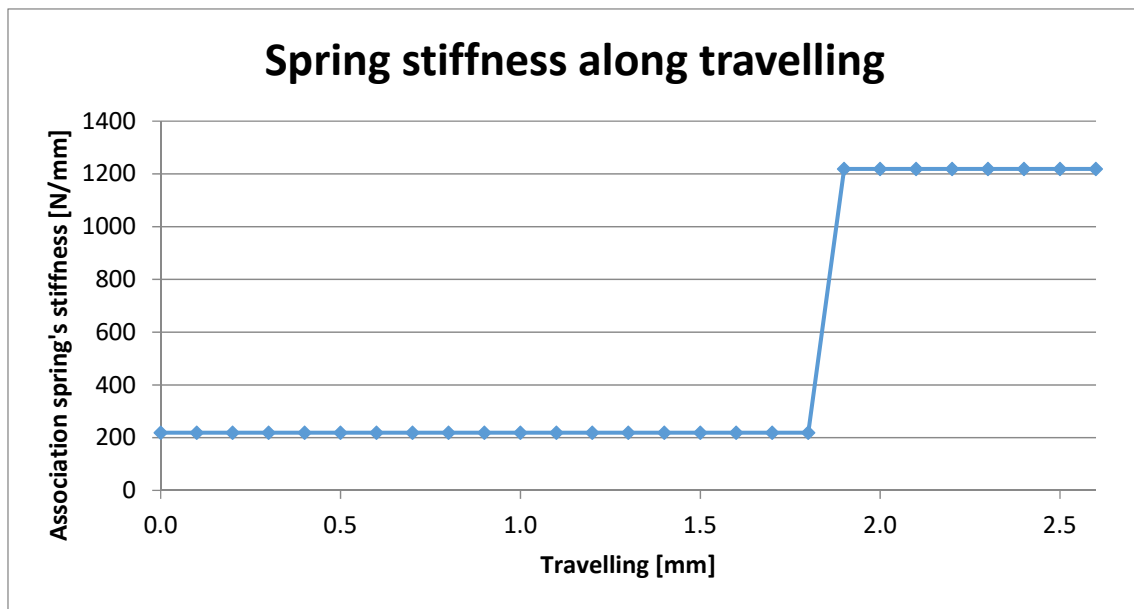


Figure 137 – Graph relating association springs' stiffness and total travelling

As explained before, at the beginning, only the spring pack is compressed and it is compressed 1.8 mm until wavy washer starts being deflected. So, it is not a surprise that from 0 to 1.8 mm travelling the stiffness is constant and equal to the spring pack stiffness. At 1.8 mm there is a rise of the stiffness. This is the point where the wavy washer starts being compressed. The value of the stiffness corresponds to the stiffness of the springs' association.

To find out what is the behaviour of the reaction force of the springs along the travelling it was created the graph of Figure 138.

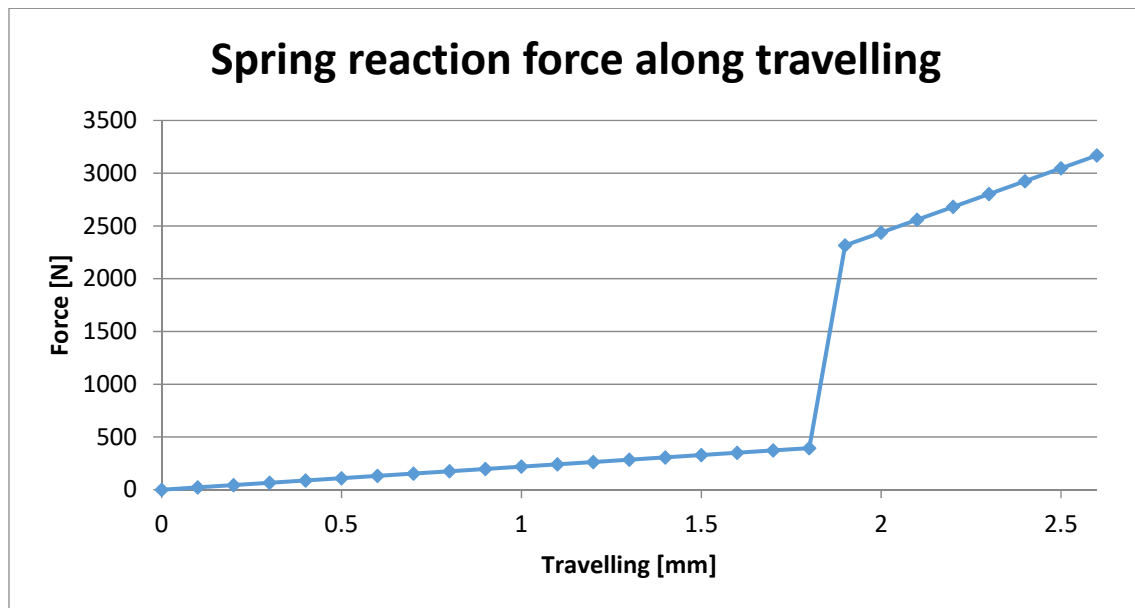


Figure 138 - Graph of travelling vs reaction force

According to the graph of Figure 138, there is a significant increase of the stiffness at 1.8 mm. There is an explanation for that: parallel springs' associations present a greater stiffness than series ones. The stiffness of the parallel association is calculated summing the values of all the springs in parallel. That is why there is a significant rise of the stiffness at 1.8 mm, where the wavy washer starts being compressed.

The table which was the base for the graphs of Figure 137 and Figure 138 can be found on Table 99 and Table 100 (Appendix), respectively. The results regarding to the SP association are presented on Table 46.

Table 46 - Results of general calculations of FWD clutch

Variable	Unit	Value
$SR_{ }$	N/mm	1218.817
$x2^{SP}$	mm	1.540
$x2^{WS}$	mm	0.8
RF_{x2}	N	975.054

3.5.5.1.1 Nominal case

In Figure 139 it is illustrated a graph relating the hydraulic pressure and the resulting axial force. The value of the axial force is the resulting force applied on the clutch plates

which means that in the calculation of this variable parameters such as the springs stiffness and hydraulic force on the FWD piston and centrifugal force were already taken in account.

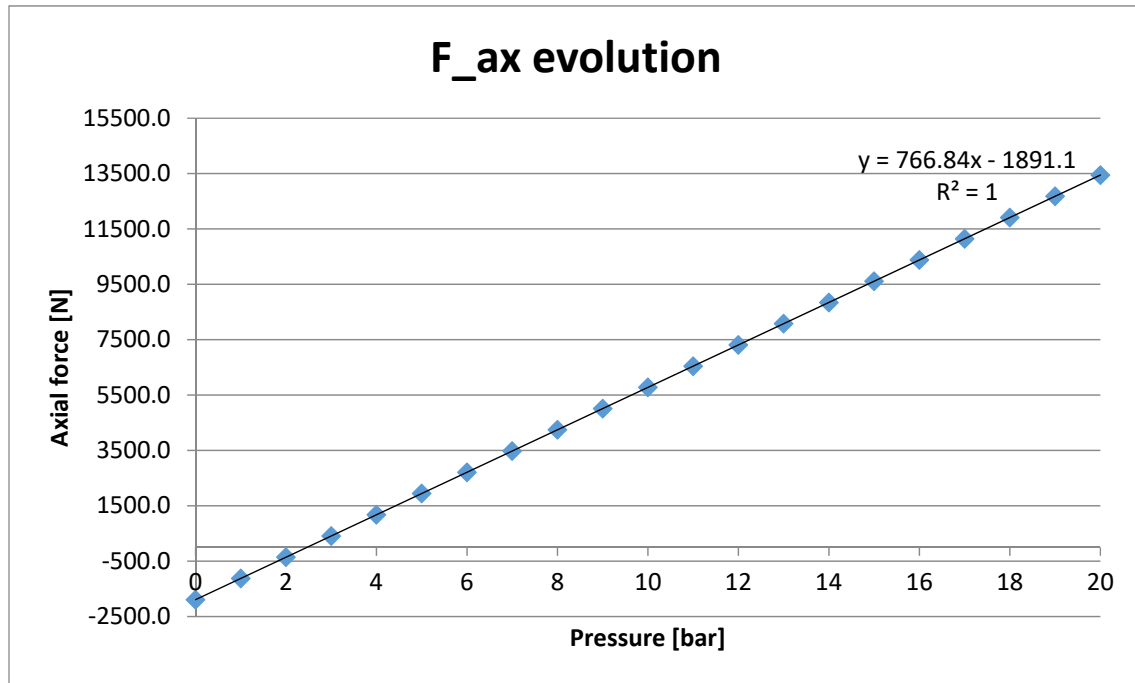


Figure 139 - Graph of pressure vs axial force of nominal scenario for FWD piston

In Table 47, it is expressed the results of the general calculations relative to the nominal scenario.

Table 47 - General calculations results for FWD clutch (nominal case)

Variable	Unit	Value
A_H	mm^2	7668.431
F_H	N	11502.647
$F_c^{FWDpiston}$	N	2051.853
F_c^{BD}	N	1644.790
F_C	N	407.063
$F^{FWDpiston}$	N	11909.710
RF_{x1}	N	1323.071
F_{x2}	N	10586.639
F^{CP}	N	9611.585

3.5.5.1.2 Second scenario

In this scenario, as explained before, the input data for these results (Table 48) have in account the tolerances of the nominal dimensions.

Table 48 - General calculations results for FWD clutch (second scenario)

Variable	Unit	Value
A_H	mm^2	7654.098
F_H	N	11481.147
$F_c^{FWDpiston}$	N	2012.826
F_c^{BD}	N	1609.920
F_C	N	402.906
$F^{FWDpiston}$	N	11884.053
RF_{x1}	N	1443.471
F_{x2}	N	10440.582
F^{CP}	N	9465.528

3.5.5.1.3 Third scenario

In Table 49, it is presented the input data for the results presented on Table 50.

Table 49 – General calculation's data input for FWD clutch set (3rd scenario)

Variable	Unit	Value
OD_H^{piston}	mm	118.050
ID_H^{piston}	mm	64.450
OR_H^{piston}	m	0.059
IR_H^{piston}	m	0.032
r_p^{piston}	m	0.029
OD_H^{BD}	mm	113.020
ID_H^{piston}	mm	0.057
OR_H^{BD}	m	63.600
IR_H^{BD}	m	0.032
r_p^{BD}	m	0.029

Table 50 - General calculations results for FWD clutch (3rd scenario)

Variable	Unit	Value
A_H	mm^2	7682.765
r_p^{piston} verification	mm	0.048
r_p^{BD} verification	mm	0.046
F_H	N	11524.147
$F_c^{\text{FWDpiston}}$	N	2056.662
F_c^{BD}	N	1633.919
F_C	N	422.743
$F^{\text{FWDpiston}}$	N	11101.404
x_1	mm	1.469
RF_{x1}	N	1114.827
RF_{x2}	N	796.361
F_{x2}	N	9986.577
SR	N/mm	936.895
F^{CP}	N	9190.216

3.5.5.1.4 Fourth scenario

In Table 51, it is presented the input data for the results presented on Table 52.

Table 51 - General calculation's data input for FWD clutch set (4th scenario)

Variable	Unit	Value
OD_H^{piston}	mm	117.950
ID_H^{piston}	mm	64.550
OR_H^{piston}	m	0.059
IR_H^{piston}	m	0.032
r_p^{piston}	m	0.029
OD_H^{BD}	mm	113.220
ID_H^{piston}	mm	63.400
OR_H^{BD}	m	0.057
IR_H^{BD}	m	0.032
r_p^{BD}	m	0.029

Table 52 - General calculations results for FWD clutch (4th scenario)

Variable	Unit	Value
A_H	mm^2	7654.098
r_p^{piston} verification	mm	0.048
r_p^{BD} verification	mm	0.046
F_H	N	11481.147
$F_c^{\text{FWDpiston}}$	N	2042.831
F_c^{BD}	N	1653.045
F_C	N	389.786
$F^{\text{FWDpiston}}$	N	11091.361
x_1	mm	2.151
RF_{x1}	N	1542.849
RF_{x2}	N	1206.984
F_{x2}	N	9548.512
SR	N/mm	1609.311
F^{CP}	N	8341.529

3.5.5.2 Transmissible torque

Transmissible torque refers to the transmitted torque from the DNR input to the clutch plates. The input of the DNR subsystem is the turbine shaft. The clutches in study are wet clutches so there is a significant loss of transmitted torque on the clutch plates due to the low friction coefficient. The friction coefficient will be taken in account on the calculations of the transmissible torque.

Transmissible torque is an important property to be calculated as there are some requirements which involve this property. For example, maximum pressure to reach a certain amount of torque, is one of the most important requirements. Once again, it will be calculated for two different scenarios: nominal and second scenario. The transmissible torque can be calculated, as eq. {2.63}.

In addition to the transmissible torque, which is a nominal value, it was calculated the maximum and minimum transmissible torque for nominal and second scenario. The only parameter that is different on minimum and maximum transmissible torque is the value of the friction coefficient. It must be used the minimum and maximum, respectively.

The value of F_{ax} is equal to F^{CP} that can be found on Table 47 and Table 48. The values of ζ can be found on Table 146 and N_s on Table 148. The values to calculate R_F (eq. {2.64}) can be found on Table 146.

The tables created on Excel® for both nominal and second cases can be found on Table 101 and Table 102 (Appendix), respectively.

Comparing the results for nominal and second scenario, it is conclusive that the difference is not significant. Forward, it will be analysed if the requirements are still fulfilled for the second scenario.

3.5.5.2.1 Nominal scenario

The value of R_F for the nominal scenario is equal to 0.062 m.

Below, it is illustrated a graph of the transmissible torque vs the hydraulic pressure. In the same graph, it is presented the maximum, minimum and the average torque. As explained before, the maximum torque has in account the maximum value of coefficient of friction. Contrary to the max. torque, minimum torque considers the minimum value of the friction coefficient. The average torque has in account the average value of the friction coefficient.

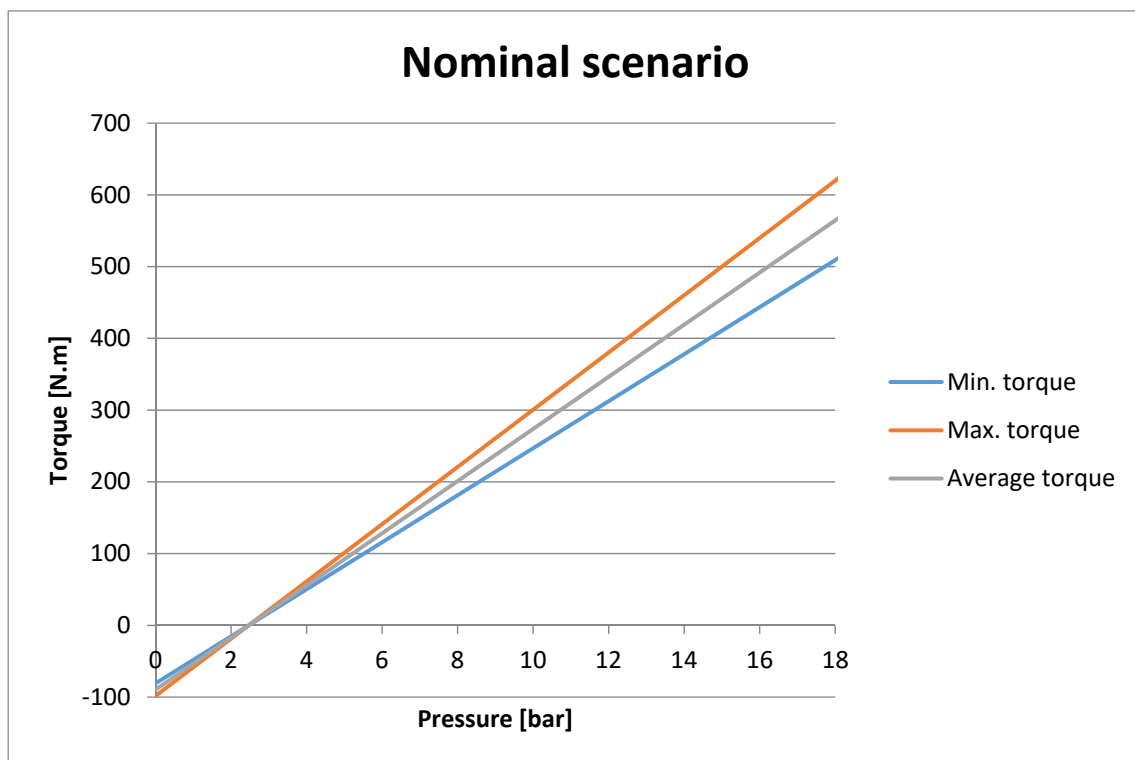


Figure 140 - Graph of transmissible torque vs pressure for the nominal case of FWD clutch

According to the graph of Figure 140, there is a significant difference between the three scenarios at the maximum pressure of 18 bar. At 15 bar, there is a difference of about 100 Nm between the min. and max. torque. Until approximately 2 bar the transmissible torque is negative which means there is no transmitted torque.

3.5.5.2.2 Second scenario

In this case, the second scenario refers to the transmissible torque taking in account the tolerances of some dimensions which is known that will prejudice the amount of transmissible torque.

The value of R_F for the nominal scenario is equal to 0.058 m.

In Figure 141, it is presented the different cases of transmissible torque according to the second scenario.

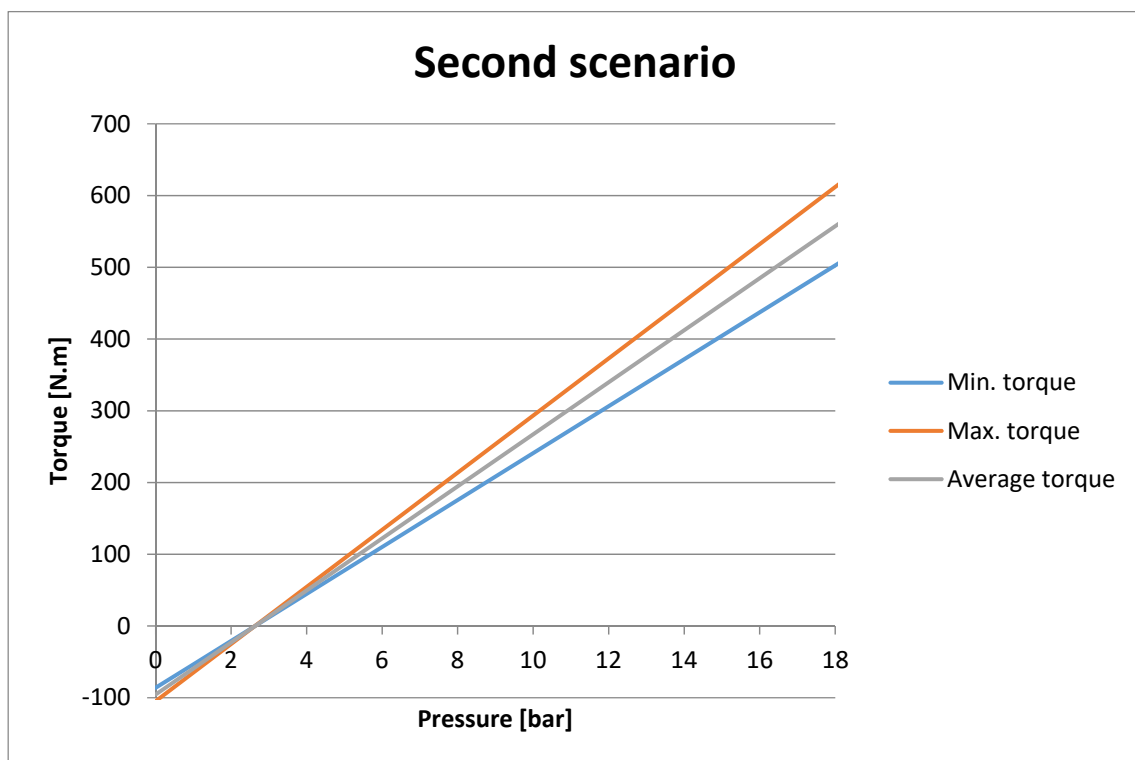


Figure 141 - Graph of transmissible torque vs pressure for the second case of FWD piston

According to the graph of Figure 141, there is a significant difference between the three scenarios at the maximum pressure of 18 bar. At 15 bar, there is a difference of about 100 Nm between the min. and max. torque as verified for the nominal scenario. Until

approximately 2 bar the transmissible torque is negative which means there is no transmitted torque.

3.5.5.2.3 Third scenario

In Table 53, it is expressed the friction area of the clutch plates considered on the calculations in order to obtain the final values.

Table 53 – Input data for transmissible torque calculation (3rd scenario)

Variable	Unit	Value
A_F^{pad}	mm^2	61.085
A_F	mm^2	1954.726

In Figure 142, it is presented the different cases of transmissible torque according to the third scenario.

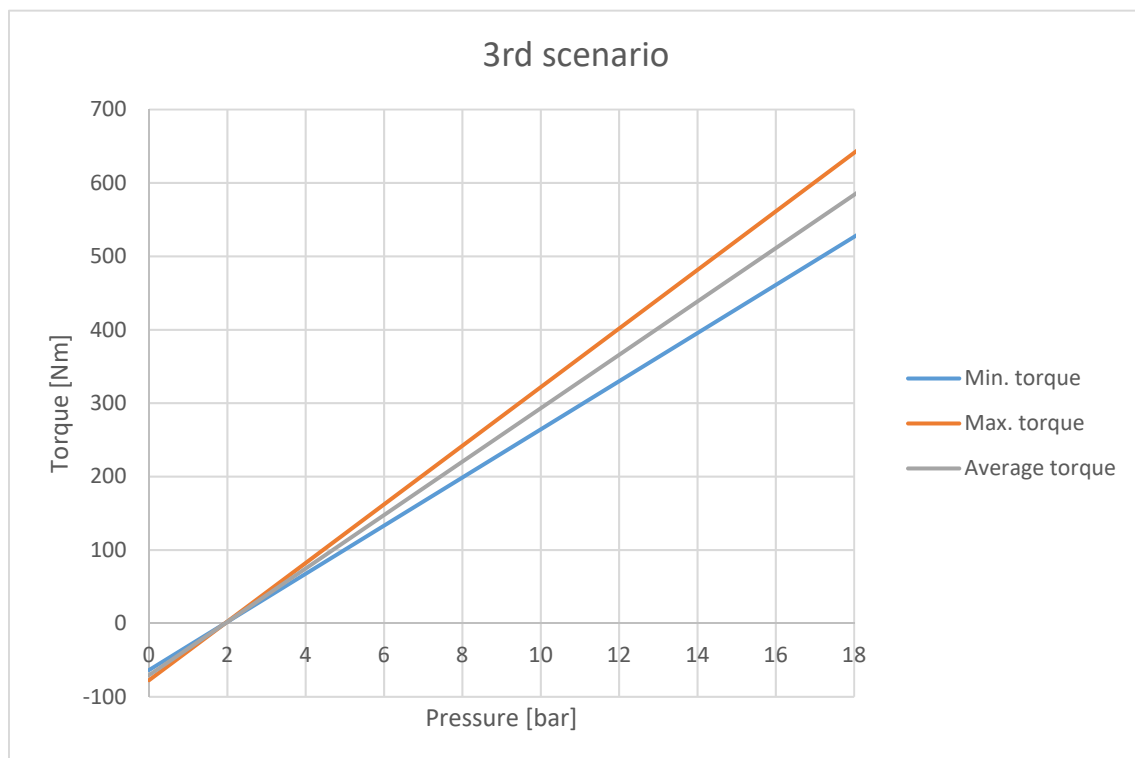


Figure 142 - Graph of transmissible torque vs pressure for the third case of FWD piston

According to the graph of Figure 142, there is a significant difference between the three scenarios at the maximum pressure of 18 bar. At 15 bar, there is a difference of about 100 Nm between the min. and max. torque as verified for the nominal scenario. Until approximately 2 bar the transmissible torque is negative which means there is no transmitted torque.

3.5.5.2.4 Fourth scenario

Table 54 – Input data for transmissible torque calculation (4th scenario)

Variable	Unit	Value
A_F^{pad}	mm ²	52.624
A_F	mm ²	1683.971

In Figure 143, it is presented the different cases of transmissible torque according to the fourth scenario.

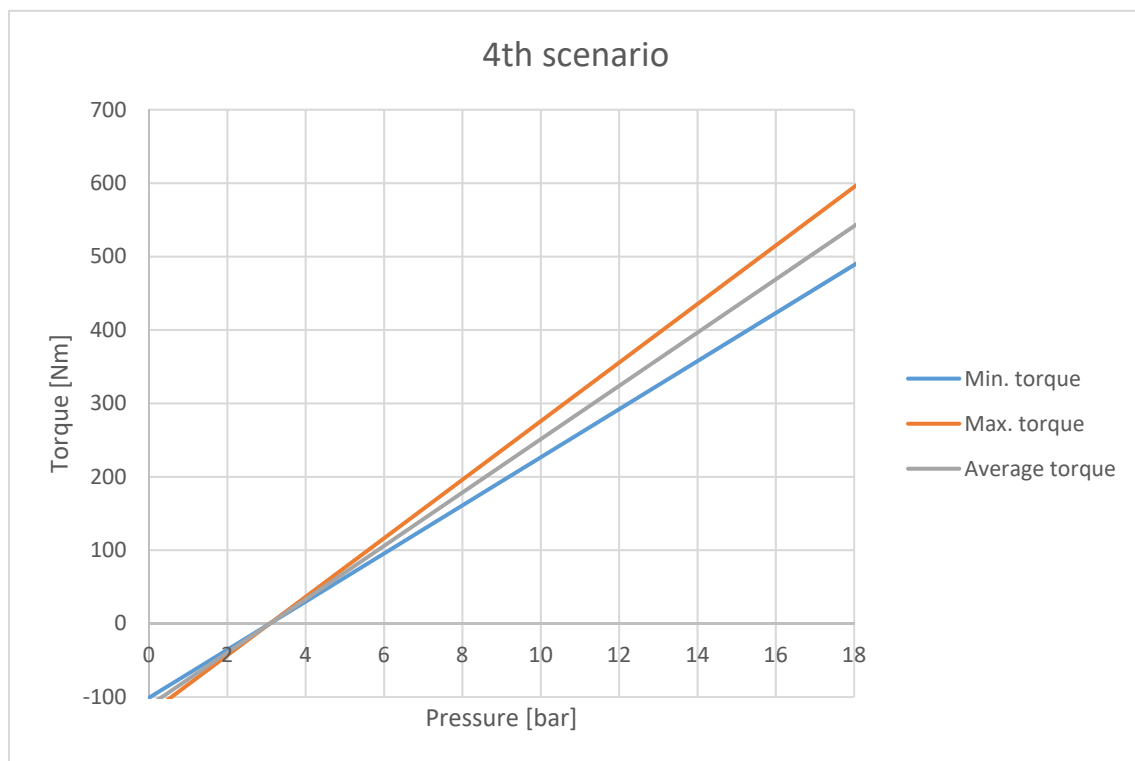


Figure 143 - Graph of transmissible torque vs pressure for the fourth case of FWD piston

According to the graph of Figure 143, there is a significant difference between the three scenarios at the maximum pressure of 18 bar. At 15 bar, there is a difference of about

100 Nm between the min. and max. torque as verified for the nominal scenario. Until approximately 2 bar the transmissible torque is negative which means there is no transmitted torque.

3.5.5.2.5 Scenarios' comparison

Based on the graphs presented before relative to the transmissible torque it is not possible to take conclusions regarding the different scenarios. For that, it is necessary to plot some graphs comparing the different scenarios.

Firstly, in the graph of Figure 144, it was studied the difference between third, fourth and nominal scenarios. As long as the third and fourth scenarios are more complex than the others, it is important to understand if there is a significant difference comparing to the nominal case. These scenarios are more complex due to the fact they take in account several factors which can influence the transmissible torque. For plotting this graph, it was considered the average torque of each scenario.

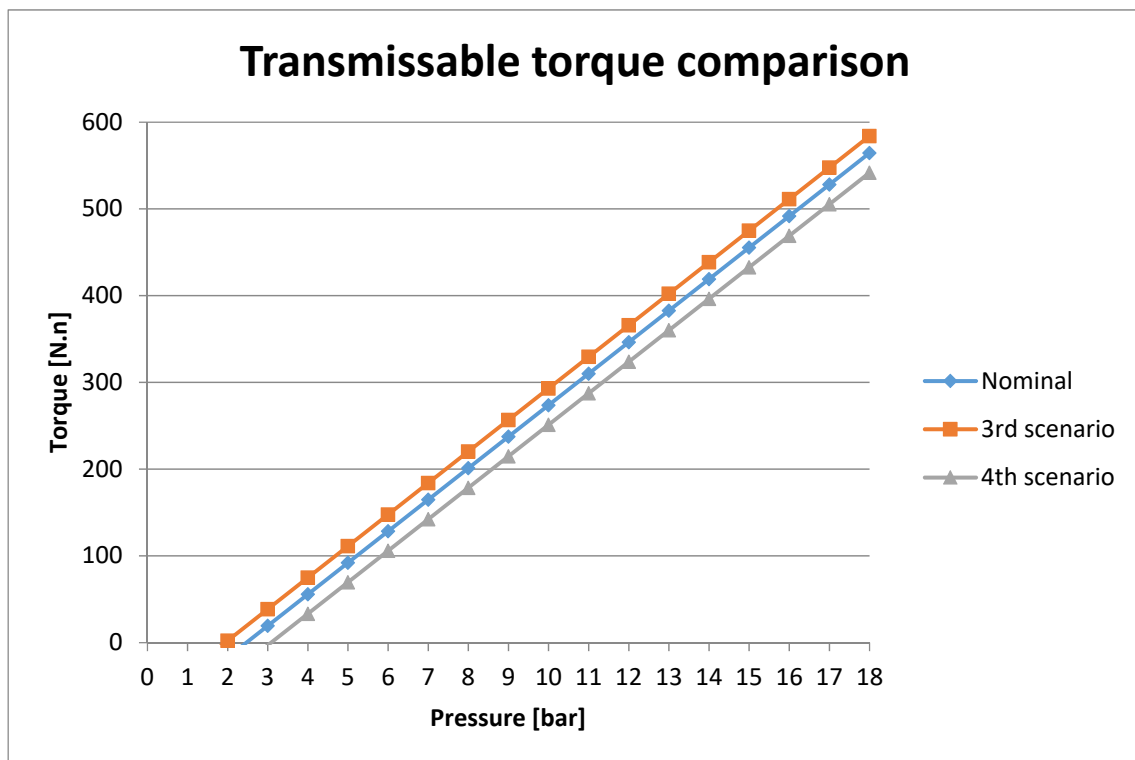


Figure 144 – Transmissible torque scenario comparison for FWD clutch set

According to the graph of Figure 144, there is a visible difference between the three lines. For example, at a pressure of 15 bar, which corresponds to the pressure whose maximum torque must be reached, there is a difference of about 50 Nm between the

scenarios. Forward, this graph should be studied again during the process of the validation of the requirements. As expected the line relative to the nominal scenario is located between both lines of the others scenarios.

In Figure 145, it is presented the graph relating the travelling of the spring association and the hydraulic pressure.

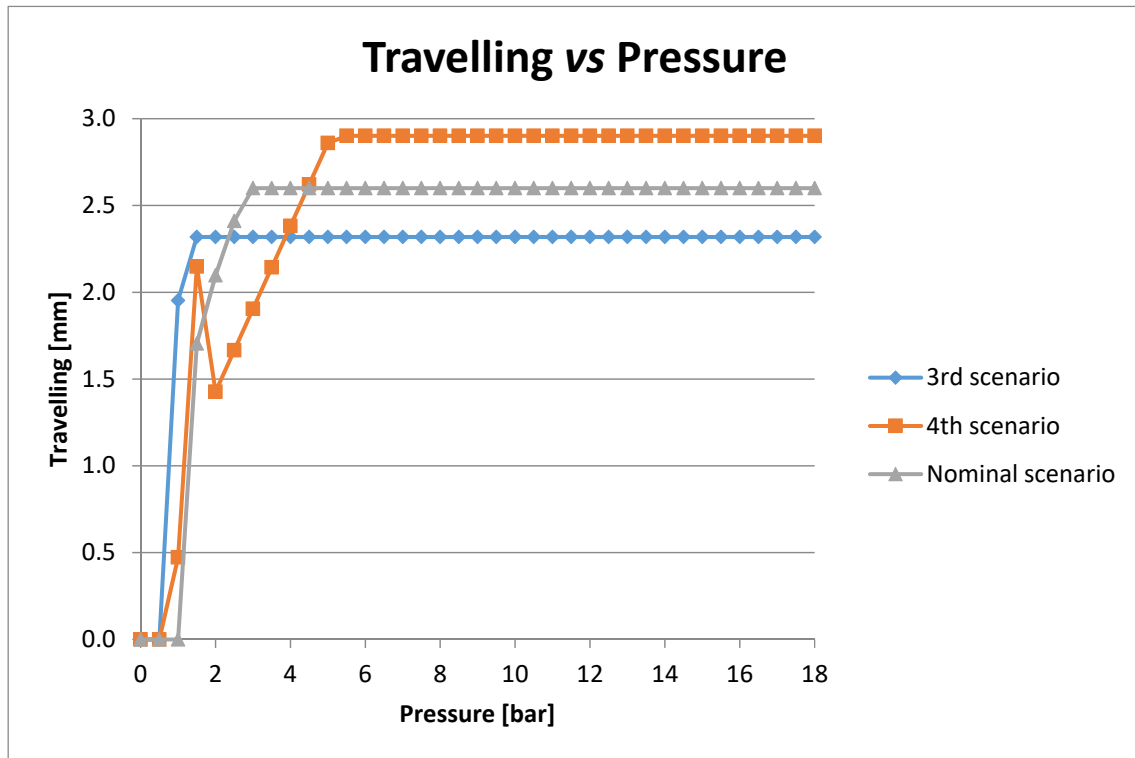


Figure 145 – Travelling vs pressure comparison between scenarios

According to the graph of Figure 145, as expected and presented before until a certain value of pressure there is any compression on the spring pack neither on the wavy washer. To start compressing the spring pack it is necessary to build a pressure which translates itself in an amount of force greater than the load of the spring. This phenomenon is visible for all the scenarios. The pressure keeps increasing as also the travelling. This is verified until a certain point. It can be seen that at a certain point the slope of the curves changes smoothly. This is the point where the wavy washer starts to be compressed. This point corresponds to different travelling values for the distinct scenarios due to the tolerances of the working heights of the spring pack considered in the different scenarios. At a certain point, the curves become horizontal and stable. This value of travelling corresponds to the maximum compression of the spring association. This means that from this point the increase of pressure will have no effect on the amount of travelling of the spring association. There is a particular phenomenon in the

4th scenario in the point of starting to compress the wavy washer. At the point of shifting of the slope of the curves, it is known the maximum pressure and respective force that can be loaded at the spring pack. At this point the spring stiffness will change because a springs association will be created since the wavy washer and the spring pack will work together in parallel. This means that the value of the spring stiffness will change. It is known and presented before that the stiffness of the parallel association of the springs of the 4th scenario is considerably higher than of the 3rd scenario. This means that when the wavy washer starts to be compressed, the stiffness of the springs association will increase significantly. Consequently, the necessary force to compress the association needs to be considerably higher. At about 1.5 bar (inflection point) the built force is not high enough to overcome the necessary load of the springs' association which will cause the expansion of the springs.

In the graph of Figure 146, it is presented the comparison between the 3rd and 4th scenarios relating the force reaction of the springs' association and the travelling. According to the graph of Figure 146, the spring pack stiffness for both scenarios is equal. This can be concluded since the force at each travelling instance is the same for both scenarios.

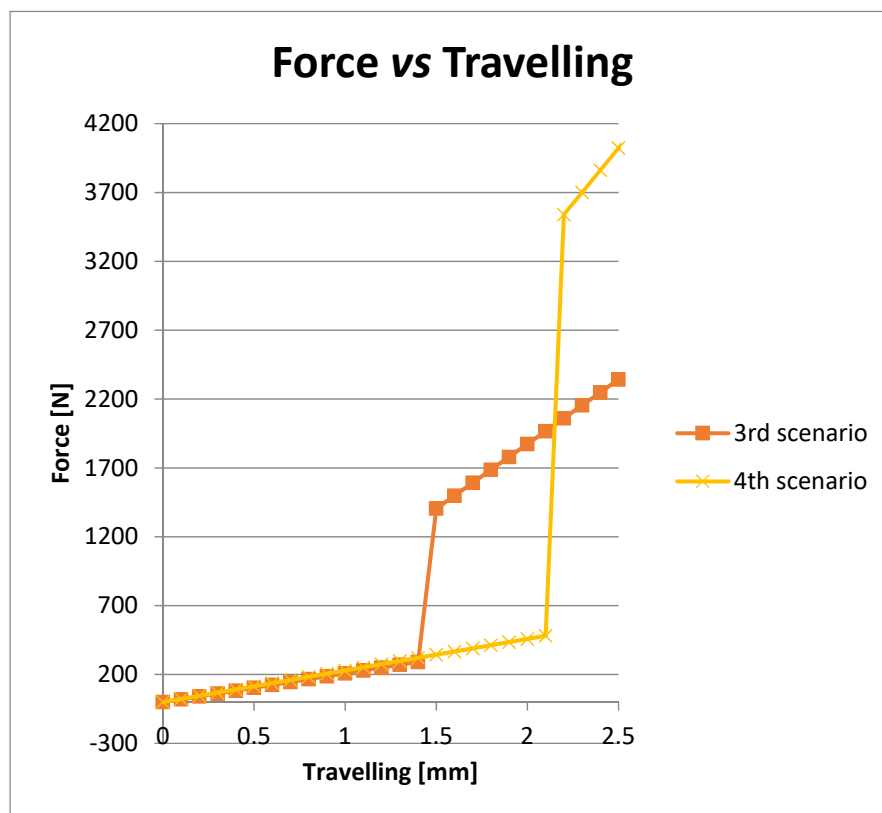


Figure 146 – Graph force vs travelling of the springs association for 3rd and 4th scenarios of FWD clutch set

At a certain travelling value (approximately 1.4 mm) the force of relative to the 3th scenario increases significantly while the force for the 4th scenario keeps increasing straightly until approximately 2.2 mm. This means that the tolerances of the SP dictates the point where the wavy washer will start to be compressed. In 3rd scenario the wavy washer is compressed earlier than in 4th. As seen before in the calculations of the wavy washer, for the 3rd and 4th scenarios the values of the spring stiffness was significantly different which means that the forces obtained for each scenario will be significantly distinct.

3.5.5.2.6 Creeping effect

In this chapter, the transmissible torque will be calculated having in account the creeping effect which is expected to have a negative effect on the transmissible torque. It was calculated the transmissible torque for two different values of slipping speed: 200 and 100 rpm. It is expected to obtain higher values of transmissible torque of the lowest value of slipping speed.

On chapter 6.11 it is presented Table 107 which is an example of a table used to calculate the resulting centrifugal force for a slipping speed of 200 rpm. The final results of the transmissible torque for the different values of the slipping speed are presented on chapter 6.12 (Table 108) and 6.13 (Table 109).

In addition to the slipping speed, it was also calculated the transmissible torque under the influence of the coefficient of friction as have been done before for the nominal scenario.

In Figure 148 and Figure 147, it is illustrated the resulting graphs of transmissible torque for different slipping speeds.

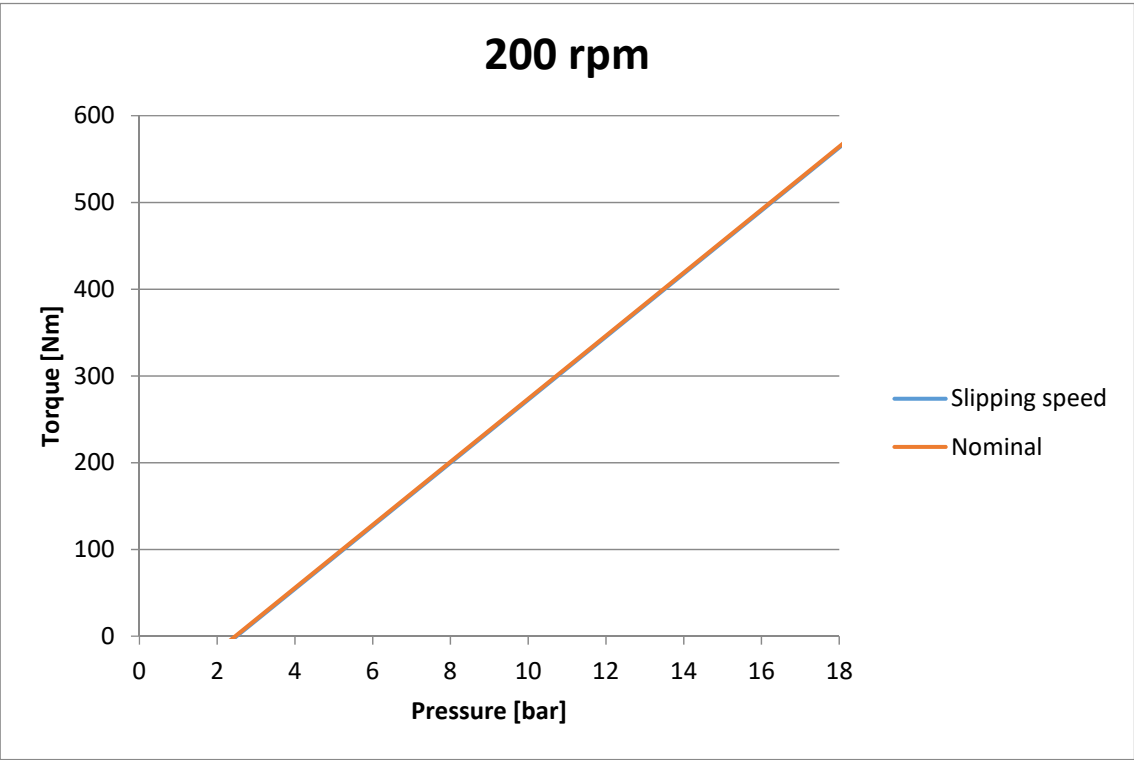


Figure 148 – Resulting chart of transmissible torque for a slipping speed of 200 rpm

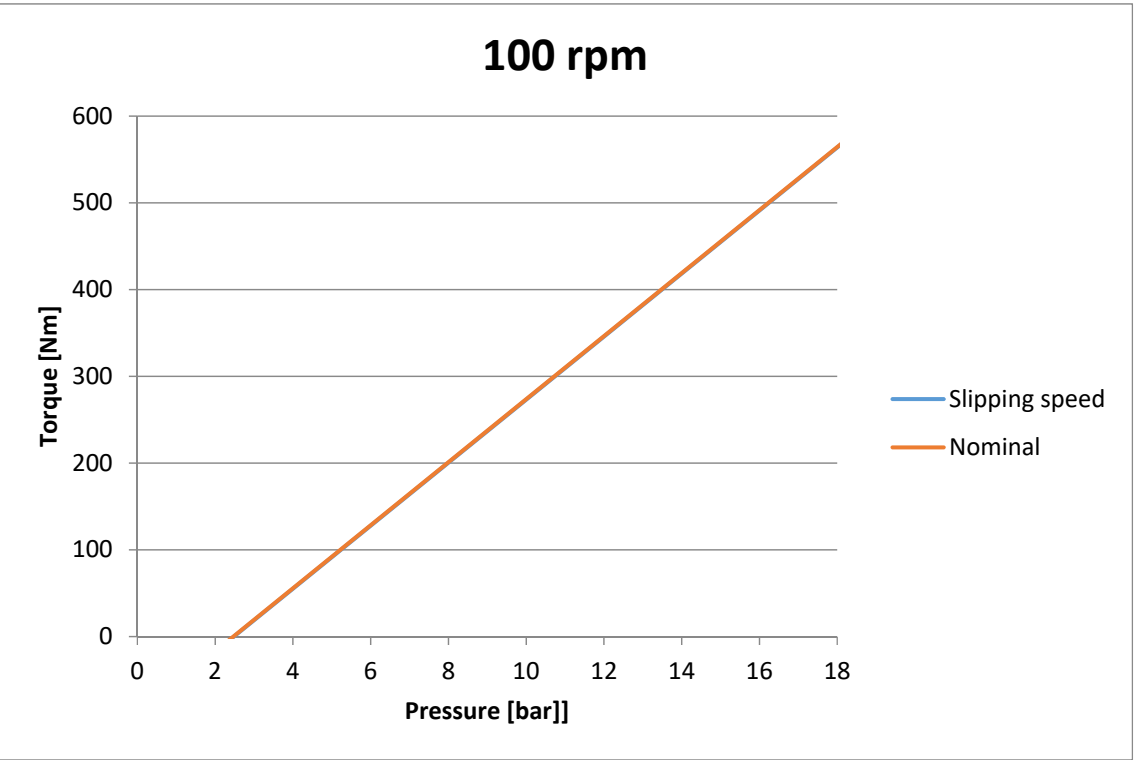


Figure 147 - Resulting chart of transmissible torque for a slipping speed of 100 rpm

Comparing both graphs, it is conclusive that there is no significant difference between the nominal transmissible torque and the transmissible torque affected by the slipping speed since the curve of the nominal scenario is overcoming the slipping speed curve.

3.5.5.2.7 Influent factors

Before studying the best and worst scenarios for the transmissible torque it is necessary to understand which variables have influence in the results of each parameter such as hydraulic force, hydraulic centrifugal force, etc. This analysis has already been done for the different springs. These results will be taken in account for the REV transmissible torque calculations however not all factors are applied to that since there are some architectural differences.

Hydraulic force

As previously discussed there is a linear correlation between the hydraulic force and the transmissible torque. The higher the hydraulic force is, the higher the transmissible torque. The hydraulic force can be increased easily by means of dimensions or hydraulic pressure. If the outer diameter of the hydraulic chamber increases and the inner diameter decreases, this will lead to the increase of the area of the chamber and consequently the increase of the hydraulic force. The increase of the hydraulic pressure will also increase the amount of transmissible torque. However, it is not desirable to deal with high values of pressure not only because of safety issues but also because of its controls and the complexity and stiffness of the parts.

Slipping speed

In Figure 149, it is illustrated a graph where the effect of different values of slipping speed on the transmissible torque are compared. As it was previously explained, usually the maximum slipping speed of a clutch is 200 rpm but most of the times the slipping speed is 100 rpm. Both values were simulated on the calculations.

According to the graph of Figure 149 it is conclusive that there is no significant difference between the resulting lines. In fact, the difference is not visible since one line is overlapping the other one which proves the low difference between them.

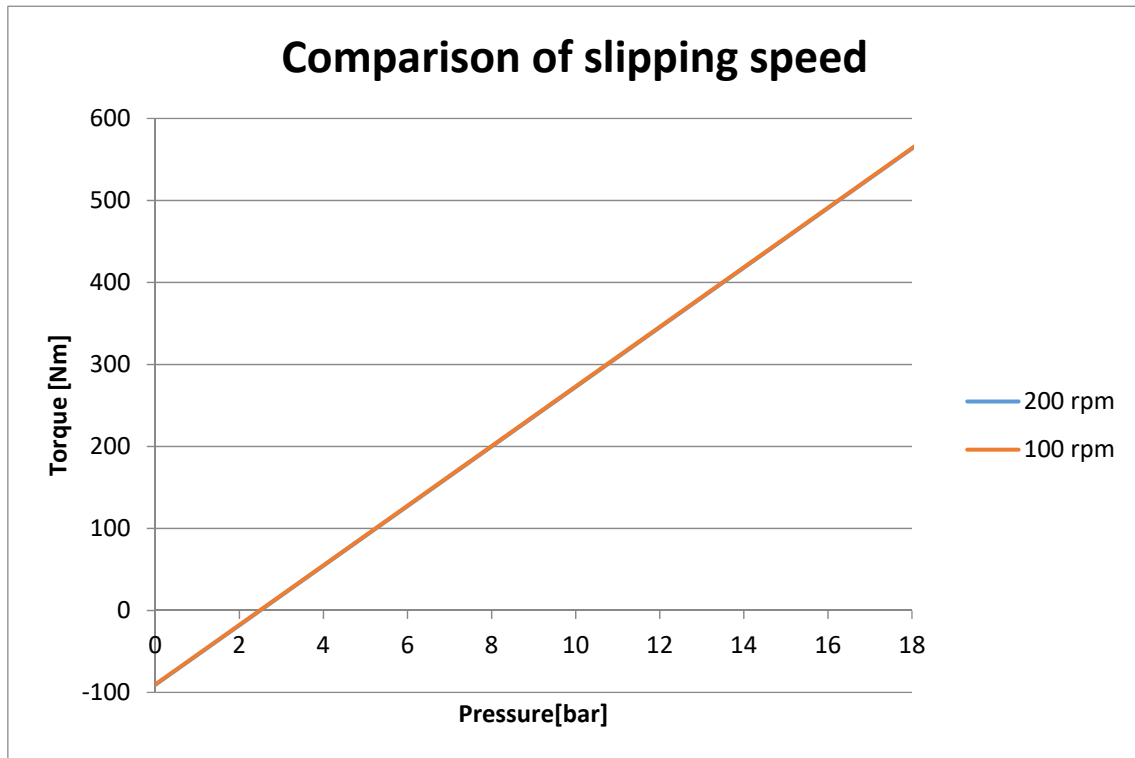


Figure 149 – Resulting graph of the comparison of transmissible torque for different slipping speeds

As seen before there is no visible difference between none transmissible torque obtained for both the slipping speeds and the nominal transmissible torque. In conclusion, slipping speed as a really low effect on the amount of transmissible torque.

Hydraulic centrifugal force

The hydraulic centrifugal force plays a considerable role in the amount of transmissible torque. In this case centrifugal force is built in both FWD piston and balance dam. Since the hydraulic area of the FWD piston is greater than the balance dam's, the centrifugal force will be always higher in the FWD piston. Furthermore, the concavities of the parts make easy to understand that the centrifugal force of the FWD piston has a positive influence, contrarily to the balance dam. Although, at the end, as the centrifugal force on the FWD piston is always higher, the final centrifugal force will be always positive and will have a positive influence in the amount of transmissible torque. However, there are several factors that dictate the value of its as presented below.

Rotating speed

As seen before, centrifugal force has a significant role on the amount of transmissible torque. If centrifugal force is very high on the BD, so the axial force applied on the clutches will be lower and consequently, transmitted torque will be lower too. On the other hand, if the centrifugal force on the FWD piston is high, the axial force and the transmitted torque will be high, too.

Due to the importance of the centrifugal force, it was decided to study which factors have influence on it. This study can then be used for future projects to help decreasing the value of the centrifugal forces.

It will be studied the centrifugal force of the BD, FWD piston and total centrifugal force. Firstly, it will be studied the effect of the rotating speed.

In Figure 150, it is illustrated the effect of rotating speed on centrifugal force of the BD. According to the graph of Figure 150, as seen before centrifugal force of BD is negative. The higher rotating speed is, the higher the centrifugal force is, in terms of absolute value. It was created a tendency line, which proves that the resulting line is parabolic due to the value of R^2 be equal to 1. It is also conclusive that the difference between nominal and worst scenario is not significant.

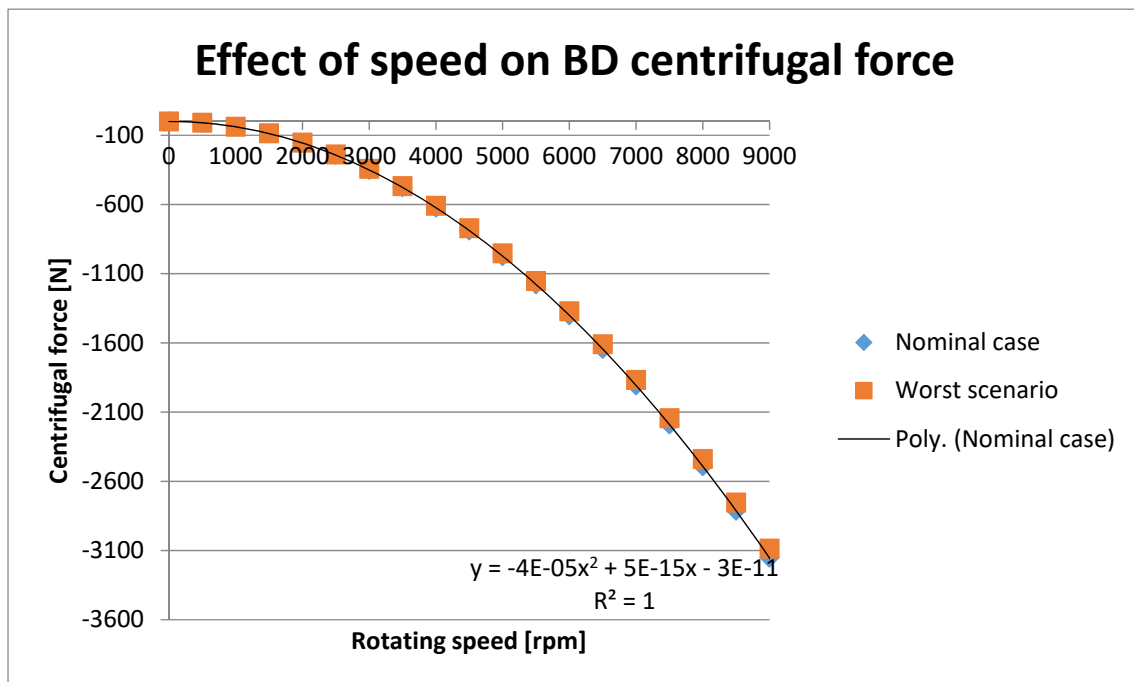


Figure 150 – Graph of the effect of rotating speed on balance dam centrifugal force

The following graph plots the effect of rotating speed on FWD piston centrifugal force.

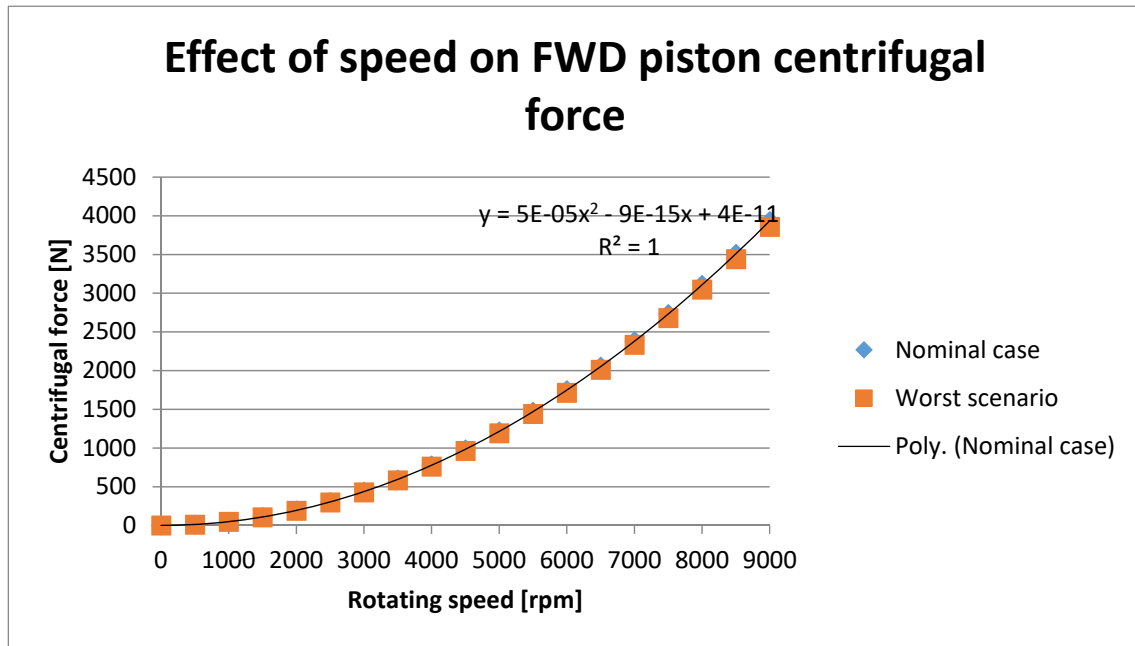


Figure 151 - Effect of rotating speed on FWD piston centrifugal force

Based on the graph of Figure 151, as seen before centrifugal force of FWD piston is positive because of the orientation of its concavity. As happens to the centrifugal force of the BD, the higher the rotating speed is, the higher the centrifugal force. It was created a tendency line, too. Once again, the parabolic curve is the one which best fits. Finally, the different between nominal and worst scenario is not significant.

On the graph presented below (Figure 152), it is illustrated the rotating speed effect of resulting centrifugal force.

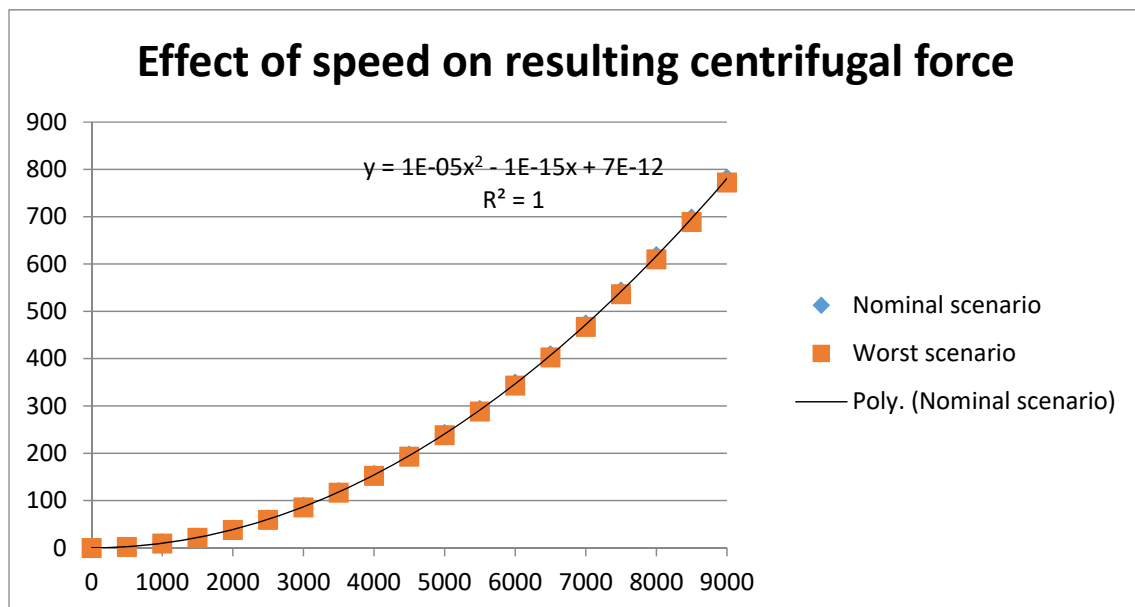


Figure 152 – Effect of rotating speed on resulting centrifugal force

As the resulting centrifugal force is sum of the centrifugal force on FWD piston and BD and taking in account the graphs it is expected that the curve of resulting centrifugal force has a parabolic shape and its value increases with the rising up of rotating speed.

Centrifugal force on BD ins always negative for the rotating speed range in study and the centrifugal force on FWD piston is always positive for the same range. The absolute value of centrifugal force of FWD piston is always greater than for BD, so it is expected that the resulting centrifugal force is always positive.

Fluid density

It is known that the density of the oil is affected by the working temperature. In order to understand how it oil density affects the centrifugal force it was plotted a graph on Figure 153. In this graph, it is illustrated the relation between oil density and centrifugal force. It was used the FWD piston for any particular reason.

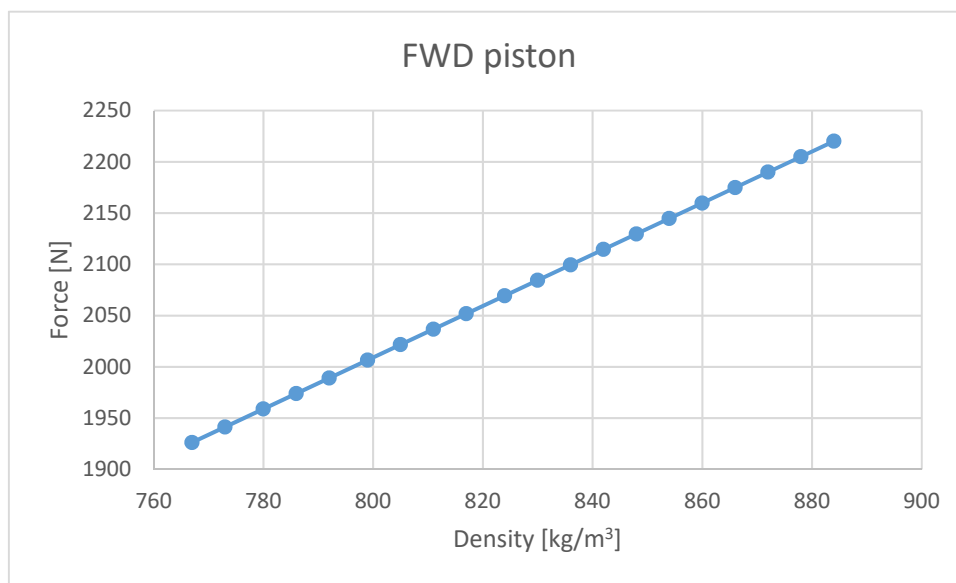


Figure 153 – Influence of oil density in the centrifugal force of FWD piston

According to the graph, the relation between centrifugal force and oil density is linear and well established.

Fluid entry radius

Fluid entry radius is one of the parameters which define the value of the centrifugal force and consequently has an influence on it. Although the effect of the influence is

unknown so in this chapter it will be studied the influence of the fluid entry radius in the centrifugal force originated in the FWD piston (Figure 154) and BD (Figure 155).

The value of the centrifugal force for the FWD piston is positive while for the BD is negative due to the fact they have contrary directions as explained before.

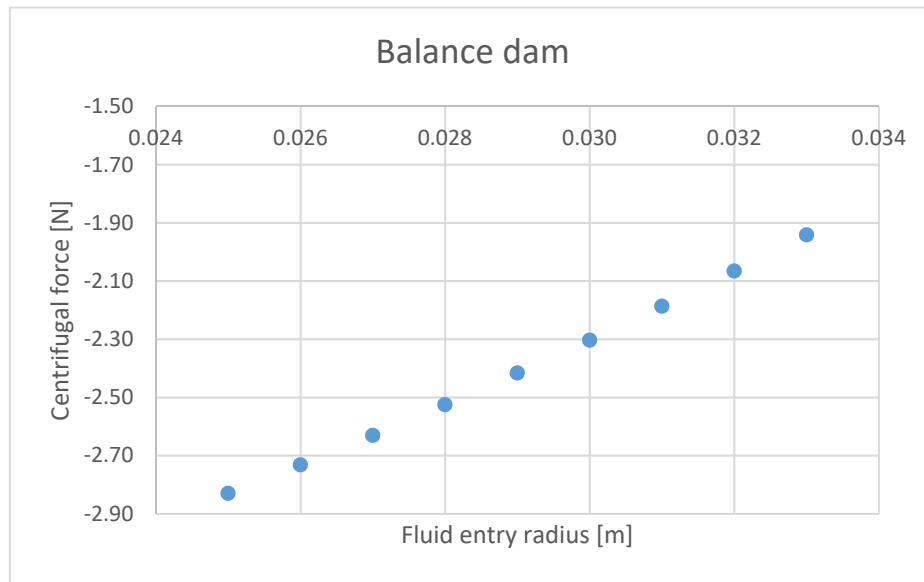


Figure 155 – Influence of fluid entry radius in the centrifugal force of BD

According to the graph of Figure 154, it is conclusive that the increase of the value of the fluid entry radius, decreases the value of the centrifugal force. On the other hand, according to the graph of Figure 155, the increase of the value of the fluid entry radius increases the value of the obtained centrifugal force.

The tables with the input values and the tables with the resulting values used to build both graphs are presented on the appendix (6.14).

3.5.5.3 Biting-point conditions

Biting-point conditions are the necessary conditions to start transmitting a torque of 8 N.m. First of all, it is necessary to calculate the tendency line's equation of the curve for the minimum transmissible torque. The graph and respective tendency curves' equation are presented for nominal and worst scenario on Figure 156 and Figure 159, respectively. By using these equations, it is possible to calculate the necessary pressure to achieve the 8 N.m torque.

After calculating the necessary pressure, it was finally calculated the necessary axial force on FWD piston based on the tendency line equation of graph which relates axial force and applied pressure. Now, it is known the necessary force to apply a torque of 8 N.m.

The final resulting values are expressed on Table 55 and Table 56 for the nominal and second scenario, respectively.

3.5.5.3.1 Nominal scenario

In the graph of Figure 156, it is presented the graph of the minimum transmissible torque. It was designed the tendency line as it will useful forwards.

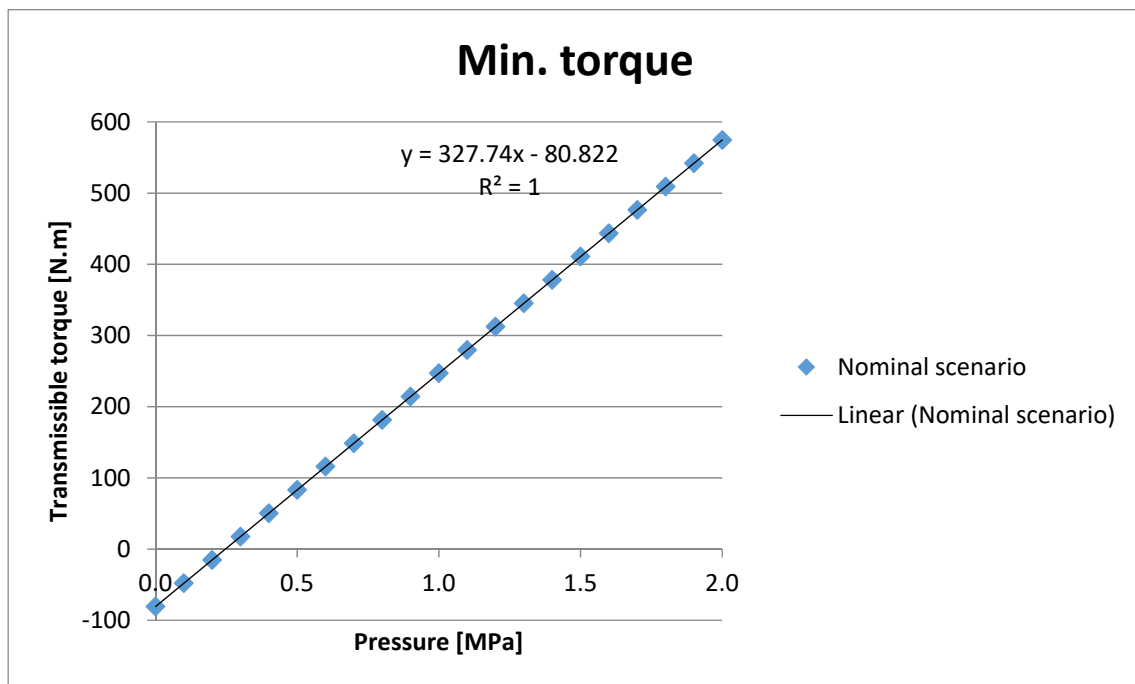


Figure 156 – Graph of minimum transmissible torque for nominal scenario of FWD clutch

As seen before the relation between the transmissible torque and the pressure is linear.

In the graph of Figure 157 it is illustrated the relation between axial force and hydraulic pressure. The axial force is the force applied on the clutch plates.

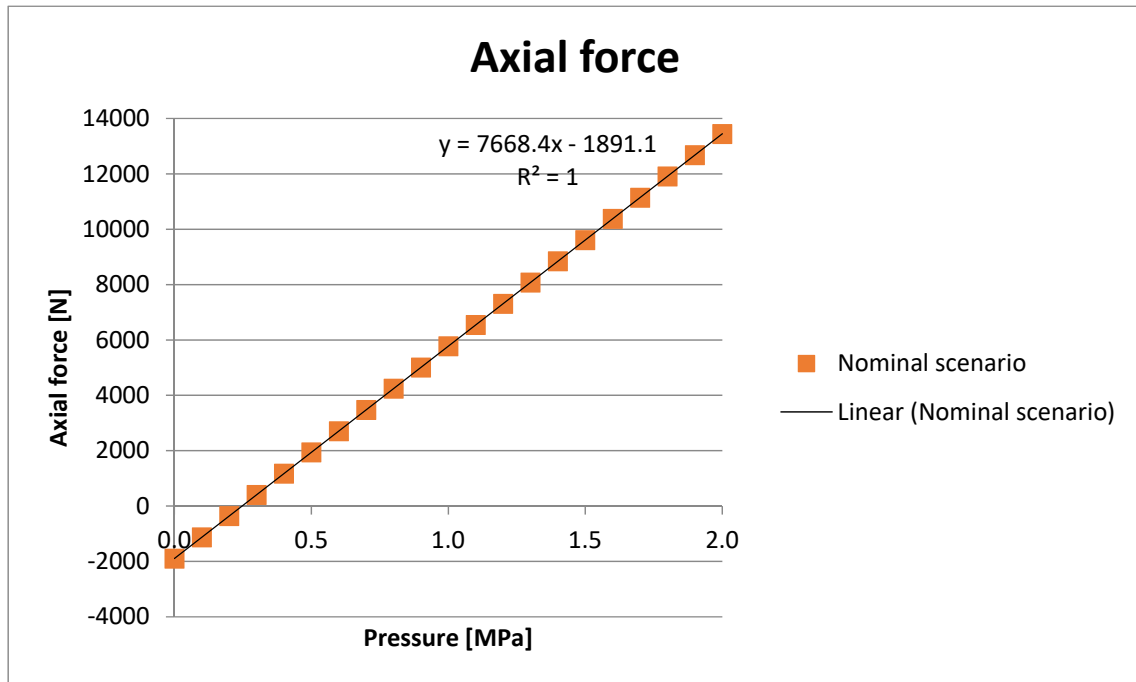


Figure 157 - Graph of axial force vs pressure for nominal scenario of FWD clutch

The results are expressed on Table 55.

Table 55 – Results of biting-pint conditions of FWD clutch set (nominal scenario)

Variable	Unit	Value
p	bar	2.710
F _{ax}	N	187.182

3.5.5.3.2 Second scenario

he method used for the second scenario is the same used for the calculations of the nominal scenario but by taking in account the dimensions' tolerances.

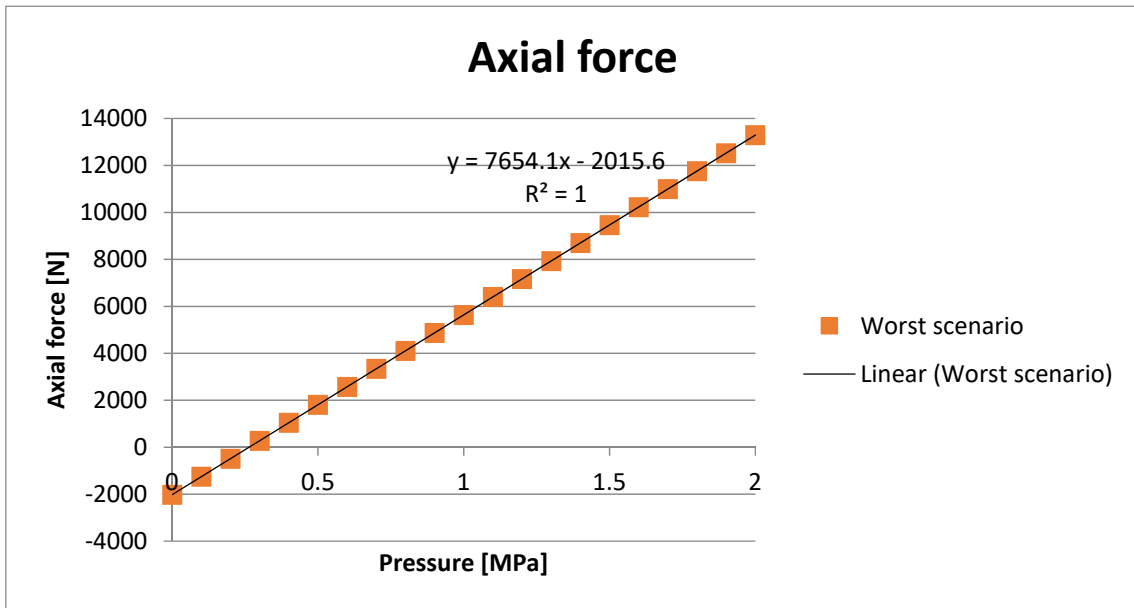


Figure 158 - Graph of axial force vs pressure for second scenario of FWD clutch

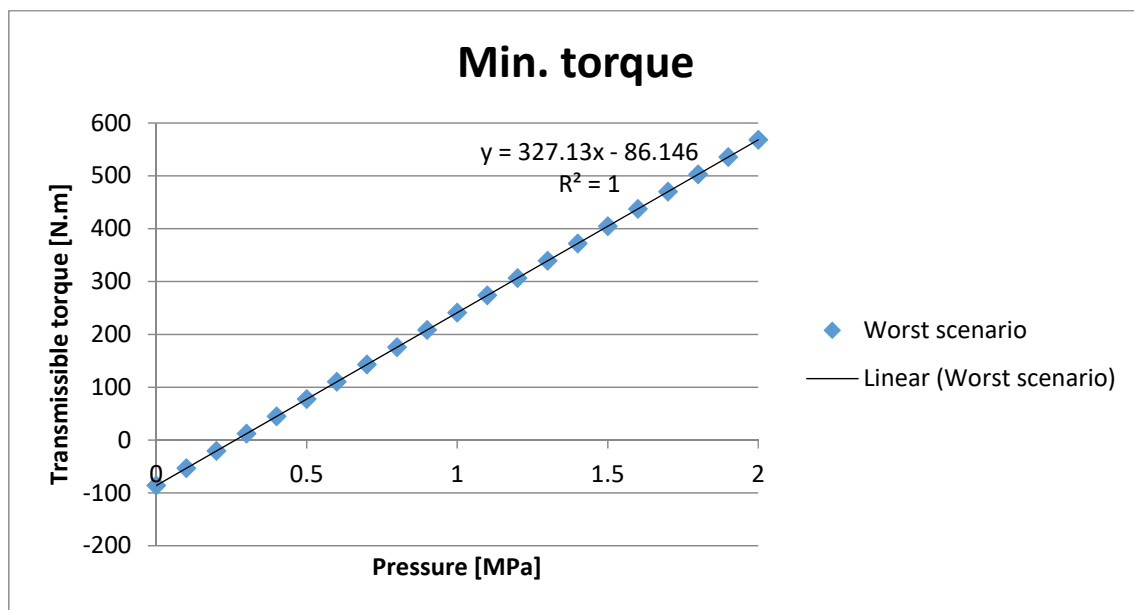


Figure 159 - Graph of minimum transmissible torque for second scenario of FWD clutch

The results regarding to the second scenario are expressed on Table 56.

Table 56 - Results of biting-pint conditions of FWD clutch set (second scenario)

Variable	Unit	Value
p	bar	2.878
F _{ax}	N	187.182

3.5.5.3.3 Third scenario

The method used for the third scenario is the same used for the calculations of the nominal scenario but by taking in more factors that can influence the amount of transmissible torque.

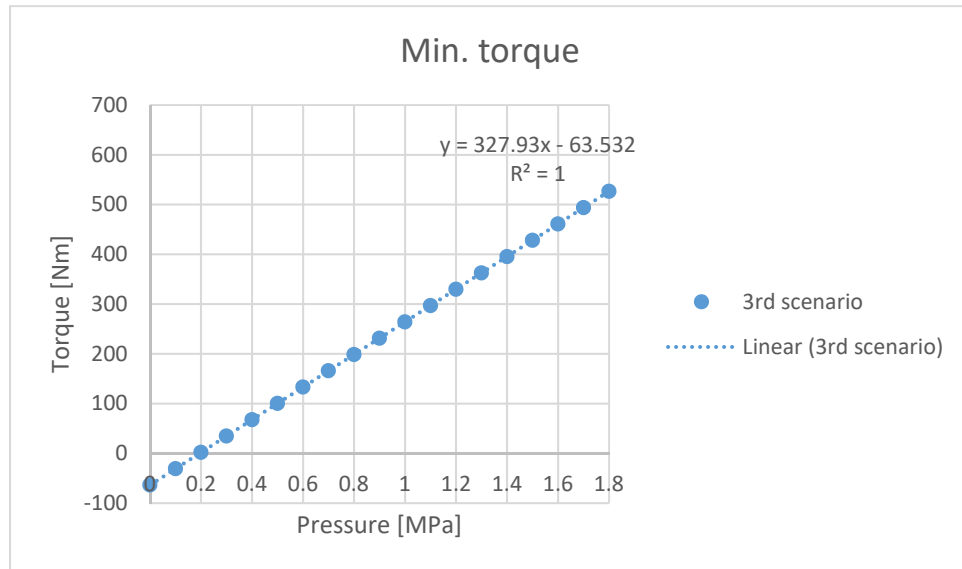


Figure 160 - Graph of minimum transmissible torque for third scenario of FWD clutch

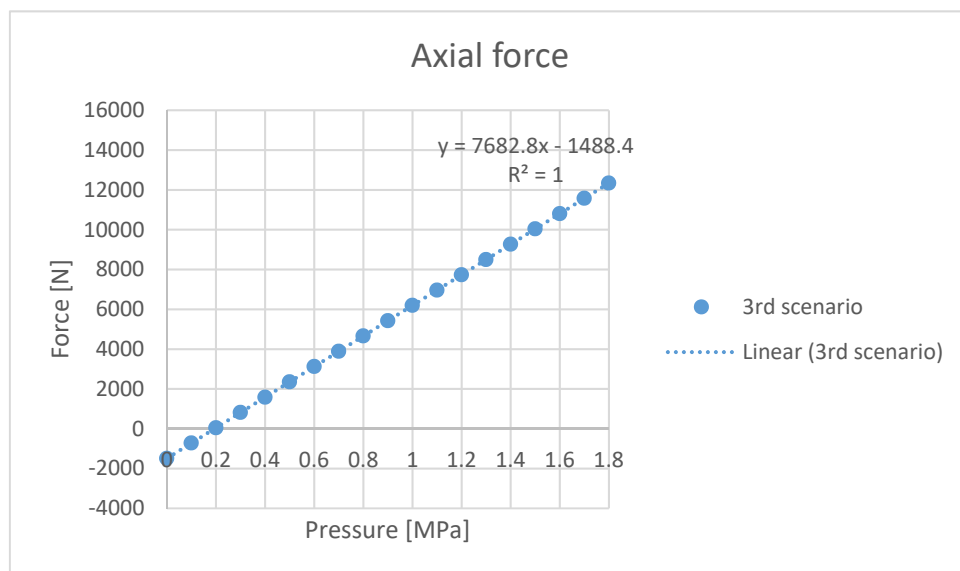


Figure 161 - Graph of axial force vs pressure for 3rd scenario of FWD clutch

The results regarding to the second scenario are expressed on Table 57.

Table 57 - Results of biting-pint conditions of FWD clutch set (third scenario)

Variable	Unit	Value
p	bar	2.181
F _{ax}	N	187.426

3.5.5.3.4 Fourth scenario

The method used for the fourth scenario is the same used for the calculations of the nominal scenario but by taking in more factors that can influence the amount of transmissible torque.

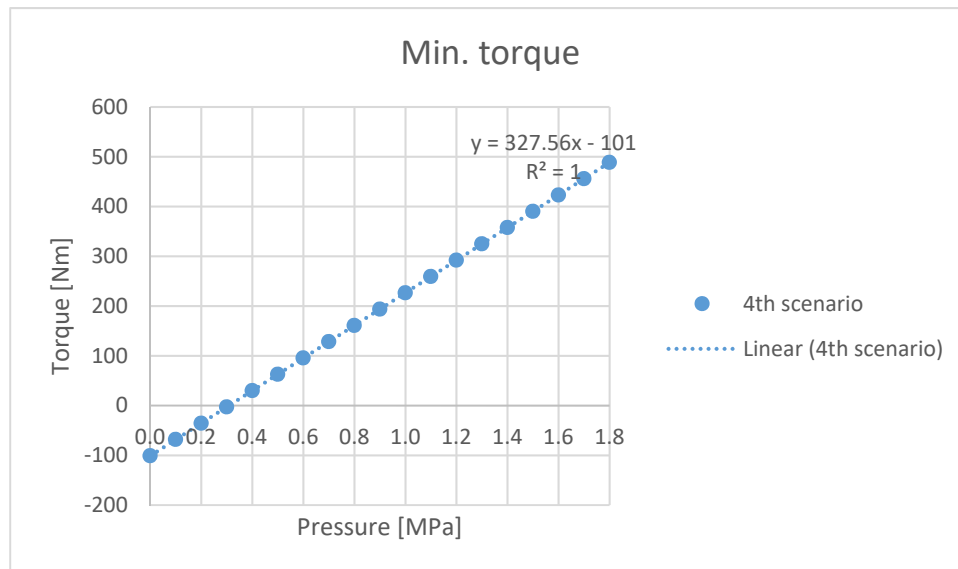


Figure 162 - Graph of minimum transmissible torque for fourth scenario of FWD clutch

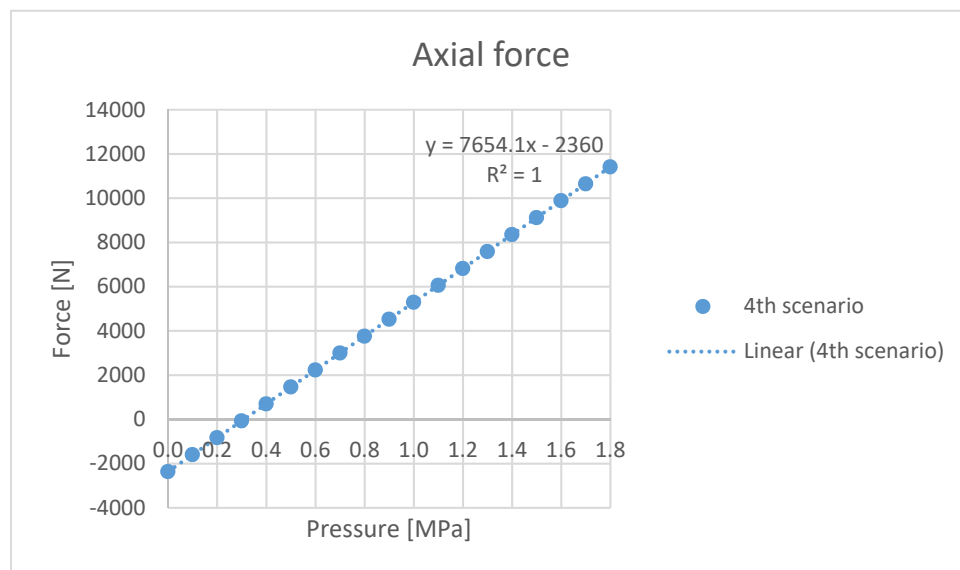


Figure 163 - Graph of axial force vs pressure for 4th scenario of FWD clutch

The results regarding to the second scenario are expressed on the table below.

Table 58 - Results of biting-pint conditions of FWD clutch set (fourth scenario)

Variable	Unit	Value
p	bar	3.328
F _{ax}	N	186.938

3.5.5.4 Drag losses

3.5.5.4.1 Model validation

In order to study the drag losses of the DNR subsystem, it was necessary to build a mathematical model in an Excel® worksheet. This makes it easier to change values, build charts and verify what is the influence of some parameters.

Before applying the parameters' values of the real subsystem, it was mandatory to validate the mathematical model. To do that, values of the article (Heyan *et al.*, 2013) were inserted in the model and then the obtained charts were compared to the ones illustrated on the article.

All the charts must be carefully analysed in order to detect any differences. It is important to state that the charts presented on the article were obtained by simulation which can return some different values due to friction of test rig components or vibrations on sensors, for example.

On Figure 165, it is illustrated a chart of the needing feed flow rate for full oil film with speed. The input data can be found on Table 8.

Comparing the chart of Figure 165 and Figure 43 it is evident that the parabolic curves for the different pressures are not so distinguishable as on the chart of Figure 43. The values of flow rate at 2500 rpm is the same for both. In conclusion, the chart of Figure 165 is acceptable.

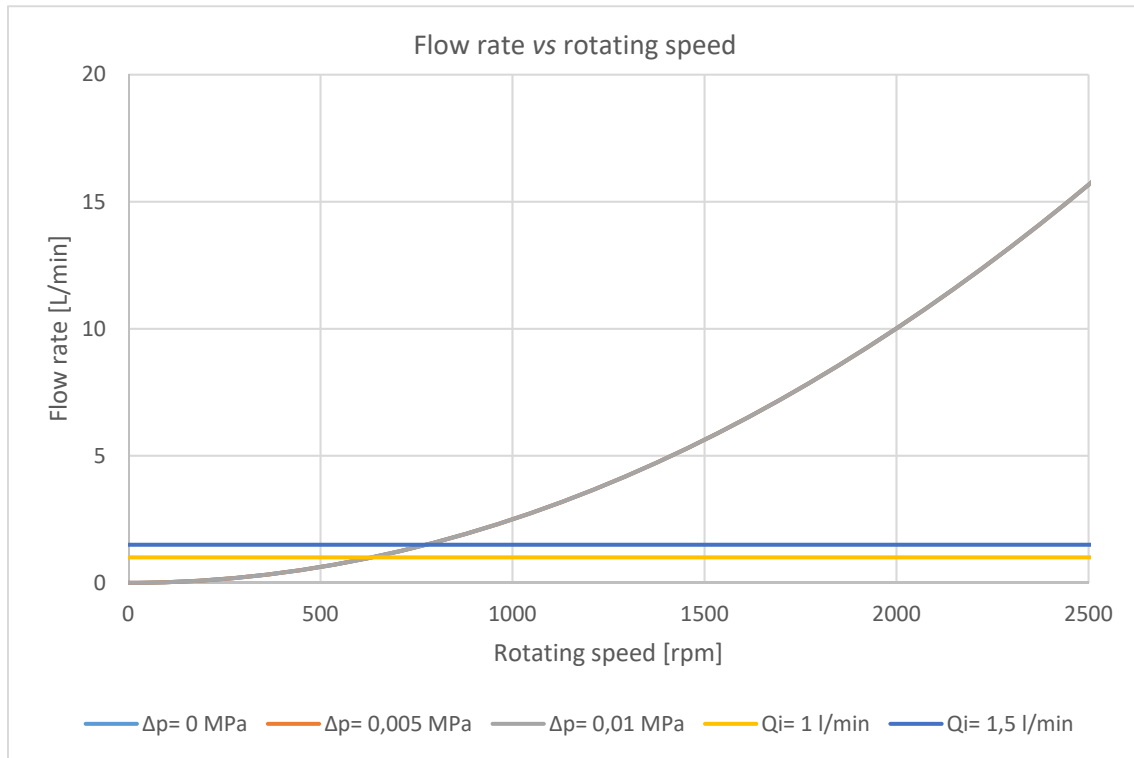


Figure 165 - Needing feed flow rate curve for full oil film of drag losses' model validation

On Figure 164 it is illustrated a graph of equivalent radius with rotating speed. The input data used to build this chart is the same used to build the chart of Figure 165.

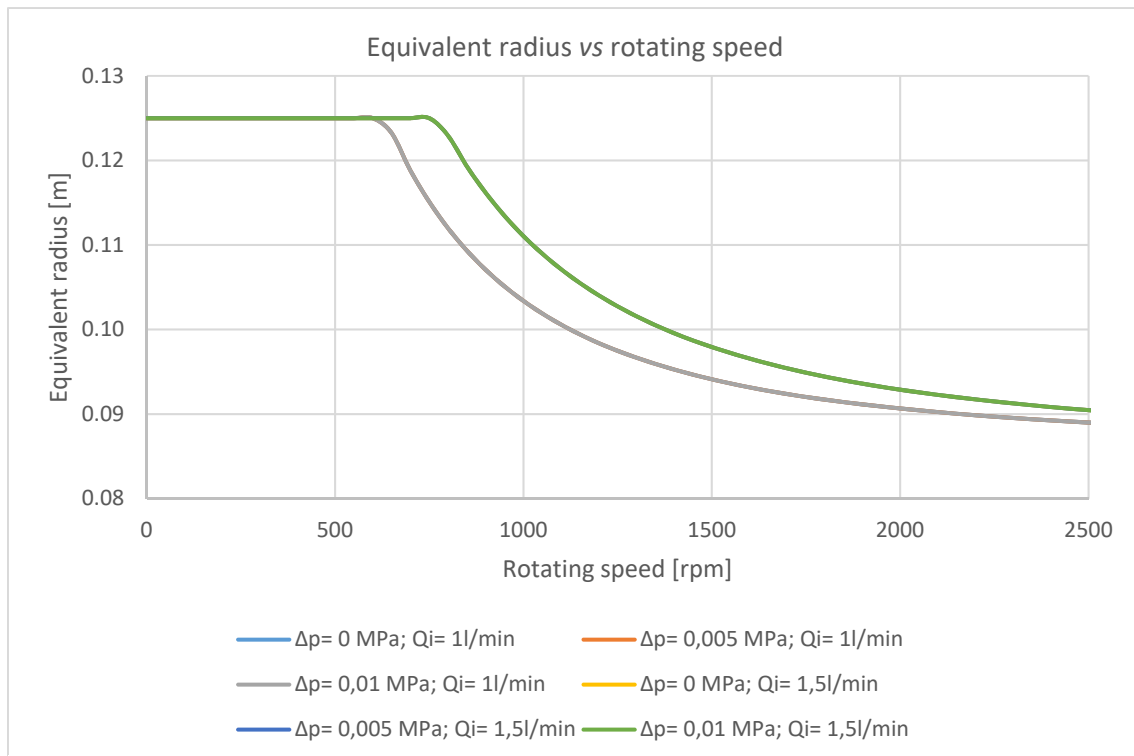


Figure 164 - Equivalent radius curve with speed of drag losses' model validation

Comparing chart of Figure 164 and the chart of Figure 45 it is evident that on the graph of Figure 45 the different curves can be easily distinguished. On the other hand, on the chart of Figure 164, it can be distinguished two groups for the two different feed flow rates. The values of equivalent radius until the beginning of the shrinking phenomenon is the same for both charts as so the values for 2500 rpm. In conclusion, the obtained graph of Figure 164 is acceptable.

On Figure 166, it is presented a chart of drag torque of a single pair with rotating speed. The input data of this chart is the same used for the previous charts.

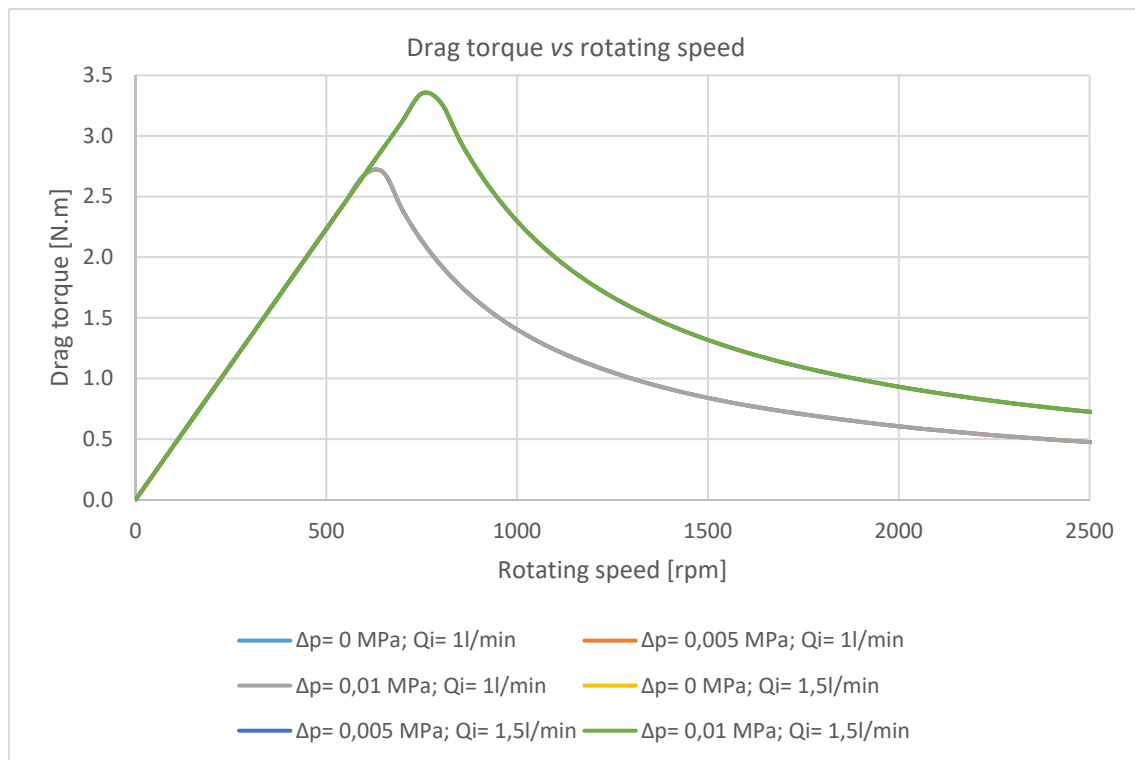


Figure 166 - Drag torque of single pair of drag losses' model validation

Comparing the graphs of Figure 166 and Figure 46, as happened for the previous graphs, for the graph obtained by simulation it is possible to distinguish all the curves. As happened for graph of Figure 164, on graph of Figure 166 the curves are organized in two groups according to feed flow rate. However, the maximum values of drag torque and the values of drag torque for 2500 rpm are quite accurate. In conclusion, the graph obtained on Figure 166 is acceptable.

Having in account that all the charts returned very good results and that all the charts were considered acceptable, it is conclusive that the mathematical model developed on Excel® is validated for the study of the real subsystem.

3.5.5.4.2 Data analysis

On this chapter, it will be studied the drag losses on FWD clutch. All the calculations methods and equations are presented on chapter 2.2.4.

During this chapter, several parameters will be changed in order to see what is its effect on the drag losses.

The nominal feeding rate flow of the DNR subsystem is 0.5 L/min., however the oil flow is not dedicated to clutches cooling. In fact, it is expected that only 60% of the rate flow is for clutches cooling purposes and the other 40% is for lubricant purposes of gears, bearings and so on. The maximum feeding flow rate is 1L/ min and it is only applied for short periods of time. This means that the nominal value is 0.3 L/min on the clutch plates.

Table 59 - Flow rate conditions of FWD clutch

Designation	Unit	Value
Nominal feeding flow rate reaching DNR subsystem	L/min	0.5
Nominal feeding flow rate for cooling of clutch plates	L/min	0.3
Maximum feeding flow rate reaching DNR subsystem	L/min	1
Maximum feeding flow rate for cooling of clutch plates	L/min	0.6

In order to understand the evolution of the different parameters, it was built several tables. The first series of tables are organized on the following way: on the first column it is presented the rotating speed of CC from 0 to 9000 in steps of 50 rpm; on the second column is presented the feeding flow rate; on the third and fourth columns are presented the outer and inner diameter pressures, respectively; on fifth column it is calculated the pressure variation between inner and outer radius; on sixth column it is calculated the necessary feeding flow rate for full oil film; and finally drag torque. An example of this table is presented on chapter 6.15. On the first series of tables it was considered the following feeding flow rates:

- 0.3 L/min;
- 0.5 L/min;
- 0.6 L/min.

Below, it is illustrated a graph of the relation between necessary flow rate to ensure full oil film and rotating speed.

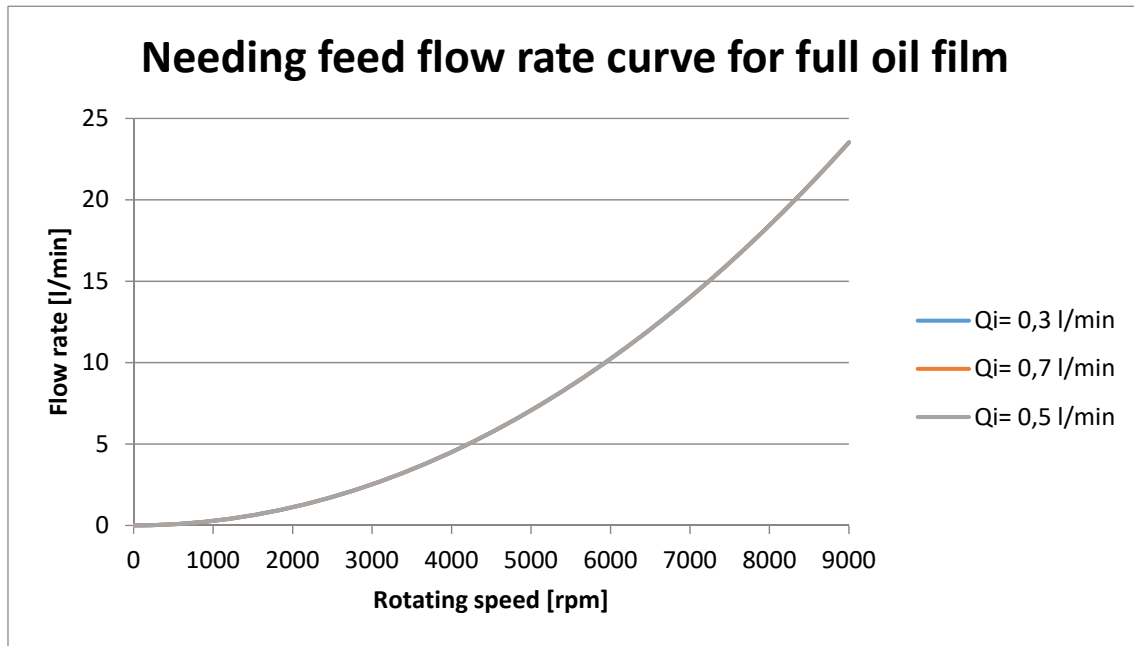


Figure 167 – Graph of needing feed flow rate for full oil film

According to the graph of Figure 167, input flow rate has no effect on needing feed flow rate for full oil film. Based on the same figure it is also possible to conclude that the greater the rotating speed, the greater the needing feed flow rate to ensure full oil film. The curve for the three values of input flow rate are overlapping each other.

On Figure 168, it is illustrated the evolution of pressure on inner and outer diameter and pressure variation for an input flow rate 0.3 L/min.

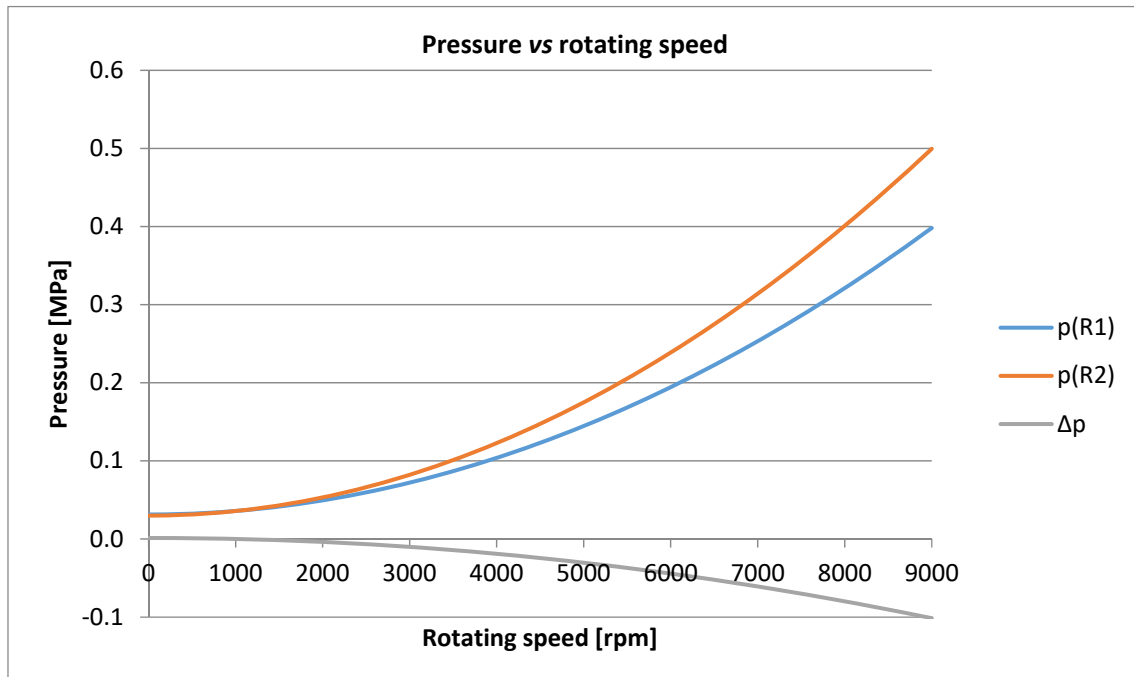


Figure 168 - Pressure portions vs rotating speed for 0,3 L/min input flow rate

According to Figure 168, when rotating speed increases, so does outer and inner diameter pressures. In contrast, the difference of pressures decreases. This is expected taking in account the equation of pressure difference and that the value of pressure on R_2 is most of the times greater than the pressure on R_1 .

On Figure 169 it is illustrated a graph similar to the of Figure 168 but for different values of input flow rate.

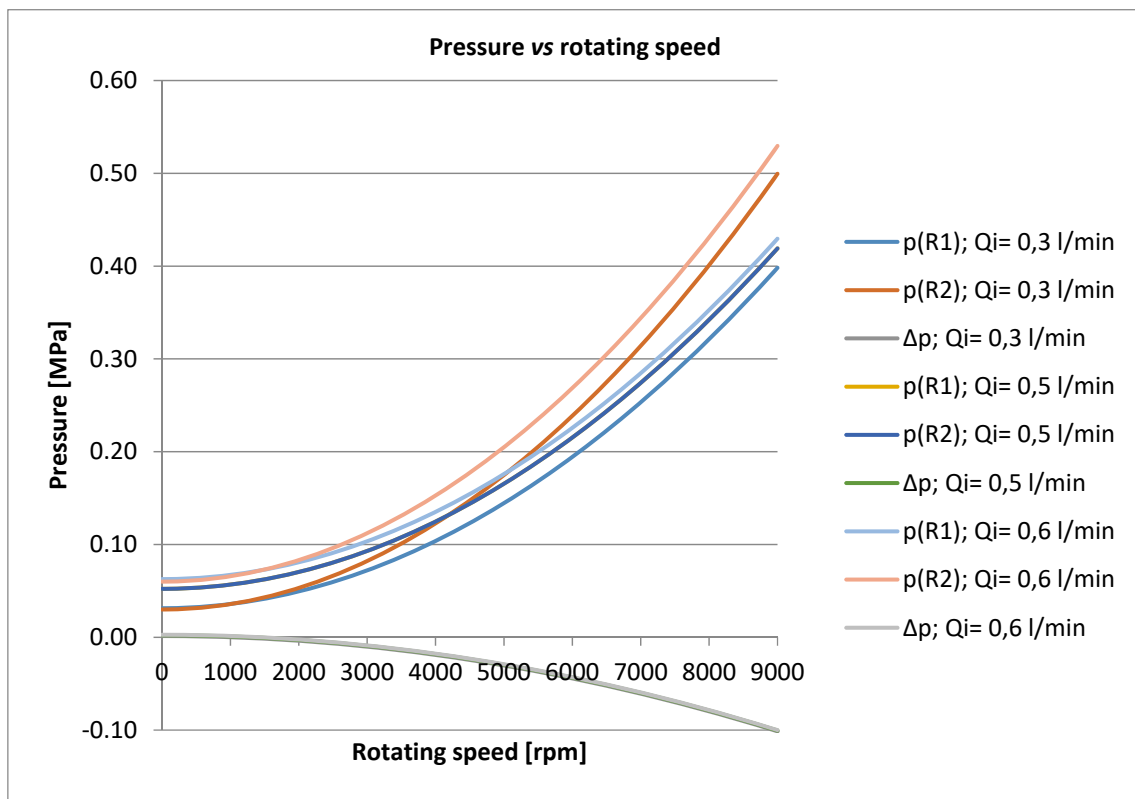


Figure 169 - Pressure portions vs rotating speed for different input flow rates

According to Figure 169, there is a visible difference between the curves of $p(R_1)$ and $p(R_2)$ for the different flow rates. Although, on the curves of Δp it is not so visible that difference. It is conclusive that all the curves' shapes are very similar.

On the graph of Figure 171, it is illustrated the drag losses of a single plate for different input flow rates. According to the graph of Figure 171 it is conclusive that there is not a direct relation between rotating speed and drag torque. On the other hand, there is a direct relation between drag torque and input flow rate: the higher the input flow rate is, the higher the drag torque is. This is only applied on the region where the input flow rate is lower than the necessary to ensure full oil film. Until that point the drag torque is equal independently of the input flow rate.

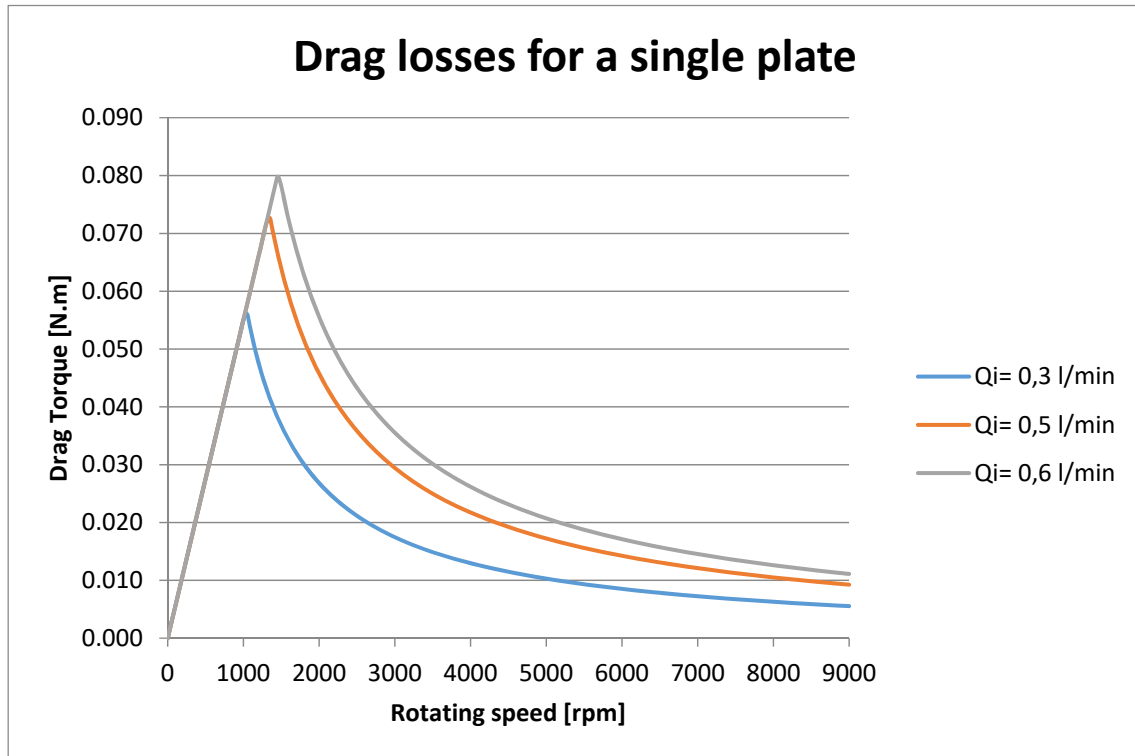


Figure 171 – Drag losses of a single plate on FWD clutch

On Figure 170 it is presented the drag torque of all plates for FWD clutch.

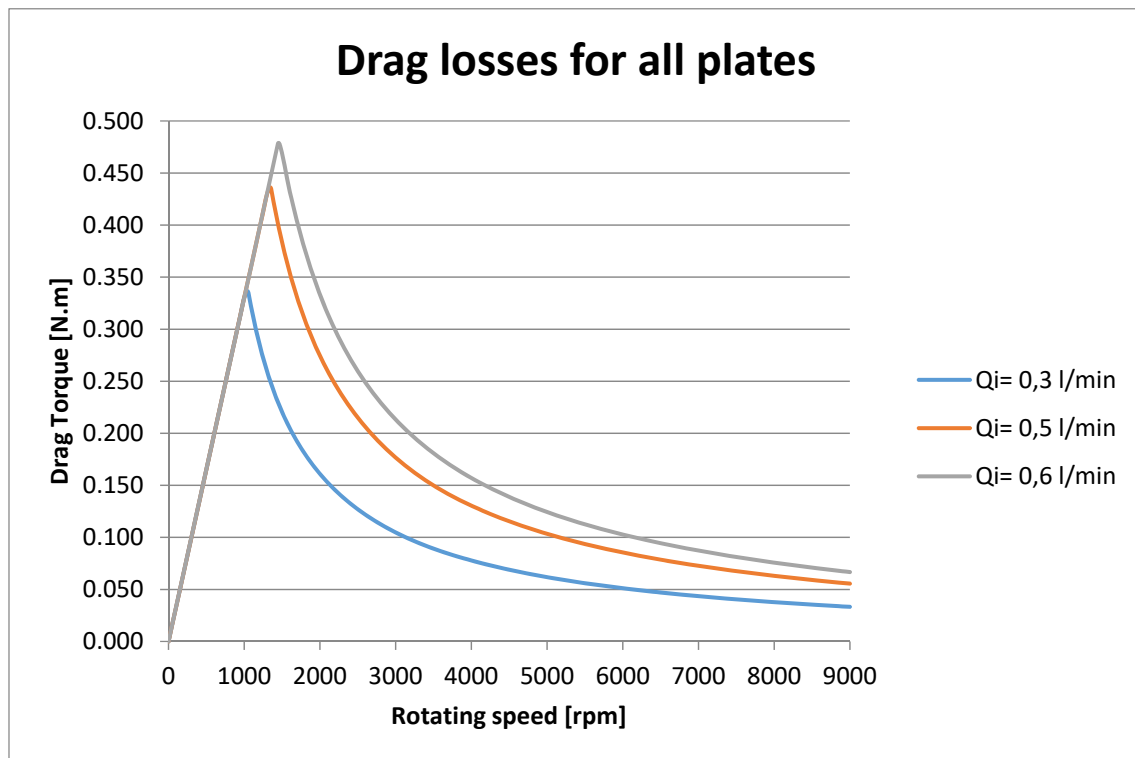


Figure 170 – Drag losses for all plates of FWD clutch

All the conclusions taken for the graph of Figure 171 are applicable to the graph of Figure 170. On Figure 172, it is presented a graph comparing the relation between needing feed flow rate and rotating speed for different values of Δp on FWD clutch.

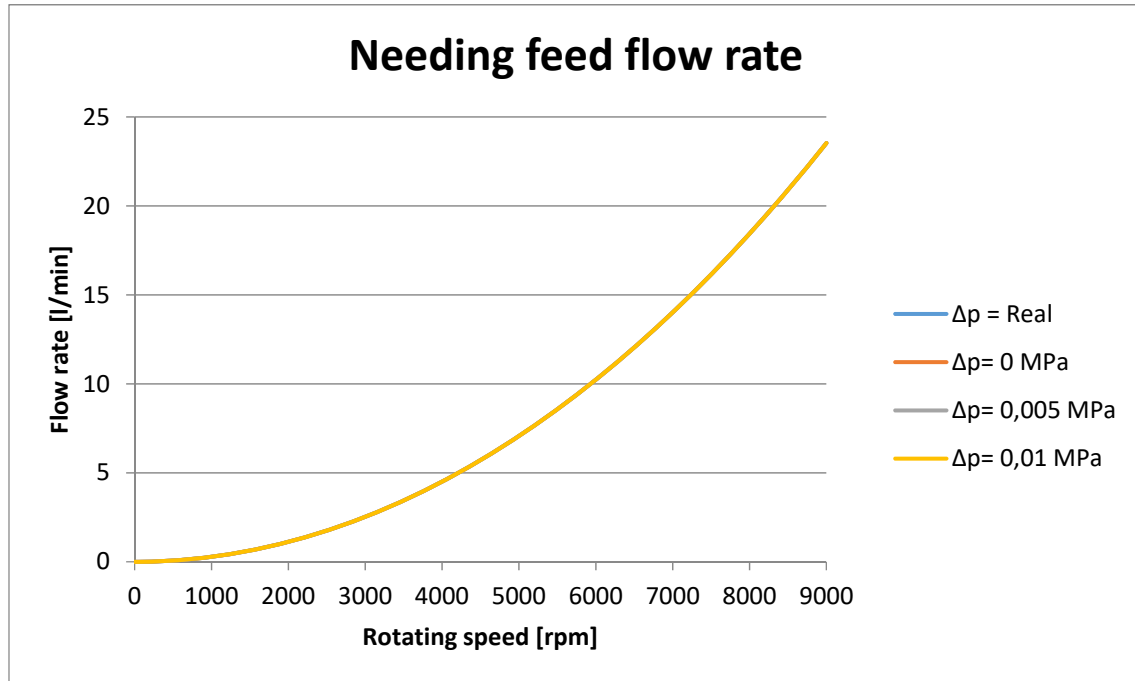


Figure 172 – Needing feed flow rate full oil film for different Δp

According to Figure 172 it is evident that there is no relation between Δp and needing feed flow rate for full oil film. The relation between rotating speed and feed flow rate was already studied above.

On Figure 173, it is presented a graph relating the equivalent radius and rotating speed for different feed flow rates. According to Figure 173, until a certain value of rotating speed the equivalent radius is common to every curve and the curve is constant and horizontal. At a certain value of rotating speed, the curves start decreasing significantly. The value of rotating speed where there is a change on the curve's shape corresponds to the value of the rotating speed where the feed flow rate is not enough to ensure full oil film. As seen before this value of rotating speed depends on the feed flow rate.

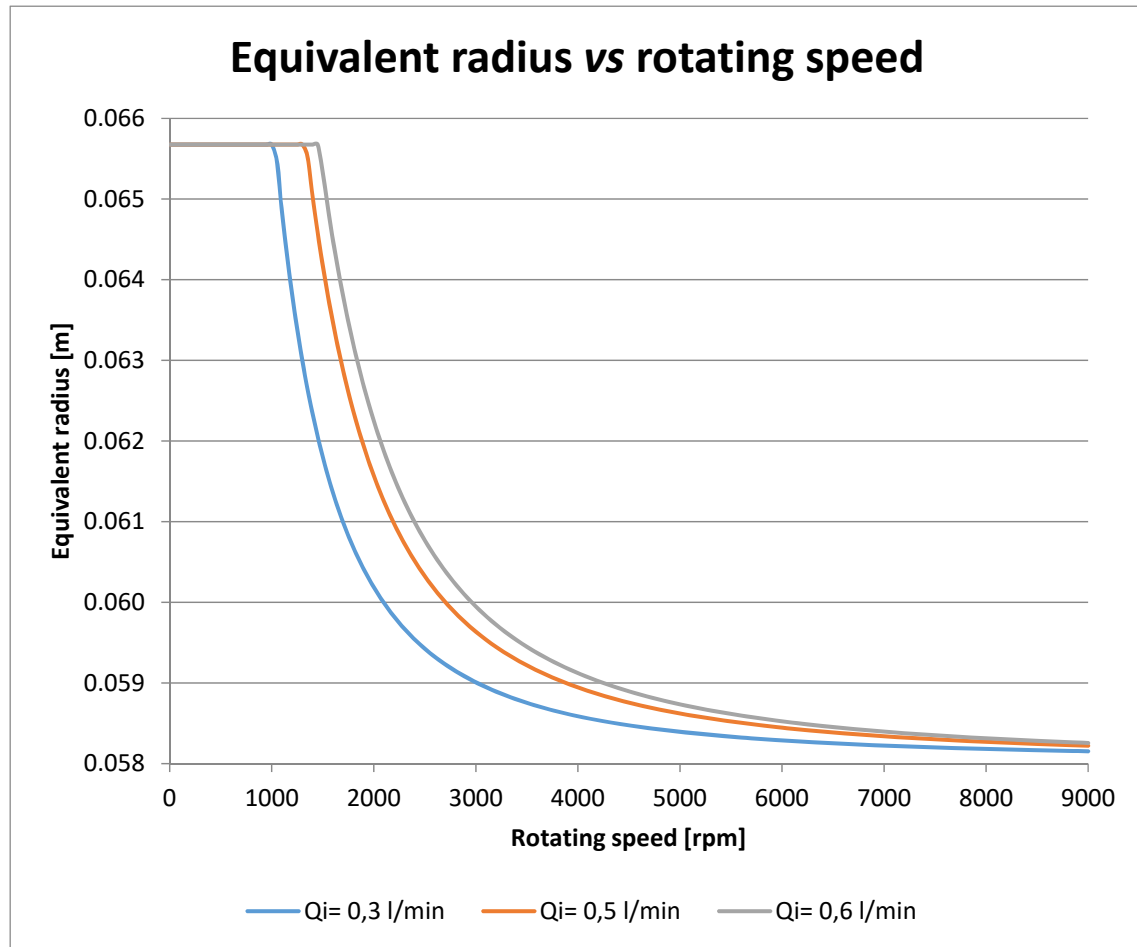


Figure 173 - Equivalent radius vs rotating speed for different feed flow rates

The second series of table is organized exactly on the same way as the first one, but the value of pressure variation was also introduced and not calculated. The values for feeding flow rate were considered to be always 0.3 L/min. An example of this table is presented on chapter 6.16. On the second series of tables it was considered the following pressure variations:

- 0 MPa;
- 0.005 MPa;
- 0.010 MPa.

On the graph of Figure 174, it is presented the relation between rotating speed and equivalent radius for different value of Δp . According to Figure 174 the value of Δp has no effect on the equivalent radius. The effect of rotating speed on equivalent radius was already explained.

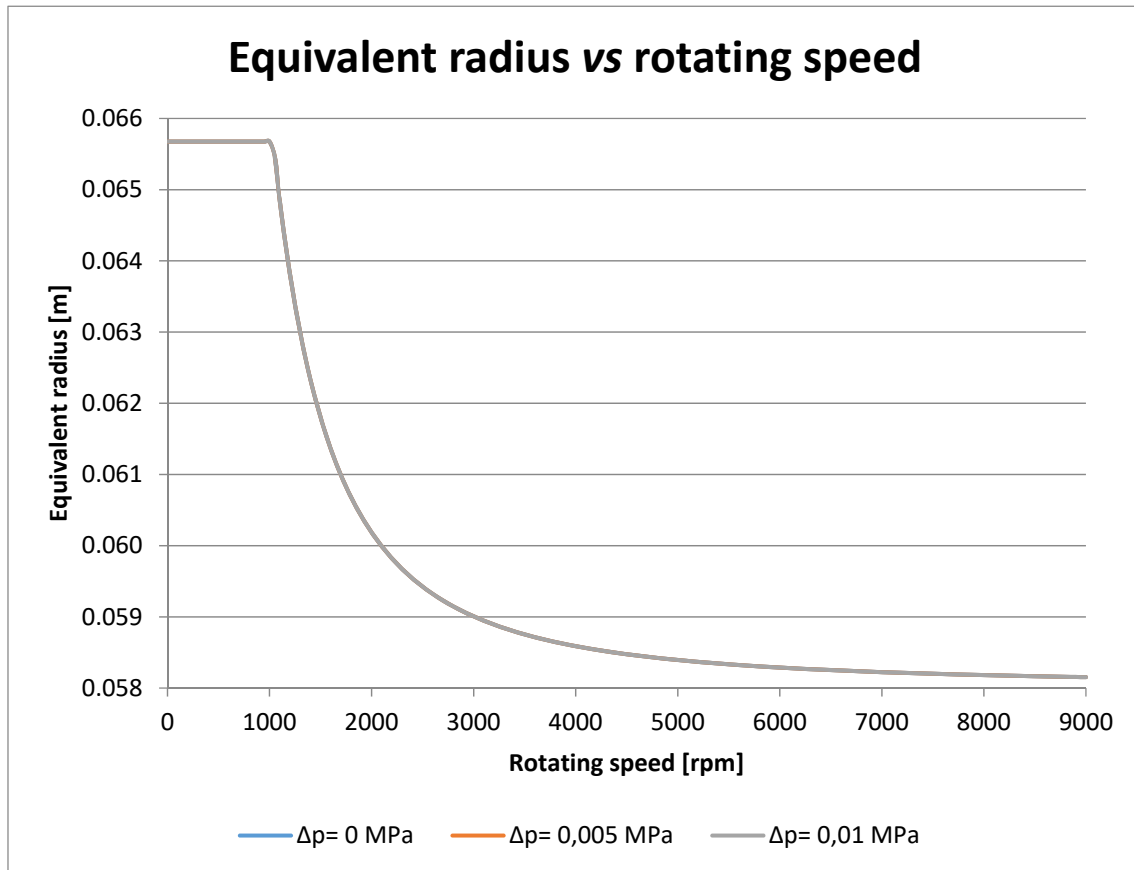


Figure 174 – Equivalent radius vs rotating speed for different values of Δp

The third series of tables is the most approximate to the real conditions due to the fact that there are two inputs. The first one is the oil temperature and the second one is the feed flow rate. The value of feed flow rate was defined as 0.3 L/min and the temperature can have different values:

- -20 °C;
- 70 °C;
- 90 °C;
- 110 °C.

An example of this table is presented on chapter 6.17. The main purpose of this series of tables is to study the effect of oil temperature on the different calculated parameters.

It is important to underline the fact the most common range of working temperatures varies from 70 to 90 °C.

On Figure 175, it is presented the relation between needing feed flow rate for full oil film vs rotating speed for different oil temperatures.

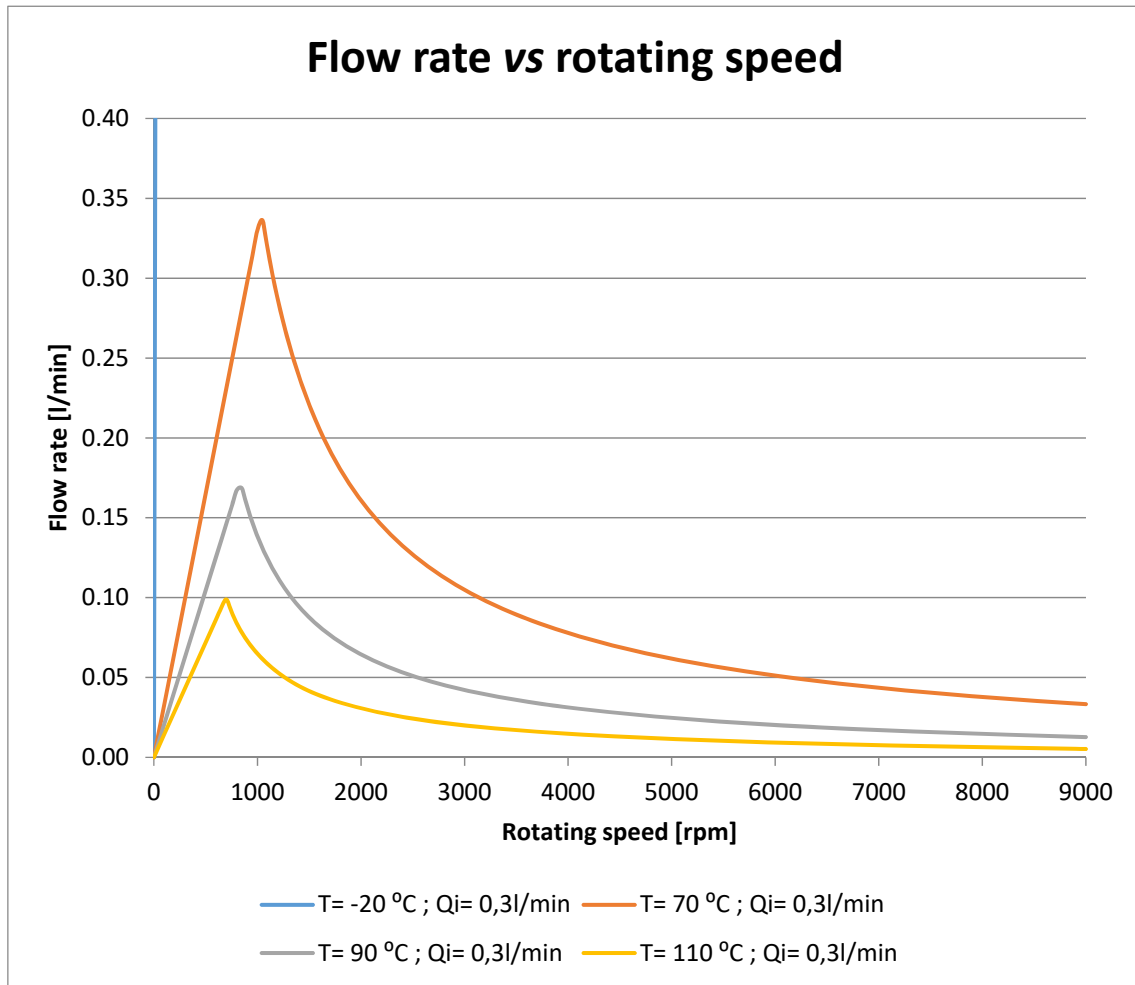


Figure 175 - Needing feed flow rate for full oil film vs rotating speed for different oil temperatures

Based on the graph of Figure 175, it is possible to conclude that the rise of temperature has a direct effect on needing feed flow rate. The higher the temperature is, the lower feed flow rate needs to be. It is also visible that the needing feed flow rate rises constantly until a certain point. After that point, the needing feed flow rate decreases until almost zero. This point is already known. The shape of the curves for 70, 90 and 110 °C are similar, however the shape for the curve of -20 °C is almost a vertical line.

On Figure 176, it is presented a graph of the dissipated power due to the drag torque vs rotating speed for different oil temperatures. The shape of the curves of the graph of Figure 176, is very similar to the ones of the graph of Figure 175. Although at higher rotating speed, the value of the dissipated power is not close to zero. For higher oil temperatures, the dissipated power at higher rotating speeds is significantly higher. At higher rotating speed, the value of dissipated power smoothly decreases.

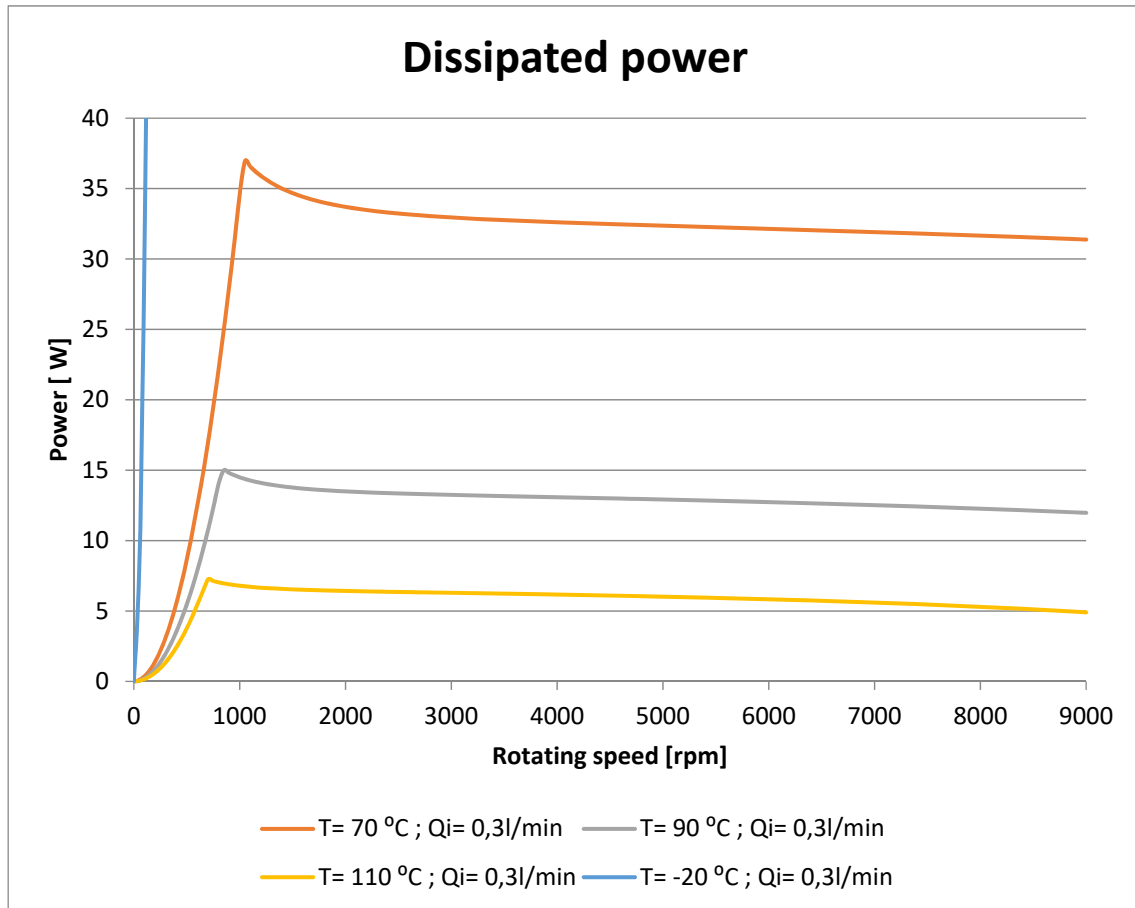


Figure 176 - Dissipated power vs rotating speed for FWD clutch

3.5.5.4.3 Working parameters

Rotating speed

Based on Figure 168, it is evident that rotating speed has an effect on pressures. The greater the rotating speed is, the greater the outer and inner diameter pressure are. On the other hand, pressure difference decreases for higher rotating speed.

Input feed flow rate

As previously seen on the graph of Figure 167, input flow rate has not any effect on needing feed flow rate for full oil film.

Pressure variation

Based on the graph of Figure 166, it is conclusive that pressure variation has no effect on the drag losses. In fact, in some literature pressure variation is considered null due to its null effect on the drag losses.

Oil temperature

According to the graph of Figure 177, it is conclusive that the oil temperature has a considerable effect on the amount of drag torque originated.

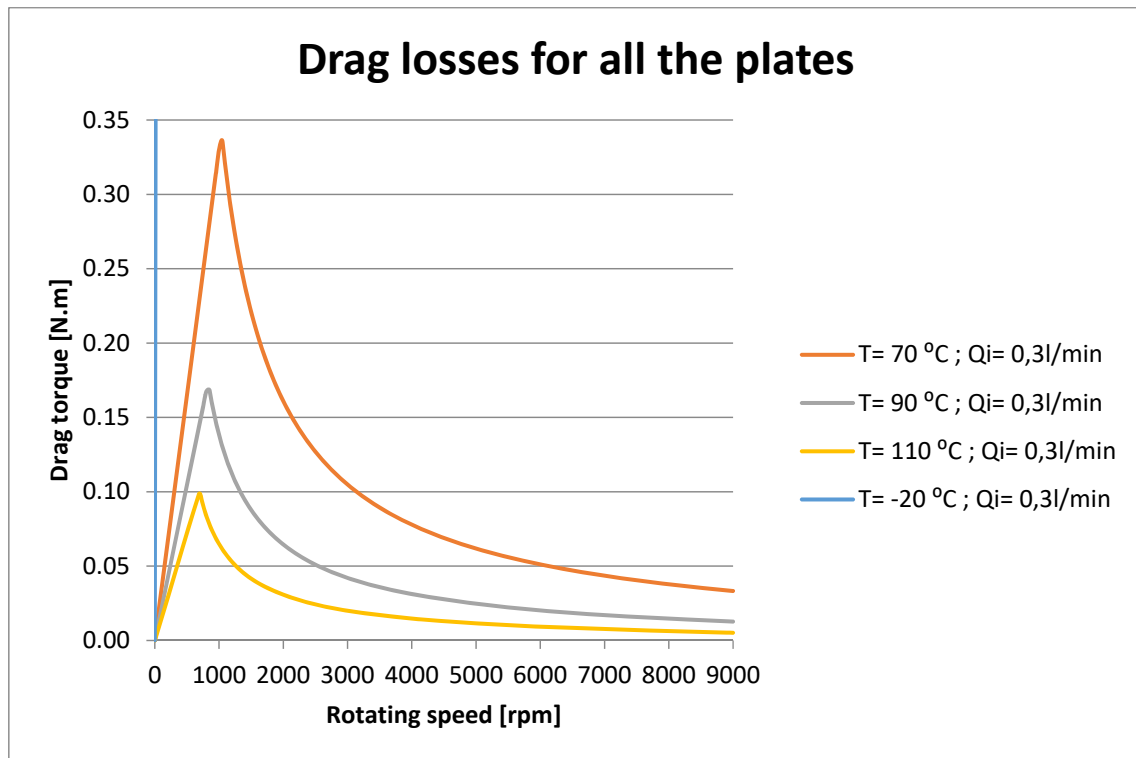


Figure 177 – Influence of oil temperature in the drag torque

3.5.6 Clutch plates

3.5.6.1 FWD clutch

One key factor related to the clutch plates is the pressure applied on them. So, to find out that value, it is necessary to find out the value of the total friction area:

$$A_F = A_F^{pad} \cdot N_p \left[mm^2 \right] \quad \{3.26\}$$

All the necessary values are presented on Table 146. The value of the pressure applied on the clutch plates can be obtained, as follows:

$$p^{CP} = F^{CP} \cdot A_F^{FWD\ CP} \quad \{3.27\}$$

Once again, it was considered the nominal and second scenario. The friction area of the clutch plates assumed for the third and fourth scenarios are presented on Table 53 and Table 54, respectively.

The results are presented on the following table.

Table 60 – Results for FWD clutch plates' pressure

Variable	Unit	Value
A_F	mm^2	1817.584
p^{CP} (nominal case)	bar	52.881
p^{CP} (second scenario)	bar	52.078
p^{CP} (third scenario)	bar	47.015
p^{CP} (fourth scenario)	bar	49.535

In this case, for the third scenario it was considered the maximum friction area. As seen before, third scenario is the one which presents more force applied on the clutch plates.

In order to simulate the worst possible scenario in terms of wear and pressure on the clutch plates, it was applied the friction area of the fourth scenario (minimum friction area) in the third scenario. By this way it is possible to simulate the maximum force to the clutch plates with the minimum friction area in order to obtain the maximum pressure and maximum wear. The value obtained of pressure was approximately 54.8 bar.

On one hand, it is desirable that the pressure on the clutches is low in order to prevent too much wear. On the other hand, the lower pressure on the clutch plates, the lower the transmissible torque is. So, maybe, worst case is not the best expression to designate this scenario.

In the graph of Figure 178 it is presented a comparison between the clutch plates' pressure for the different scenarios.

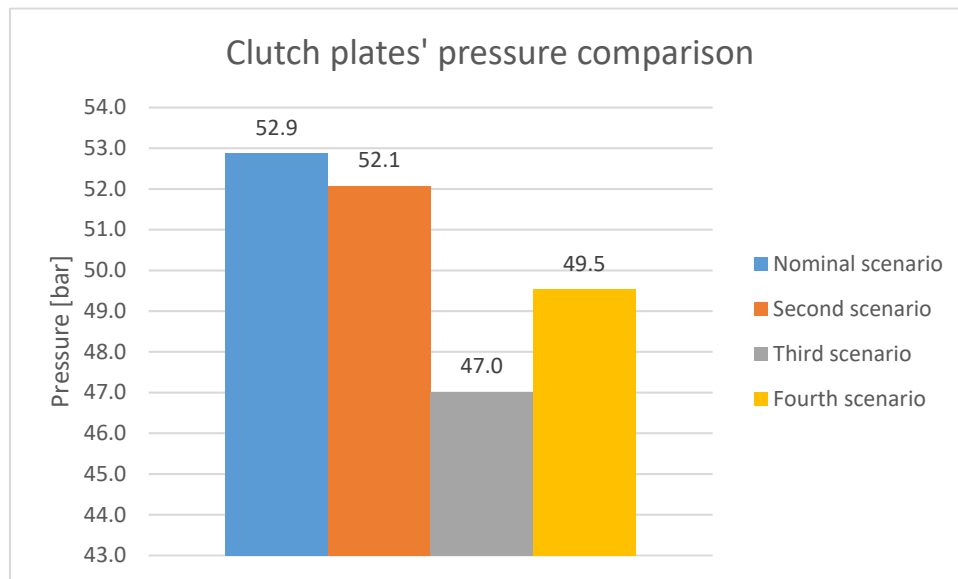


Figure 178 – Graph of comparison of clutch plates' pressure of FWD clutch set

According to the graph of Figure 178, as explained before, third scenario presents a lower pressure than fourth scenario. Nominal scenario it the scenario which presents the highest clutch pressure.

3.5.6.2 REV clutch

The method to calculate the pressure on the REV clutch plates is the same applied on the FWD clutch plates set.

The necessary values to calculate eq. {3.26} are expressed on Table 147. The friction area was considered the same for all the scenarios.

In Table 61, it is presented the results for the different scenarios.

Table 61 - Results for REV clutch plates' pressure

Variable	Unit	Value
p^{CP} (nominal case)	bar	56.120
p^{CP} (second scenario)	bar	57.130
p^{CP} (third scenario)	bar	55.111

In the graph of Figure 179, it is presented a comparison between the clutch plates' pressure for the different scenarios.

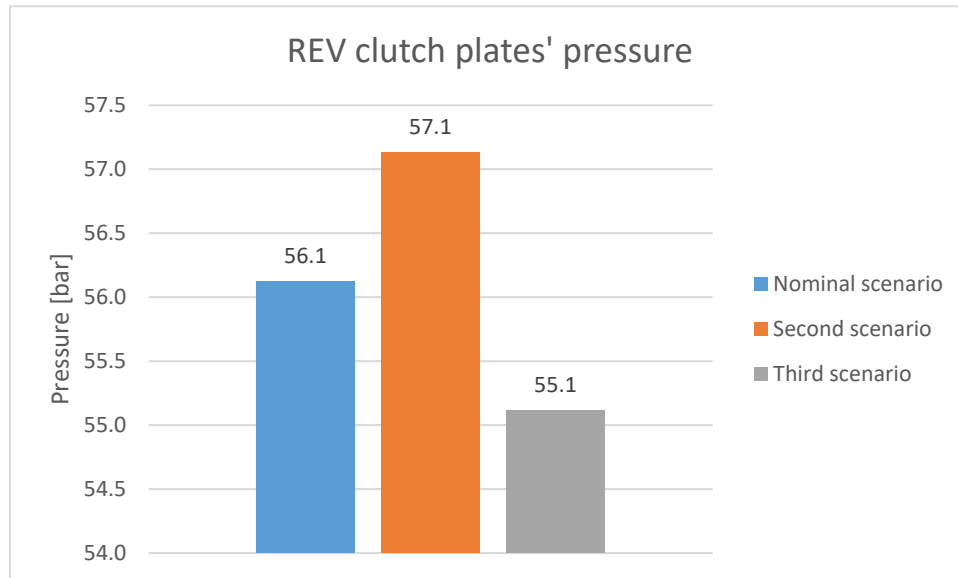


Figure 179 - Graph of comparison of clutch plates' pressure of REV clutch set

Since the friction area is the same for all the scenarios, it is expected that the pressure of the second scenario is higher than the third's because the force applied on the clutches is higher.

3.5.7 REV clutch set

The convention of signs used for REV clutch set calculations is identical to the one used for FWD system.

3.5.7.1 General calculations

The main difference in the general calculations between the REV and FWD clutch sets is the inexistence of centrifugal force on the REV clutch set because there is no rotational motion. For the following calculations it was considered a value of pressure of 15 bar and a rotational speed of 6500 rpm. This means that hydraulic area and respective force are calculated on the same way, according to eq. {3.16} and eq. {3.17}. All the necessary values can be found on Table 145.

As there is no centrifugal force so the force on the REV piston is calculated as follows:

$$F^{REVpiston} = F_H [N] \quad \{3.28\}$$

The schematics of the sprigns' association is similar to the FWD however the first stage compression is only 1.1 mm and instead of a wavy washer there is a disc spring.

The value of SR and F_{hmax} and $F_{maxhmax}$ can be found in Table 30. The value of SR^{SP} can be obtained on Table 30 and the value of SR^{BW} can be obtained on Table 43. The result of $SR_{||}$ is presented on Table 62.

Regarding to the second stage of the compression of the parallel association there are two possible hypotheses: the complete compression of the REV spring pack or the complete compression of the disc spring. In order to figure out what is the compression of the second stage of the parallel association it is necessary to calculate the maximum compression of each springs.

To calculate the travelling of the 2nd stage, eq. {3.23} must be used by replacing WS by BW (Belleville washer), resulting on:

$$\begin{cases} x_2^{SP} = d_{max}^{SP} - \Delta_{x1} \\ x_2^{BW} = FL^{BW} - b^{BW} \end{cases} [mm]$$

The value of the maximum compression of the REV SP can be found on Table 30. The values for the second equation of the equations' system can be found on Table 142.

The results of the general calculations are presented on Table 62.

As it can be seen, the lowest resulting value is 0.750 mm, that corresponds to the complete compression of the disc spring. The springs association cannot be compressed more than 0.750 mm on the 2nd stage, however the spring pack could still be compressed 1.700 mm but not the disc spring.

The main goal of the graph of Figure 180 is to better understand the spring's association. By means of this graph it is possible to understand at which point the disc spring starts to be compressed and what are the values of the spring stiffness at each of the two stages.

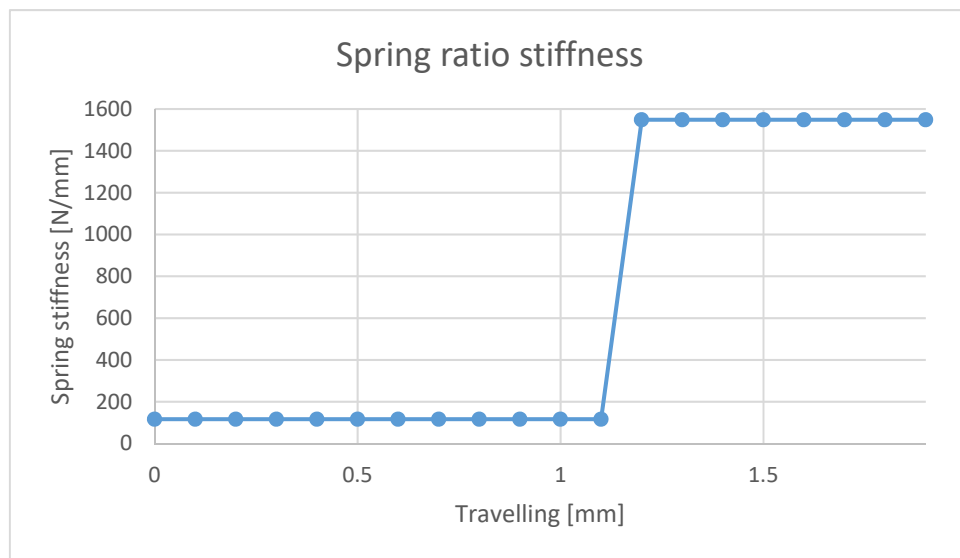


Figure 180 – Graph relating springs' association's stiffness and total travelling

As expected, until 1.1 mm of travelling, REV SP is being compressed which means that the spring stiffness on the graph corresponds to the spring stiffness of the REV SP. After that, Belleville washer starts to be compressed which means that the stiffness of the spring's association will increase significantly. The spring stiffness remains stable until approximately 1.9 mm travelling.

In the graph of Figure 181 it is possible to visualize what is the behaviour of the reaction force of the springs' association along the travelling.

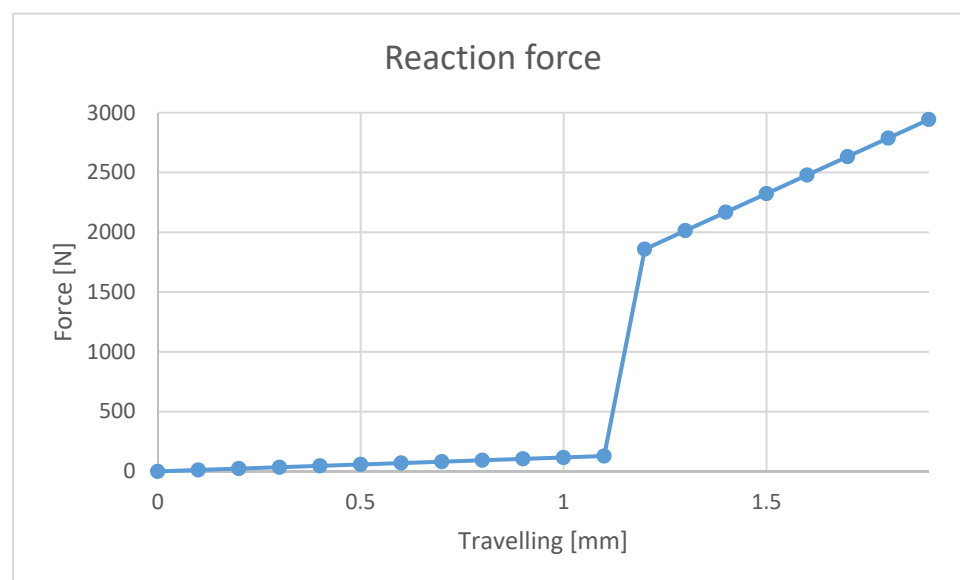


Figure 181 – Graph relating travelling and reaction force of the springs' association

The point where there is an instantaneous increase of the reaction force corresponds to the point where the BW starts to be compressed. At the second stage, it is expected to there be larger reaction forces due to the higher spring stiffness of the springs' association.

The table which was the base for the graphs of Figure 180 and Figure 181 can be found on chapter 6.21.

Table 62 – Results of general calculation of the REV clutch set (nominal scenario)

Variable	Unit	Value
$SR_{\parallel}$	N/mm	1548.518
$x2^{SP}$	mm	1.700
$x2^{BW}$	mm	0.750
RF_{x2}	N	1161.388

3.5.7.1.1 Nominal case

For the following calculations inclusive for the other scenarios it was considered a hydraulic pressure of 15 bar. For the calculations of the nominal scenario it was considered the nominal parameters.

The results of general calculations relative to the nominal scenario are expressed on Table 63.

Table 63 – Results of general calculations for REV clutch set (nominal scenario)

Variable	Unit	Value
A_H	mm^2	10687.698
F_H	N	16031.547
$F^{FWDpiston}$	N	16031.547
RF_{x1}	N	656.420
F_{x2}	N	15375.128
F^{CP}	N	14213.739

3.5.7.1.2 Second scenario

The values of the different parameters used for the calculations relative to the second scenario are presented on Table 64.

Table 64 – General calculations data input for REV clutch set (second scenario)

Variable	Unit	Value
OD_H^{piston}	mm	183.070
ID_H^{piston}	mm	140.930

The results of general calculations relative to the second scenario are presented on Table 65.

Table 65 - Results of general calculations for REV clutch set (second scenario)

Variable	Unit	Value
A_H	mm ²	10723.324
F_H	N	16084.986
$F^{\text{FWDpiston}}$	N	16084.986
RF_{x1}	N	588.389
F_{x2}	N	15496.597
F^{CP}	N	14469.325

3.5.7.1.3 Third scenario

The values of the different parameters used for the calculations relative to the third scenario are presented on Table 66.

Table 66 - General calculations data input for REV clutch set (third scenario)

Variable	Unit	Value
OD_H^{piston}	mm	182.930
ID_H^{piston}	mm	141.070

The results of general calculations relative to the third scenario are presented on Table 67.

Table 67 - Results of general calculations for REV clutch set (third scenario)

Variable	Unit	Value
A_H	mm^2	10652.073
F_H	N	15978.109
$F^{\text{FWDpiston}}$	N	15978.109
RF_{x1}	N	724.450
F_{x2}	N	15253.659
F^{CP}	N	13958.154

3.5.7.2 Transmissible torque

In this case the transmissible torque refers to the amount of torque that the clutch can hold when REV gear is selected. As explained before, when R is selected, clutch plates get close. Clutch plates do not rotate as so the ring gear. So, the clutch plates need to hold the torque transmitted by the ring gear. The transmissible torque will be calculated for three different scenarios.

The transmissible torque must be calculated according to eq. {2.63}. In addition to the nominal torque, it was also calculated the maximum and minimum torque. The difference between both is the coefficient of friction. The value of F_{ax} is equal to F_{CP} that can be found on Table 63, Table 65 and Table 67. The values of ζ can be found on Table 147 and N_s on Table 149. The values to calculate R_f (eq. {2.64}) can be found on Table 147.

The tables created for the three scenarios can be found on Table 119, Table 120 and Table 121 (Appendix).

3.5.7.2.1 Nominal scenario

The value of R_f for the nominal scenario is equal to 0.088 m.

In the graph of Figure 182 it is illustrated the relation between transmissible torque and hydraulic pressure of the nominal scenario. In the same graph it is presented the maximum, minimum and nominal torque. The maximum transmissible torque has in account the maximum value of the friction coefficient while in the minimum

transmissible torque is the opposite. The average transmissible torque takes into account the average value of the friction coefficient.

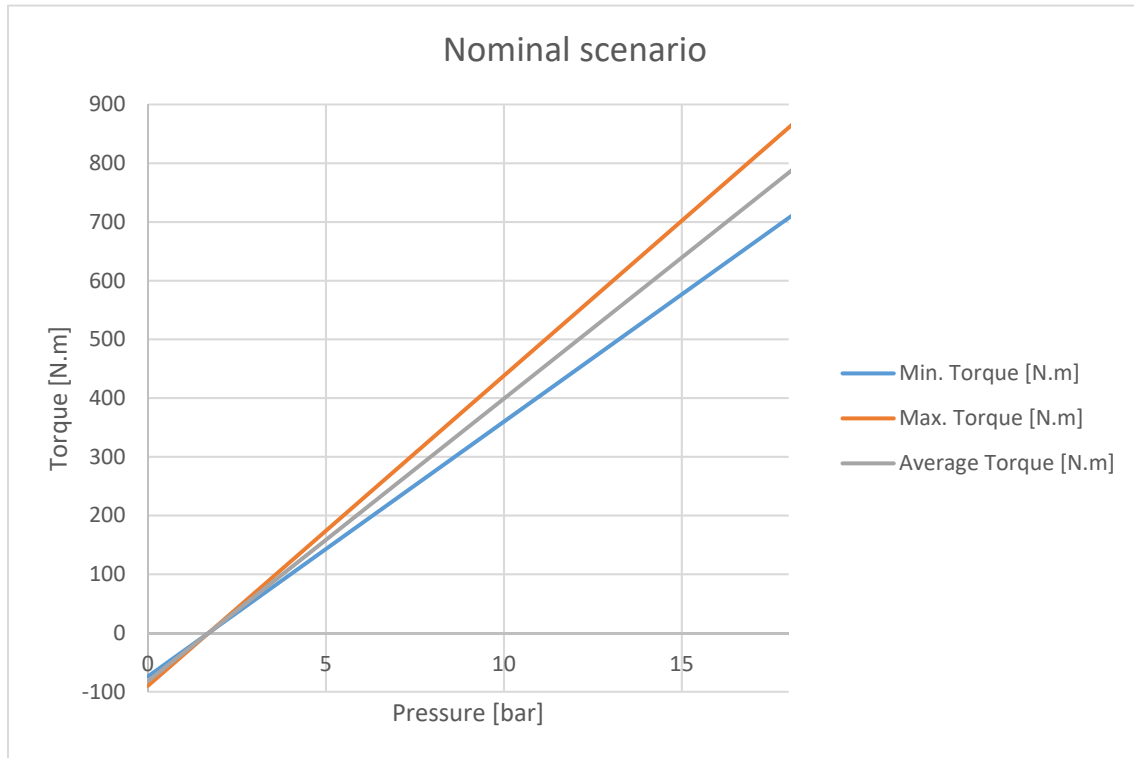


Figure 182 – Graph of transmissible torque vs pressure for the nominal scenario of REV clutch set

According to the graph of Figure 182, there is a significant difference between the three curves. In fact, the higher the pressure, the larger the difference is. At 15 bar, the difference between the curves is about 50 N.m which is a considerable value. As seen for the FWD clutch set, until approximately 2 bar the torque is negative which means that no torque is transmitted. The data of this graph was obtained analytically by calculations. It was not measured.

3.5.7.2.2 Second scenario

In the case of the second scenario, it was considered the tolerances of some parameters that is known will further the amount of transmissible torque. In Figure 183 it is illustrated a graph with the three different cases of transmissible torque for the second scenario.

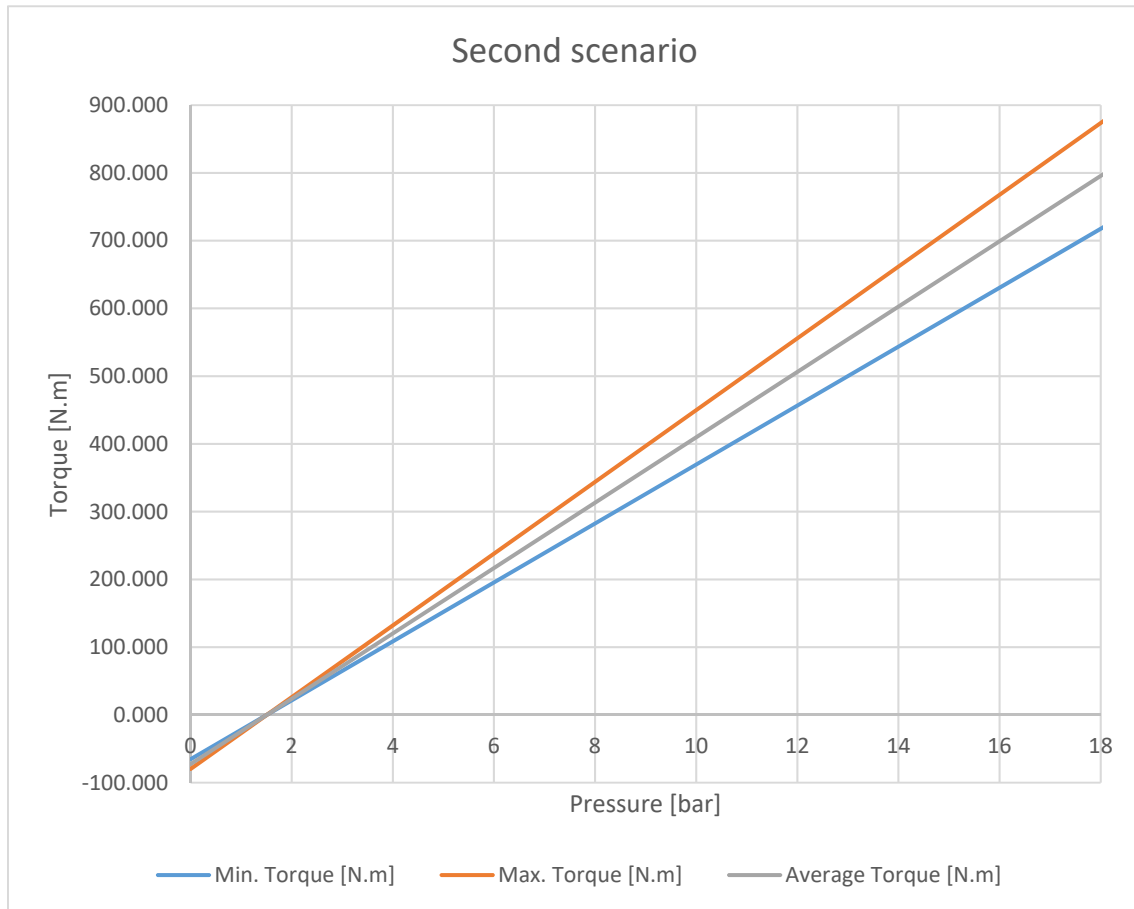


Figure 183 - Graph of transmissible torque vs pressure for the second scenario of REV clutch set

Based on graph of Figure 183, it is conclusive that there is a significant difference between the three curves. At 15 bar, the difference is slightly greater than 50 N.m. The amount of transmissible torque is greater when compared to the nominal scenario as so the difference between curves.

3.5.7.2.3 Third scenario

In the case of the third scenario, it was considered the tolerances of some parameters that is known will prejudice the amount of transmissible torque. In Figure 184 it is illustrated a graph with the three different cases of transmissible torque for the third scenario.

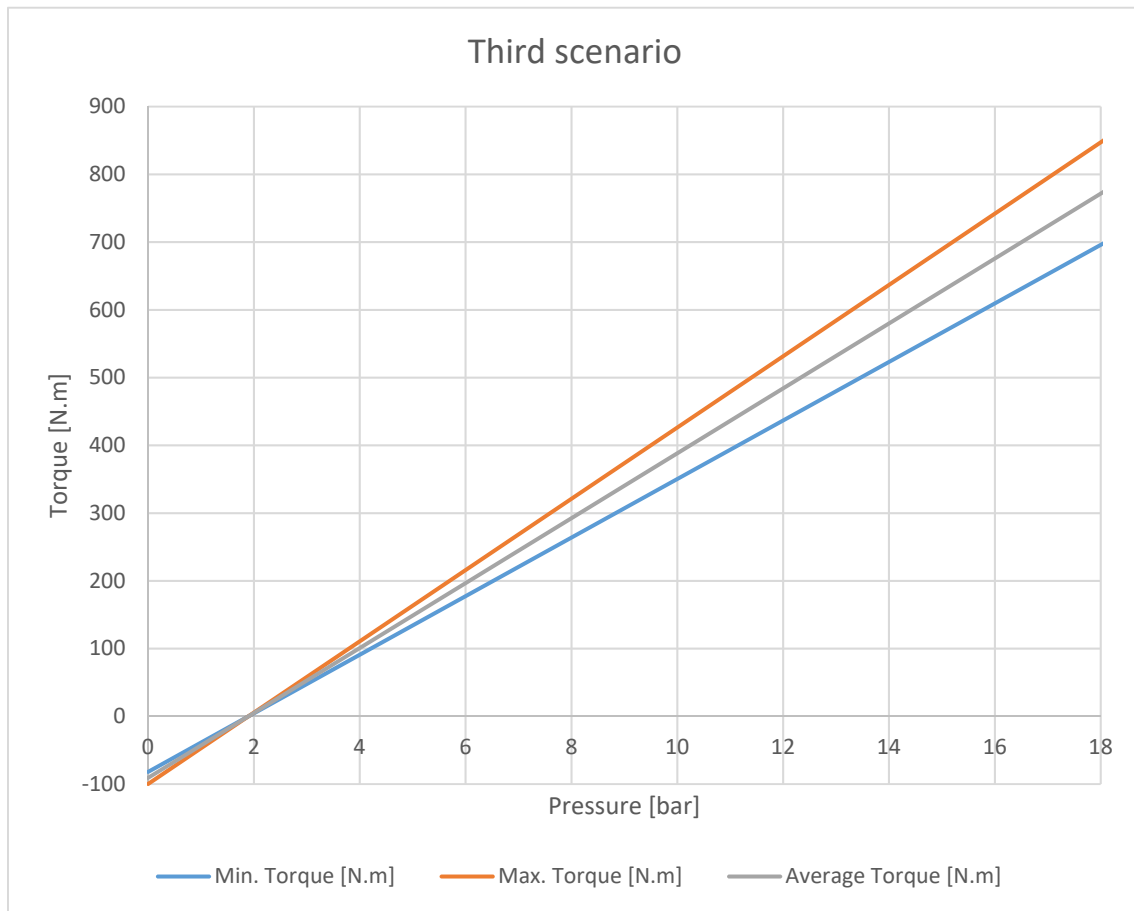


Figure 184 - Graph of transmissible torque vs pressure for the third scenario of REV clutch set

According to the graph the difference between curves, at 15 bar, is slightly lower than 50 Nm. Furthermore, the amount of transmissible torque is lower than both nominal and second scenario.

3.5.7.2.4 Scenarios' comparison

According to the graphs presented before it was conclusive that there were smooth differences between scenarios however in order to better conclude this, it is better to plot a graph comparing the three scenarios. Average transmissible torque was used in the graph for the distinct scenarios.

According to the graph of Figure 186 is visible that second scenario is the one which presents the higher transmissible torque. On the other hand, the third scenario is the one which presents the lower transmissible torque. It was already expected to see the nominal scenario between the other curves.

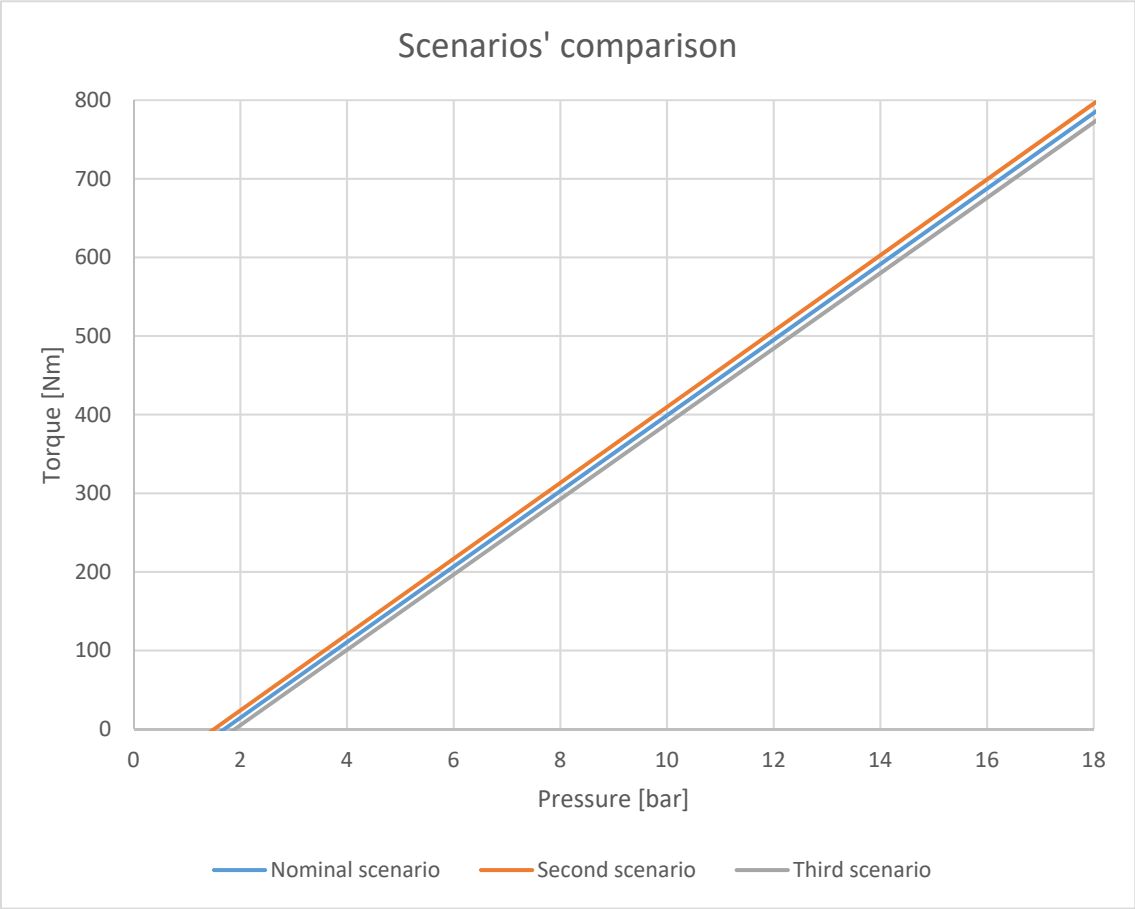


Figure 186 – Transmissible torque scenarios’ comparison for REV clutch set

In Figure 185, it is illustrated a graph comparing the springs’ association stiffness for all the different scenarios.

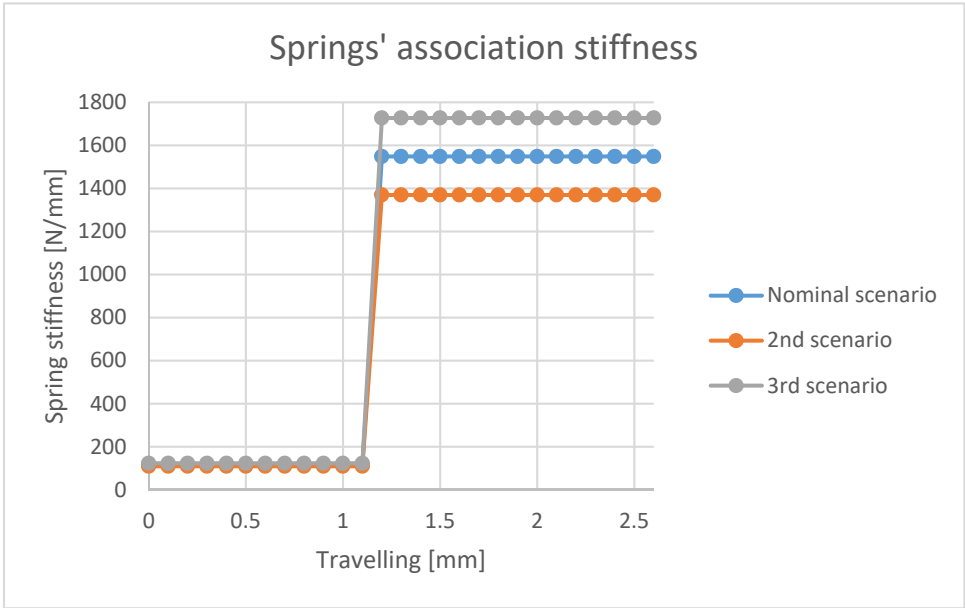


Figure 185 – Graph spring stiffness vs travelling for all the scenario of REV clutch set

Based on the graph of Figure 185, it is evident that at the first stage the stiffness is approximately equal for the different scenarios. However for the second stage there is a significant difference.

In Figure 187 it is plotted a graph comparing the reaction force of the different springs' associations.

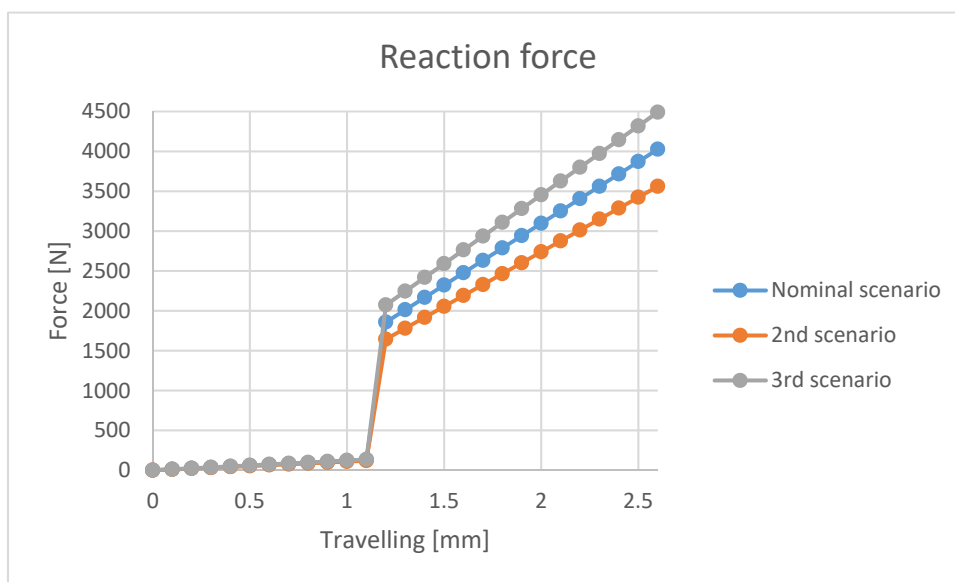


Figure 187 – Graph force vs travelling of the springs' association for all the scenarios of REV clutch set

Based on the graph of Figure 187 it is possible to conclude that the previous phenomenon is verified too. This is expected since this graph derives from the graph of Figure 187.

3.5.7.3 Biting point conditions

The method to calculate the necessary conditions to start transmitting a torque of 8N.m is exactly the same used for FWD clutch set on chapter 3.5.5.3. In first place, it was calculated the tendency line's equation of the curve for the minimum transmissible torque. This first graph relates the minimum transmissible torque and the hydraulic pressure. The main goal of this graph and its tendency line's equation is to know what is the necessary hydraulic pressure to reach a torque of 8 N.m.

After calculating the necessary hydraulic pressure, it is necessary to know what is the axial force which corresponds to the hydraulic pressure previously calculated. For that, it was plotted a graph relating the hydraulic pressure and the originated axial force. At

this point, it is already known the necessary axil force to reach a transmissible torque of 8 N.m.

3.5.7.3.1 Nominal scenario

In the graph of Figure 188, it is presented the graph of the minimum transmissible torque vs the hydraulic pressure. It was designed the tendency line because it will useful afterwards. The data of this graph was obtained analytically by calculations. It was not measured.

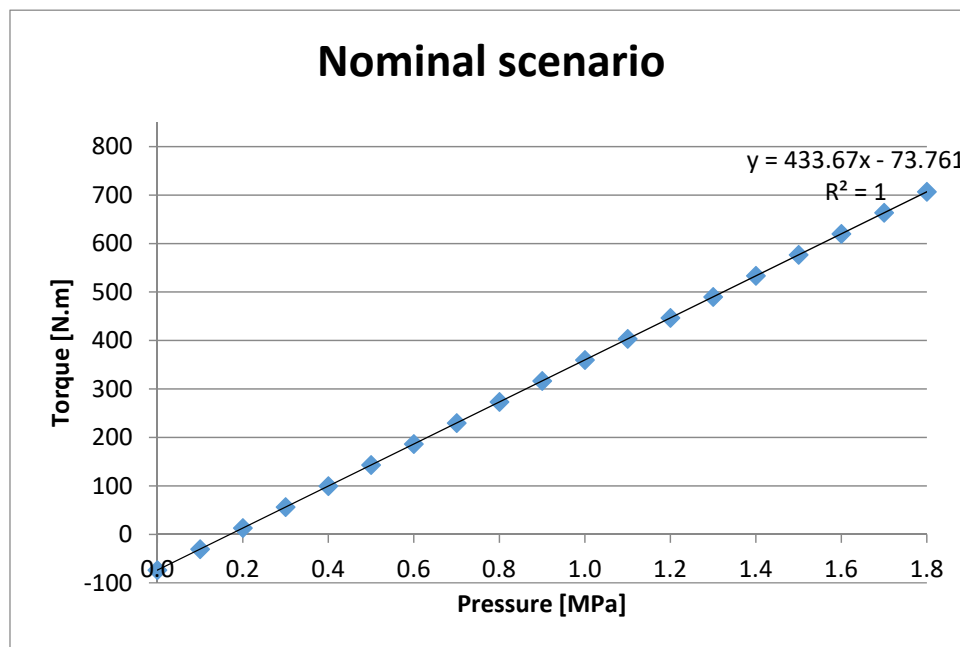


Figure 188 – Graph of minimum transmissible torque vs hydraulic pressure for nominal scenario of REV clutch set

As illustrated on the graph the relation between transmissible torque and hydraulic pressure is linear.

In Figure 189, it is plotted the graph relating the axial force and the hydraulic pressure.

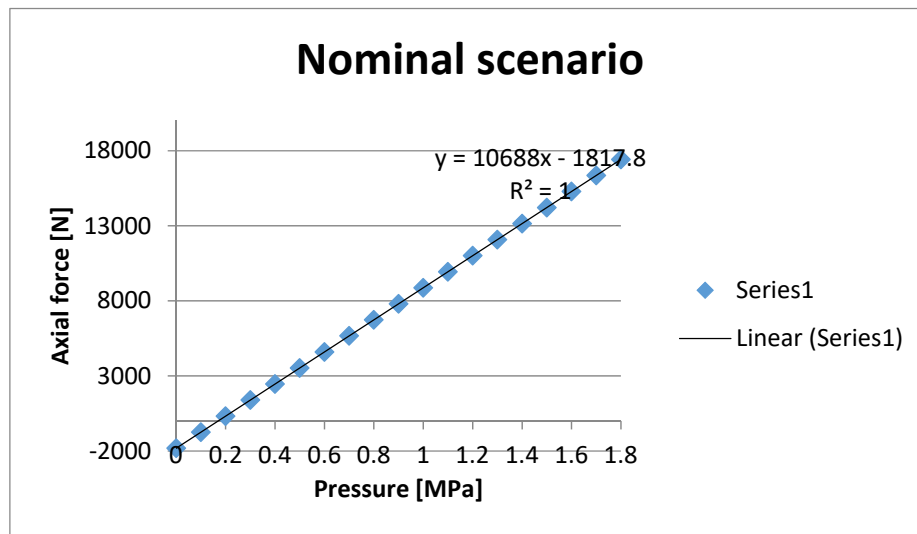


Figure 189 – Graph of axial force vs hydraulic pressure for nominal scenario of REV clutch set

The results for the nominal scenario are presented on Table 68.

Table 68 – Biting point conditions of nominal scenario of the REV clutch set

Variable	Unit	Value
p	bar	1.885
F _{ax}	N	197.157

3.5.7.3.2 Second scenario

In the graph of Figure 190, it is presented the graph of the minimum transmissible torque vs the hydraulic pressure.

The results for the nominal scenario are presented on Table 69.

Table 69 - Biting point conditions of second scenario of the REV clutch set

Variable	Unit	Value
p	bar	1.691
F _{ax}	N	197.157

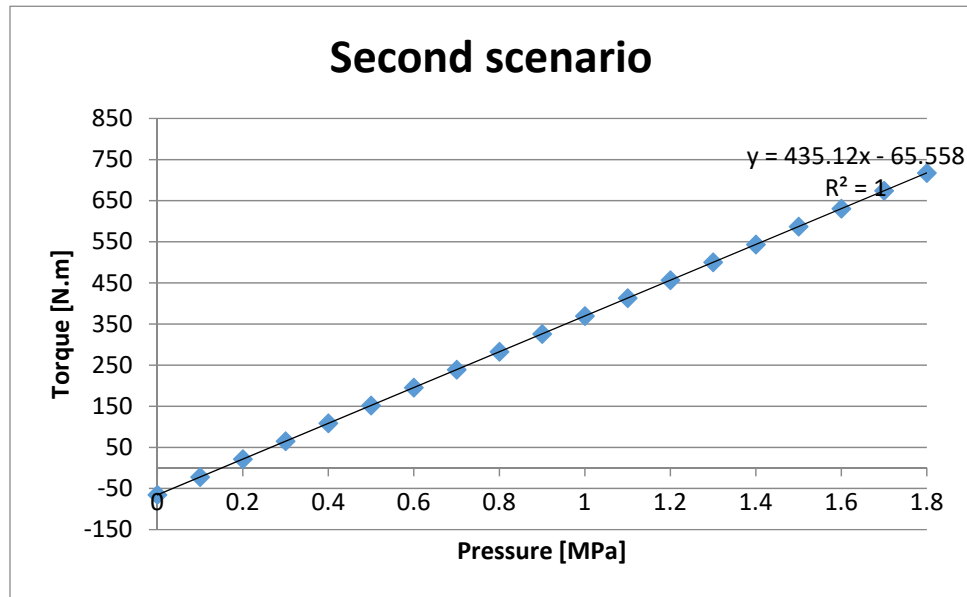


Figure 190 - Graph of minimum transmissible torque vs hydraulic pressure for second scenario of REV clutch set

In Figure 191, it is plotted the graph relating the axial force and the hydraulic pressure.

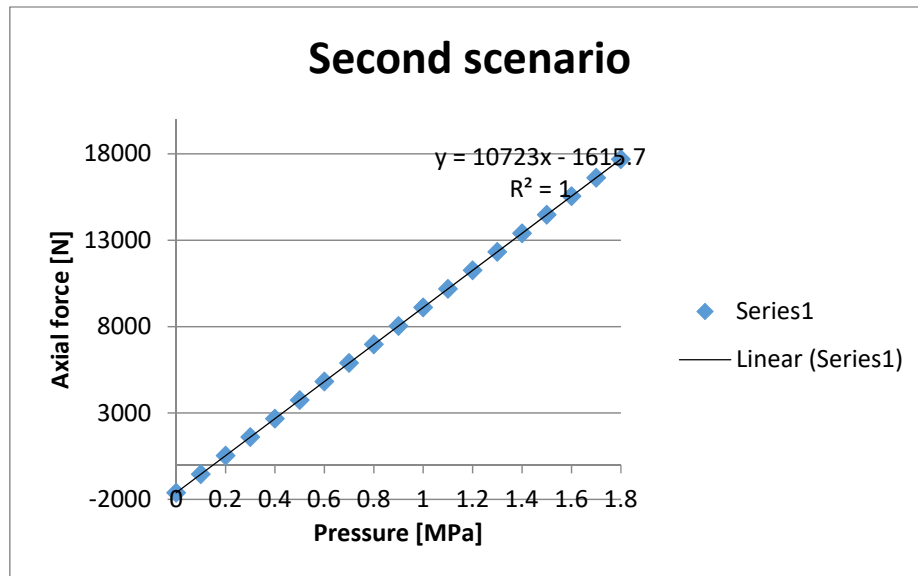


Figure 191 - Graph of axial force vs hydraulic pressure for second scenario of REV clutch set

3.5.7.3.3 Third scenario

In the graph of Figure 193, it is presented the graph of the minimum transmissible torque vs the hydraulic pressure.

In Figure 192, it is plotted the graph relating the axial force and the hydraulic pressure.

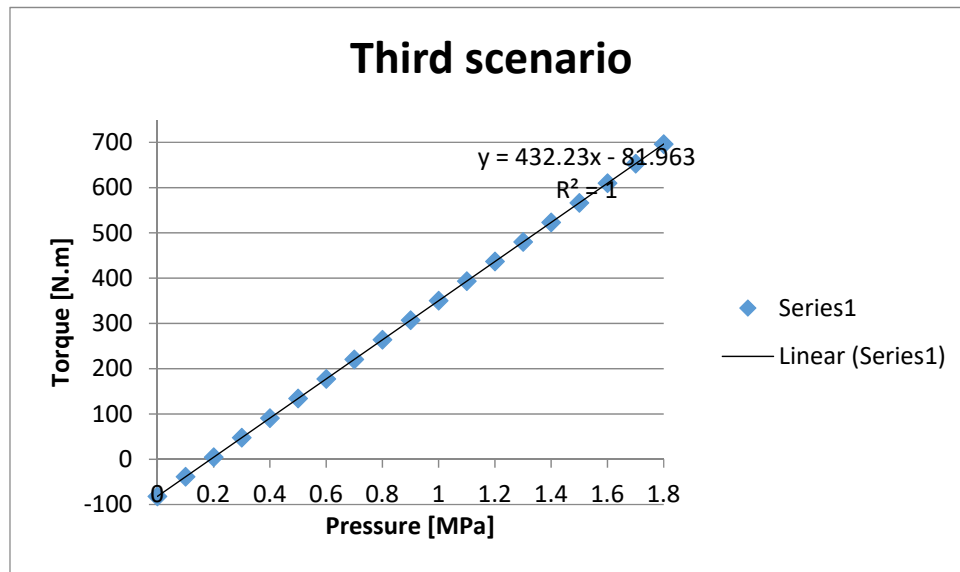


Figure 193 - Graph of minimum transmissible torque vs hydraulic pressure for third scenario of REV clutch set

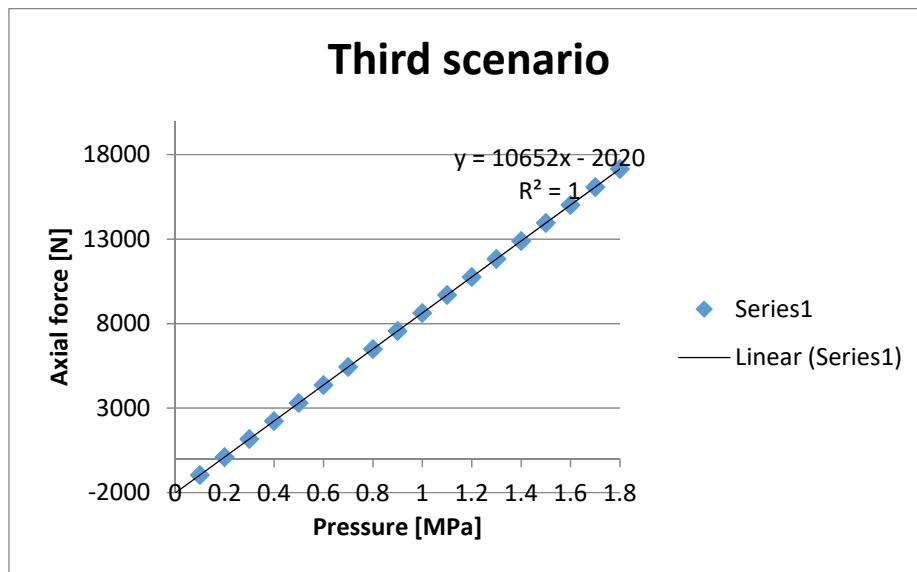


Figure 192 - Graph of axial force vs hydraulic pressure for third scenario of REV clutch set

The results for the nominal scenario are presented on Table 70.

Table 70 - Biting point conditions of third scenario of the REV clutch set

Variable	Unit	Value
p	bar	2.081
F _{ax}	N	197.157

3.5.7.4 Drag losses

In the case of REV clutch set the model that will be used for the calculation of the drag losses is the same used for the calculations of FWD clutch set. So, as the model has already been validated, it will be used will not be verified again.

3.5.7.4.1 Data analysis

On this chapter, it will be studied the drag losses referent to the REV clutch set. The calculation's model is presented on chapter 2.2.4. As mentioned for the FWD set, the nominal oil feeding flow for the DNR subsystem is 0.5 L/min, however only 90% of this value is for cooling purposes of the clutch plates (0.45 L/min). This means that only 10% of the flow rate which comes to the DNR subsystem is for lubricant purposes (gears, bearings, etc.). The maximum feeding flow rate is 1 L/min for short periods of time. The flow rate conditions are presented on Table 71.

Table 71 – Flow rate conditions for REV clutch

Designation	Unit	Value
Nominal feeding flow rate reaching the DNR subsystem	L/min	0.5
Nominal feeding flow rate for cooling clutch plates	L/min	0.45
Maximum feeding flow rate reaching DNR subsystem	L/min	1
Maximum feeding flow rate for cooling of clutch plates	L/min	0.9

Several tables were built in order to understand the behaviour of different parameters. The first series of tables are organized on the following way: on the first column it is presented the rotating speed of CC from 0 to 9000 rpm in steps of 50 rpm; on the second column is presented the feeding flow rate; on the third and fourth columns are presented the outer and inner diameter pressures, respectively; on fifth column it is calculated the pressure variation between inner and outer radius; on sixth column it is calculated the necessary feeding flow rate for full oil film; and finally drag torque results are presented for only one plate and for all the plates as also drag power losses. An excerpt of this table can be found on Table 122 on chapter 6.25. On the first series of tables it was considered the following values of feeding flow rates:

- 0.45 L/min;
- 0.7 L/min;
- 0.9 L/min.

In Figure 194, it is plotted a graph of the relation between the necessary feeding flow rate to ensure full oil film and the rotating speed of the turbine shaft assy.

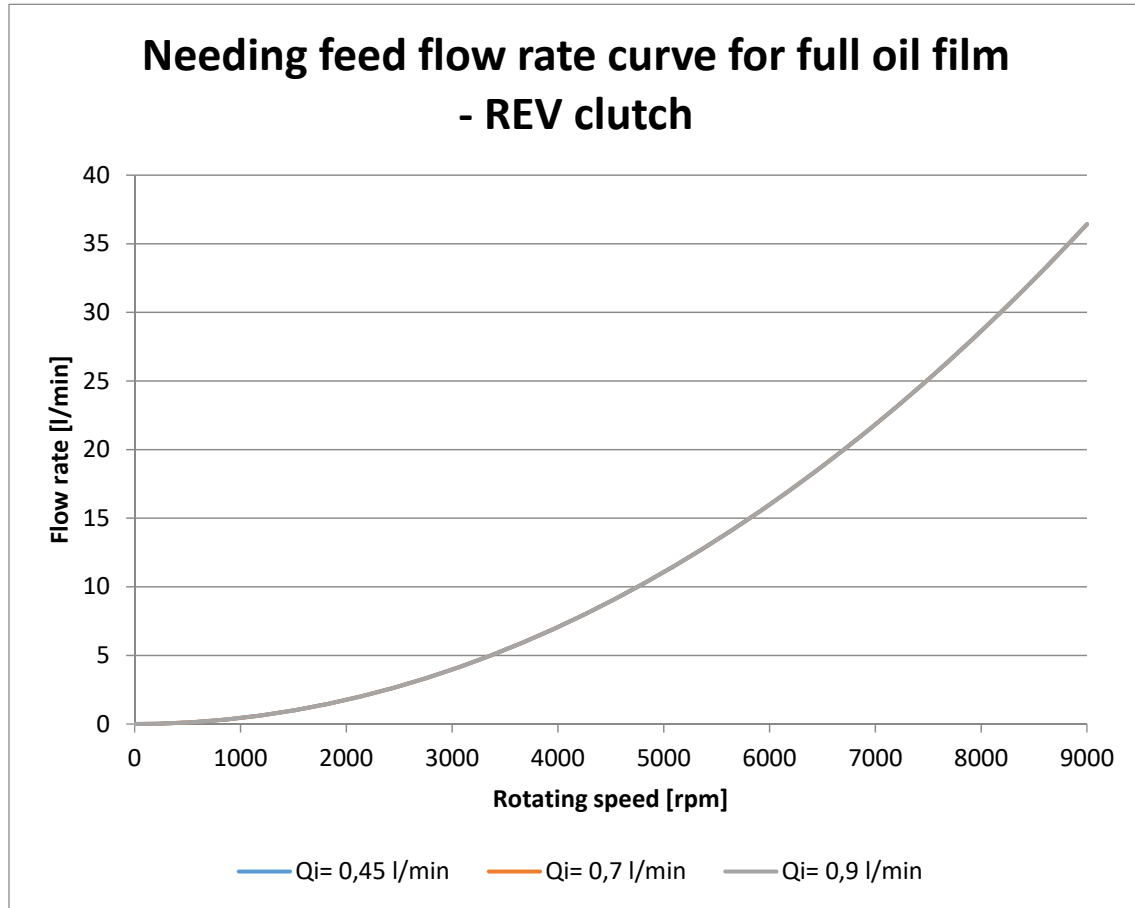


Figure 194 – Graph of needing feed flow rate for full oil film of REV clutch

According to the graph of Figure 194, it is not visible a clear difference between the curves of the distinct input flow rates. As concluded for the FWD clutch, the higher the rotating speed, the higher the needing feed flow rate is.

In Figure 195, it is illustrated a graph with the evolution of the different portions of the pressure: inner and outer diameter and resulting pressure variation for a feeding flow rate of 0.45 L/min.

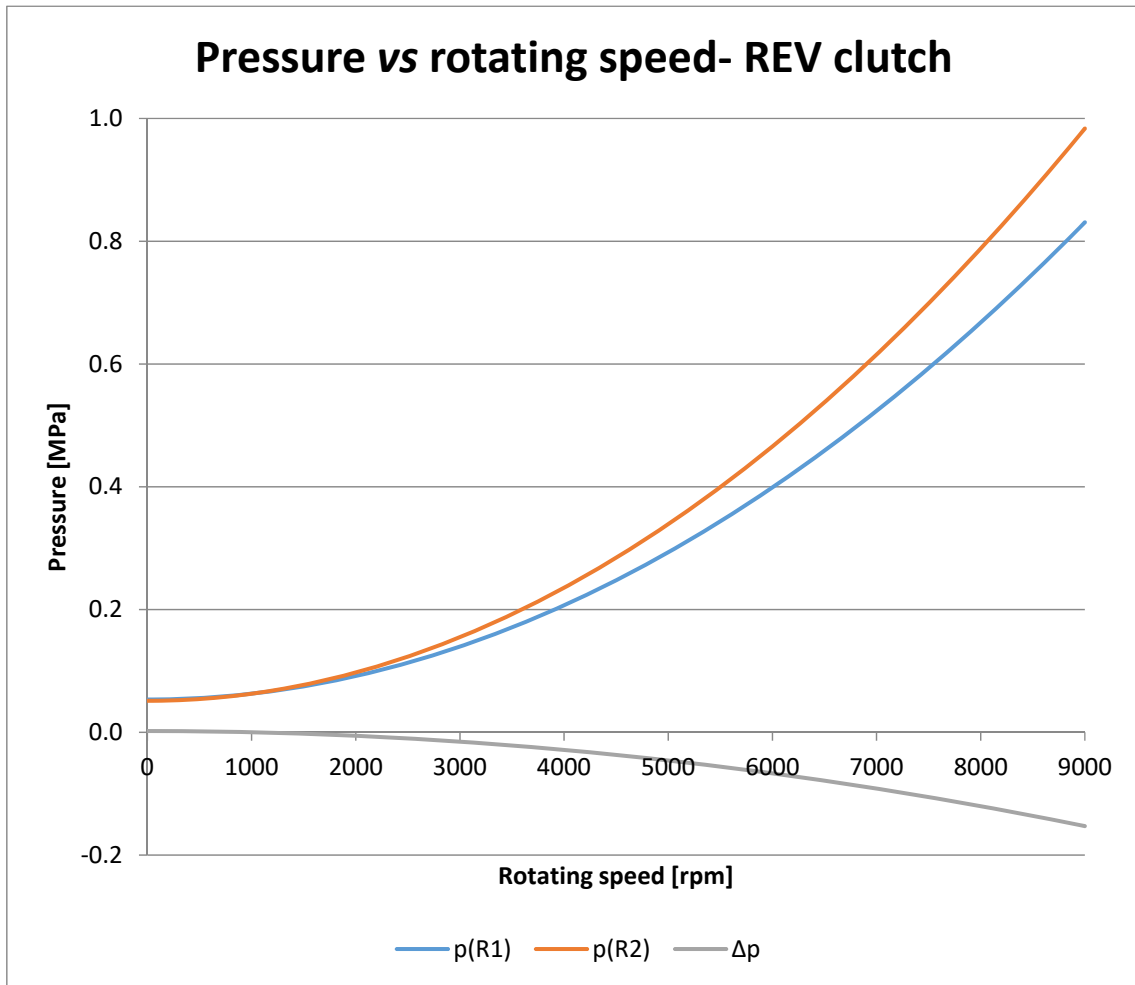


Figure 195 – Pressure portions vs rotating speed for 0,45 L/min input flow rate of REV clutch

As already concluded for the FWD clutch set, according to the graph of Figure 195, when rotating speed increases, so does outer and inner diameter pressures. In contrast, the difference of pressures decreases. This is expected taking in account the equation of pressure difference and that the value of pressure on R2 is most of the times greater than the pressure on R1.

In Figure 196, it is plotted a graph similar to the one of Figure 195 but for more input flow rates. According to the graph of Figure 196, there is a visible difference between the curves of p(R1) and p(R2) for the different flow rates. Although, on the curves of Δp it is not so visible that difference. In fact, the input flow rate has not got any effect on this parameter. It is conclusive that all the curves' shapes are very similar.

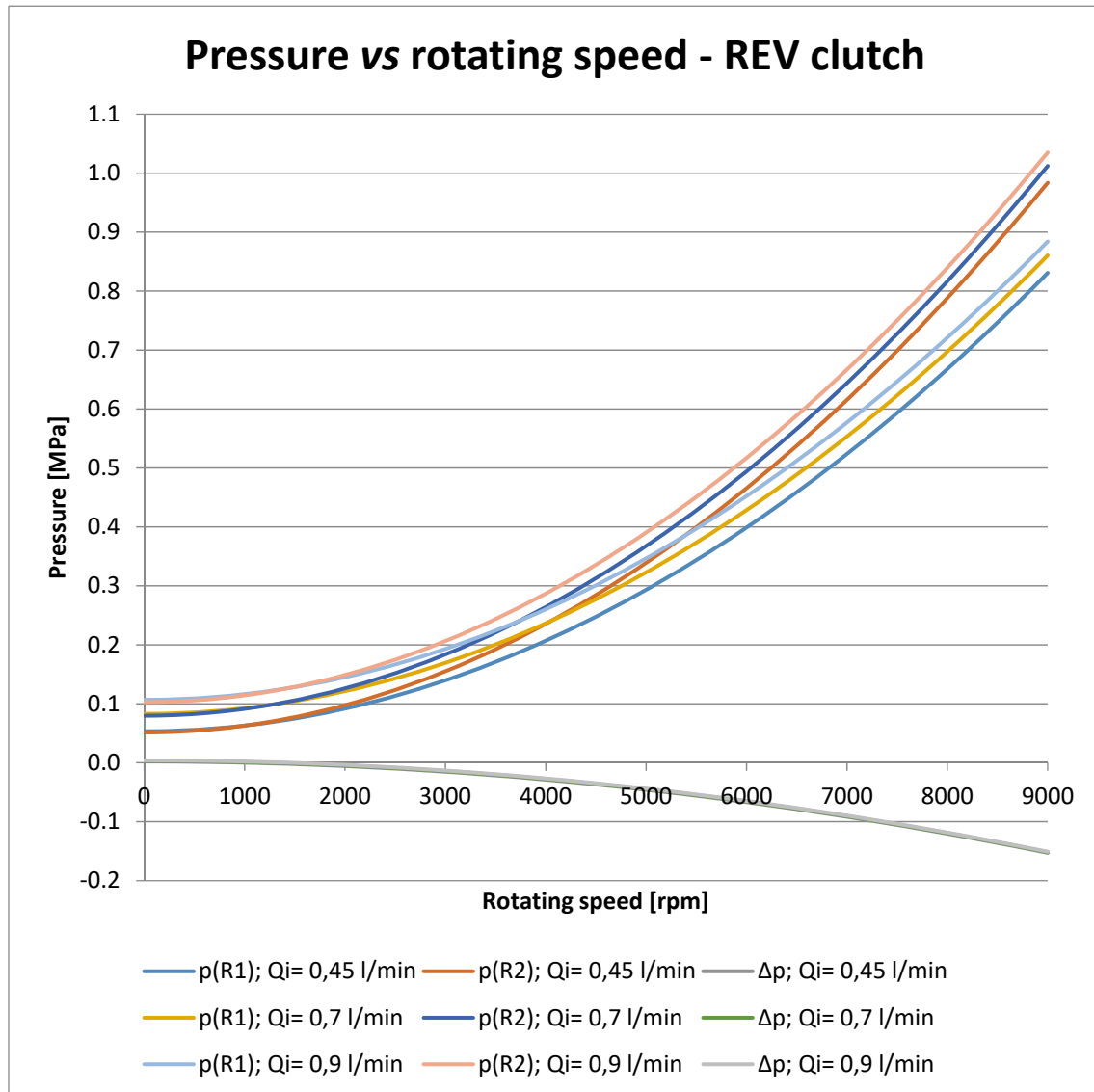


Figure 196 – Pressure portions vs rotating speed for different input flow rates

In Figure 197, it is illustrated the drag losses of a single plate for different input flow rates calculated on the REV clutch.

There is a direct relation between drag torque and input flow rate: the higher the input flow rate is, the higher the drag torque is. This is only applied on the region where the input flow rate is lower than the necessary to ensure full oil film. Until that point the drag torque is equal independently of the input flow rate.

Due to the existence of a ratio on the PGS for the REV set, it is expected that the drag torque calculated and measured on the REV clutch is different from the one measured on the primary shaft.

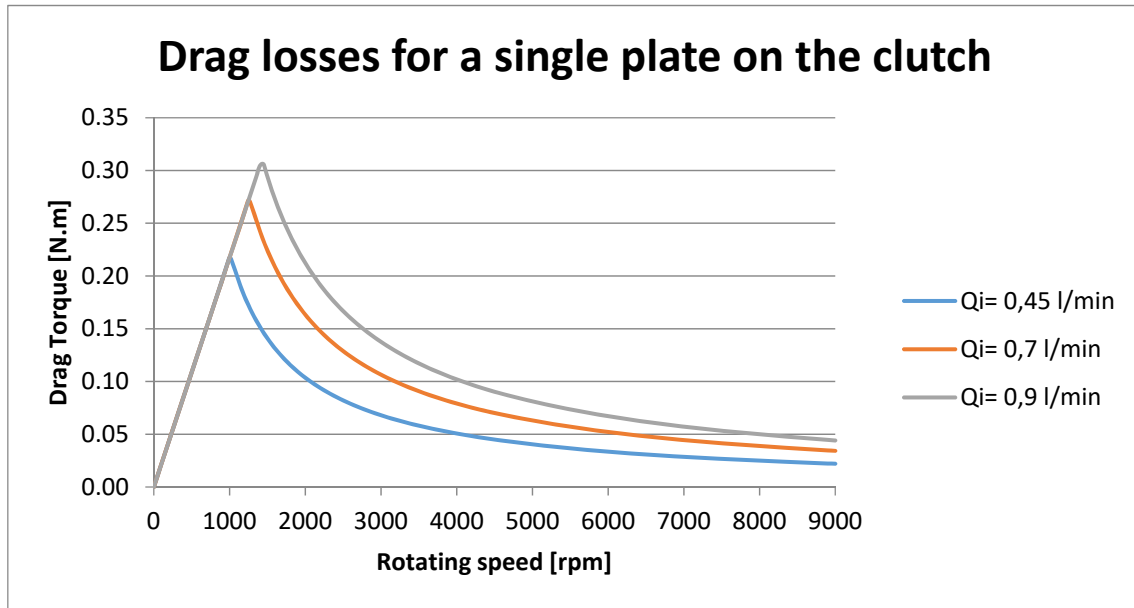


Figure 197 – Graph of drag losses for a single plate calculated on the REV clutch

In Figure 198, it is presented a graph of the drag losses of REV clutch calculated on the primary shaft.

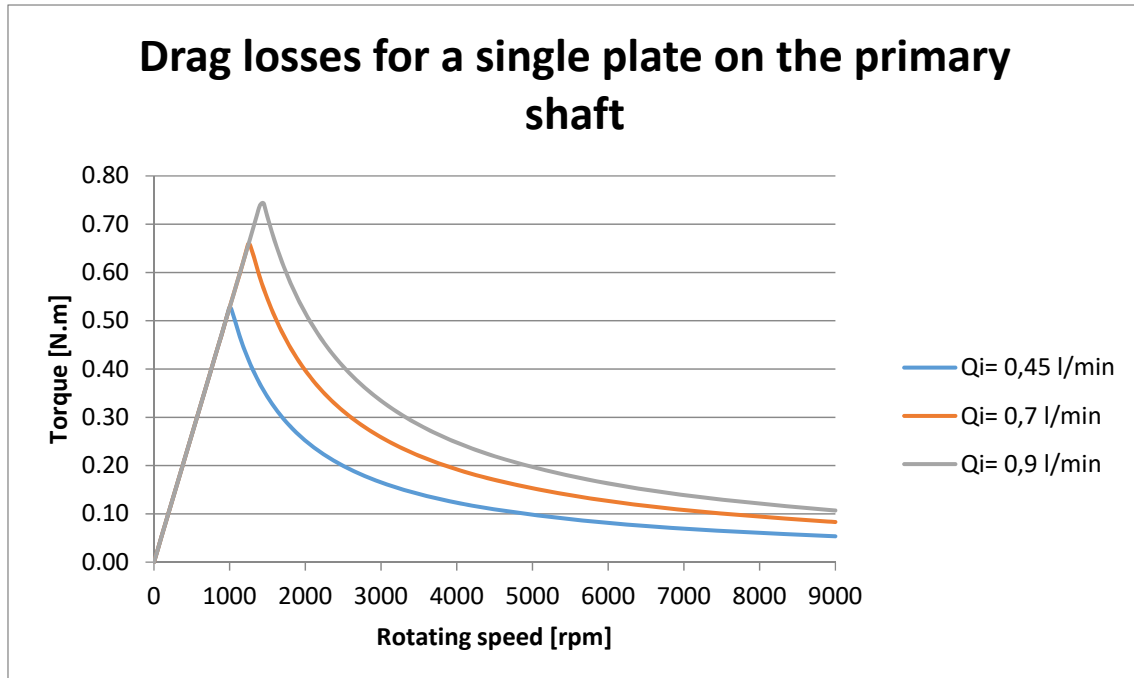


Figure 198 – Graph of drag losses for a single plate calculated on the primary shaft

Comparing the graph of Figure 198 to the one of Figure 197, it is conclusive that the maximum drag losses are higher on the first one as also at the stabilization area.

In Figure 199 it is presented the drag torque of all plates for REV clutch calculated on the REV clutch set.

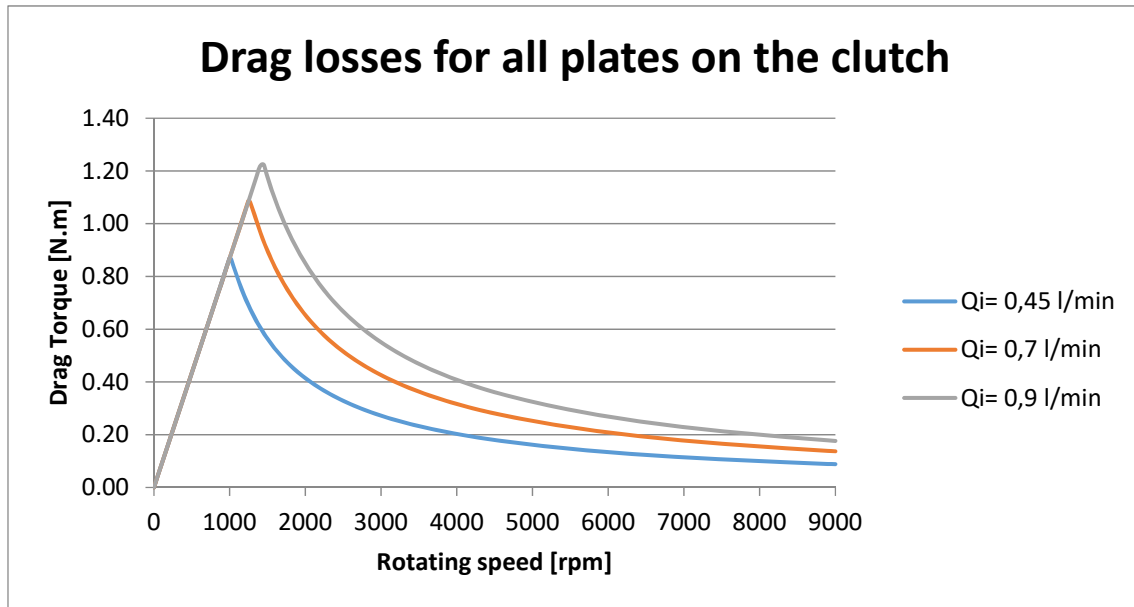


Figure 199 – Graph of drag losses of all the plates calculated on the REV clutch set

The same conclusions done to the graph of Figure 197 are applied for this graph.

In Figure 200, it is illustrated the graph of drag losses for all the plates calculated on the primary shaft.

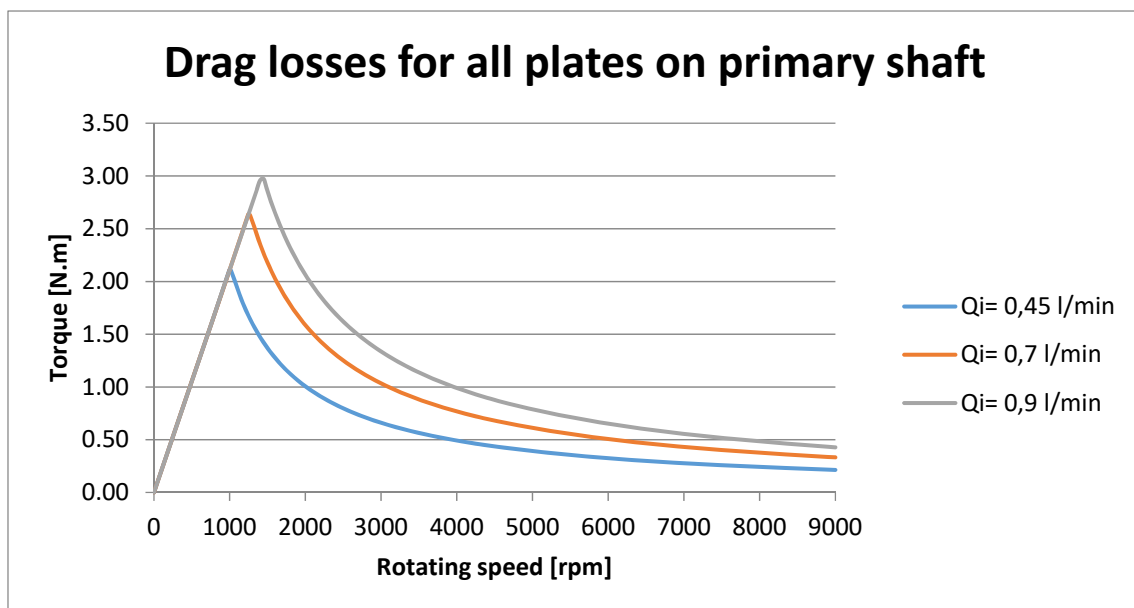


Figure 200 – Graph of drag losses for all the plates calculated on the primary shaft

The same conclusions of Figure 198 are applied to this graph.

In Figure 201, it is presented a graph comparing the necessary feeding flow rate to guarantee a full oil film for different values of Δp .

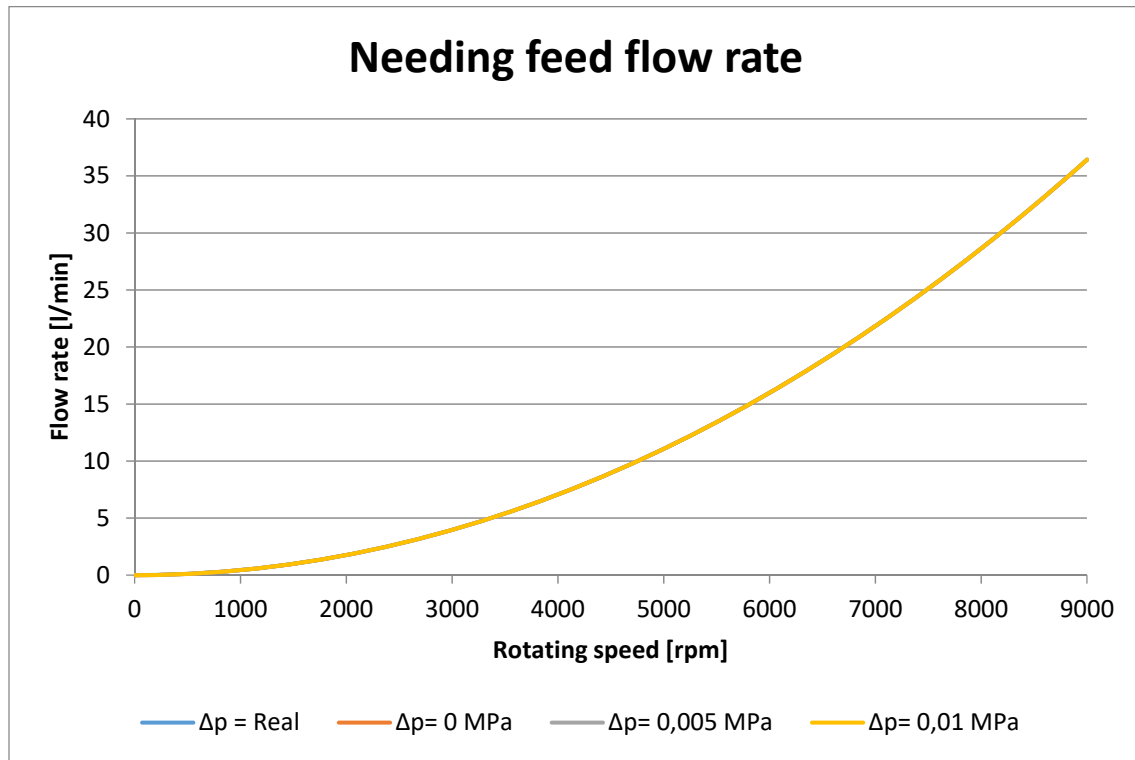


Figure 201 – Needing feed flow rate for different values of Δp

Based on the graph of Figure 201, it is evident that there is not visible difference between the curves of the different Δp . So, in conclusion, the variation of Δp has no effect on the needing feed flow rate.

The graph of Figure 202 illustrated the relation between the equivalent radius and rotating speed for different feeding flow rates.

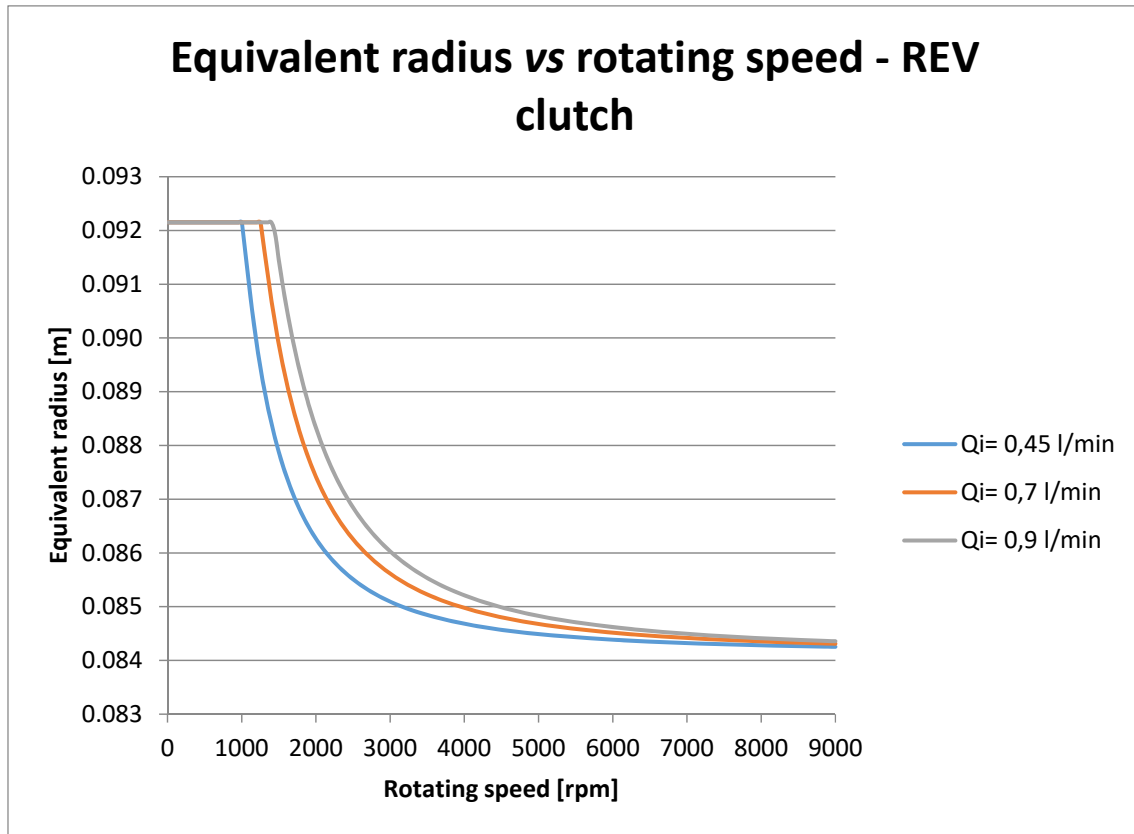


Figure 202 – Equivalent radius vs rotating speed for different feed flow rates of REV clutch

Similar to the FWD clutch, at the beginning and until a certain point, the equivalent radius keeps stable. At a certain point, the curves start decreasing significantly. The value of rotating speed where there is a change on the curve's shape corresponds to the value of the rotating speed where the feed flow rate is not enough to ensure full oil film. As seen before this value of rotating speed depends on the feed flow rate.

The second series of tables is exactly the same as the first however the value of pressure variation is introduced by the user and not calculated as happens for the first series. The value of the input flow rate for the second series was considered to be 0.45 L/min. In the second series of tables the following values of pressure variation were considered:

- 0 MPa;
- 0.005 MPa;
- 0.010 MPa.

In the graph of Figure 203 it is illustrated the relation between equivalent radius and rotating speed for different values of pressure variation.

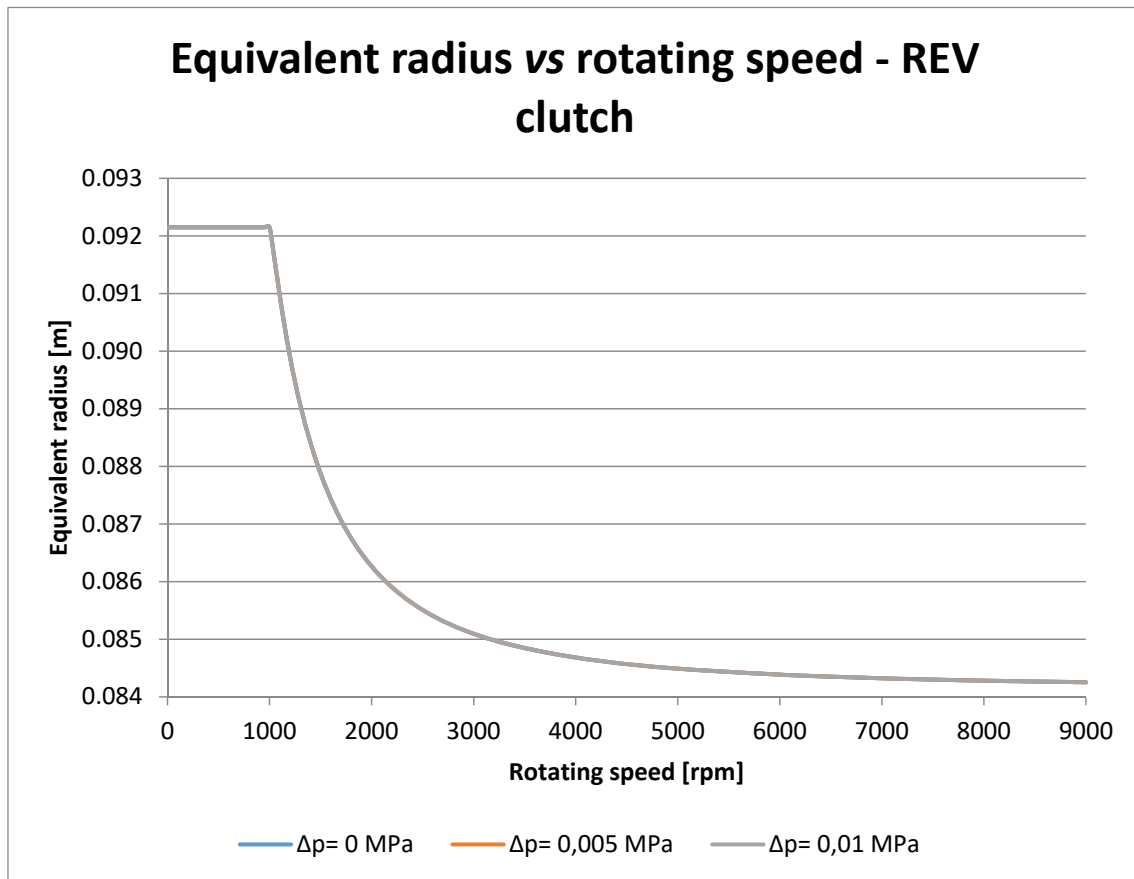


Figure 203 – Equivalent radius vs rotating speed for different values of Δp for REV clutch set

It is evident the pressure variation has no effect in the equivalent radius.

The series of tables which is the most approximated to the real conditions is the third one. In this series of tables there are two inputs: oil temperature and feed flow rate. The value of feed flow rate will be considered constant with a value of 0.45 L/min. The temperature values are the following:

- -20 °C;
- 70 °C;
- 90 °C;
- 110 °C.

The main goal of this series of tables is to allow to simulate and understand the behaviour of the different parameters according to the variation of temperature. The most common range of temperature varies between 70 and 90 °C.

The relation between necessary feeding flow rate to guarantee full oil film and rotating speed is plotted in Figure 205

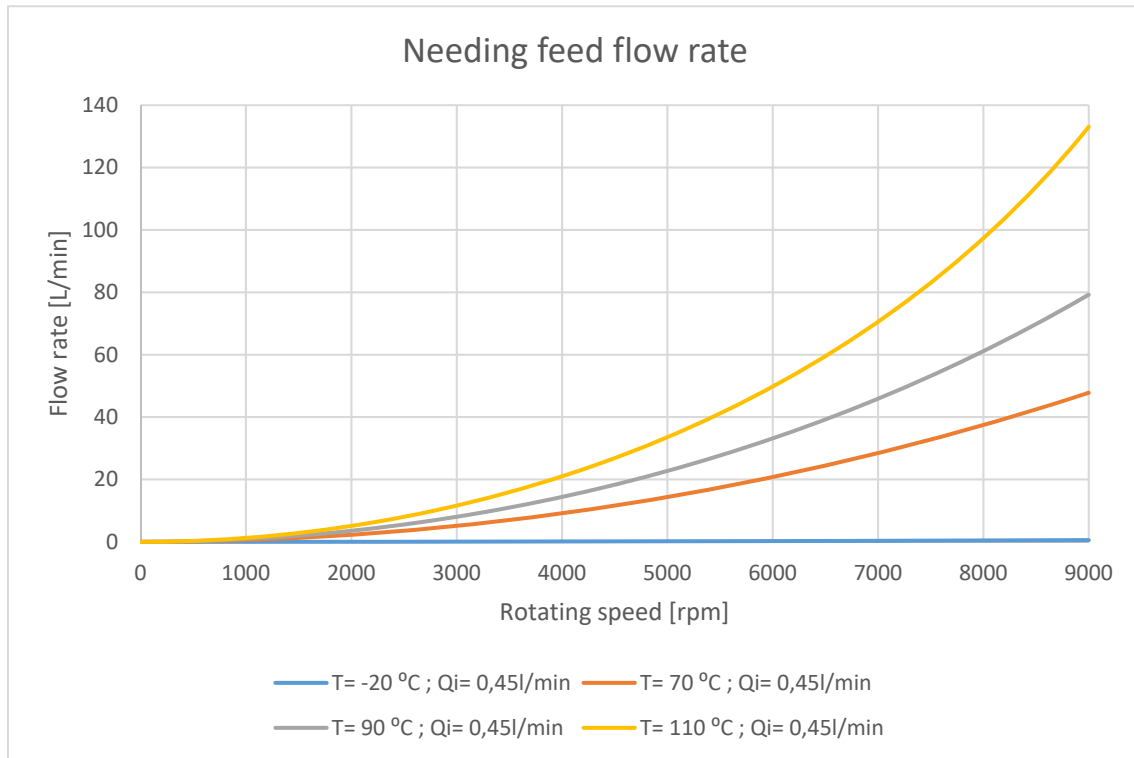


Figure 205 – Needing feed flow rate to ensure full oil film affected by different working temperatures in REV clutch

According to the graph there is a visible evolution of the curves for the positive temperatures. Regarding to the negative temperature, there is a rising in the needing feed flow rate at higher rotating speed but it is not significant as on the other temperatures. The needing feed flow rate for the positive temperatures is very high.

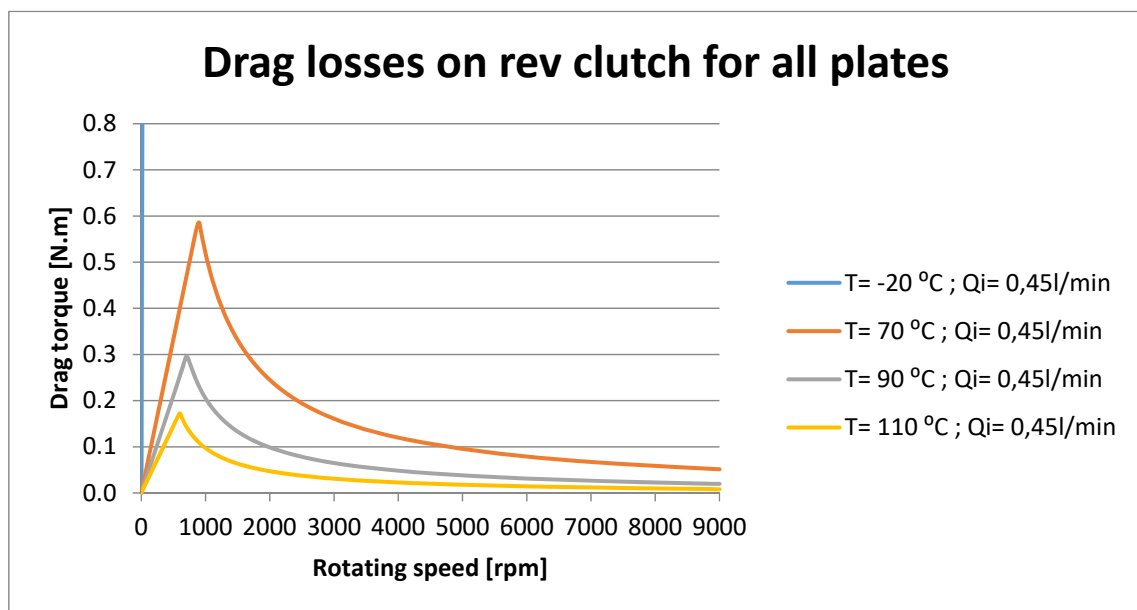


Figure 204 – Drag losses for all plates in REV clutch

In Figure 204, it is plotted the relation between drag torque and rotating speed for different working temperatures and the same feeding flow rate.

Based on the graph of Figure 204, it is evident that for the negative temperature, the drag torque is significantly higher when compared to the positive working temperatures. The slope of the curve is incredibly high. For the positive temperatures, the same of the curve is identic for all of them. Until a certain point, a straight line can be seen and after that it decreases smoothly. The inverting point corresponds to the point where the feeding flow rate is equal to the necessary flow rate to guarantee full oil film. From that point, the feeding flow rate is not enough to guarantee this condition.

In Figure 206, a graph similar to Figure 204 is plotted but at this time calculated at the primary shaft.

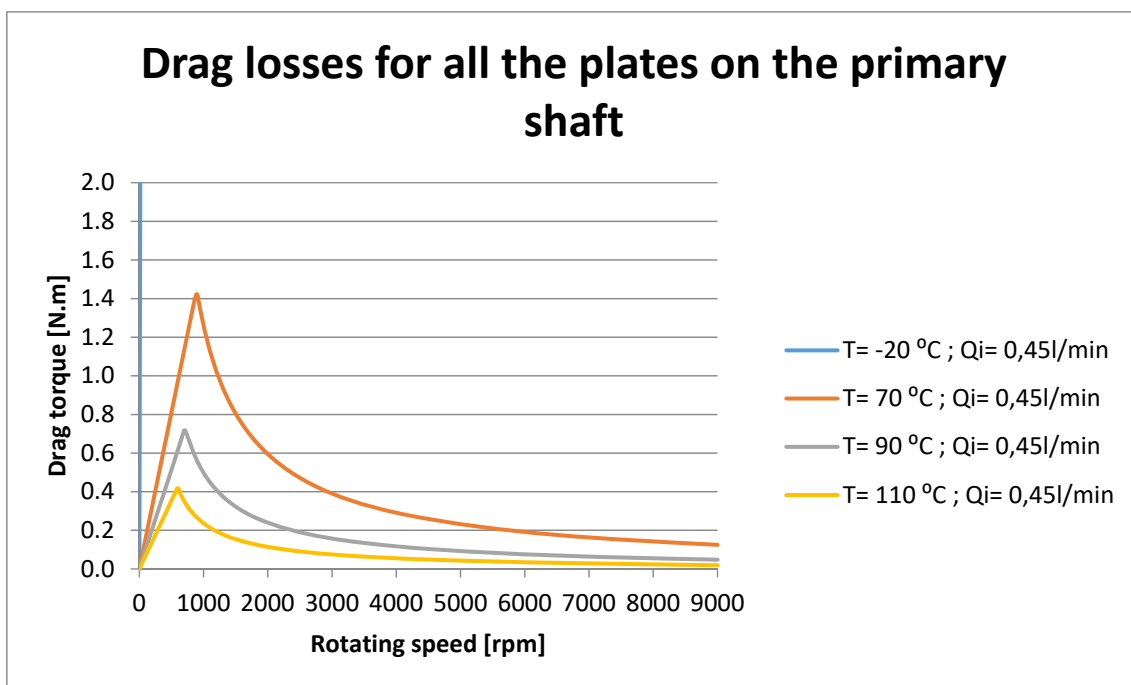


Figure 206 – Drag losses for all plates calculated on the primary shaft for REV clutch

The conclusions for this graph are the same of the graph of Figure 206. However, in this case, as it was calculated from the primary shaft, the values of drag torque for the same temperatures are considerable higher on the graph of Figure 206.

In Figure 207, it is illustrated a graph with the dissipated power through drag losses. The graph of Figure 206 was the base for this one. Both are calculated at the primary shaft.

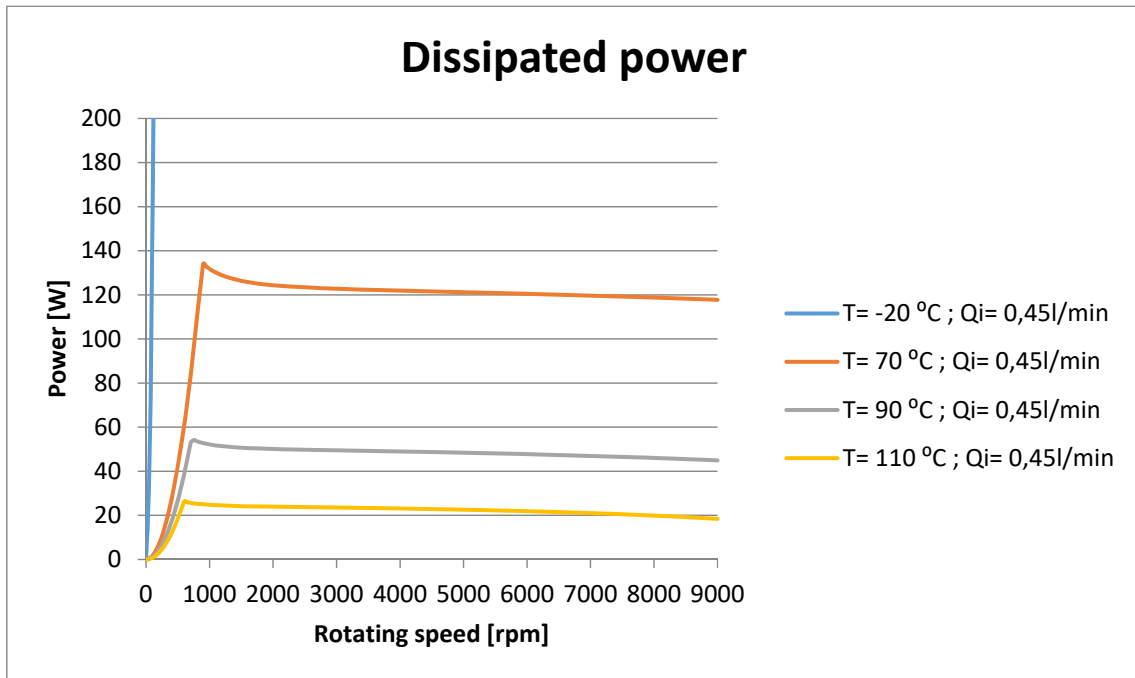


Figure 207 – Dissipated power through drag losses for all the plates calculated on the primary shaft for REV clutch

The aim goal of this graph is to have a notion of the amount of power which is dissipated on the REV clutch as a consequence of drag losses. It is visible a significant difference between positive working temperatures curves. The dissipated power when the oil temperature is equal to -20 °C is very high.

3.5.7.4.2 Working parameters

Rotating speed

Based on the graph of Figure 195, it is conclusive that the influence of the rotating speed on the different portions of pressure in REV clutch set is similar to the influence verified in FWD clutch set.

Input feed flow

According to the graph of Figure 194, input flow rate hasn't got any effect on the needing feed flow rate for full oil film.

Pressure variation

The conclusion relative to this parameter is the same found for the FWD clutch set.

Oil temperature

The conclusion regarding to this parameter is the same stated for the FWD clutch set.

3.5.8 Tolerance chain stack up analysis

In this chapter, it will be presented all the analysis done relative to tolerance chain stack up. Two main analysis were done: the first one was relative to the FWD clutch set and the other one to REV clutch set. For both, the purpose of this analysis was to find out what was the play between the components of each clutch set in order to verify what was the amount of travelling that will be done by the spring pack.

The tolerance chain stack up analysis will be based on the RSS (Root sum square) model. This method is suitable for large tolerance chains that is the case in study. This model evaluated the variation of the dimensions of the parts having in account the tolerances. In this model it is assumed that most of the components fall to the mid of the tolerance zone rather than at the extreme ends based on the normal distribution.

For the calculation of the final tolerance range it is necessary to insert the nominal dimension and respective tolerance of the number of the dimension. After that it is necessary to insert the signal of the dimension (positive or negative). When the dimension is defined from left to right is defined as negative, otherwise is considered positive.

3.5.8.1 FWD clutch set

As previously explained, the main goal of the tolerances stack up analysis of the FWD clutch set is to enquire what is the amount of play between the different components such as clutch endplates, clutch plates, wavy washer and endplates. The existing play between these components defines the amount of travelling that the FWD spring pack will compress before the wavy washer starts compressing. In the case of the FWD clutch it was considered the RSS analysis at 20 °C. It was considered this temperature because this value will be useful not only for calculations of the clutch sets but also for

calculations regarding to the showcase model. As the showcase model will work at environment temperature of about 20 or 25 °C it is not necessary to predict worse scenarios.

As can be seen in Figure 208, each dimension corresponds to a number. The final gap between components is calculated having in account the tolerances of each component and the tolerance of other dimensions. The tolerances' chain starts at number 1 which means that it goes from left to right (1 to 8), inverts at 9 and then from 11 to 12 goes again from left to right. All the numbers correspond to a dimension of a component except the dimension number 9 and 10. This dimension (9) is clearly defined in the technical drawing as also its tolerance. The dimension 10 corresponds to the total dimension of all the components. At the end the gap between components will be returned. In this case, dimension number 9 and 10 are considered positive. All the others are considered to be negative. Dimension number 9 at a first sight is not necessary. However, on a deeper look it is possible to realise that the dimension number 8 corresponds to a snap ring that is inside a groove.

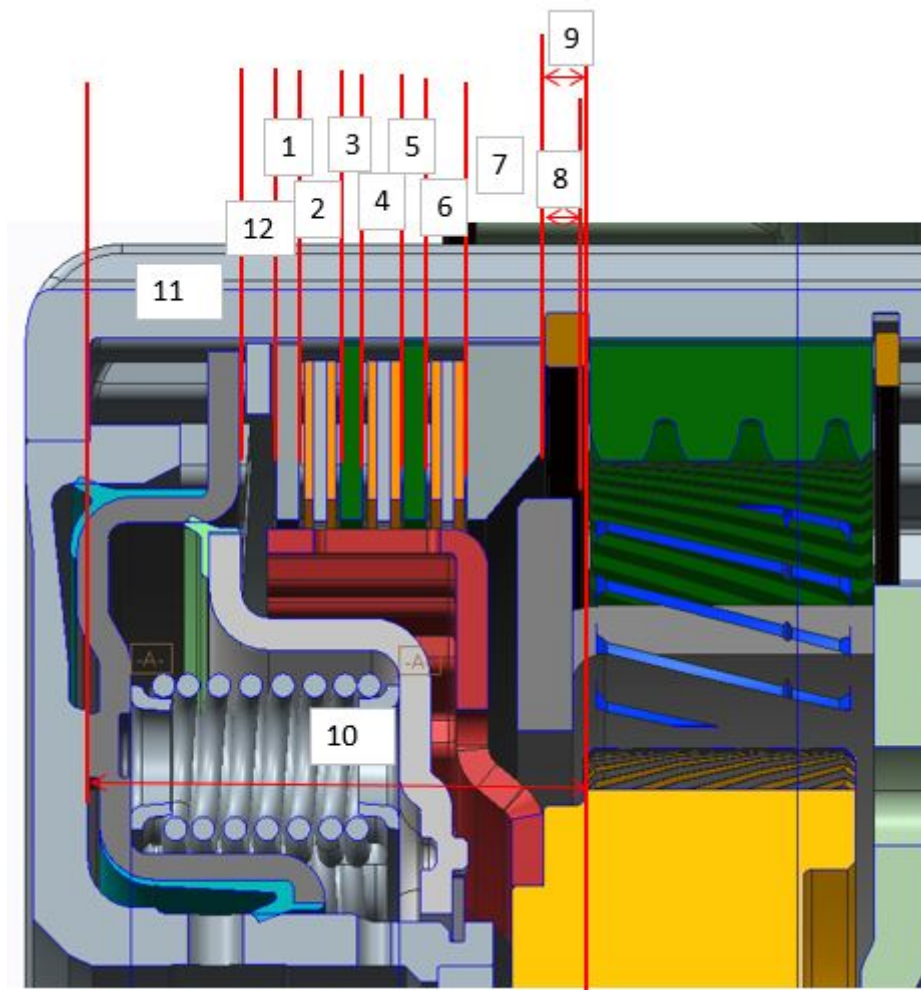


Figure 208 - Tolerance chain stack up of FWD clutch set

The dimensions and mainly the tolerance of the groove is important since it affects the play between components and after all the travelling of the FWD spring g pack.

The obtained results can be found on Table 72

Table 72 – Results of tolerance chain stack up analysis of FWD clutch set

Minimum case	Nominal case	Worst case	Tolerance
1.469	1.810	2.151	0.681

For all the calculations it was considered the nominal case but a rounded value of 1.8 mm.

3.5.8.2 REV clutch set

As stated before, the main goal of the tolerances stack up analysis of the REV clutch set is to find out what is the amount of play between the different components such as clutch endplates, clutch plates, disc spring and endplates. The existing play between these components defines the amount of travelling that the REV spring pack will compress before the disc spring starts compressing. For the REV clutch it was considered the RSS analysis at 20 °C. It was chosen this scenario for the same reason of FWD clutch set.

As can be seen in Figure 209, each dimension corresponds to a number. The final gap between components is calculated having in account the tolerances of each component and the tolerance of other dimensions. All the numbers correspond to a dimension of a component except the dimension number 1 and 2. This dimension (1) is clearly defined in the technical drawing as also its tolerance. The dimension 2 corresponds to the total dimension of all the components. At the end the gap between components will be returned. In this case, dimension number 2 is considered positive. All the others are considered to be negative.

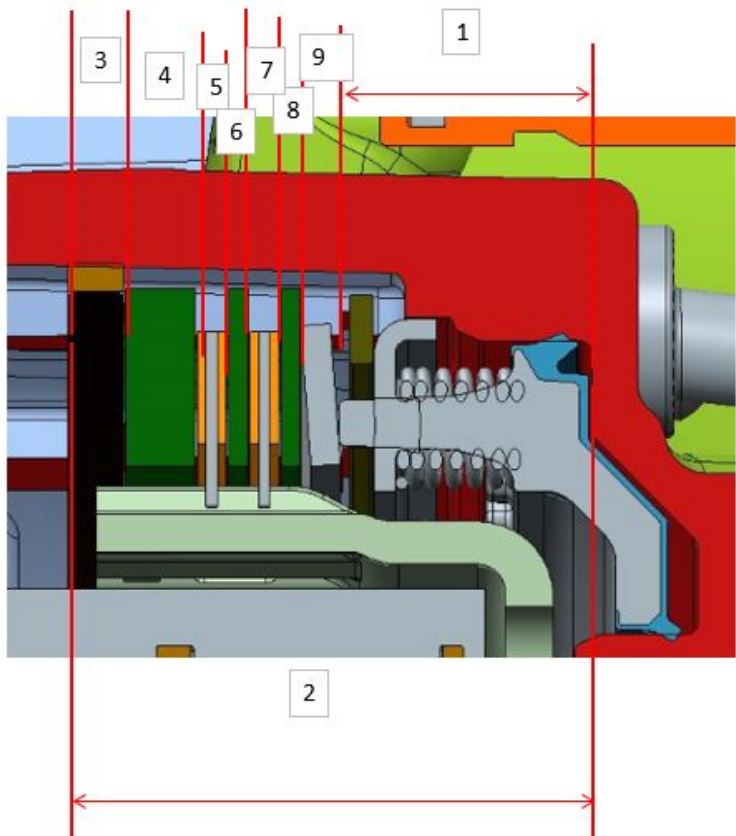


Figure 209 - Tolerance chain stack up of REV clutch set

Table 73 - Results of tolerance chain stack up analysis of REV clutch set

Minimum case	Nominal case	Worst case	Tolerance
0.839	1.045	1.251	0.411

For all the calculations it was considered the nominal case but a rounded value of 1.1 mm.

3.5.9 PGS

3.5.9.1 Transmission ratio

3.5.9.1.1 D gear

In order to obtain the ratio of drive gear it is not necessary to calculate anything. As seen before (3.3.6.1), the sun gear rotates with the CC and at the same time is connected to

the variator. This means that the sun gear transmits the motion and torque from the turbine shaft assy directly to the primary shaft with no ratio.

In conclusion, the ratio in D gear is equal to 1.

3.5.9.1.2 REV gear

In contrast to the D gear, to obtain the ratio of REV it is necessary to do some calculations and to attentively look to the DNR subsystem. For this chapter, it is necessary to take a look at chapter 3.3.6.3, chapter 2.2.5, Figure 126 and Table 9.

In REV, the PGS carrier is steady so it works as a frame. Ring gear receives the motion and torque from the turbine shaft assy. Sun gear is connected to the primary shaft so it works as an output.

Now, it is necessary to consult Table 9. Having in account all these factors, b) is the state of motion in use and the result for the ratio is:

$$i^{REV} = -\frac{79}{113} = 0,699 \quad \{3.29\}$$

3.5.9.2 Kinematics analysis

3.5.9.2.1 Drive gear

In this chapter, it will be studied the kinematics of the different gears of the PGS when D gear is selected. Firstly, it was studied the angular speed of each gear around the centre of the PGS. It is known that sun gear and the ring gear rotate at the same time since both of them are connected to the clutch carrier. When D gear is selected, FWD clutch plates connect the sun gear and the CC while the ring gear is always connected to the CC. If they are connected to the CC, it means that they rotate at the same rotating speed of the turbine shaft.

As the sun gear and the ring gear are rotating at the same speed, it is expected the planet gears to rotate around the centre of the PGS but not around themselves. The reason for this statement is the following: according to Figure 210, there are two contact points for the planet gears. One point is the interface between the planet gear and the ring gear and the other one is the intersection between sun gear and planet gear. Since the sun gear and ring gear rotate at the same speed, a line can be drawn from the centre of the PGS and intersecting both red points of Figure 210. This line will always intersect both red points. As there is no angular speed variation, both points are always within the line.

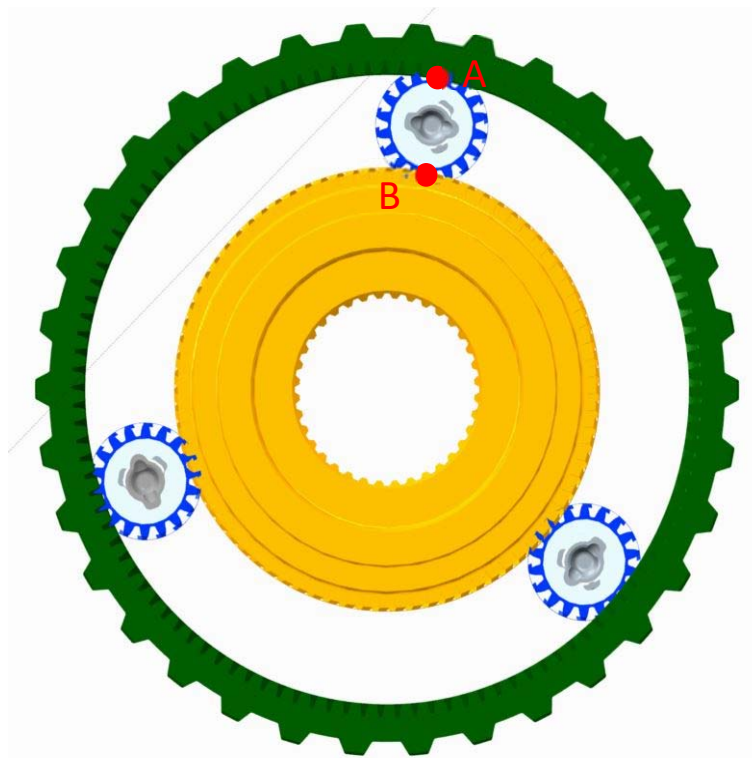


Figure 210 – PGS contact points between planet gears and sun and ring gears

As can be seen on the graph of Figure 212, all the gears rotate around the centre of the PGS at the same speed, including the planet carrier. Planet carrier is fixed to the planet gears through a pin. As the planet gears rotate at the same speed as the sun and ring gears, so does the planet carrier.

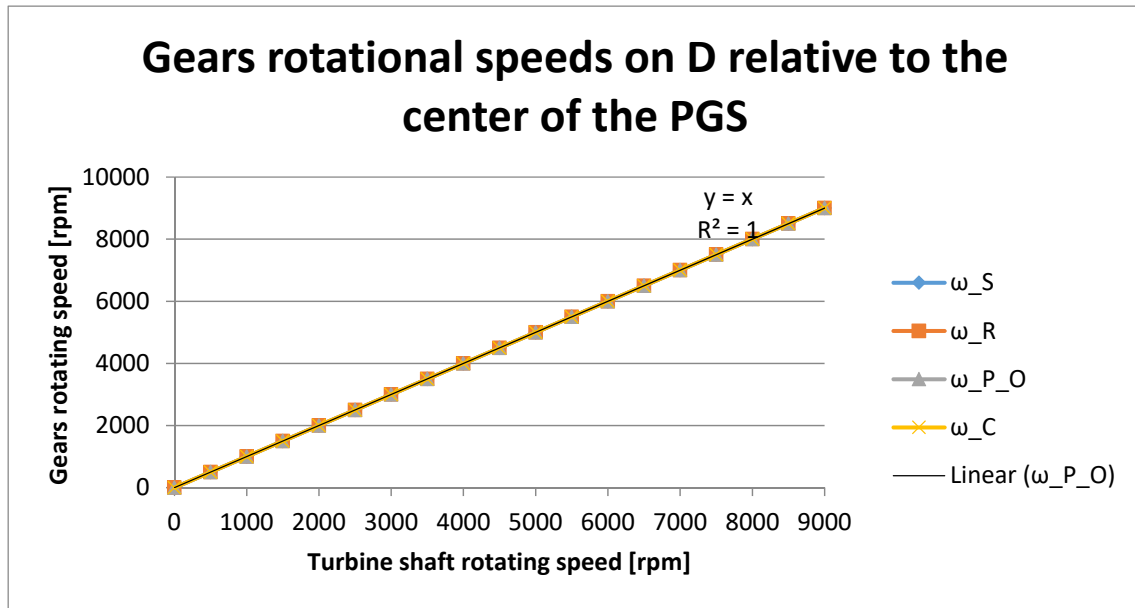


Figure 212 – Rotating speeds on D relative to the centre of the PGS in D gear

In the graph of Figure 211 it is illustrated the linear speed of the centre of each gear vs the rotating speed of the turbine shaft. As all the gears rotate at the same speed around the centre of the PGS, it is expected that the linear speed of point A is higher than the one on the centre of the planet gear and higher than the one of point B according to this relation $A > P > B$. Point A and point B are shown in Figure 210 and P refers to centre of the planetary gear. The relation between the linear speeds on the different points can be seen on the graph of Figure 211.

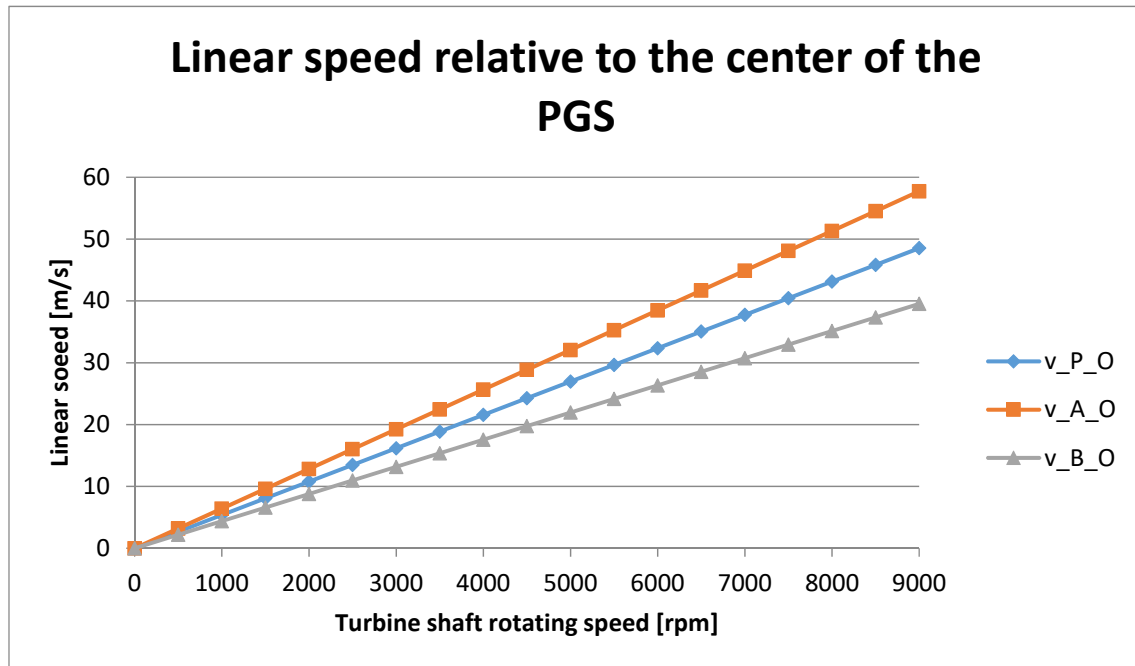


Figure 211 – Linear speed relative to the centre of the PGS in D gear

According to the graph of Figure 211, linear speed of point A is higher than point B. This is because point A is placed in a circle (ring gear) that has a greater radius than the one of point B (sun gear). In D, all the gears rotate together. The linear speed of point P, that corresponds to the centre of the planet gear has a value between the linear speed of point A and point B because this point is located in a radius between both points.

3.5.9.2.2 Neutral gear

In this subsection, it will be studied the kinematics of the PGS in Neutral gear. When N is selected, none of the clutches is engaged; sun gear and planet carrier rotate freely and ring gear rotates with the CC.

For this study, three scenarios were considered. In the first one it was considered a null rotating speed of the sun gear while for the second and third scenarios it was considered rotating speeds of +100 rpm and -100 rpm, respectively.

In N gear, the engine is rotating, so does turbine shaft and consequently the ring gear. If the driver presses the accelerator pedal, there is a variation in the rotating speed on the engine but the vehicle does not move.

Sun gear is directly connected to the primary shaft that is connected to the variator. This means that if the car moves there is variation in the rotating speed of the sun gear. This speed depends on the speed of the vehicle and not on the rotating speed of the engine. The car can roll back or forward. Evidently, the speed is not significant comparing to the rotating speed that can be performed by the engine. In this study, it was considered a maximum rotating speed of the sun gear of 100 rpm when the car moves back or forward.

In the graph of Figure 214, it is presented the rotating speed of all the gears of the PGS when the sun gear speed is equal to zero (when the car is steady). According to Figure 214, as expected, the rotating speed of the CC and ring gear are exactly the same. That is the reason why the lines are overlapping each other. As the sun gear was considered to have no rotating motion, the speed is null independently of the angular speed of the engine.

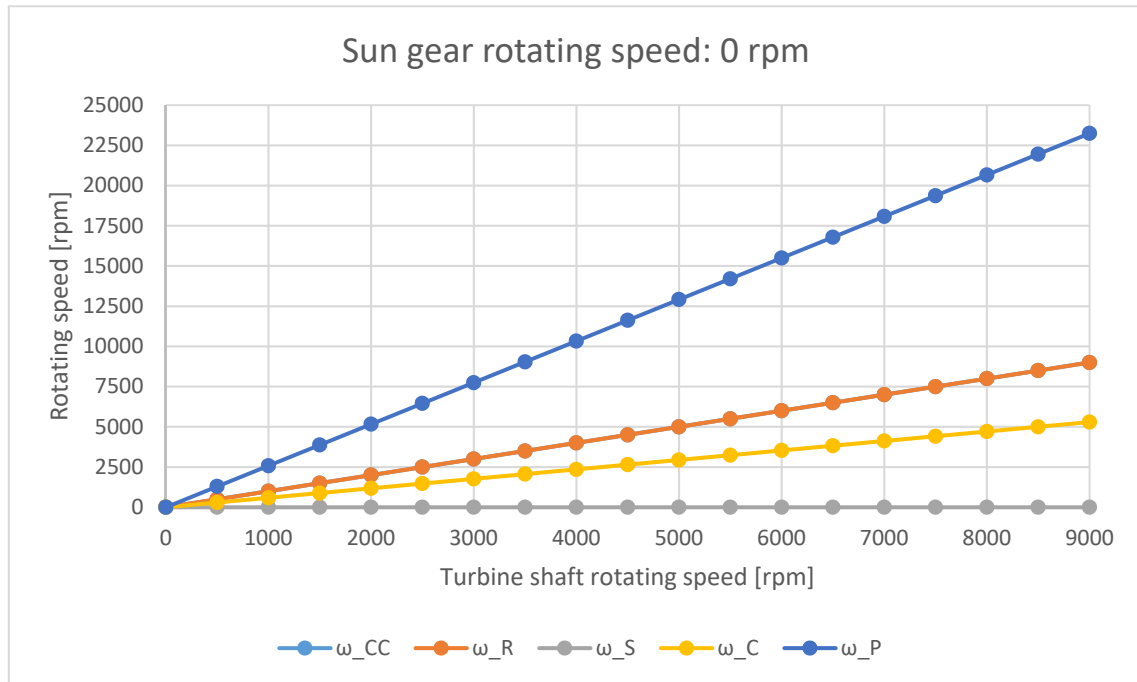


Figure 214 – Graph of rotating speed of PGS gears vs the rotating speed of the turbine shaft in N

In Figure 213, it is plotted the rotating speed of all the gears of the PGS for a rotating speed of 100 rpm.

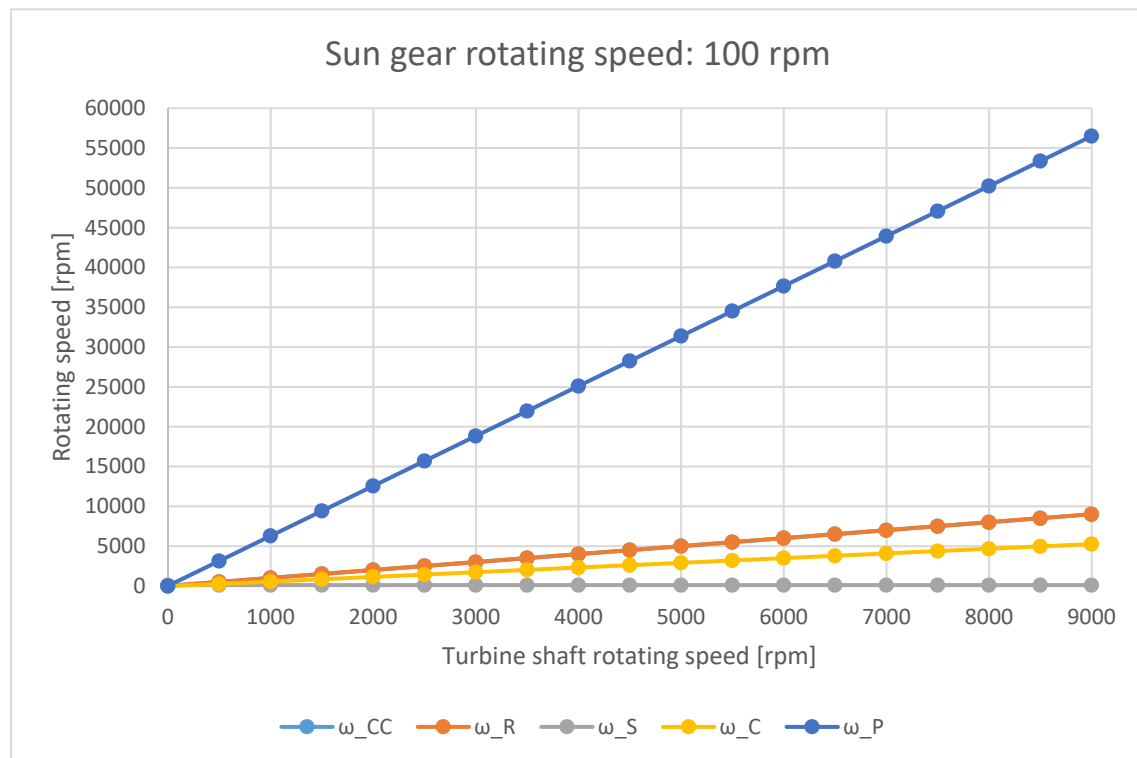


Figure 213 – Graph of rotating speed of PGS gears vs rotating speed of the turbine shaft in N

As can be seen on the graph of Figure 213, as expected the rotating speed of the sun gear is constant and equal to 100 rpm. Clutch carrier and ring gear rotate at the same speed that is why ring gear curve is overlapping the one of CC. Planet gears rotate significantly faster than the other gears. All the gears rotate to the positive direction.

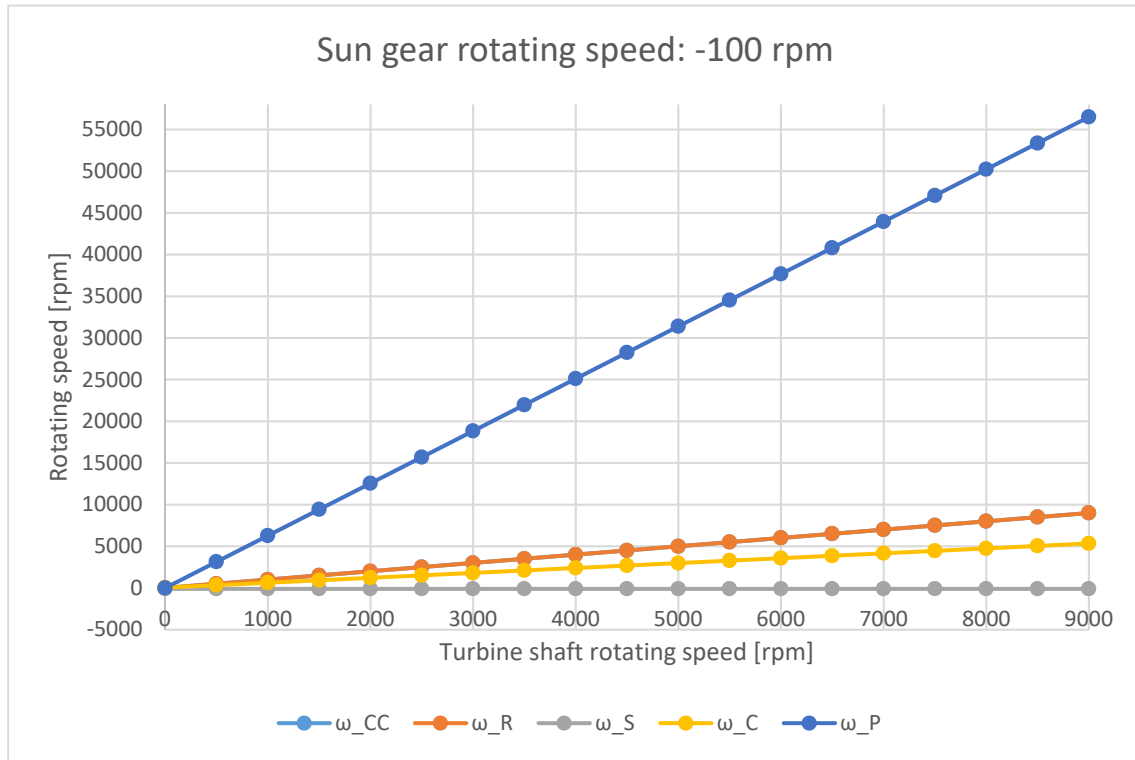


Figure 215 - Graph of rotating speed of PGS gears vs rotating speed of the turbine shaft in N

The conclusions related to the graph of Figure 215 are the same of the graph of Figure 213 but in this case the sun gear rotates to the negative direction.

3.5.9.2.3 REV gear

In this chapter, it will be studied the kinematics of the PGS in REV gear.

When R is selected, REV clutch is engaged but not the FWD clutch; ring gear rotates solidary with the CC, sun gear and planet carrier rotate freely.

In Figure 216, it is presented the angular speeds of each gear relative to their own centre of rotation. According to the graph of Figure 216, sun gear rotates to an opposite direction of planet gears and ring gear. Planet gears are with no doubts the gears which roll faster.

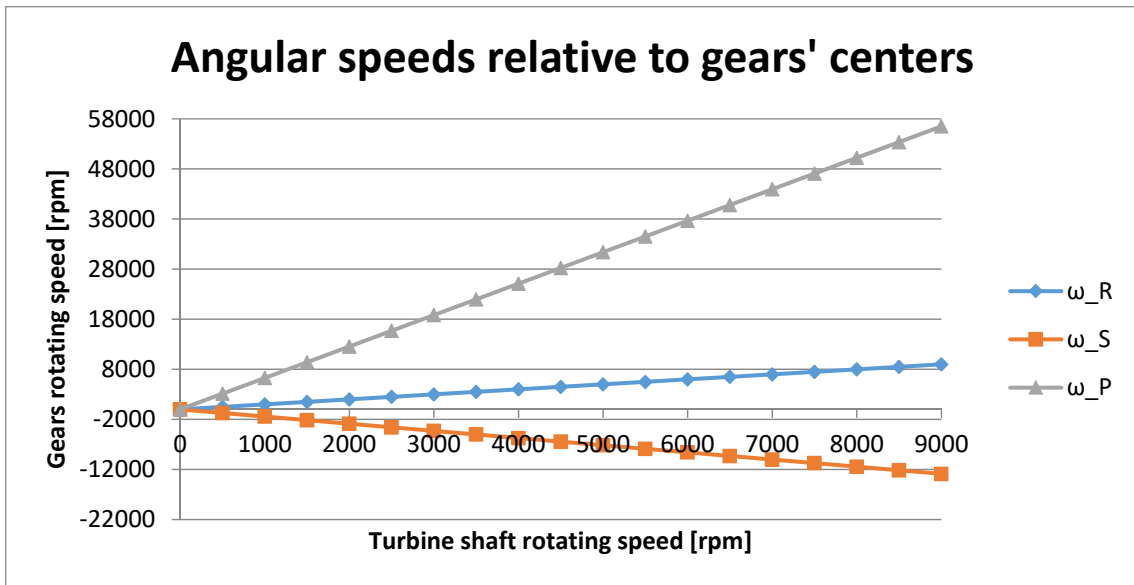


Figure 216 – Angular speeds relative to gears' centres of R

In the graph of Figure 217, it is plotted the angular speed of each gear relative to the centre of the carrier which is the centre of the PGS.

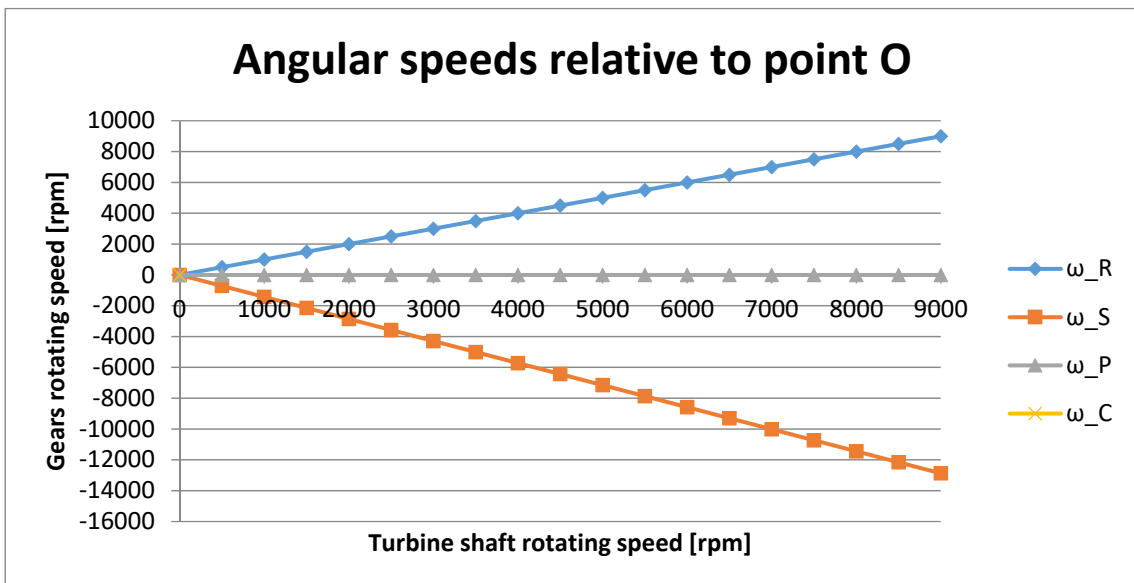


Figure 217 – Angular speeds relative to point O in R

Having in account that point O corresponds to the centre of the PGS, it is expected the angular speed of the carrier is zero since it is rotating around itself. Sun gear is rotating in the opposite direction of ring gear. Sun gear roll faster than all the other gears.

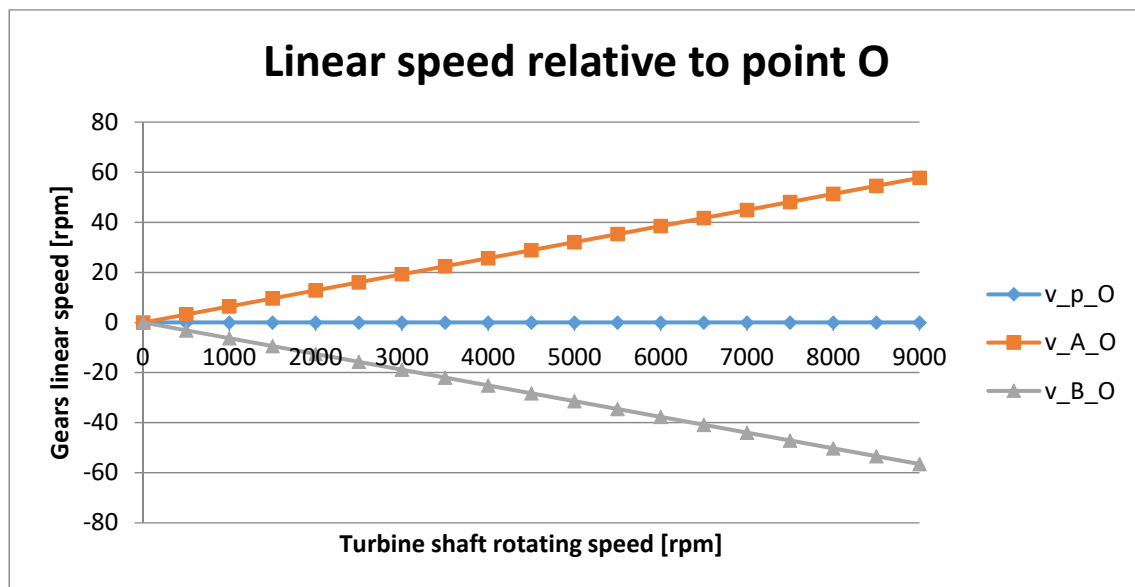


Figure 218 – Planet gears' linear speed relative to point O in R

In Figure 218, it is illustrated the relation between the linear speed of each gear relative to the centre of the PGS (point O) and the turbine shaft rotating speed.

As can be seen in the graph of Figure 218, the linear speed of point A and B is exactly the same but in opposite directions. This means that the planet gear is rotating around its own centre of rotation. The linear speed of the centre of the planet gear is constant and null.

3.5.9.3 Torque analysis

In this chapter, it will be done an analysis of the torque generated and transmitted on the PGS. This will be useful for the calculations of the loads applied on the gears of the PGS.

3.5.9.3.1 Drive gear

One of the requirements is that the maximum torque value that can be applied or transmitted to the DNR subsystem is 360 N.m. The torque from the torque converter is

transmitted to the turbine shaft assy. As explained before in Drive mode both ring gear and sun gear rotate at the same time with the same speed because both of them are connected to the clutch carrier that is part of the turbine shaft assy. This means that the torque from the turbine shaft assy is transmitted to the sun gear and ring gear with the same value.

In Figure 219 it is presented a schematic of the PGS on D with the representation of the torque on ring and sun gears. Ring gear is on green, planetary gear on blue and sun gear on orange.

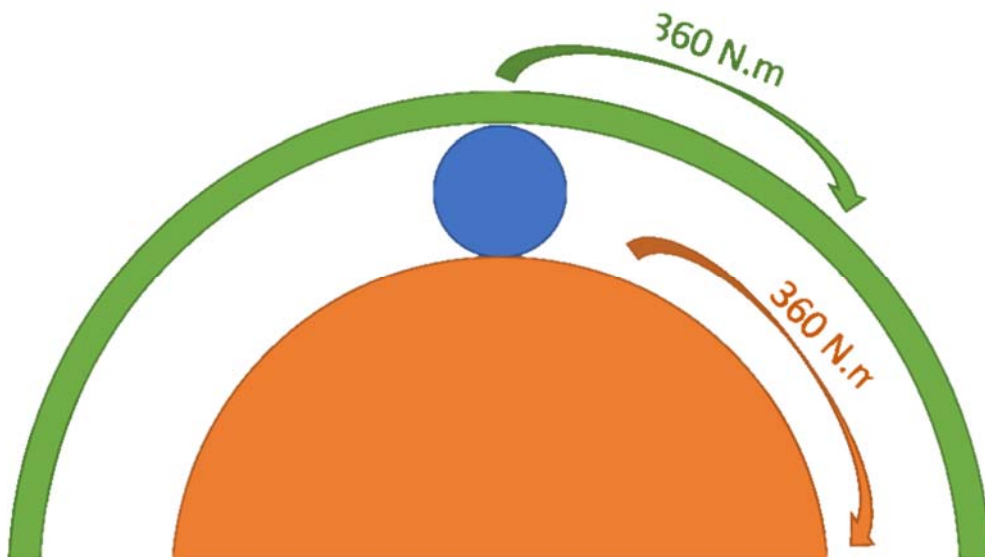


Figure 219 - Illustration of torque orientation on the PGS in D

3.5.9.3.2 Neutral gear

In Neutral gear there is no transmission of torque since this gear is selected when the vehicle is steady and it is not desired to drive it.

3.5.9.3.3 REV gear

A requirement regarding to the REV gear is that the maximum torque applied on the sun gear must be 175 N.m. Due to the ratio of PGS, to obtain a torque of 175 N.m on the sun gear it is necessary to be applied a torque of approximately 250 N.m on the CC.

$$T^{CC} = \frac{T^S}{R} = \frac{175}{0.7} = 250 \text{ N.m} \quad \{3.30\}$$

The amount of torque on the carrier can be calculated as follows:

$$\begin{aligned} \sum T &= 0 \Leftrightarrow \\ \Leftrightarrow T^R + T^S + T^C &= 0 \Leftrightarrow \\ \Leftrightarrow 175 + 250 + T^C &= 0 \Leftrightarrow \\ \Leftrightarrow T^C &= -425 \text{ N.m} \end{aligned} \quad \{3.31\}$$

As the SF that must be considered is equal to 1.2:

$$T^C = -425 \times 1,2 = -510 \text{ N.m}$$

In Figure 220, it is illustrated a schematic of the PGS with the representation of the obtained torques on each gear. The red arm corresponds to the PGS carrier.

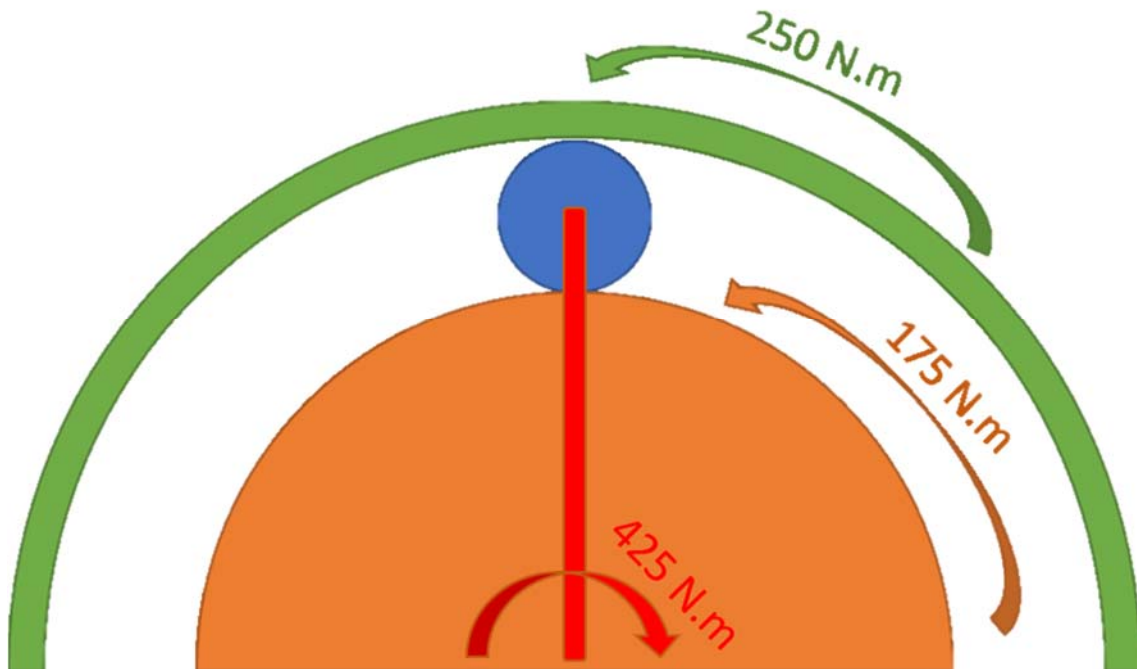


Figure 220 – Schematic of the resulting torques on R in the PGS

3.5.9.4 Loads

In this chapter it will be studied the loads applied on each gear of the PGS. In D there is torque transmission but there is no counter-torque or reaction torques. In N there is no torque transmission. In R there is a considerable transmission of torque. This means that sun gear, ring gear, planetary gear and planetary pins will be submitted to considerable loads and consequent stresses.

In Figure 221, it is presented the free body diagram of the planetary gear where R corresponds to the ring gear, S to the sun gear and C to the PGS carrier.

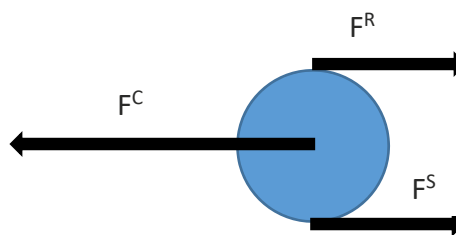


Figure 221 – FBD of planetary gear

It is necessary to calculate the value of the reaction force of the sun gear and ring gear. The reaction force of the ring gear can be calculated as follows:

$$F^R = \frac{T^R \times SF}{IR^R} = \frac{250 \times 1,2}{61,258 \times 10^{-3}} = 4897,320N$$

The reaction force of the sun gear can be obtained as follows:

$$F^S = \frac{T^S \times SF}{OR^S} = \frac{175 \times 1,2}{41,944 \times 10^{-3}} = 5006,676N$$

The value of F^C can be calculated as follows:

$$F^C = \frac{T^C \times SF}{R^C} = \frac{-510}{51,5 \times 10^{-3}} = -9902,913N$$

The value of F^C can be confirmed according to the FBD (Free body diagram) of the planetary gear:

$$\sum F = 0 \Leftrightarrow 4897,320 + 5006,676 - 9902,913 = 0$$

3.5.10 Planetary pin

It is known that there are three planetary gears and so three planetary pins. By this way, the force applied on each planetary pin can be calculated as follows:

$$F^{pin} = \frac{F^C}{N^{pins}} = \frac{9902,913}{3} = 3300,971N$$

The shear stress can be calculated as follows:

$$\tau = \frac{3300.971}{2 \times 50.2} = 32.9MPa \quad \{3.32\}$$

Where the area of the cross section of the pin is 50.2 mm^2 and τ corresponds to the shear stress in MPa. Since this pin is made of steel and the yield strength is about 250 MPa this component can be validated with a wide safety margin.

3.6 Results comparison between FWD and REV clutch

The main purpose of this chapter is to have a notion of what are the differences between FWD and REV clutch sets in terms of the distinct parameters such as transmissible torque, drag losses, clutches' pressure and so on.

3.6.1 Transmissible torque

In the graph of Figure 222, it is presented the comparison between the average transmissible torque of FWD and REV with the increase of the hydraulic pressure.

As can be seen in the graph, the amount of transmissible torque is always higher on REV clutch set. One of the reasons for this is the fact that the dimensions of the clutch plates of REV are clearly higher than FWD.

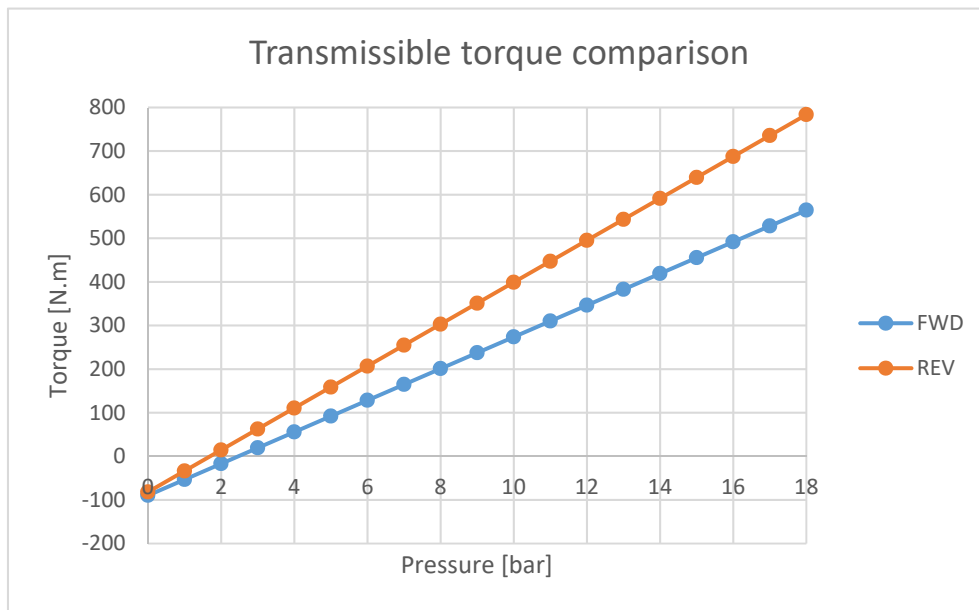


Figure 222 – Comparison of average transmissible torque between FWD and REV clutch sets

3.6.2 Drag losses

In Figure 223, it is plotted a graph where the drag losses of FWD and REV clutches are compared. These values were obtained considering the nominal feeding flow rate of each clutch set. The nominal feeding flow rate of FWD is 0.3 L/min and 0.45 L/min for REV.

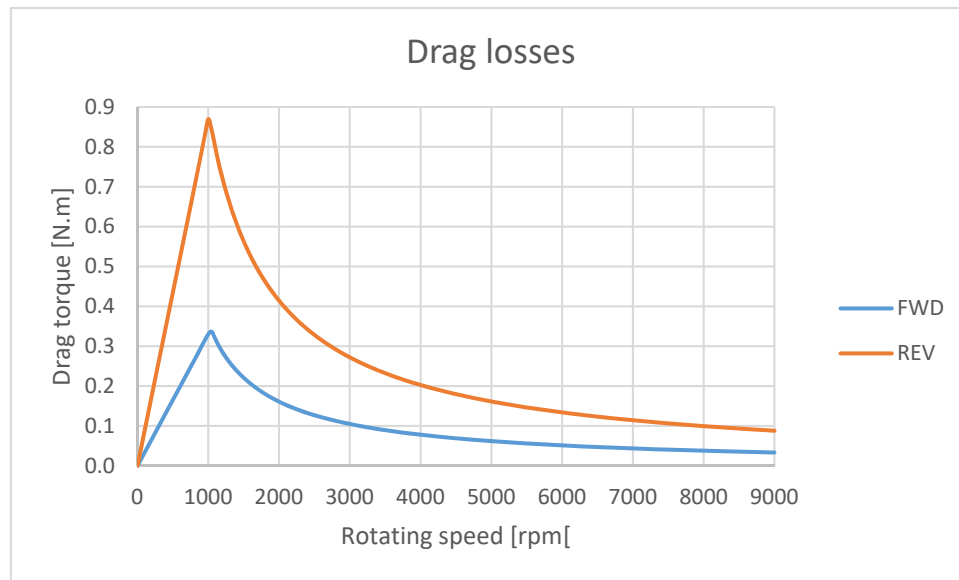


Figure 223 – Comparison of drag losses between FWD and REV clutch sets for nominal feed flow rate

Based on the graph of Figure 223, it is conclusive that the drag losses on REV clutch are considerably higher than on FWD. Once again, a reason for this fact is the larger dimensions of the clutch plates of REV clutch set but also the fact that the nominal feeding flow rate for REV is higher than for FWD.

In Figure 224, it is illustrated the comparison between the drag losses of FWD and REV calculated on the primary shaft.

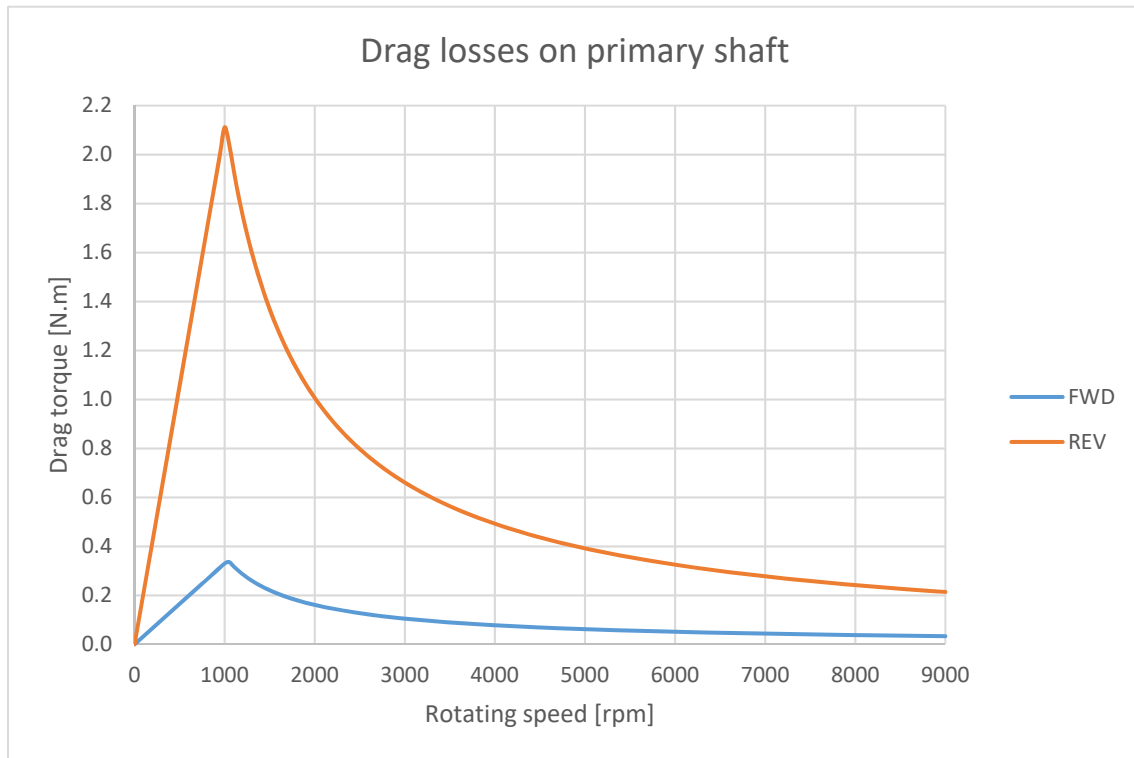


Figure 224 – Comparison of drag losses calculated on the primary shaft between FWD and REV clutch sets for nominal feed flow rate

As expected, in the graph of Figure 224, the curve of FWD is exactly the same as the graph of Figure 223. The reason for this fact is that the PGS ratio has an influence in the drag losses calculated on the primary shaft. Since in D the PGS ratio is equal to 1, the drag losses calculated on the clutch itself is the same of the one calculated on the primary shaft. On the other hand, in R PGS presents a ratio that will affect considerably the drag losses calculated on the primary shaft. However, in this case, the drag losses suffer an increase when calculated on the primary shaft, which is unfavourable.

In Figure 225, it is plotted the comparison between FWD and REV drag losses for a maximum feeding flow rate. Comparing the graph of Figure 225 to the graph of Figure 223, it is perceptible an increase in the amount of drag losses. This means that, as previously explained, the feeding flow rate as a negative effect on the drag losses.

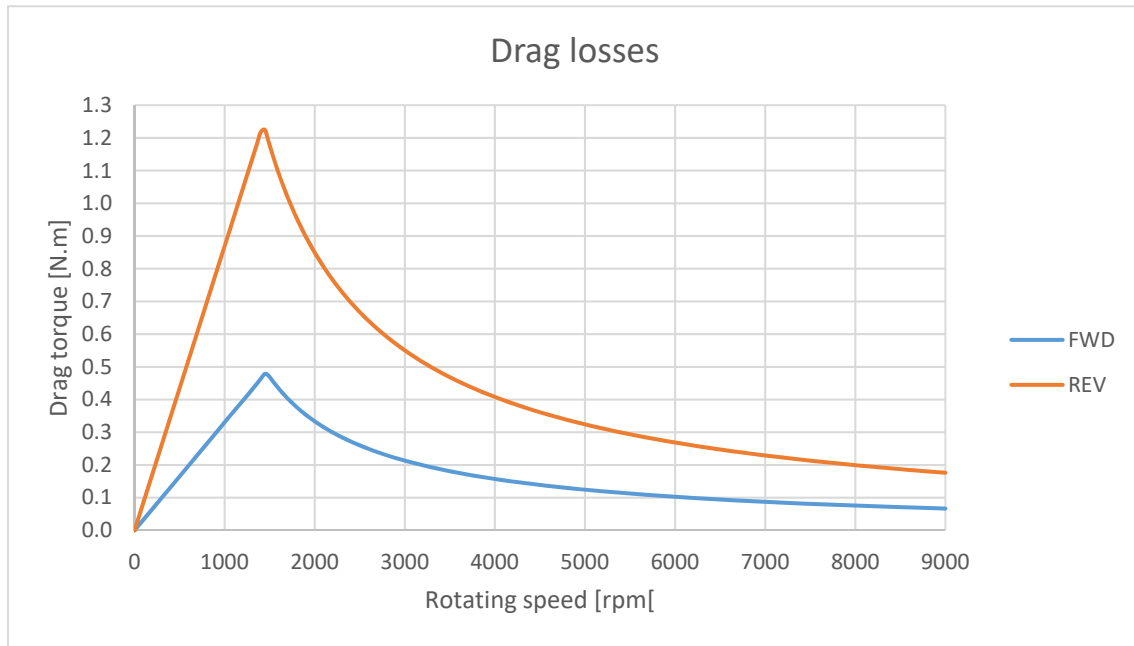


Figure 225 - Comparison of drag losses between FWD and REV clutch sets for maximum feed flow rate

In Figure 226, it is illustrated the drag losses calculated on the primary shaft for FWD and REV.

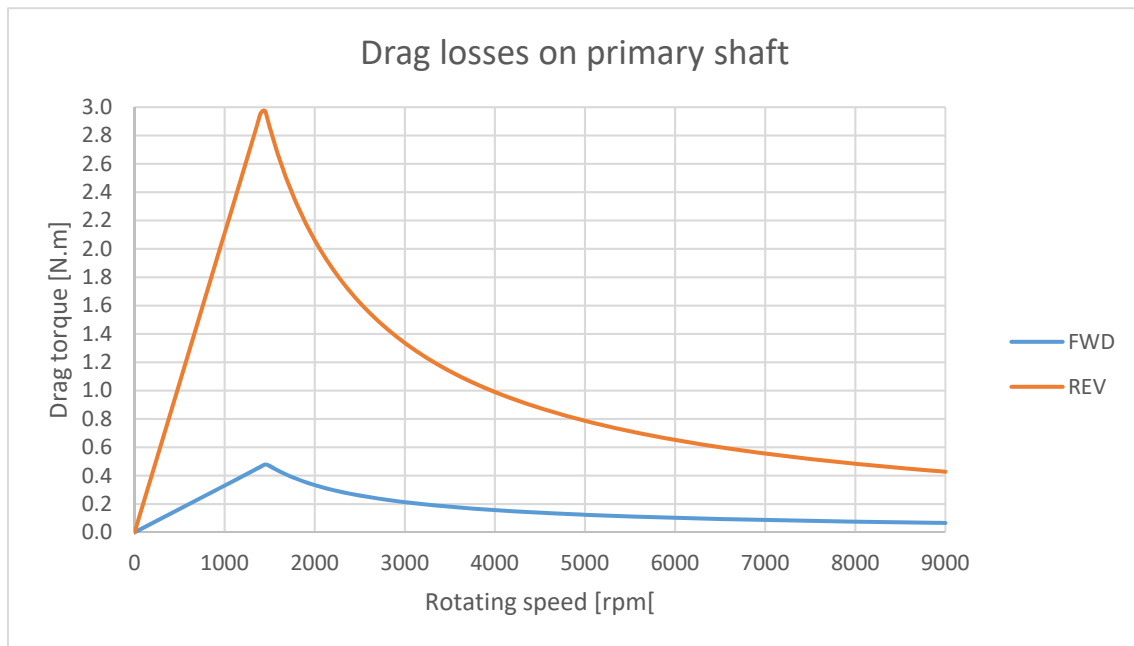


Figure 226 - Comparison of drag losses calculated on the primary shaft between FWD and REV clutch sets for maximum feed flow rate

Once again, there is a significant increase in the amount of drag losses when calculated on the primary shaft while the curve of FWD keeps the same.

3.6.3 Clutches' pressure

In Figure 227, it is illustrated the comparison between the clutch plates' pressure of FWD and REV. For this graph it was considered the nominal scenarios.

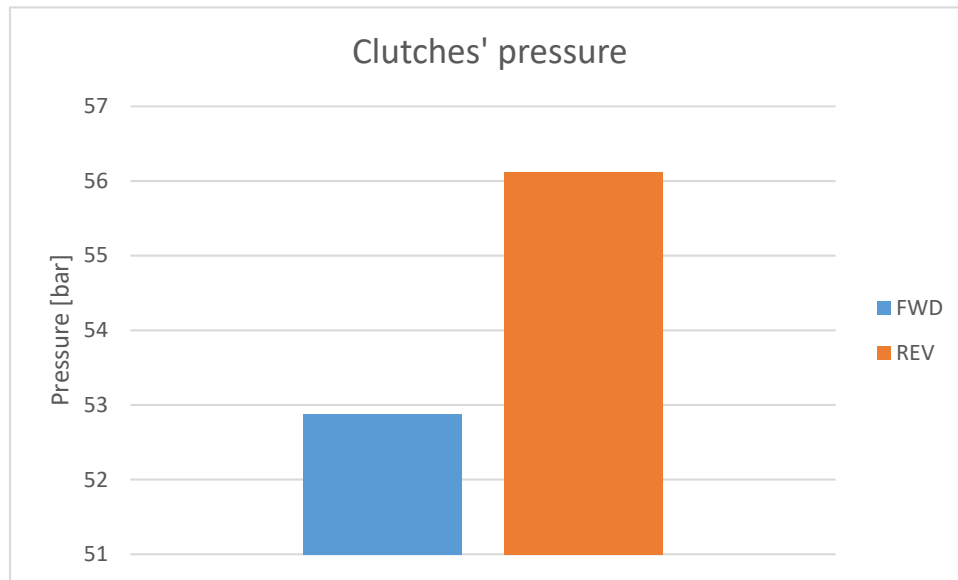


Figure 227 – Comparison of clutch plates' pressure between FWD and REV clutch sets

According to the graph it is visible that there is a difference of pressure of about 3 bar between FWD and REV where the REV is the one which presents the higher pressure. At a first sight it is expected that the pressure on REV clutch plates are lower due to the larger dimensions however the amount of transmissible torque and axial force is higher.

3.7 Ramp-down test

3.7.1 Description

The main purpose of this test is to evaluate the drag torque of the DNR subsystem on FWD and REV clutches, separately, measured on the input shaft. After obtaining the values of the drag torque, they will be compared with the calculated analytically on chapter 3.8.

Three configurations of the DNR subsystem were applied during the test:

- Neutral gear with both clutch packs installed;
- Neutral gear with FWD clutch pack installed;
- Neutral gear with REV clutch pack installed.

By “clutch pack” are considered the clutch discs and endplates.

3.7.2 Procedure

Firstly, it was installed the DNR subsystem with none of the clutch packs installed. By this way it is possible to measure the friction torque created by all the components of the test rig (bearings, seals, etc.). Only after that, the three configurations were applied one by one. Tests run five times for each configuration, including the initial one (none of the clutch packs installed). With this procedure, it is possible to achieve more reliable results. Once the DNR is installed in the test rig, the electric motor is activated. It is set a rotating speed for the motor which will be the same on the turbine shaft during a certain time until it is completely stable. After that, the motor is switched off. The period that the all system takes to completely stop is measured. With the mass moment of inertia, initial rotating speed and period of time it is possible to measure the drag torque in each configuration. All the calculations are expressed on the following chapter.

3.7.3 Calculations

Before plotting the graphs, it is necessary to refine the returned data from the results. The results are returned for each timestamp which means that the values of drag torque are quite accurate and any interference in the measurement sensors are presented on the results. In each timestamp there are several instances. For each instance corresponds a certain value of rotating speed and drag torque. Firstly, it was rounded the value of the timestamp and then calculated the values of drag torque and rotating speed which correspond to that timestamp. It was calculated the average of all values which correspond to the same timestamp. After that it was already possible to plot the desired graphs.

3.7.4 Results

The obtained results for the different configurations are presented below.

3.7.4.1 FWD clutch set installed

The test was done with an oil temperature of 37.1 °C and an oil flow on the DNR of 1.1 L/min.

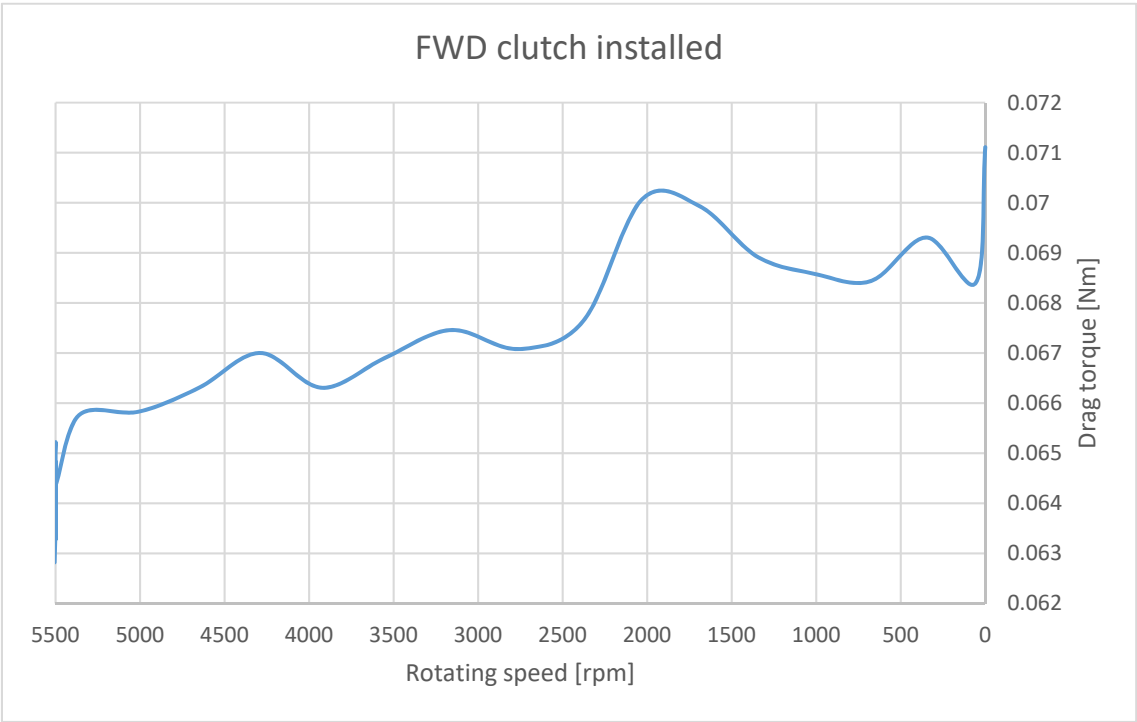


Figure 228 – Results of the ramp-down test for FWD clutch installed

3.7.4.2 REV clutch set installed

The test was done with an oil temperature of 30 °C and an oil flow on the DNR of 1 L/min.

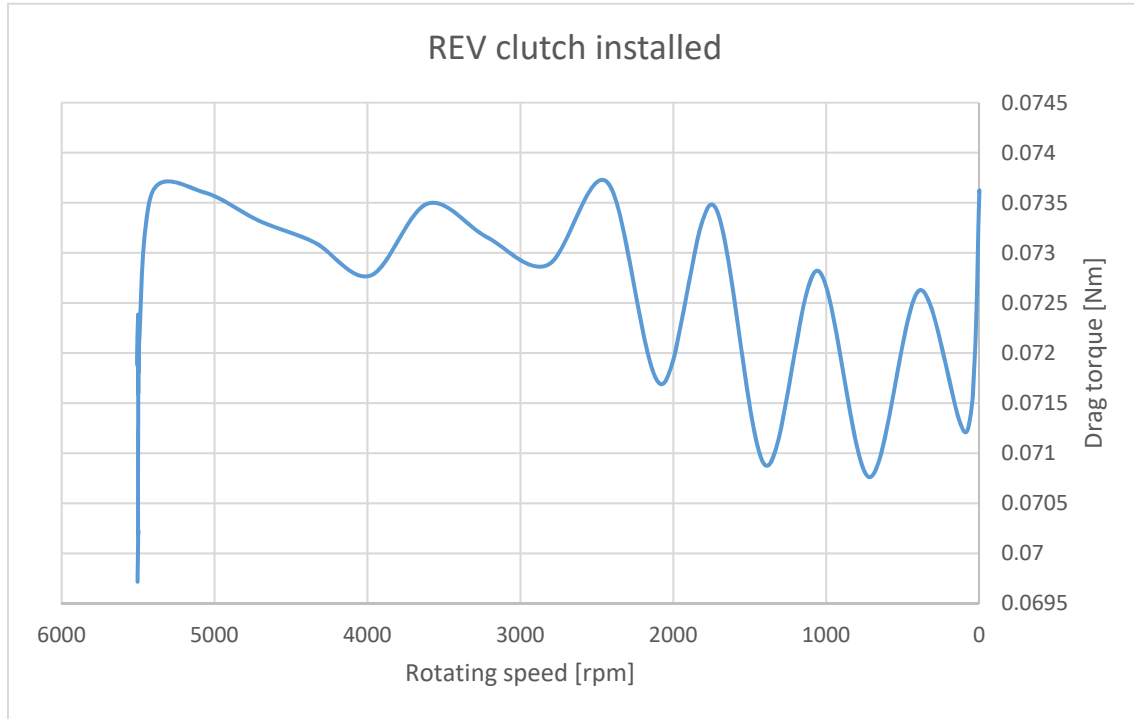


Figure 229 – Results of the ramp-down test for REV clutch installed

3.7.4.3 Both clutch sets installed

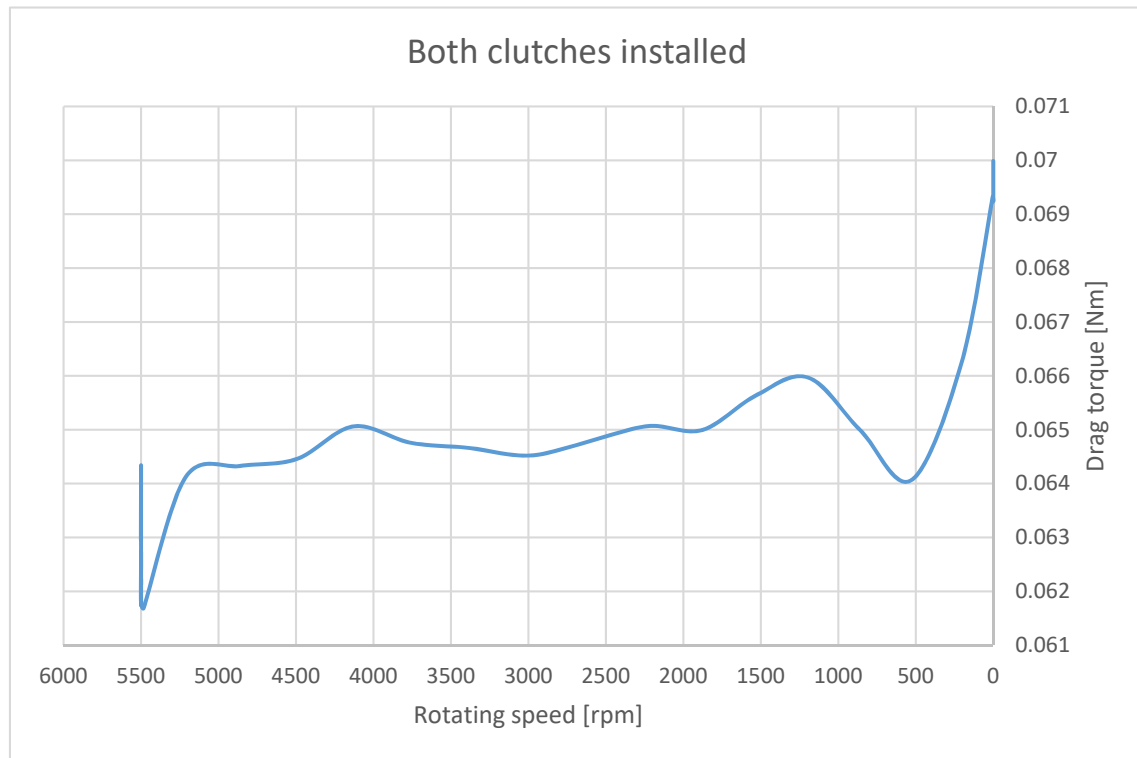


Figure 230 – Results of the ramp-down test for both clutches installed

3.8 Comparison between tests and analytical calculations

In this chapter it will be compared the values obtained in the calculations with the obtained in the ramp-down test.

3.8.1 FWD clutch set installed

By comparing the graph of Figure 228 with the graph of Figure 231 it is conclusive that the drag torque and also the shape of the graph is not the same. The maximum obtained value for the drag torque of the test was approximately 0.07 Nm while in the calculations was 0.75 Nm, for the same conditions of temperature and oil flow. As expected, the obtained graph on the tests presents low drag torque at high rotating speeds however the shape of the curve is not as well defined as the theoretical curve.

In conclusion, the obtained results of the ramp-down test are not validated since the calculations model has already been validated.

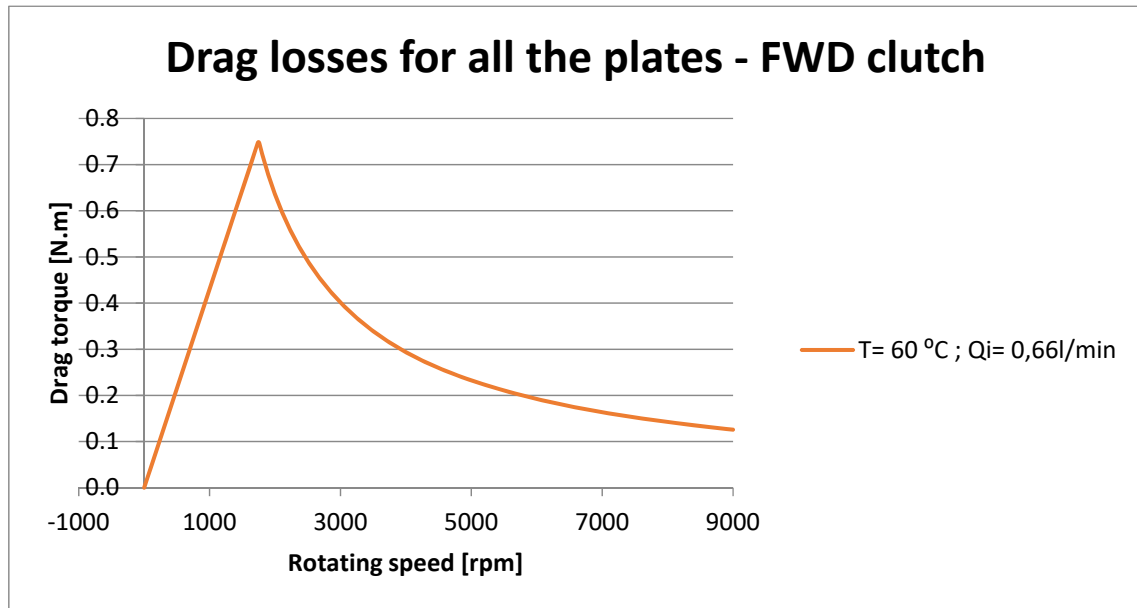


Figure 231 – FWD clutch drag torque for comparison with ramp-down test

3.8.2 REV clutch set installed

By comparing the graph of Figure 229 with the graph of Figure 232 it is conclusive that the drag torque and also the shape of the graph are not matching. The maximum obtained value for the drag torque of the test was approximately 0.074 Nm while in the calculations was 1.4 Nm, for the same conditions of temperature and oil flow.

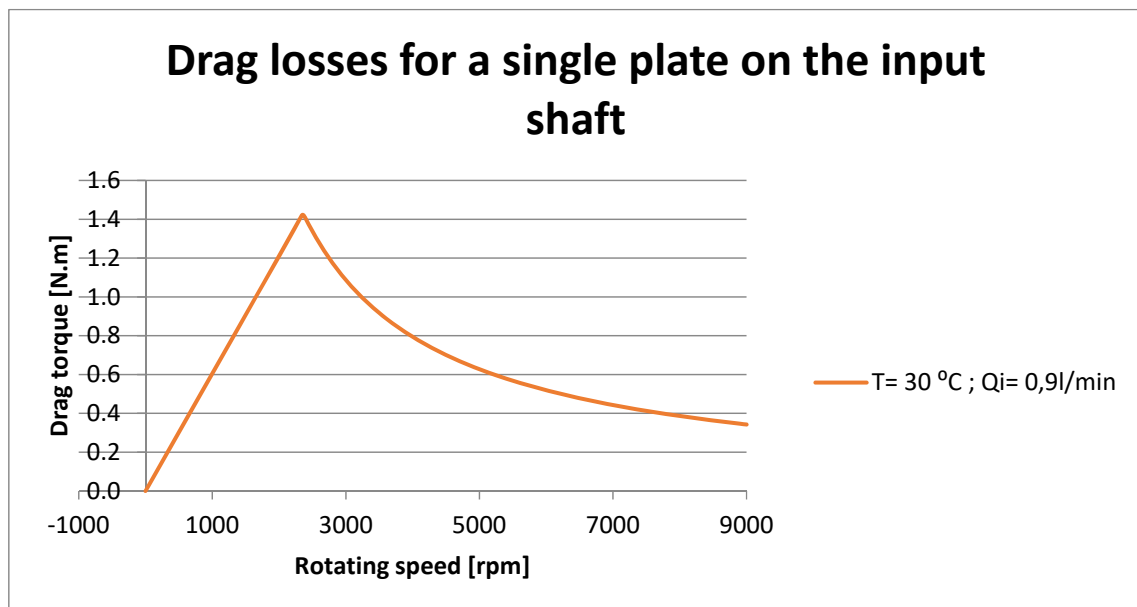


Figure 232 - REV clutch drag torque for comparison with ramp-down test

In addition, the drag torque at high speeds is higher than at low speeds for the graph of the ramp down test which is completely opposite to the theoretical results which is expected to be lower at low speeds.

In conclusion, the obtained results of the ramp-down test are not validated since the calculations model has already been validated.

3.9 Requirements description

3.9.1 Requirement #1 - Max Clamping Torque in FWD Clutch

The full range of clamping torque of the FWD clutch must be 360 Nm clutch capacity multiplied by a safety margin of 1.10 (i.e., $360 \times 1.10 = 400$ Nm) for the primary functional operating range (PFOR). This means that the DNR must be able to continuously transmit a total torque of 400 N.m to the primary shaft.

3.9.2 Requirement #2 - Max Clamping Torque in REV Clutch

The Reverse clutch must be able to continuously transmit a torque between 0 and the equivalent of 193 Nm at the primary shaft multiplied by the safety factor of 1.10 = 175×1.10 , for all speeds and temperatures within the PFOR (i.e., 467 Nm at R-clutch).

3.9.3 Requirement #3 - Drive ratio range

The Drive gear ratio range should be equal to one. This means that the rotating speed on the primary shaft should be the same as the one on turbine shaft. This applies to the torque, too.

3.9.4 Requirement #4 - Reverse ratio range

Reverse ratio range should be equal to $0.7 \pm 0.5\%$ (Table 74). This means that the rotating speed of the primary shaft comparing to the speed of the CC is lower.

Table 74 – REV ratio range

Minimum	Nominal	Maximum
0.697	0.700	0.704

3.9.5 Requirement #5 - Drive clutch zero-slip current-to-torque sensitivity

The zero-slip current to clutch torque sensitivity in Drive must be less than 45 Nm/bar evaluated at the clutch. This means that for each variation in the pressure of 1 bar, the torque increase or decrease must be at the maximum 45 Nm.

3.9.6 Requirement #6 - Maximum guaranteed clutch pressure to reach maximum clamping torque in clutches

The maximum clamping torque for each clutch (FWD and REV) should be achieved at a maximum pressure of 15 bar. The values of the maximum clamping torque for FWD and REV are defined on 3.9.1 and 3.9.2 , respectively. For FWD set, at 15 bar of hydraulic pressure, the transmissible torque should be at least 400 N.m. For REV, for a pressure of 15 bar, a torque of at least 193 Nm must be achieved. In case of being measured on the primary shaft, the value of the torque must be at least 467 Nm.

3.9.7 Requirement #7 - Oil compatibility

Material and other liquids (rust inhibitor, release agent, etc.) must be compatible with the transmission oil.

3.9.8 Requirement #8 - Maximum input torque in Drive

Maximal torque that can be seen by the DNR in Drive is equal to 360Nm. This is the maximum torque that can be transmitted from the torque converter to the turbine shaft. This parameter is not controlled by the DNR subsystem. It is just a base for the design and calculations of the DNR.

3.9.9 Requirement #9 - Cooling/lubrication budget

The cooling and lubrication of the DNR system shall function with a combined flow of maximum 1.0 L/min at engine speeds beyond 600 rpm, for all ambient temperatures and without lubrication during towing for 200 km and vehicle speed of 80 km/h. The desired flow rate for cooling is 0.5 L/min. A flow rate of 0.5 l/min must be used as long as possible.

3.9.10 Requirement #10 - Safety Factor Torque at primary shaft

For design calculations, must be considered a safety factor for actuation of 1.1.

3.9.11 Requirement #11 - Drag torque of Drive clutch (in Reverse and Drive/Open)

The drag torque of the Drive clutch in Reverse and Drive/Open must be less than 1 Nm, evaluated at the Drive clutch itself:

- For 60° C sump oil temperature;
- For all applicable DNR cooling flows;

3.9.12 Requirement #12 - Drag torque of Reverse clutch (in Drive and Reverse/Open)

The drag torque of the Reverse clutch in Drive must be less than 0.26 Nm for a single plate, evaluated at the DNR input shaft:

- For engine speed of 1500 rpm;
- For 60 °C sump oil temperature;

3.9.13 Requirement #13 - Drag torque in Neutral

The combined drag torque of both DNR clutches in neutral must be less than -2.7 to 2.7 Nm (evaluated at the primary shaft) for the FFOR (Fully functional operating range).

3.9.14 Requirement #14 - Safe state 1 - Open all DNR clutches

The safe state must be performed by the release of springs pushing the pistons to NO state.

3.9.15 Requirement #15 - Sufficient separation between Open mode and preload pressure

The difference between the pressure target for Open mode and the lowest preload pressure must be larger than the maximum total inaccuracy at 0 Nm of 0.36 bar, both in D and R;

3.9.16 Requirement #16 - Normally Neutral

It must not be required any energy to keep both DNR clutches open.

3.9.17 Requirement #17 - Sufficient separation between preload pressure and biting point

The mechanical module must make sure that the smallest difference between preload pressure and kiss point pressure is at least 0.1 bar for R and 0.4 bar for D.

3.9.18 Requirement #18 - Reverse clutch zero-slip current-to-torque sensitivity

The zero-slip current to brake torque sensitivity in Reverse must be less than 55 Nm/bar evaluated at the clutch.

3.9.19 Requirement #19 - Maximum input torque in Reverse

Maximal torque that can be seen by the DNR in Reverse is equal to 250 Nm.

3.9.20 Requirement #20 - Max allowed clutch pressures

The maximum allowed pressure applied on the DNR is 18 bar. The electronic module has to guarantee this parameter.

3.9.21 Requirement #21 - Piston Stroke Volume D

The piston stroke volume must be less than 21cm^3

3.9.22 Requirement #22 - Piston Stroke Volume R

The piston stroke volume must be less than 24cm^3 .

3.9.23 Requirement #23 - Max. Pressure on FWD clutch plates

The max. allowed pressure on the FWD clutch friction surface is 3.71 MPa.

3.9.24 Requirement #24 – Max. pressure on REV clutch plates

The max. allowed pressure on the REV clutch friction surface is 3.95 MPa.

3.9.25 Requirement #25 – Max. rotational speed of the Needle bearing

The maximum allowed rotating speed of the needle bearing is 30000 rpm.

3.9.26 Requirement #26 – Max. rotating speed of the needle roller thrust bearing

The maximum allowed rotating speed of the needle roller thrust bearing is 12000 rpm.

3.9.27 Resume

Table 75 – Requirement's resume

Requirement #	Parameter
1	> 400 Nm
2	> 467 Nm
3	1
4	0.7±0.5%
5	< 45 Nm/bar
6	> 400 Nm at 15 bar (FWD clutch) > 467 Nm at 15 bar (REV clutch)
7	Datasheet and tests results
8	360 Nm
9	Max. 1 L/min and nominal 0.5 L/min
10	1.1
11	< 1 Nm at 60 °C for the
12	0.26 Nm at 60 °C at 1500 rpm
13	-2.7 to 2.7 Nm
14	Springs pulling the piston to NO state
15	> 0.36 bar
16	Clutches normally opened
17	> 0.4 bar for D > 0.1 bar for R
18	< 55 Nm/bar
19	250 Nm
20	18 bar
21	< 21 cm ³
22	< 24 cm ³
23	< 3.71 MPa
24	< 3.95 MPa
25	< 30000 rpm
26	< 12000 rpm

3.10 Design validation

3.10.1 Requirements verification

3.10.1.1 Resume

Below it is presented a table which is a resume of the validation of the requirements. This table was filled in at the same time that the requirements were validated which means it was progressively filled in.

Table 76 - Requirements' validation resume

Requirement nr.	Validation	Justification
1	Validated	488.604 N.m at max. pressure (18 bar)
2	Validated	696.046 N.m at max. pressure (18 bar)
3	Validated	Inexistence of variation of transmission ratio. Sun gear is connected to the input and output shaft
4	Validated	0.699
5	Validated	39.899 Nm/bar
6	Not validated for FWD clutch; Validated for REV clutch	390.337 Nm at 15 bar (FWD clutch) 566.378 Nm at 15 bar (REV clutch)
7	Validated	Datasheet and aspect of parts after testing
8	Not verified	Not enough information
9	Not verified	Controlled by the hydraulic unit, not DNR subsystem
10	Validated	Incorporated in requirement nr. 1, 2 and 6
11	Validated	0.7 Nm (maximum feed flow rate) 0.4 Nm (nominal feed flow rate)
12	Not validated	0.3 Nm (maximum feed flow rate) 0.24 Nm (nominal feed flow rate)
13	Validated	1 Nm
14	Validated	Pistons are pulled by the springs
15	Validated	0.502 bar (FWD SP) 0.436 bar (REV SP)
16	Validated	SP pulling the pistons
17	Validated	1.679 bar (FWD clutch set)

		1.254 bar (REV clutch set)
18	Validated	52.971 Nm/bar
19	Not verified	Controlled by the hydraulic unit
20	Not verified	Controlled by the hydraulic unit
21	Validated	19.9 cm ³
22	Validated	17.2 cm ³
23	Not validated	5.29 MPa
24	Not validated	5.71 MPa
25	Not validated	56500 rpm
26	Not validated	12900 rpm

3.10.1.2 Requirement #1

Considering the fourth scenario which is the most conservative and considering the minimum torque of this scenario it is obtained a value of 408.604 N.m at the maximum pressure of 18 bar (Table 103) which is greater than 400 Nm. All the results of transmissible torque for all the scenarios are presented on chapters 6.5, 6.6 and 6.7.

3.10.1.3 Requirement #2

Based on the third scenario which corresponds to the more conservative scenario it is obtained a transmissible torque of 696.046 N.m for a hydraulic pressure of 18 bar (Table 121) which is greater than 467 Nm at primary shaft. All the results of transmissible torque of FWD clutch set for the three scenarios are presented on chapters 6.22, 6.23 and 6.24.

3.10.1.4 Requirement #3

As explained before, inside FWD mechanism there is no multiplication. i.e. all the gears are rotating at the angular speed as the shaft. Furthermore, the sun gear is connected at the same time to the turbine shaft (input shaft) (which is the component which transmits the motion from the torque converter) and to the primary shaft (which is the output shaft of the DNR subsystem). In conclusion, the transmission ratio in D is equal to 1.

3.10.1.5 Requirement #4

The transmission ratio for REV gear is calculated in chapter 3.5.9.1.2. The obtained value is 0.699 which is within the allowed range (Table 74).

3.10.1.6 Requirement #5

The maximum value is 39.899 Nm/bar for the maximum transmissible torque of the nominal scenario which is less than 45 Nm/bar. The value of Drive clutch zero-slip current-to-torque sensitivity was calculated for both scenarios as expressed on chapters 6.19 and 6.20.

3.10.1.7 Requirement #6

For FWD clutch, for the fourth scenario and the min. frictional coefficient it is obtained a transmissible torque of 390.337 for a hydraulic pressure of 15 bar. It is important to underline the fact that this scenario has in account a lot of parameters that influence negatively the amount of transmissible torque. This means that for the FWD clutch this requirement is not fulfilled since it is less 400 Nm. The tables with the amount of transmissible torque for all the scenarios of FWD clutch are presented on chapters 6.5, 6.6 and 6.7.

Regarding to the REV clutch the minimum transmissible torque at 15 bar is verified on the third scenario for the minimum coefficient of friction where an amount of 566.378 Nm is obtained. This scenario has in account some parameters that influence negatively the amount of transmissible torque. This means that for the REV clutch this requirement is fulfilled since it is greater than 467 Nm. The tables with the amount of transmissible torque for all the scenarios of REV clutch are presented on chapters 6.22, 6.23 and 6.24.

3.10.1.8 Requirement #7

This requirement can be verified in two different ways and in different stages. At a first stage it is necessary to read the technical datasheet of the oil and analyse if there is any constraint regarding to the metal or other materials in contact with the oil. The other stage that is the most important is about analysing the parts that are in contact with the transmission oil and verify if there are anything wrong with the aspect of the parts. On

the several parts that were submitted to tests it was not detected any problem. Having this in account the oil can be validated.

3.10.1.9 *Requirement #8*

First of all, this is a requirement that cannot be controlled by the DNR subsystem because the amount of torque that is transmitted to the turbine shaft (input shaft of the DNR subsystem) comes from the torque converter which belongs to another subsystem. This parameter can be controlled by the hydraulic unit of the CVT transmission. The maximum torque of 360 Nm is a reference for the design of the DNR subsystem. In order to check this requirement is necessary to submit the DNR subsystem to this load constant and cyclically in order to also check the fatigue behaviour of the different components. Another way to check the feasibility of the components is to analyse the FEA of the different components. By doing this as a first stage a lot of time and money can be saved. Having this account there aren't enough data to check this requirement.

3.10.1.10 *Requirement #9*

Once again, this parameter cannot be controlled by the DNR subsystem. Actually, it is controlled by the hydraulic unit. These values are the reference for the calculations of the drag losses for example. Having in account that there is no possible control on this factor by the DNR subsystem it cannot be verified.

3.10.1.11 *Requirement #10*

This is a requirement related to the transmissible torque. This value of SF was already taken in account on the requirement of the transmissible torque which was already studied. Having this account this requirement is fulfilled.

3.10.1.12 *Requirement #11*

In order to verify this requirement, it was necessary to plot the graph of the drag losses of all the plates of FWD clutch for a temperature of 60 °C. It was considered two feed flow rates: the maximum and the nominal. It is known that only 60% of the feeding flow rate of the DNR flows objectively to the clutches.

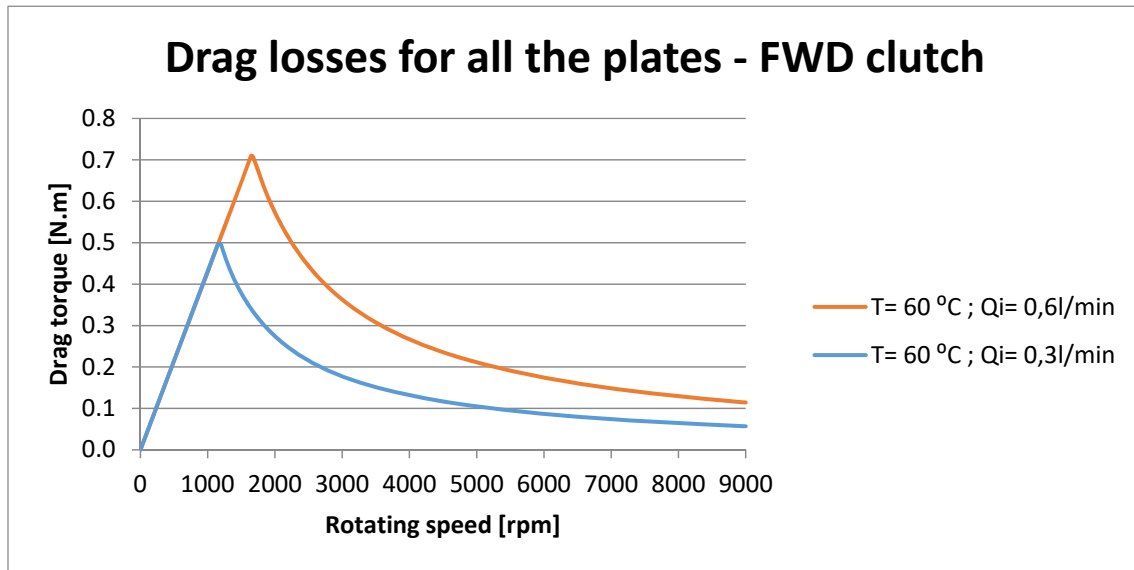


Figure 233 – Drag losses on FWD clutch in D for the validation of the respective requirement

Based on the graph of Figure 233, for both feeding flow rates the requirement is fulfilled. For the maximum feeding flow rate, the maximum drag torque is about 0.7 and 0.5 for the nominal feeding flow rate. In conclusion, this requirement can be validated.

3.10.1.13 Requirement #12

With the purpose of verify this requirement it was necessary to plot the graph of the drag losses of a single plate of the REV clutch for the maximum and nominal feed flow rates.

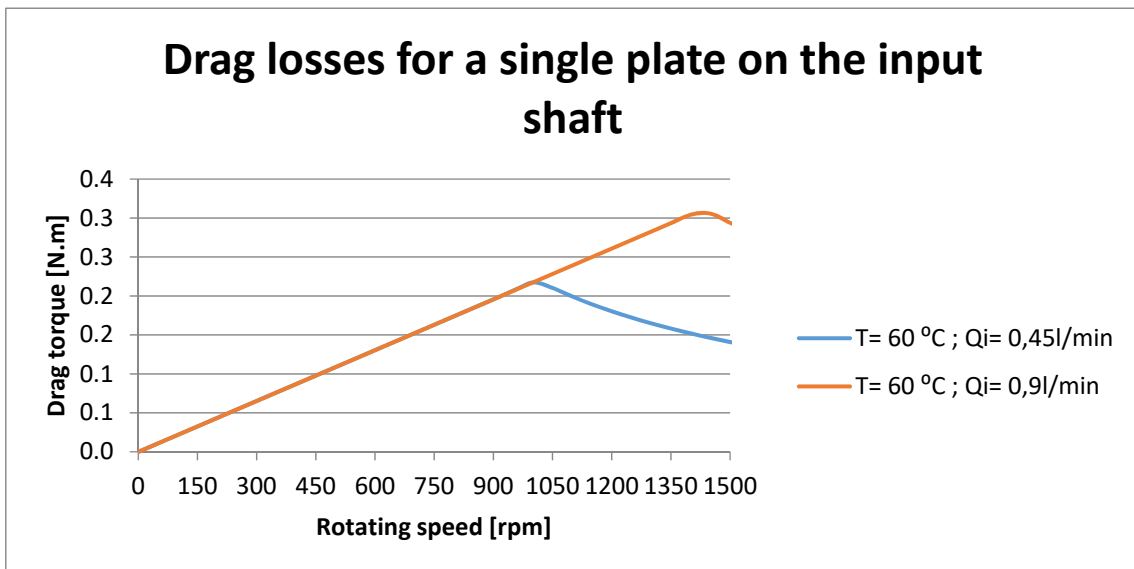


Figure 234 – Graph of the drag losses on REV clutch calculated on the input shaft for a single plate

According to the graph of Figure 234, it is evident that at 1500 rpm only for the nominal feed flow rate is fulfilled the requirement. For a feed flow rate of 0.45 L/min the the drag torque at 1500 rpm is equal to approximately 0.24 Nm. On the other hand, the drag torque for a feed flow rate of 0.9 L/min at 1500 rpm is approximately 0.3 Nm which is greater than 0.26 Nm. In conclusion, this requirement cannot be validated since at the maximum feed flow rate the drag torque is greater than 0.26 Nm.

3.10.1.14 *Requirement #13*

By adding the maximum values of drag torque of the graphs of Figure 233 and Figure 234 it is obtained a result of approximately 1 Nm which is within the range -2.7 Nm to 2.7 Nm. In conclusion, this requirement can be validated.

3.10.1.15 *Requirement #14*

This requirement is easily verified by analysing a cross section of the DNR subsystem. As previously explained the springs of both clutch sets are compressed when the pistons are submitted to hydraulic pressure. When the pressure is released, the springs pull the pistons in order to reach their free state which implies to completely pull the pistons back, disengaging the clutches.

3.10.1.16 *Requirement #15*

This requirement can be verified by the pre-load conditions of the spring pack of each clutch set. Regarding to the FWD SP, the lowest necessary pressure to compress it is 0.502 bar which corresponds to the third scenario. All the calculated parameters can be consulted on chapter 3.5.2.1.3. Regarding to the REV SP, the lowest necessary pressure to compress the it is 0.436 bar which corresponds to the second scenario. All the calculated parameters can be found on chapter 3.5.2.3.3. In conclusion, both values are greater than 0.36 bar which means that this requirement can be validated.

3.10.1.17 *Requirement #16*

This requirement is linked to requirement number 15. Requirement number referred that the SP of both clutch sets should be responsible for pulling the pistons to NO state.

When the pistons are on the NO state, it means that the clutches are disengaged or opened. In conclusion, having this account this requirement is fulfilled.

3.10.1.18 Requirement #17

To verify this requirement is necessary to consult the calculations regarding to the biting point and preload conditions for both clutch sets. The calculations related to the biting point of FWD clutch set are presented on chapter 3.5.5.3. To verify the kiss point pressure, it is necessary to consult the chapter of preload conditions of FWD SP (3.5.2.1.3). The difference between biting and kiss point for FWD is expressed in Table 77.

Table 77 - Pressure difference between biting and kiss point for D

Scenario	Pressure [bar]
Nominal	2.186
Second	2.348
Third	1.679
Fourth	2.466

Based on Table 77, it is visible that the lowest difference corresponds to the third scenario although this value is still greater than 0.4 bar.

The calculations related to the biting point of the REV clutch are presented on chapter 3.5.7.3. To verify the kiss point pressure, it is necessary to consult the chapter of preload conditions of REV clutch set (3.5.2.3.3). The difference between biting and kiss point for REV is expressed in Table 78.

Table 78 – Pressure difference between biting and kiss point for R

Scenario	Pressure [bar]
Nominal	1.391
Second	1.254
Third	1.529

Based on Table 78, it is visible that the lowest difference corresponds to the second scenario although this value is still greater than 0.1 bar.

In conclusion, this requirement is fulfilled for both FWD and REV clutch sets.

3.10.1.19 Requirement #18

The maximum value is 52.971 Nm/bar for the maximum transmissible torque of the second scenario which is less than 55 Nm/bar. The value of Reverse clutch zero-slip current-to-torque sensitivity was calculated for all the scenarios and are expressed on chapters 6.26, 6.27 and 6.28.

3.10.1.20 Requirement #19

This requirement cannot be controlled by the DNR subsystem but for the hydraulic unit. This value is a reference for the design of the parts and to analyse the PGS torque conversion in the worst scenario. So, this requirement cannot be checked.

3.10.1.21 Requirement #20

This requirement is directly controlled by the hydraulic unit. This means that this value is just a reference for the study of the different parameters. In conclusion, this requirement cannot be checked.

3.10.1.22 Requirement #21

Firstly, it was calculated in PTC Creo® the volume of the chamber between the piston and the clutch carrier. Secondly it was calculated the volume of the chamber between the piston and the clutch carrier but for a travelling of the piston of 1.8 mm. Finally, it was subtracted the second volume to the first one. All the values are expressed on Table 79.

Table 79 – FWD piston volumes

Designation	Units	Value
Max volume	mm ³	31777.8
Min volume	mm ³	11839.9
Stroke Volume	mm ³	19937.9
Stroke Volume	cm ³	19.9379

In conclusion, as the resulting volume is less than 21 cm³ the requirement can be validated.

3.10.1.23 Requirement #22

In the first place it was measured the volume of the chamber between the REV piston and the transmission housing in PTC Creo®. After that the REV piston was moved 1.1 mm which is the amount of travelling in nominal working conditions. It was measured the resulting volume. At the end, the value of the second volume was subtracted to the first one. All the values are expressed on Table 80.

Table 80 - REV piston volumes

Designation	Units	Value
Max volume	mm ³	29211
Min volume	mm ³	12025.1
Stroke Volume	mm ³	17185.9
Stroke Volume	cm ³	17.1859

In conclusion, as the resulting volume is less than 24 cm³ the requirement can be validated.

3.10.1.24 Requirement #23

In chapter 3.5.6.1 it is calculated the pressure applied on the FWD clutch plates. The highest pressure is 5.29 MPa. This means that this requirement fails and because of that cannot be validated.

3.10.1.25 Requirement #24

In chapter 3.5.6.2 it is calculated the pressure applied on the REV clutch plates. The highest pressure is 5.71 MPa for the nominal scenario. This means that this requirement fails and because of that cannot be validated.

3.10.1.26 Requirement #25

The needle bearing is assembled inside the planetary gears. According to the kinematics calculations of the PGS (chapter 3.5.9.2) the planetary gears can rotate at 56500 rpm on N or R when the engine is rotating at 9000 rpm. This means that this requirement cannot be validated since the rotating speed of the needle bearings is higher than the allowed speed.

3.10.1.27 Requirement #26

Needle roller thrust bearings are assembled on the clutch carrier, sun gear and PGS carrier. The maximum speed of the clutch carrier is 9000 rpm. Sun gear in R can reach 12900 rpm. Having this in account this requirement cannot be validated.

3.11 Development of the showcase model

3.11.1 Showcase function

The main function of the showcase is to help people to understand the working principle of a DNR subsystem of a CVT transmission. This requirement is fulfilled by allowing people to interact with the showcase. The user can simulate gear shifting by pushing or pulling the shifting lever which corresponds to the shifting lever of an automobile. There are three possible positions: Drive, Neutral and Reverse. The user can also interact with the showcase by rotating the handwheel which corresponds to the engine. So, by rotating the handwheel the user is simulating the rotational motion of the engine. With the showcase is also possible to look into the different components through some cross sections. These cross sections allow the user to better understand what changes and what is the motion chain when the different gears are selected.

3.11.2 Design

In this chapter, it will be presented all the analytical calculations and decisions which have been taken during the development process. During the design of the showcase and in order to take some decisions it was built some Excel® worksheets which makes

easier the interpolation process. At the end, during the 3D design all the calculated values were adjusted and verified via Mechanism application of PTC Creo®.

One important detail is that when the shifting lever is moved from one position to another the clutch of the previous gear must be disengaged. In this aspect, the cams' system plays a vital role. The desired behaviour of the cams' system is presented on the graph of Figure 235 where the light blue line corresponds to the REV cam and the dark blue line corresponds to the FWD cam.

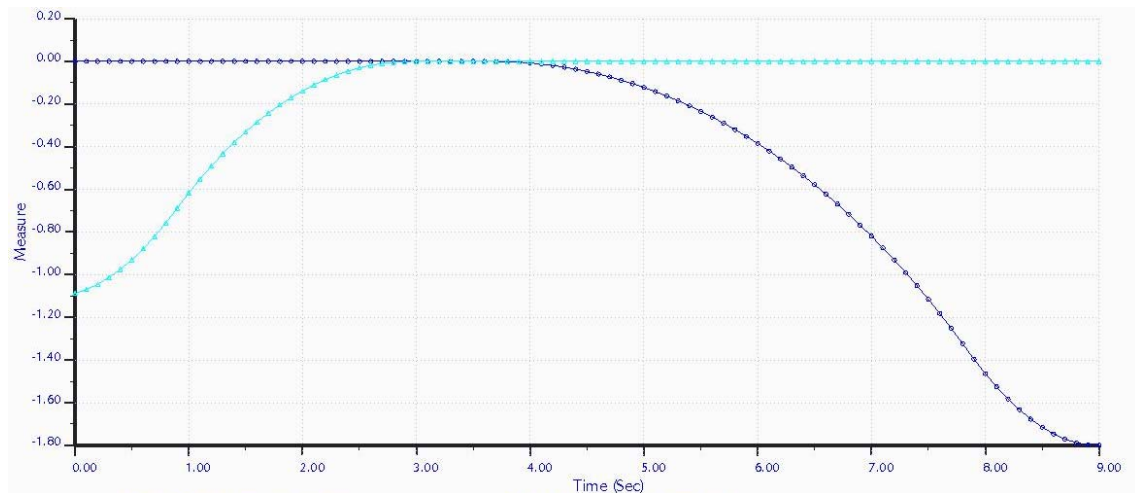


Figure 235 – Graph of travelling of the cams vs time of the showcase

It is known that the desired travelling of the REV cam is 1.1 mm and the travelling of the FWD cam is 1.8 mm. At the beginning of the graph, the cam is being actuated and is at its maximum travelling. On the other hand, the FWD cam is not being activated as can be seen for the line which is the zero travelling. This ensures that when the REV cam is being actuated the FWD cam not and the FWD clutch is disengaged. Between approximately 3 and 4 seconds both cams are at the zero position. Each second on the graph corresponds approximately to 5 degrees of rotation of the shifting lever. During almost one second or 5 degrees both cams are at the zero position. This range corresponds to the Neutral gear position. At this position it is desirable that both clutches are disengaged and to guarantee that it is necessary that none of the cams is activated which means that they need to be at the zero position. From 4 seconds to the end, the FWD cam starts to be rotated. It is evident that at the end the desired travelling is achieved and that the REV cam is at the zero position.

It is important to underline the fact that it was not allowed the use of pneumatic or hydraulic actuation.

3.11.2.1 FWD

This chapter is related to all the components which belong to the FWD clutch set.

3.11.2.1.1 Lever

During the design of the lever it was developed an Excel® worksheet with the following inputs:

- L_1 ;
- L_2 ;
- $\Delta\alpha$.

In Figure 236, it is presented a schematic and the terminology used for the calculations of the lever.

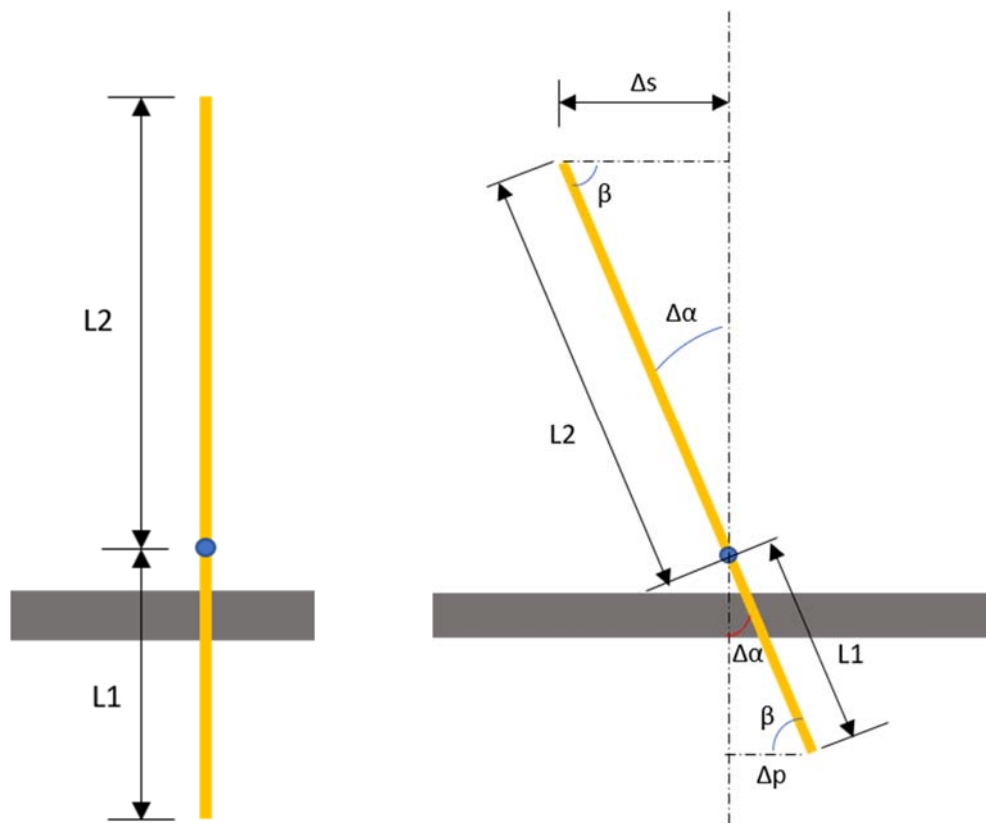


Figure 236 – Terminology for the calculations of FWD lever of the showcase

In Figure 236, the blue circle corresponds to the axis of rotation of the lever and the grey rectangle corresponds to the main plate of the showcase. On yellow, it is presented the lever. Below, all the necessary equations for the calculations related to the lever are presented:

$$\sin \Delta\alpha = \frac{\Delta s}{L_2} \quad \{3.33\}$$

$$\begin{cases} \cos \beta = \frac{\Delta p}{L_1} \\ \cos \beta = \frac{\Delta s}{L_2} \end{cases} \quad \{3.34\}$$

$$\begin{aligned} 180 &= \beta + 90 + \Delta\alpha \Leftrightarrow \\ \Leftrightarrow \beta &= 180 - 90 - \Delta\alpha = 90 - \Delta\alpha \end{aligned} \quad \{3.35\}$$

$$\sin \Delta\alpha = \frac{\Delta p}{L_1} \Leftrightarrow \Delta\alpha = \arcsin\left(\frac{\Delta p}{L_1}\right) \quad \{3.36\}$$

The main purpose of the worksheet developed is to find out an optimum value. This value implies the minimum value of $\Delta\alpha$, L_1 and L_2 . However, it is evident that the lower the value of L_2 , the higher the value of necessary force to activate FWD clutch is.

Auxiliary tables which were used for the design of FWD shifting lever are presented in chapter 6.29. The necessary angle ($\Delta\alpha$) to activate the FWD clutch is 25° . The final dimensions of the lever are presented on the technical drawings.

3.11.2.1.2 Cam

Despite the final design of the cam not being exactly equal to a spiral or an eccentric cam, for the calculations during the design it was considered the calculations' method of an eccentric cam as this was the initial shape of the cam.

It is known that the necessary travelling to make all the components of FWD clutch set touch each other and by this way starting to transmit torque is 1.8 mm. This amount of travelling will be guaranteed by the cam.

After defining the necessary step of the travelling it is necessary to understand how the motion will be transmitted from the lever to the cams. This system affects the amount of rotational motion that needs to be travelled by the cam in order to pull the pressing plate 1.8 mm smoothly. The motion will be transmitted to the cam according to the schematic of Figure 237. The green circle corresponds to the centre of rotation of the cam; the blue circle corresponds to the cam; the yellow circle corresponds to the shoulder screw that transmits the motion of the red component to the cam. It is expected that the red component is connected to the shifting lever. As illustrated in the Figure 237, the red component moves to left and right, and the yellow point when the cam is rotating moves in the perpendicular direction. The white slot corresponds to the slot where the shoulder screw will slide.

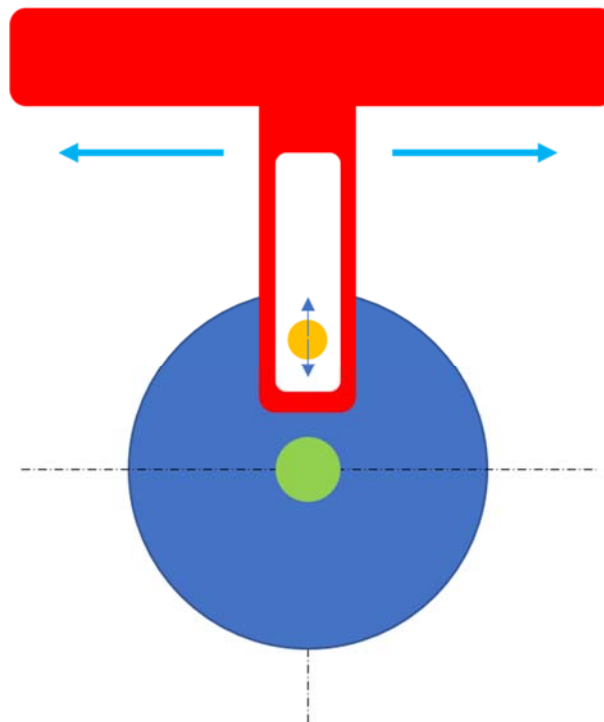


Figure 237 – Illustration of transmission of motion to the cam

In Figure 239, it is presented the FWD cam mechanism where the shoulder screw (grey circle), cam (purple) and transmitting component (blue) are presented. As can be seen, the cam does not touch directly on the pressing plate. It was screwed a stopper in the FWD and REV pressing plates in order to guarantee as less wear as possible. The cam, in fact, touches a stopper that is a standard hardened part. Both, stopper and the cam, are hardened and have the same hardness. All the details about the stopper can be

consulted on the technical datasheet. A perspective view of the stopper can be seen in Figure 238.

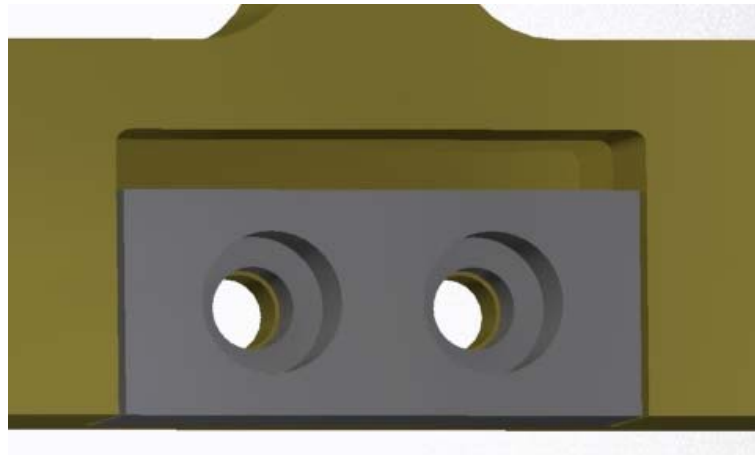


Figure 238 - Perspective view of FWD stopper

3.11.2.1.3 FWD load

Regarding to the loads, it is known that the FWD SP was already calculated. However, it was calculated for all the springs. In fact, in the FWD SP that will be applied on the showcase there are only 4 springs installed. It was reduced the number of installed springs in order to decrease significantly the necessary force to compress the spring pack since it will be compressed by human force and not by hydraulic pressure.

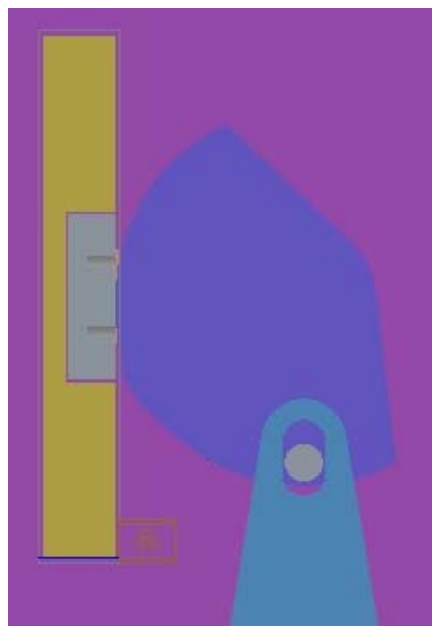


Figure 239 – FWD cam mechanism

The value of SR of each spring can be found on Table 19. The necessary travelling to make all components to touch each other is 1.8 mm, as seen before.

Table 81 – FWD load input data of showcase

Variable	Unit	Value
N	---	4
SR ^{per spring}	N/mm	8.493
x ₁	mm	1.8

With the data of Table 81 the value of load obtained for the FWD SP is 61.150 N.

It was also considered a compression of the wavy washer of 0.2 mm in order to make sure that the torque is really transmitted. Initially, the wavy washer SR is 1000 N/mm but as it will be cut out the SR will decrease significantly. It was considered that the final SR of the wavy washer is approximately two thirds of the initial SR: approximately 670 N/mm. With the added compression of 0.2 mm a parallel association of springs will be verified as seen before.

The final results are presented on Table 82.

Table 82 – Obtained results for FWD load of showcase

Variable	Unit	Value
F _{x1}	N	61.150
Δx ₂	mm	0.2
SR//	N/mm	703.792
F	N	201.939

The final reaction load by the springs is equal to approximately 202 N.

3.11.2.1.4 FWD guiding blocks

As the guiding blocks are standard components it is necessary to find a model or reference which fulfils the requirements in terms of loads. In order to calculate the loads applied on the guiding plates it was sketched the FBD of the FWD pressing plate system. All the loads, reactions forces and dimensions are expressed on Figure 240. The vertical yellow rectangle corresponds to the pressing plate while the horizontal purple rectangle corresponds to the guiding plate where the guiding blocks are screwed.

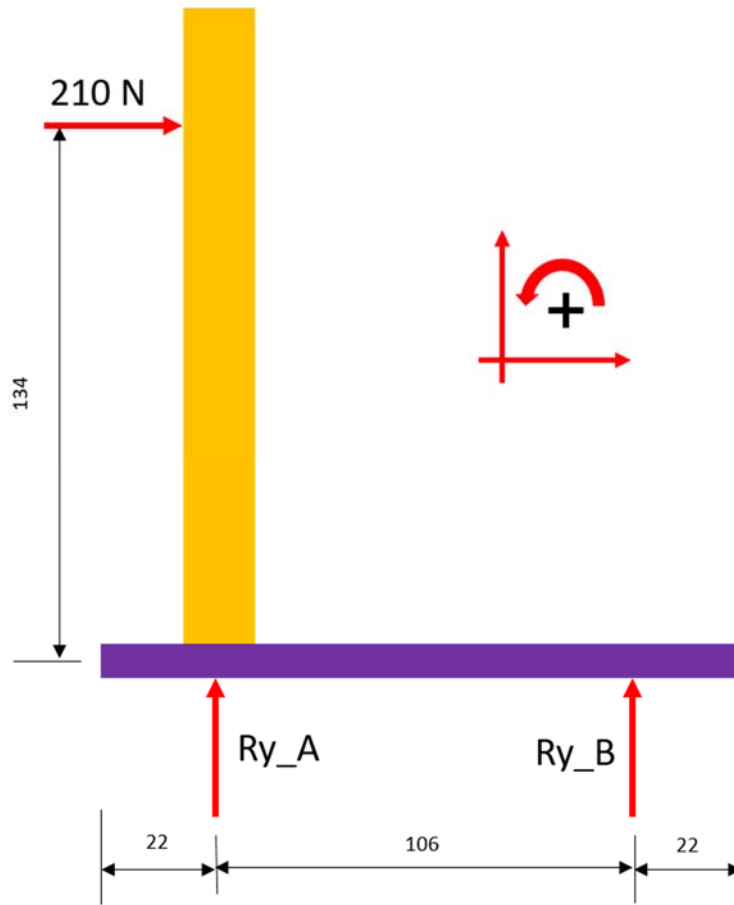


Figure 240 – FBD of FWD pressing plate system

The following conditions must be fulfilled to guarantee a static equilibrium.

$$\begin{cases} \sum M_A = 0 \\ \sum F_y = 0 \end{cases}$$

By substituting the values:

$$\begin{cases} -210 \times 134 + R_y^B \times 106 = 0 \\ R_y^A + R_y^B = 0 \end{cases} \Leftrightarrow \begin{cases} R_y^B = 265,472 \text{ N} \\ R_y^A = -265,472 \text{ N} \end{cases}$$

According to the technical datasheet of the guiding blocks, for a shaft of 16 mm of diameter, it is allowed a dynamic load of 775 N. By dynamic load it is intended to be a

load applied on the blocks during the travelling. Having this in account the SF is equal to 2.9 which is reasonable. Having in account the guiding blocks in use can be validated.

3.11.2.2 REV

3.11.2.2.1 Lever

For the calculations of the lever which are related to REV activation of the clutch it was used the same Excel® worksheet developed for FWD.

The obtained value for the angle $\Delta\alpha$ was -20° . It was expected to obtain a negative value since that to activate the REV clutch it is necessary to pull the lever to the opposite direction.

3.11.2.2.2 Cam

According to the tolerance chain stack up analysis previously presented the necessary travelling to make all the components of REV clutch set touch each other and by this way starting to transmit torque is approximately 1.1 mm. This amount of travelling will be guaranteed by the cam. The motion from the shifting lever will be transmitted to the REV cam by the same way of FWD system.

In Figure 241, it is presented the REV cam mechanism where the shoulder screw (grey circle), cam (green) and transmitting component (blue) are presented. As can be seen, the cam does not touch directly on the pressing plate (pink).

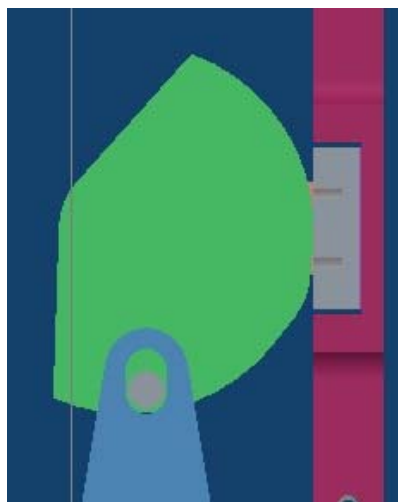


Figure 241 – REV cam mechanism

3.11.2.2.3 REV load

All the calculations regarding to the REV SP are spring disc were already presented. However, it was calculated for all the springs. In fact, in the REV SP that will be applied on the showcase there are only 18 springs installed. It was reduced the number of installed springs in order to decrease significantly the necessary force to compress the spring pack since it will be compressed by human force and not by hydraulic pressure. The value of SR of each spring can be found on

Table 35. The necessary travelling to make all components to touch each other is 1.1 mm, as seen before.

Table 83 – REV load input data for showcase design

Variable	Unit	Value
N	---	18
SR ^{per spring}	N/mm	3.859
x ₁	mm	1.1

With the data of Table 83 the value of load obtained for the REV SP is 76.408 N.

It was also considered a compression of the disc spring of 0.2 mm in order to make sure that the torque is really transmitted. As the spring disc will be cut out the SR will decrease significantly. It was considered that the final SR of the disc spring is approximately 700 N/mm. With the added compression of 0.2 mm a parallel association of springs will be verified as seen before.

The final results are presented on Table 84.

Table 84 – Obtained results for FWD load of showcase

Variable	Unit	Value
F _{x1}	N	76.408
Δx ₂	mm	0.2
SR//	N/mm	769.462
F	N	230.300

The final reaction load by the springs is equal to approximately 230 N.

3.11.2.2.4 REV guiding blocks

In order to calculate the loads applied on the guiding plates it was sketched the FBD of the REV pressing plate system. All the loads, reactions forces and dimensions are expressed on Figure 242. The vertical purple rectangle corresponds to the pressing plate while the horizontal dark blue rectangle corresponds to the guiding plate where the guiding blocks are screwed.

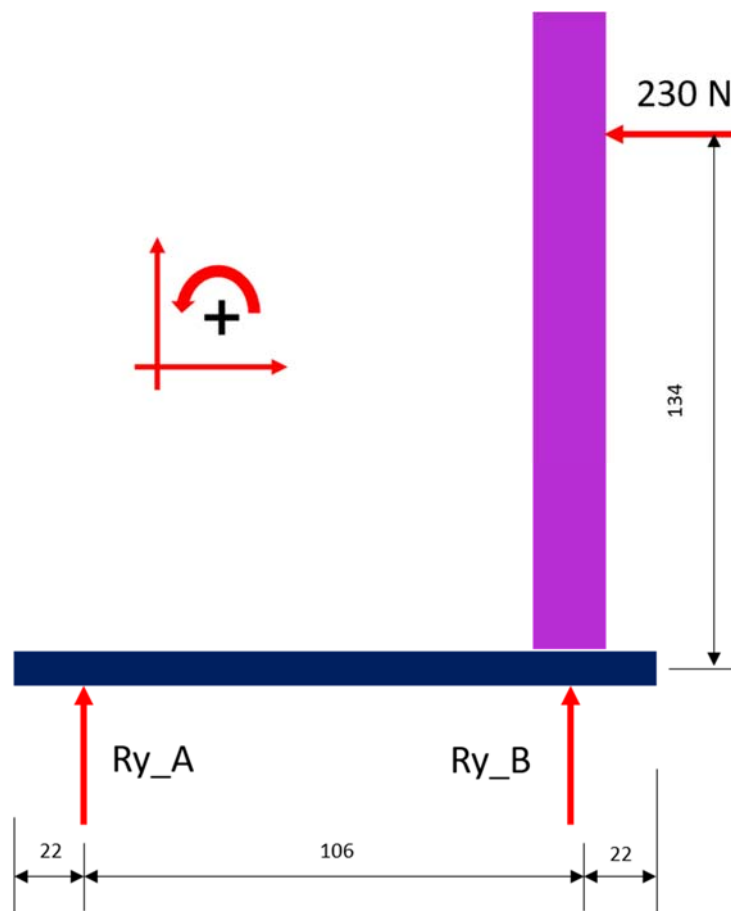


Figure 242 – FBD of REV pressing plate system

The following conditions must be fulfilled to guarantee a static equilibrium.

$$\begin{cases} \sum M_B = 0 \\ \sum F_y = 0 \end{cases}$$

By substituting the values:

$$\begin{cases} 230 \times 134 - R_y^A \times 106 = 0 \\ R_y^A + R_y^B = 0 \end{cases} \Leftrightarrow \begin{cases} R_y^A = 290,755 \text{ N} \\ R_y^B = -290,755 \text{ N} \end{cases}$$

Both guiding systems (FWD and REV) slide on the same shafts which means that the guiding blocks of both systems must be identical. This means that the maximum allowed dynamic load on the blocks is 775N that is quite higher than the applied load. Having this in account the SF is equal to 2.6 which is reasonable. Having in account the guiding blocks in use can be validated

The technical datasheet of the guiding blocks is presented in the Appendix chapter.

3.11.2.3 Ball transfer units

The number of ball transfer units assembled on the showcase is six. This means that the load applies in each of them is equal to 35 N. According to the datasheet of the ball transfer units the maximum allowed load supported by this component is equal to 150 N. There were other options with lower maximum allowed loads however it was decided to risk on a higher SF just in the case it was decided at the last minute to increase the number of springs on the FWD spring pack which would increase significantly the applied load on each ball transfer unit. Another solution would be to increase the amount of applied ball transfer units but there wasn't enough room for that. The final SF is 4.3.

3.11.2.4 Cam

In order to understand the amount of force necessary to actuate the cam for a load of 300 N, it was built a table where it is possible to see the relation between the different angles and the amount of necessary force. The table is illustrated on chapter 6.31. For the calculations it was considered an eccentric cam and all the equations are expressed on 2.2.7.

The input data for the calculations is expressed on Table 85.

Table 85 – Input data for the calculations of the forces on the cam

Variable	Unit	Value
L	mm	30
H	mm	50
r2	mm	20
e	mm	0
P	N	300
μ_1		0.18
μ_2		0.05

3.11.2.5 Lever's load

At the end it is necessary to know the amount of force that was necessary to apply on the shifting lever to actuate the cams. In order to find out that value it was built the table of chapter 6.32. In this table is expressed the value of the force related to the amount of degrees. In the same table it is expressed the values of reaction forces of the shoulder screws and the shifting lever. The maximum amount of force that needs to be applied on the shifting lever is amount 7 N. This seems to be an allowable value. After assembling the complete showcase, it was tested and in fact the amount of necessary force to actuate the cams was achievable by everyone. It is important to underline the fact that in these calculations it was not considered friction between metallic components. It was only considered the coefficient of friction between the cam and the stopper.

3.11.2.6 Fittings

In this chapter, it will be explained all the decisions regarding to fittings of the components. All the fittings were defined according to (Krulikowski, 2010).

3.11.2.6.1 Cam and bearing

The cam will be assembled in the bearing via a press fitting. It was chosen this type of fitting because it is easy to do, cheap and it guarantees that there is no motion between the cam and the inner ring of the bearing. This means that the inner ring of the bearing and the cam rotate solidary.

In Figure 243, it is presented the assembly of the cam and bearing.

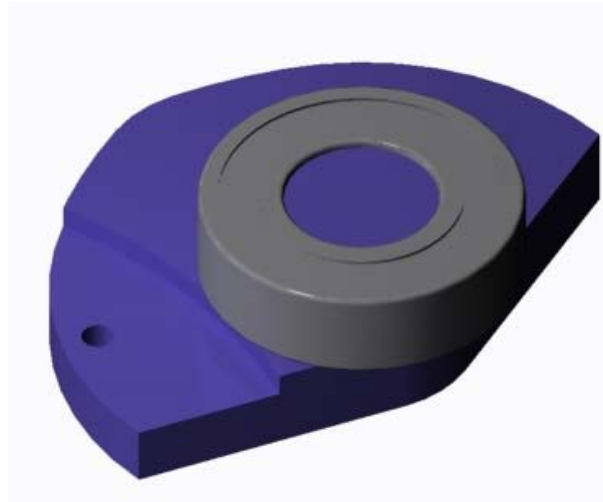


Figure 243 – Fitting between FWD cam and bearing

In Table 86, it is presented all the calculations regarding to the fitting. The hole corresponds to the bearing and the shaft corresponds to the cam. By cam it means the cylindrical surface designed on purpose to fit inside the inner ring of the bearing. The nominal diameter is 20 mm. The resulting fitting is H7/s6 (hole basis). Besides being a standard component, the tolerance of the inner ring can be selected from a range of options.

Table 86 – Fitting between cam and bearing

Hole (bearing)	Shaft (cam)	Fit
19.990	20.015	-0.038
20.000	20.028	-0.015

3.11.2.6.2 Bearing and plate

In Figure 244, it is illustrated the fitting between the bearing and the main plate. It was machined a hole for housing the bearing in the main plate. Both components will be glued. At a first sight, they could be pressed fit however, as previously explained, the cam was assembled in the bearing by press fit. This assembly process is done firstly and

then the assembly cam-bearing is assembled in the main plate. The only possible way to press the bearing into the main plate would be applying the press load on the cam. This would force the inner ring of the bearing and could damage the bearing. In order to avoid damaging the bearing it was decided to glue the assembly in the main plate. By this way it is not necessary to apply too much force on the bearing. In fact, it was previewed a certain amount of gap in order to be possible to apply the glue and not be necessary to apply too much force on the bearing.

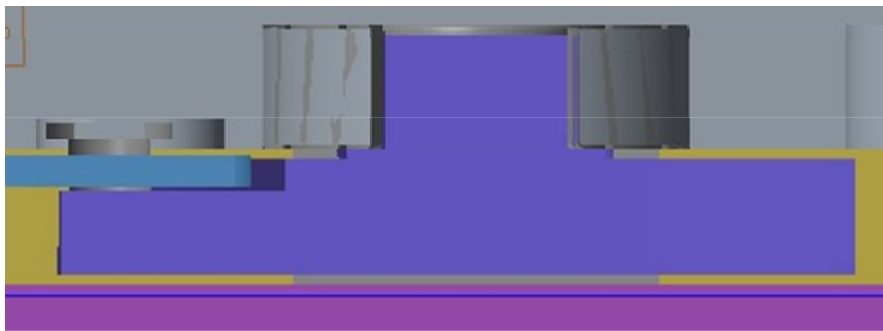


Figure 244 – Illustration of fitting between bearing and main plate

In Table 87, it is presented all the calculations regarding to the fitting. The hole corresponds to the main plate and the shaft corresponds to the bearing. The nominal diameter is 42 mm. The resulting tolerance for the main plate is H7. The tolerance of the bearing is presented on the datasheet since it is a standard component.

Table 87 – Fitting between bearing and main plate

Hole H9	Shaft	Fit
42.062	42.000	0.073
42.000	41.989	0.000

These calculations are important to understand if there is enough gap for the glue. According to the datasheets of the available glues (6.31 and 0) the minimum gap that ensures a correct bonding interface is 0.05 mm. Both glues are suitable for bonding of cylindrical fitting parts.

3.11.2.6.3 Handwheel and turbine shaft

The handwheel and the turbine shaft will be screwed to guarantee that there is no relative motion. However, it is necessary to guarantee that both components fit together. In Figure 245, it is illustrated the fitting between both components. The primary shaft needs to be machined so that both nominal dimensions of the diameter are the same. As it is necessary to assemble the handwheel in the turbine shaft smoothly and easily it was selected a free running fitting. In a hole basis, the fitting is H9/d9. The tolerance of the handwheel hole is based on the datasheet of the component. In this case the hole corresponds to the handwheel and the shaft to the turbine shaft.

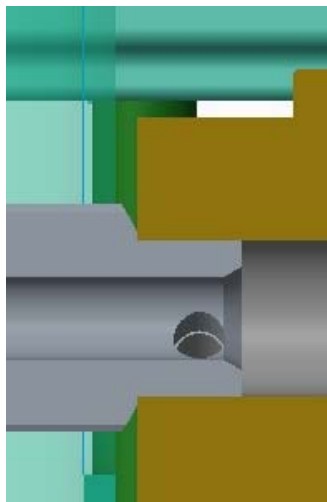


Figure 245 – Fitting between bushing support and handwheel

3.11.2.6.4 Ball transfer units' support and DNR FWD piston

Due to the complexity and lack of room, the only option which seem to be suitable is to glue the adapter to the piston. It will not be glued on the cylindrical surface but on the flat surface where the adapter faces the piston. The fitting in the cylindrical surface is quite important as it determines the correct position of the adapter. The perpendicular is defined by the flat surface. In order to guarantee a easy assembly it was selected a free running fit such as H9/d9 (hole basis). In this case the shaft corresponds to the adapter and the hole to the FWD piston.

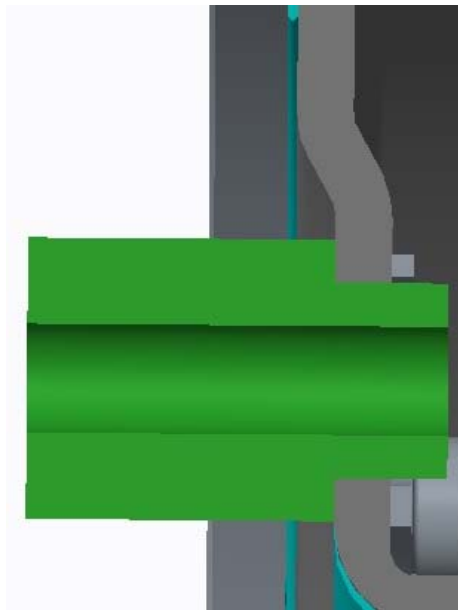


Figure 246 – Fitting between ball transfer unit's adapter and DNR FWD piston

3.11.2.6.5 Gear selector's joining and shoulder screw

The yellow part of Figure 247 corresponds to the joining and the horizontal screw is the shoulder screw.

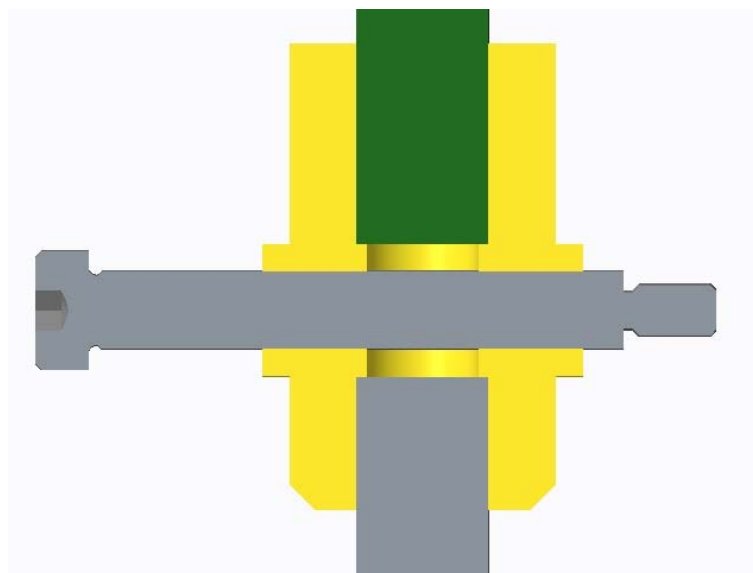


Figure 247 – Fitting between gear selector's joining and shoulder screw

When the shifting lever is pushed or pulled the joining will rotate around the shoulder screw. This means that the shoulder screw is the axis of rotation. It is desirable that the shoulder screw does not rotate. This means that it is necessary a gap between the joining and the shoulder screw so that the joining can rotate freely. In addition, the gap cannot not be exaggerated since the rotating motion needs to be accurate because the correct travelling must be guaranteed on the cams. Having this in account it was selected a clearance fit, H8/g6 (hole basis). In this case the hole corresponds to the joining and the shaft to the shoulder screw.

3.11.2.6.6 Gear selector's support and shoulder screw

As seen before, it is expected that the shoulder screw keeps steady. The thread of the shoulder screw that is placed in the gear selector's support guarantees this condition. However, a correct positioning of the shoulder screw is important since it defines the perpendicular of the joining (yellow). Having this account, it was selected a sliding fit, H7/g6 (hole basis). In this case the hole corresponds to the support and the shaft to the shoulder screw.

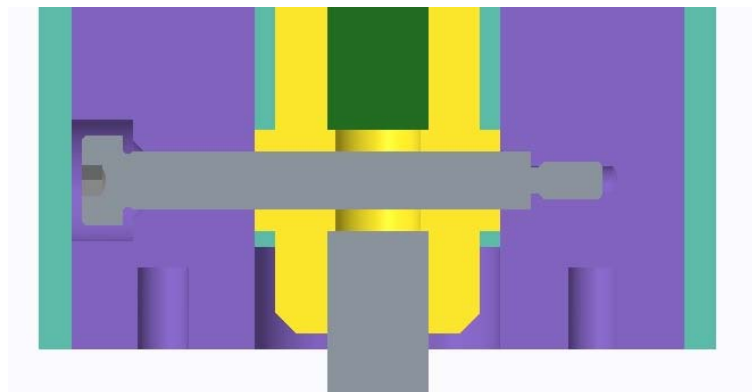


Figure 248 – Fitting between gear selector's support and shoulder screw

3.11.2.6.7 Ring and FWD pressing plate

The ring is where the ball transfer units touch when the cam is pulling the FWD pressing plate. This ring is hardened and made of steel as the ball transfer units are hardened too. As the pressing plate is made of aluminium alloy in order to save some weight it was necessary to insert a part made of steel to prevent wear. The ring is assembled in the FWD pressing plate with a medium drive fit. It is glued in the cylindrical surface and then pressed into the pressing plate in order to guarantee the correct position. The

tolerance is H7/s6 (hole basis). In this case the hole corresponds to the ring and the shaft to the FWD pressing plate.



Figure 249 - Fitting between ring and FWD pressing plate

3.11.2.6.8 Bushing support and bushing

The bushing support is a component that is used on the CVT transmission in study. As the goal of the showcase is to show people the working principle of the DNR subsystem it was decided to use as many as possible components from the CVT transmission. The bushing will be assembled inside the support and then assembled on the turbine shaft. Having this in account, it was decided to machine both components according to the tolerances h7/s6 (hole basis) providing a press fit. In this case the hole corresponds to the support and the shaft to the bushing.

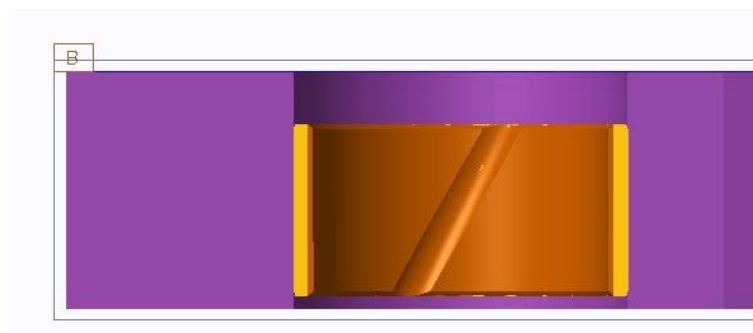


Figure 250 – Fitting between bushing support and bushing

3.12 Preliminary assembly feasibility study

The detailed preliminary study of the feasibility of the assembly process is expressed on chapter 6.35.

3.13 Showcase FEA

For the FEA using Abaqus® it was used the same system of units that was used for the DNR analysis (Table 88).

3.13.1 Abaqus® system of units

Table 88 - Abaqus® system of units' convention

Dimension	Indicator (Abaqus)	Unit
Length	L	mm
Mass	M	kg
Time	T	s
Temperature	Θ	°C
Force	F	N
E	Deformation	---
U	Displacement	mm
S	Stress	MPa

Table 89 – Abaqus® resulting units

Dimensions	Indicator (Abaqus)	Unit
Stress	σ	MPa (N/mm^2)
Density	ρ	kg/mm^3
Deformation	E	---
Displacement	U	mm
Mass moment of inertia	J	$kg \cdot mm^2$
Gravity acceleration	g	mm/s^2
Moment of inertia	I	mm^4

3.13.2 Pin (PNSA3401_2)

The pin is installed in the REV pressing plate (Figure 251). Its main purpose is to transmit the motion for the REV pressing plate to the REV piston. The pin is screwed in the pressing plate.

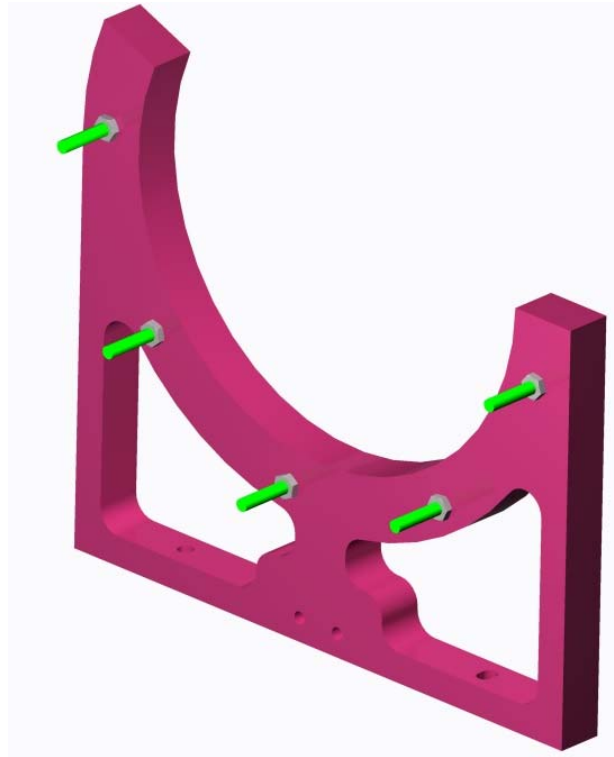


Figure 251 – REV pressing plate

Each pin presses the REV piston in a cavity purposely machined for this.

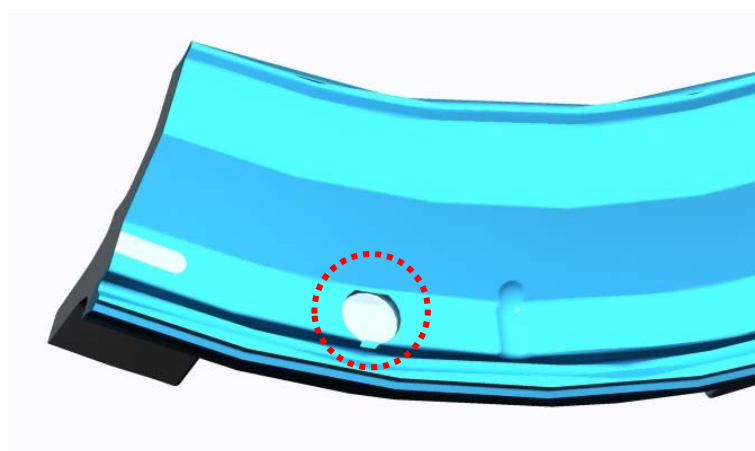


Figure 252 – Cavities of REV piston

When the REV pressing plate is being actuated by the cam, the spring pack compresses as so the disc spring as explained on the calculations. This means that this pin is subjected to the reaction of the springs to pull the pressing plate back.

In the software Abaqus®, it was simulated the working conditions of the pin. On the upper face it was applied a load of 210 N (pink arrows of Figure 253) which corresponds to the total reaction load of REV springs (REV spring pack + disc spring). The load is applied on this face because, as previously explained, this is the face which touches the cavities of the REV piston. In terms of boundary conditions, it was simulated the thread of the pin by preventing the displacement of the pin in three directions (x, y and z) as can be seen in Figure 253 (orange arrows). The input data is presented on Table 90.

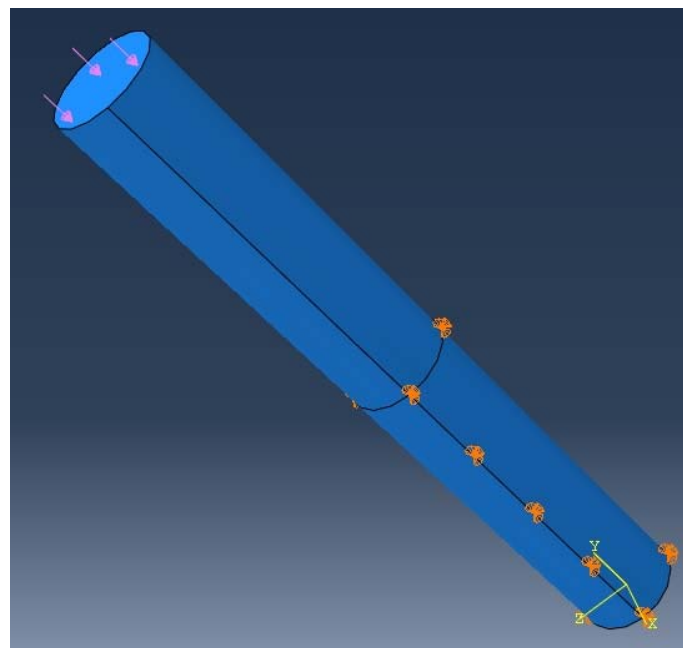


Figure 253 – Schematic of simulation conditions of the component PNSA3401_2

Table 90 – Input data of FEA of the component PNSA3401_2

Property	Value
Type of element	C3D8R (8-node linear brick, reduced integration, hourglass control)
Element size	0.4 mm
Nr. Of elements	9120
Load	230 N
Material's Young's Modulus	70 GPa
Poisson's ratio	0.33
Yield strength	240 MPa

C3D8R is a continuum element, solid element. It is Three-dimensional hexahedral element. Comparing to other tetrahedral elements, C3D8R usually provides a solution of equivalent accuracy at less cost (Simulia, 2010).

Figure 254 shows the region of the max. stress which corresponds to the red region. This region has a shape of a ring all around the pin. The value of the stress on this region is presented on

Table 91. This region corresponds to a region of stresses concentration.

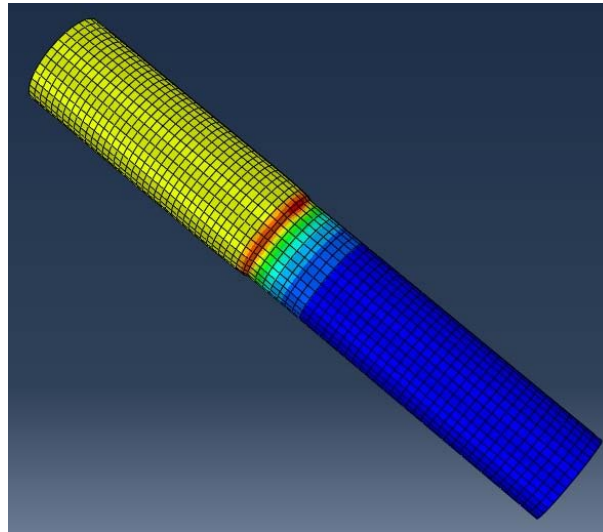


Figure 254 – Location of Von Mises max. stress area

Table 91 - Results of FEA of the component PNSA3401_2

Property	Value
S (Von Mises, Max. Principal)	25.2 MPa
U (Magnitude)	4.093×10^{-3} mm
E (Magnitude)	1.532×10^{-4}
U2	-4.078×10^{-3} mm
SF	9.54

Having in account the yield strength of the material (Table 90) and the generated Von Mises stress (

Table 91) it is conclusive that the value of the yield strength of the material was not exceeded. In fact, the SF is considerably high. In this particular application, max. stress was not the only important property. Displacement of the part along its axis is crucial. If this displacement is more than 0.1 mm, more than one shim needs to be installed.

Fortunately, the value of U_2 is quite far from this limit. In conclusion, the pin can be validated.

3.13.3 Ball transfer units' adapter (PNSA1203_3)

As showed in Figure 255, this adapter transmits the load from the FWD piston the ball transfer units. In the FWD piston is applied the loads from the FWD spring pack and wavy washer. The ball transfer units will transmit the load to the ring of the FWD pressing plate. The ball transfer unit is screwed in the inner threaded hole of the adapter.

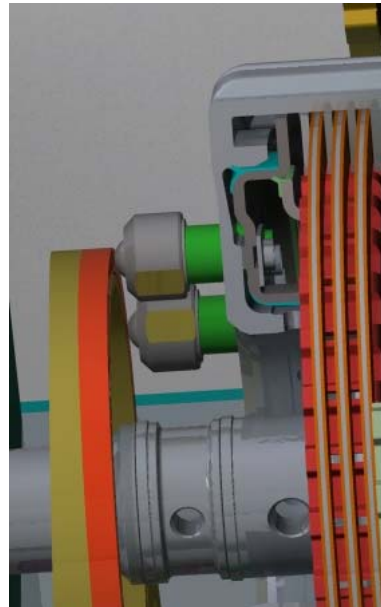
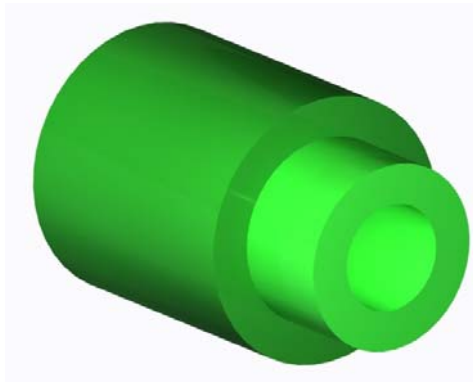


Figure 255 – Ball transfer units' adapter (left) and application (right)

There are six ball transfer units and consequently six adapters. This means that the load from the FWD springs is distributed equally for the adapters. The resulting load is presented on Table 92. Based on Figure 255, the piston presses the adapter on the contact surface. It is in this face that will be applied the load. In the left picture of Figure 256 it is illustrated the face where the load is applied (purple arrows distributed on the face). In the right picture it is illustrated the face where the boundary conditions are applied. This face corresponds to the threaded hole of the adapter.

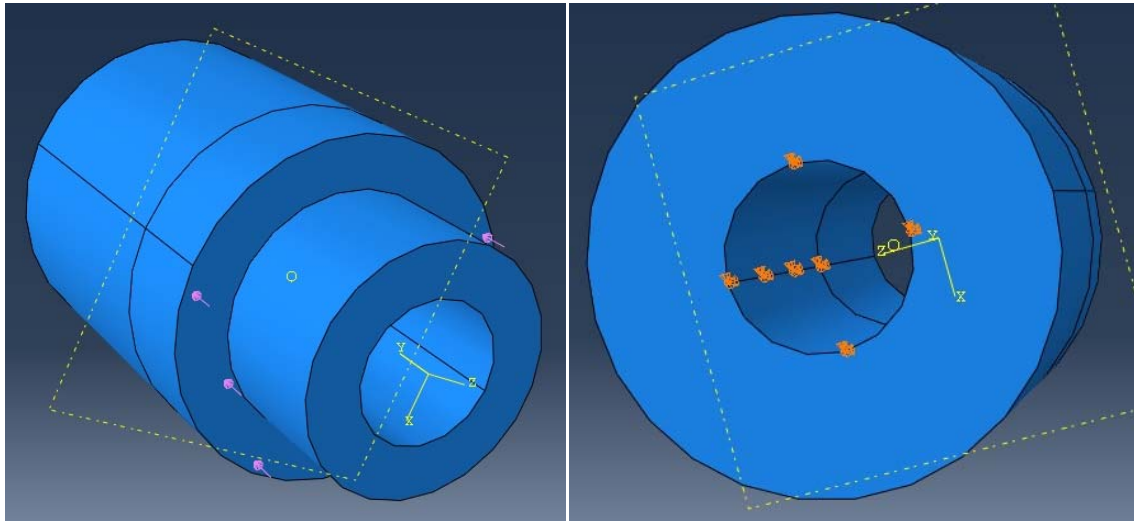


Figure 256 – Boundary conditions of component PNSA1203_3

Figure 257 shows the region of the Von Mises max. stress (left picture) which corresponds to the red region. This region has a shape of a ring all around the pin. The value of the stress on this region is presented on Table 93. This region corresponds to a region of stresses concentration on the interface between the threaded and non-threaded regions of the hole. On the right picture it is illustrated the deformed shape of the adapter.

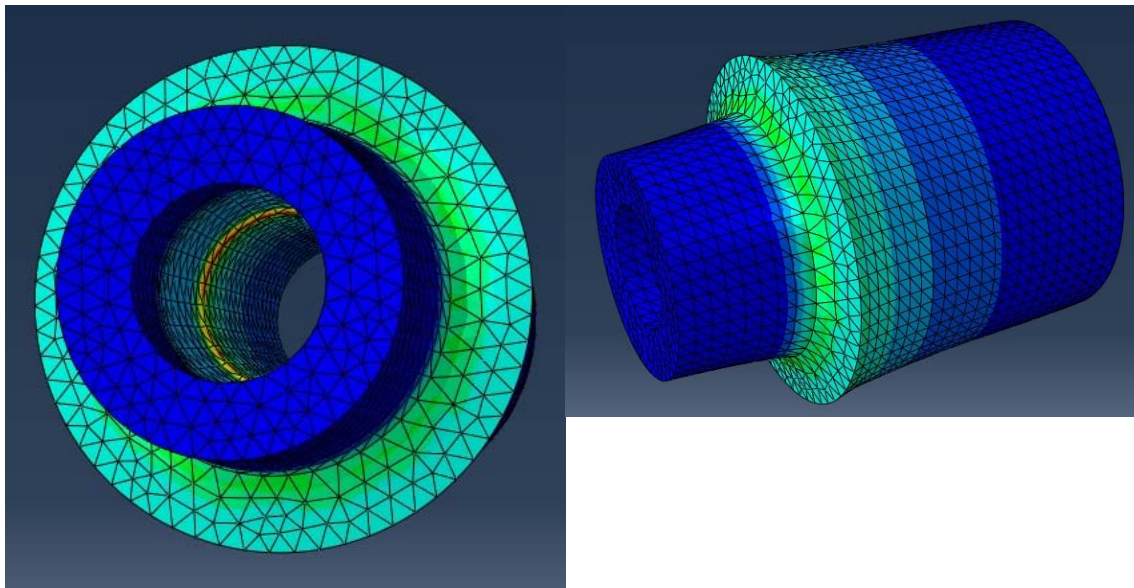


Figure 257 – Deformed shape of the adapter (scale: 2.3×10^4)

Table 92 - Input data of FEA of the component PNSA1203_3

Property	Value
Type of element	C3D10: A 10-node quadratic tetrahedron
Element size	0.5 mm
Nr. Of elements	45156
Load	35 N
Material's Young's Modulus	70 GPa
Poisson's ratio	0.33
Yield strength	240 MPa

C3D10M is a continuum element, solid element. It is a modified tetrahedral element. Comparing to the other elements of the same type, C3D10M mesh is slightly stiffer (Simulia, 2010).

Table 93 – Results of FEA of the component PNSA1203_3

Property	Value
S (Von Mises, Max. Principal)	2.675 MPa
U (Magnitude)	6.678×10^{-5} mm
E (Magnitude)	2.760×10^{-5}
U2	6.349×10^{-5} mm
SF	89.7

Having in account the yield strength of the material (Table 92) and the generated Von Mises stress (Table 93) it is conclusive that the value of the yield strength of the material was not exceeded. In fact, the SF is considerably high. In this particular application, max. stress was not the only important property. Displacement of the part along its axis is crucial. If this displacement is more than 0.1 mm, more than one shim needs to be installed. Fortunately, the value of U2 is quite far from this limit. In conclusion, the adapter can be validated.

3.13.4 FWD pressing plate (PNSA3302_5)

According to Figure 258, on the left side is located the stopper where the cam applies its load. On the upper right side is located the ring which is the interface between the ball transfer units and the FWD pressing plate. In the lower and horizontal face of the FWD pressing plate it is located four holes: two of them are for the screws that fix the pressing plate do the guiding plate.



Figure 258 – FWD pressing plate assy

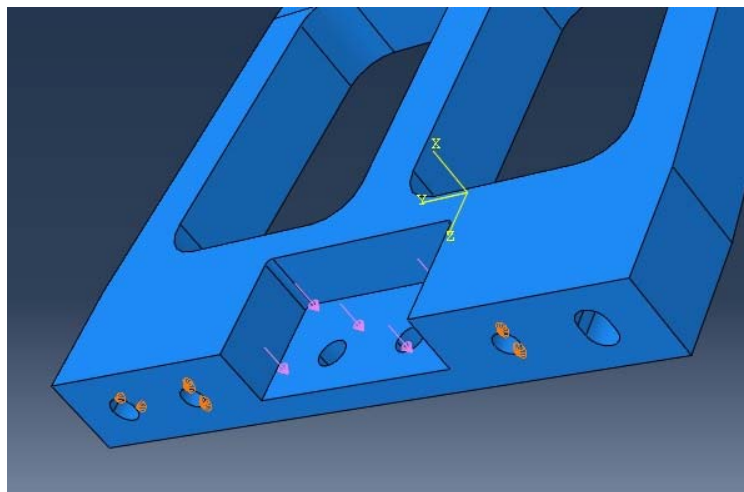
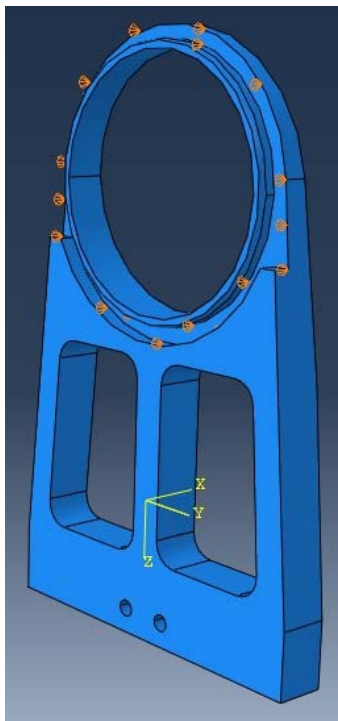


Figure 259 – Boundary conditions of component PNSA3302_5

In this case it was considered the scenario where the ring is already touching the ball transfer units and that these do not allow any kind of displacement, but the cam is still pressing the pressing plate. This means that on the interface with the ring will be not allowed any kind of displacement along the x axis. On the holes for fixation, will not be allowed any displacement along the z and y axis. The load will be applied on the interface with the stopper or the shims. All these boundary conditions are illustrated in Figure 259.

As expected and confirmed on Figure 260, the region of stress concentration is located on the holes for fixing the pressing plate to the guiding plate. On the left picture of Figure 260 it is visible that most of the component is blue and the holes are coloured of green. On a deeper look at inside the holes (left picture of Figure 260) it is evident that the concentration of stress is in the edge of the hole. The value of the maximum stress is presented on

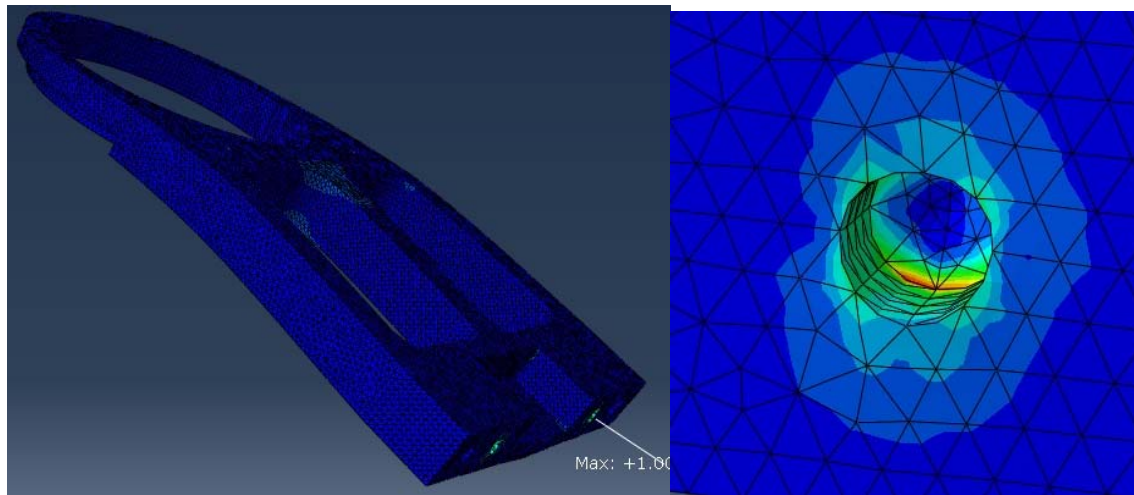


Figure 260 – Max. stress regions of FEA of component PNSA3302_5 (scale: 2.2×10^2)

Table 94 - Input data of FEA of the component PNSA3302_5

Property	Value
Type of element	C3D10: A 10-node quadratic tetrahedron
Element size	1.5
Nr. Of elements	198135
Load	210 N
Material's Young's Modulus	70 GPa
Poisson's ratio	0.33
Yield strength	240 MPa

C3D10 is solid, quadratic tetrahedral element. It is a general purpose tetrahedral element.

Table 95 - Input data of FEA of the component PNSA3302_5

Property	Value
S (Von Mises, Max. Principal)	101 MPa
U (Magnitude)	8.310E-02 mm
E (Magnitude)	2.199E-03
SF	2.3

Having in account the yield strength of the material (Table 94) and the generated Von Mises stress (C3D10 is solid, quadratic tetrahedral element. It is a general purpose tetrahedral element.

Table 95) it is conclusive that the value of the yield strength of the material was not exceeded. In fact, the SF is higher than 1.5 that was the requirement. In conclusion, the FWD pressing plate can be validated.

3.13.5 REV pressing plate (PNSA3401_1)

As illustrated in Figure 261, on one side it is placed the stopper where the load of the cam is applied.



Figure 261 - REV pressing plate assy

On the other side pins are screwed to the plate. These pins are the interface between the pressing plate and the REV piston. This means that the load of the REV spring is applied on the pins.

In order to simulate the behaviour of the REV pressing plate in the working conditions it was applied a load on the face where the stopper is screwed (pink arrows). In terms of boundary conditions, it was defined that the threaded holes on the bottom of the pressing plate (which will be used to screw the pressing plate to the guiding plate) do not allow any motion along the x and z axis while in the threaded holes of the pins the displacement along the y axis is constraint.

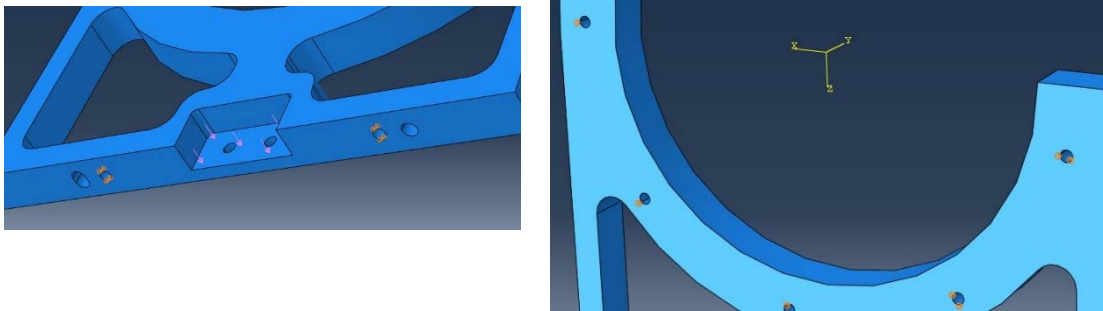


Figure 262 – Boundary conditions of the FEA of component PNSA3401_1

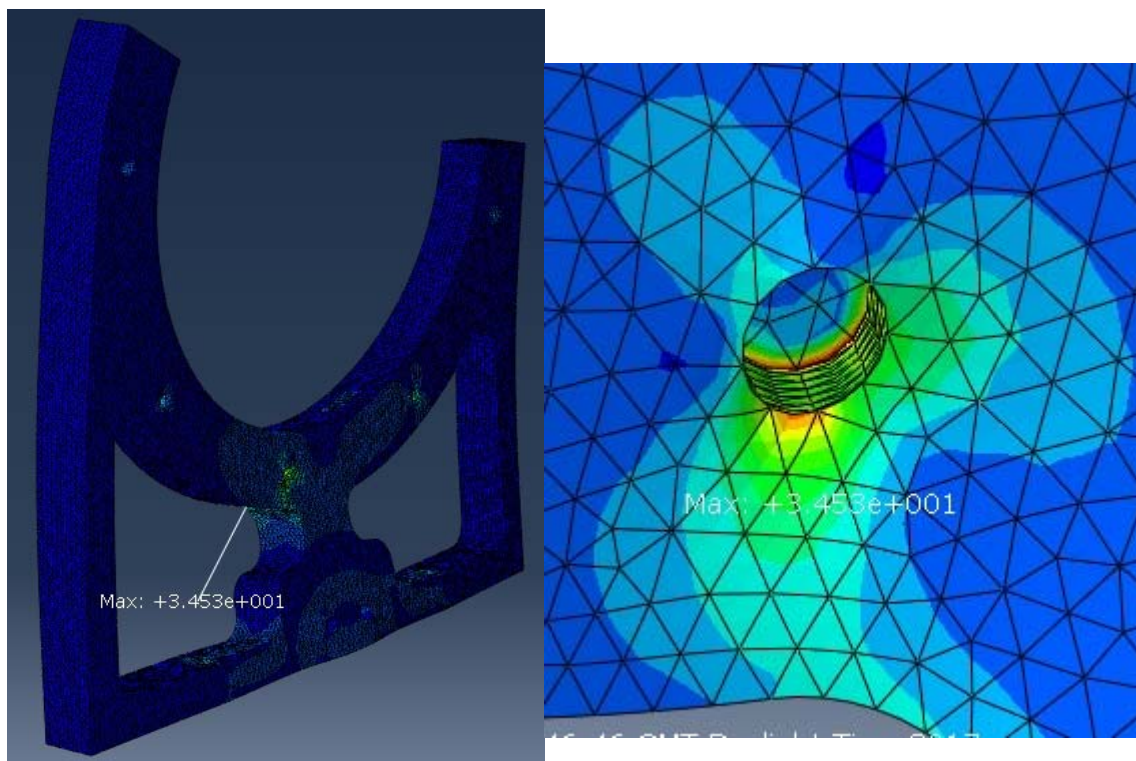


Figure 263 - Max. stress regions of the FEA of component PNSA3401_1

According to Figure 263, the max. stress region is located in the threaded holes of the pins but only on one of them (green hole of left picture). In the right picture is visible the red zones where it is the max. stress.

Table 96 - Input data of FEA of the component PNSA3401_1

Property	Value
Type of element	C3D10: A 10-node quadratic tetrahedron
Element size	1.5
Nr. Of elements	308793
Load	230 N
Material's Young's Modulus	70 GPa
Poisson's ratio	0.33
Yield strength	240 MPa

Table 97 – Results of FEA of the component PNSA3401_1

Property	Value
S (Von Mises, Max. Principal)	34.5 MPa

U (Magnitude)	3.514E-02 mm
E (Magnitude)	5.963E-04
SF	6.9

Having in account the yield strength of the material (Table 96) and the generated Von Mises stress (Table 97) it is conclusive that the value of the yield strength of the material was not exceeded. In fact, the SF is higher than 1.5 that was the requirement. In conclusion, the REV pressing plate can be validated.

3.14 Drawings

All the technical drawings of the showcase are in the appendix chapter. For all the machined parts it was used the respective standard (Standardization, 1989). All the threads were based on (BRITISH STANDARD, 1981). It is important to underline the fact the due to the lack of time for quotation it was asked to prepare the drawings only with the main dimensions, dimensions and location of holes and nonstandard tolerances of machined parts since the STEP file would be delivered to the supplier. A complete BOM of the showcase is also in the appendix chapter.

3.15 Poster

In appendix 0 it can be found a poster whose goal is to help people who interact with the showcase model understanding the main functions and principles of a DNR subsystem and how the showcase model works.

CONCLUSIONS

4.1 CONCLUSIONS

4.2 FUTURE WORKS PROPOSAL

4 CONCLUSIONS AND FUTURE WORKS PROPOSAL

4.1 CONCLUSIONS

Regarding to the calculations of the DNR subsystem the following conclusions were achieved:

- The spring rate reference values provided by the supplier of the FWD SP can be validated because the errors for the three scenarios are less than 1%, which is acceptable;
- The higher the rotating speed, the lower the necessary hydraulic pressure applied on FWD SP in order to compress it is (Figure 129) due to, mainly, the resulting centrifugal force;
- In terms of pre-load conditions of the FWD SP there is not any significant difference between nominal and second scenarios when relating the hydraulic pressure and rotating speed of the CC (Figure 129);
- Fourth scenario of FWD clutch set corresponds to the maximum necessary hydraulic pressure to start compressing the FWD SP with a value of 0.862 bar (Table 23) while 3rd scenario presents the lower pressure of 0.502 bar (Table 22). The main reason for this fact is that for the calculations according to the 4th and 3rd scenarios it was considered the worst and best conditions to transmit the torque, respectively. That is the reason why they have opposite results;
- In terms of fatigue of FWD SP, it was achieved acceptable values when used the Sines criteria obtaining a value of SF equal to 1.315 (that is considerably greater than one) (Table 24). On the other hand, for calculations based on Goodman and Gerber criteria the obtained values are lower than 1 although they are close to 1 (Table 24). The first justification for this fact is that these two criteria are more conservative than the first one. Besides this, the FWD SP can be validated (in terms of fatigue) because in these calculations heat treatments were not considered and in fact these components are submitted to heat treatments. The stresses originated by the manufacturing processes can be harmful. These components are obtained by plastic deformation (Ashby, 2005) which induces mechanical hardening in the part. A solution is a stress relief annealing.
- Regarding to the SR of the wavy washer based on the different scenarios it is conclusive that there is a significant difference between the obtained values for the different scenarios (Table 26, Table 27 and
- Table 28). The reason for this phenomenon is that there is a wide range of tolerances in the loads of the spring and in the different scenarios the tolerances were taken in account;
- Wavy washer and all the other springs are subjected to dynamic loads and in this field, it was calculated that the wavy washer is expected to be prepared for a 10^7 life cycle since the value stress calculated for the spring corresponds to 32% of the tensile strength of the material (Table 29). This is an acceptable life cycle;
- In terms of SR of the REV SP there is a significant disparity between the results for the different scenarios (Table 33, Table 34 and
- Table 35). This parameter affects directly the pre-load conditions as also the fatigue behaviour;
- In terms of pre-load pressure, the discrepancy between the maximum and minimum obtained values are in the same magnitude as FWD SP where the maximum value (0.553 bar) (Table 39) corresponds to the 3rd scenario and the minimum value (0.436 bar) (Table 38) corresponds to the 2nd scenario;

- Similar to FWD SP it was only obtained a SF greater than 1 for the Sines criteria (Table 40). For the two other criteria (Gerber and Goodman) it was obtained results lower than 1 although close to (Table 40). Once again, heat treatments were not taken in account in the calculations. It is conclusive that in terms of fatigue REV SP can be validated;
- The loads' tolerance range of the disc springs is wide, and it translates itself in a considerable variance between scenarios for the value of SR (Table 43). The third scenario presents the greatest, 1603.840 N/mm while the second scenario presents the lowest value, 1260.160 N/mm;
- For the disc spring it is expected an infinite life (Figure 131) due to the low stresses applied on the part;
- Regarding to the transmission oil, relating the oil density and temperature, the higher the temperature, the lower the density is. The relation is linear. By relating the oil kinematic viscosity and oil temperature it is visible that with the increase of the temperature the viscosity decreases significantly. From 60 °C the viscosity becomes stable and close to zero. From -40 °C until 60 °C there is a large variation in the resulting parabolic curve. The behaviour of the curve which related the oil dynamic viscosity and oil temperature is identical to the previous one;
- In the FWD clutch set, fourth scenario corresponds to the lowest applied force on clutch plates, 8341.529 N (Table 52), while the nominal scenario presents the highest axial force, 9611.585 N (Table 47). Besides this, the third scenario corresponds to the highest transmissible torque while the lowest transmissible torque belongs to the fourth scenario. The main difference between transmissible torque of 3rd and 4th scenarios is the SR which translates in considerable difference in the loads (Figure 146);
- According to the graphs of Figure 148 and Figure 147 it is conclusive that the influence of the slipping speed in the amount of FWD transmissible torque is very low;
- Centrifugal force is a key factor in the amount of transmissible torque of FWD clutch and rotating speed of the CC has a primordial effect in the amount of generated centrifugal force: the higher the rotating speed the higher the resulting centrifugal force is (Figure 152). The resulting curve is parabolic. The resulting centrifugal force is the sum of centrifugal force generated on the FWD piston and the one generated on the balance dam. The centrifugal force generated in the BD increases negatively with the rising of the rotating speed (Figure 150). The centrifugal force generated in the compensation chamber (balance dam) is not enough to null the centrifugal force created in the FWD piston so the resulting centrifugal force will favour the applied force on the clutch plates. The higher the rotating speed, the higher the centrifugal force in the FWD piston is (Figure 151);
- There are some parameters that affect the centrifugal force. The first one is the fluid density. The relation between centrifugal force and oil density is linear and increases with the rising of the density (Figure 153). The other parameter is the fluid entry radius and once again the relation between centrifugal force and the radius is linear, but the slope is negative for the FWD piston (Figure 154) and positive for the BD (Figure 155);
- In terms of FWD biting-point conditions, fourth scenario represents the highest necessary pressure (3.328 bar) (Table 58) to guarantee a transmissible torque of 8Nm which is equivalent to an axial force of 186.938 N. On the other hand, third scenario represents the minimum necessary pressure (2.181 bar) which is translated in an axial force of 187.426 N (Table 57);
- Feeding oil flow rate has really low effect on the amount of necessary flow rate to guarantee full oil film (Figure 167 and Figure 194). On the other hand, rotating speed has a significant effect. The higher the rotating speed, the higher the needing feeding flow rate is (Figure 167 and Figure 194). The pressure variation between inner and outer diameter has no effect on the needing feed flow rate (Figure 172 and Figure 201). The higher the oil temperature, the higher the needing feed flow rate is (Figure 175 and Figure 205);

- The increase of the rotating speed, increases the pressure in inner and outer diameter (Figure 169 and Figure 195);
- Drag losses are higher when the feeding flow rate is higher too, for high rotating speeds (Figure 171 and Figure 197). At low speeds the curves are coincident until the inversion point where the feeding flow rate is lower than the necessary feeding flow rate to ensure full oil film. Oil temperature has a negative effect on the drag losses since the increase of the fluid temperature increases the amount of drag losses (Figure 177);
- Equivalent radius is affected by the rotating speed of the CC from the inversion point. Until that point the curve is horizontal (Figure 173 and Figure 202). After the inversion point feeding flow rate has some effect on the equivalent radius. For the same rotating speed and different feeding flow rates, the equivalent decreases with the decrease of the feeding flow rate; On the other hand, pressure variation between inner and outer radius has no effect in the equivalent radius (Figure 174 and Figure 203);
- Dissipated power through drag losses is significantly affected by the fluid temperature. The higher the fluid temperature, the higher the amount of dissipated power is (Figure 176 and Figure 207);
- The highest pressure on the FWD clutch plates is found for the nominal scenario (52.881 bar) (Table 60) while the lowest pressure corresponds to the third scenario (47.015 bar). In the case of REV clutch plates, the highest pressure corresponds to the second scenario (57.130 bar) (Table 61) and the lowest pressure corresponds to the third scenario (55.111 bar);
- The highest axial force applied on the REV clutch plates is obtained for the second scenario (14469.325 N) (Table 65);
- In terms of transmissible torque of REV clutch set, the highest amount of transmissible torque is obtained for the second scenario while the lowest corresponds to the third scenario (Figure 186);
- In the first stage of the springs' association of REV clutch set there is not a visible distinction in the spring stiffness between the three scenarios, however for the second stage is visible a difference between the different scenarios. 3rd scenario corresponds to the highest spring's stiffness while the 2nd scenario corresponds to the lowest one (Figure 185);
- In terms of REV biting-point conditions, 3rd scenario corresponds to highest necessary pressure to ensure a transmissible torque of 8 Nm (2.081 bar) (Table 70) while the 2nd scenario represents the lowest pressure (1.885 bar) (Table 68);
- In D, all the gears rotate together (Figure 212);
- In N, planetary gears rotate around their own axes faster than the other gears. Ring gears and CC rotate together (Figure 214, Figure 213 and Figure 215);
- In R, planetary gears rotate faster than the other gears. Ring gear rotates in the same direction as the planetary gears while the sun gear rotates to the opposite direction (Figure 216);
- The drag losses on REV are considerably higher than on FWD (Figure 223);
- Clutch plates' pressure is higher on REV plates (about 3 bar) (Figure 227);
- In terms of validation, the following requirements were not validated:
 - o 6;
 - o 13;
 - o 24;
 - o 25;
 - o 26;
 - o 27.

Regarding to the showcase, it was obtained the following conclusions:

- All FEA prove that the parts can be validated not only in terms of stress but also in terms of displacement;

- After assembled, the showcase was tested, and it was verified that it was possible to clearly understand and see all the details of the working principle of the DNR. The necessary force to actuate the cams was easily achievable.

There are some reasons why some requirements were not validated. The first one is the fact that the datasheets necessary for the verification of these requirements were probably not up-to-date and secondly, the fact that some parts were being modified which means that some reinforcements and even the main dimensions were modified. Furthermore, the complete subsystem is tested in laboratory and in the car which means that before being launched the complete transmission is tested under real working conditions. This means that the quality, safety and quality of all the components is guaranteed.

As previously stated, the drag losses graphs resulting from the tests of the DNR subsystem were not similar to the theoretical graphs. This should have happened since the theoretical graphs were compared to graphs from scientific articles where analytical and testing graphs were presented. The reason for this, is the fact that the output connections of the sensors were probably changed which means that the provided data is not correct. In addition, an error during the conversion of the raw file may have happened. It is important to underline the fact that for the calculations it was considered the graphs obtained analytically whose calculation model is totally validated.

During the development of this thesis it was clear that there was some lack of literature regarding to automotive transmissions and related components (for example springs) with an engineering approach and with a deeper point of view.

At the end, the internship at Punch Powertrain proved to be a great experience not only by giving me a new perspective of the development process behind the automotive industry but also because it was possible to consolidate the knowledge obtained in the university. During the internship it was possible to learn theoretical subjects more related with calculations and also more practical details related to some manufacturing processes. 3D design with PTC Creo® and FEA with the software Abaqus® were two major skills acquired during the internship.

4.2 FUTURE WORKS PROPOSAL

Given the lack of validated information I suggest the study of the following fields:

- Geometry parameters' influence in the Belleville washers' behaviour;
- Influence of geometrical dimensions on the mechanical properties of a wavy washer.

REFERENCES AND OTHER SOURCES OF INFORMATION

5.1 SCIENTIFIC JOURNALS' ARTICLES and LITERATURE

5.2 CATALOGUES

5.3 STANDARDS

5.4 WEBSITES

5 REFERENCES AND OTHER SOURCES OF INFORMATION

5.1 SCIENTIFIC JOURNALS' ARTICLES and LITERATURE

- (ACEA), E. A. M. A. (2017). *The Automobile Industry Pocket Guide*.
- Akilesh Yamsani. (2014). Gradeability for Automobiles. *IOSR Journal of Mechanical and Civil Engineering (IOSR-JMCE)*, 11(2), 35–41.
- Ashby, M. F. (2005). *Materials Selection in Mechanical Design*. Design. ISBN:0750661682.
- Avallone, E. A., III, T. B., & Sadegh, A. M. (2007). *Marks' Standard Handbook for Mechanical Engineers* (11th ed.). ISBN:9780071428675.
- Brown, A. A. D. (1981). *Mechanical Springs* (1st ed.). ISBN:0-19-859181-0.
- Brown, T. H. (2005). *Marks' calculations for machine design*. ISBN:ISBN-10: 0-07-143689-8.
- Carvil, J. (1993). *Mechanical Engineer's Data Handbook*. Butterworth Heinemann. ISBN:0750619600.
- Crolla, D. A. (2009). *Automotive engineering: powertrain, chassis system and vehicle body*. Elsevier publications and Book AID International (1st ed.). ISBN:1856175774.
- Crolla, D., Foster, D. E., Kobayashi, T., & Vaughan, N. (2014). *Encyclopedia of Automotive Engineering*. *Encyclopedia of Automotive Engineering*. ISBN:9781118354179.
- Gustafsson, F. (2014). *Wet clutch load modeling for powershift transmission bench tests*.
- H. U. Jibin, Peng Zengxiong, Y. S. (2009). Drag torque prediction model for wet clutches. *Chinese Journal of Mechanical E*, 22.
- Halderman, J. (2012). *Automotive Technology : Principles, Diagnosis and Service* (4th ed.). ISBN:9780132542616.
- Henriksen, E. K. (1973). *Jig and Fixture Design Manual*. ISBN:9780831110987.
- Herdman, R. et al. C. (OTA). (1995). *Advanced Automotive Technology: Visions of a Super-Efficient Family Car*.
- Heyan, L., Qi, J., & Biao, M. (2013). Modeling and Parametric Study on Drag Torque of Wet Clutch - Proceedings of the FISITA 2012 World Automotive Congress. *Automotive Concept Modelling: Optimization of the Vehicle NVH Performance*.
- Hillier, V. A. W. (2012). *Hillier's Fundamental of Motor Vehicle Technology* (6th ed.). ISBN:978 1 4085 1518 1.
- Juvinall, R. C., & Marshek, K. M. (2012). *Fundamentals of Machine Component Design*

- (5th ed.). ISBN:9781118012895.
- Krulikowski, A. (2010). *Alex Krulikowski's ISO Geometrical Tolerancing Reference Guide*. ISBN:0924520175.
- Liu, W. (2017). *Hybrid Electric Vehicle System Modeling and Control* (2nd ed.). ISBN:978-1-119-27932-7.
- Maki, R. (2005). *Wet clutch tribology: friction characteristics in limited slip differentials*.
- Mashadi, B., & Crolla, D. (2012). *Vehicle Powertrain Systems*. ISBN:9780470666029.
- Müller, H. W. (1982). *Epicyclic Drive Trains: Analysis, Synthesis, and Applications*.
- Naunheimer, H., Bertsche, B., Ryborz, J., & Novak, W. (2010). *Automotive Transmissions: Fundamentals, Selection, Design and Application* (2nd ed.). ISBN:9783642162138.
- Shigley, J. E., Mischke, C. R., & Budynas, R. G. (2002). *Shigley's Mechanical Engineering Design* (9th ed.). ISBN:9780070568990.
- Simulia, D. (2010). *Abaqus 6.10 Analysis User's Manual - Volume IV: Elements*.
- White, F. M. (2010). *Fluid Mechanics*. McGraw-Hill, New York. ISBN:0077422414.
- William D. Callister, J., & David G. Rethiwsch. (2010). *Materials science and engineering: An introduction. Materials & Design* (8th ed.). ISBN:978-0470419977. [https://doi.org/10.1016/0261-3069\(91\)90101-9](https://doi.org/10.1016/0261-3069(91)90101-9)
- Yuan Shihua, Peng Zengxiong, J. C. (2010). Experimental Research and Mathematical Model of Drag Torque in Single-Plate Wet Clutch. *Chinese Journal of Mechanical Engineering*, 23.

5.2 CATALOGUES

- Baosteel, A. (n.d.). Baosteel auto UHSS 70.
- Corporation, J. F. E. S. (n.d.). HOT ROLLED STEEL SHEET.
- Datenblatt Saerstahl. (n.d.). *Datasheet - 18CrMo4*.
- Dendoff Springs. (n.d.). Characteristics, Magnetic Steels, Carbon Alloys, Inconel Steels, Stainless Alloys, Titanium Modulus, Elastic Magnetic, Non Magnetic, Non, <<http://www.dendoff.com/cms/wp-content/uploads/201>.
- Disc springs - theory and practice*. (n.d.).
- Foundries de Brousseval et Montreuil. (n.d.). Fonderies de brousseval et montreuil.
- Kyocera. (2017). Kyocera Technical Information.
- Nippon Steel & Sumitomo Metal. (2012). Hot-Rolled Steel Sheets and Coils.

Products, C. R. (n.d.). *isd dunaferf Product catalogue*.

Rods, C., Clamps, H., Various, P., Various, R., Code, C., Sizes, S., ... Mpa, Y. S. (1991). 1045 MEDIUM TENSILE CARBON STEEL BAR Chemical Composition.

Saarstahl. (2011a). Material specification sheet, 4, 7227.

Saarstahl. (2011b). Material specification sheet, 7227.

Sheet, D. (2006). Catamold ® 100Cr6, (October 2000), 1–2.

Stena Aluminium. (2010). Aluminium alloy EN AB-46000, 46000.

Tata Steel. (2010). Ympress ® S315MC Strong , consistent and highly formable.

Terms, T. (n.d.). Electrical steel strip.

5.3 STANDARDS

ASTM International. (2003). ASTM D 341 - Standard Test Method for Viscosity-Temperature Charts for Liquid Petroleum Products.

BRITISH STANDARD. (1981). Specification for ISO metric screw threads, 1981(2), 60.

DIN 2093. (1992).

National Standard of the People's Republic of China. (2004). Structural Steels Subject to End-quench Hardenability Requirements.

Standardization, I. O. of. (1989). ISO 2768 - International standard.

5.4 WEBSITES

Cambridge Dictionary. (n.d.). Barge meaning. Retrieved August 28, 2017, from <http://dictionary.cambridge.org/dictionary/english/barge>

Centrifugal force in rotating hydraulic cylinders - MATLAB. (n.d.). Retrieved August 16, 2017, from https://www.mathworks.com/help/physmod/hydro/ref/centrifugalforceinrotatingcylinder.html?s_tid=gn_loc_drop

Corp, N. S. (n.d.). Presetting Coil Springs | Coil Spring Manufacturing | Newcomb Spring. Retrieved August 11, 2017, from http://www.newcombspring.com/article_presetting_springs.html

Dictionary, C. (n.d.). Gridlock definition. Retrieved August 15, 2017, from <http://dictionary.cambridge.org/dictionary/english/gridlock>

Formulas for Compression Spring Fatigue Design. (n.d.). Retrieved August 23, 2017, from <http://www.amesweb.info/MechanicalSprings/Tables/FormulasForCompressionSpringFatigueDesign.aspx>

Spring, L. (2011). Spring-I-Pedia - The Complete Guide To Spring Engineering. Retrieved August 5, 2017, from <http://springipedia.com/>

APPENDIX

6.1 Extra theoretical aspects

6.2 Table 98 – Resulting centrifugal force of nominal and second scenario for FWD clutch

6.3 Table 99 – Auxiliary table for the study of springs' association of FWD clutch

6.4 Table 100 - Auxiliary table for calculations of the load along the travelling of the FWD springs' association

6.5 Table 101 – Results of transmissible torque of the nominal case for FWD clutch

6.6 Table 102 - Results of transmissible torque of the second scenario for FWD clutch

6.7 Table 103 - Results of transmissible torque of the third and fourth scenarios for FWD clutch

- 6.8 Table 104 - Resulting centrifugal force of third and fourth scenarios for FWD clutch
- 6.9 Table 105 – Auxiliary table to plot the graph of Error!
Reference source not found.
- 6.10 Table 106 – Auxiliary table to the calculations of pre-load conditions
- 6.11 Table 107 – Creeping effect on FWD clutch set
- 6.12 Table 108 – Resulting transmissible torque for a constant slipping speed of 200 rpm
- 6.13 Table 109 - Resulting transmissible torque for a constant slipping speed of 100 rpm
- 6.14 Auxiliary tables for the calculation of the influence of fluid entry radius in the centrifugal force
- 6.15 Table 113 – Excerpt of first series of tables for the calculation of FWD drag losses
- 6.16 Table 114 – Excerpt of second series of tables of FWD drag losses
- 6.17 Table 115 - Excerpt of third series of tables of FWD drag losses
- 6.19 Table 116 - Drive clutch zero-slip current-to-torque sensitivity expressed in Nm/bar for the nominal scenario

6.20 Table 117 - Drive clutch zero-slip current-to-torque sensitivity
expressed in Nm/bar for the second scenario

6.21 Table 118 - Auxiliary tables of graphs of Figure 180 and Figure
181

6.22 Table 119 – Results of the transmissible torque for REV clutch
set (nominal scenario)

6.23 Table 120 - Results of the transmissible torque for REV clutch
set (second scenario)

6.24 Table 121 - Results of the transmissible torque for REV clutch
set (third scenario)

6.25 Table 122 – Excerpt of first series of tables of drag losses
calculations of REV clutch

6.26 Table 123 - Reverse clutch zero-slip current-to-torque
sensitivity expressed in Nm/bar for the nominal scenario

6.27 Table 124 - Reverse clutch zero-slip current-to-torque
sensitivity expressed in Nm/bar for the second scenario

6.28 Table 125 - Reverse clutch zero-slip current-to-torque
sensitivity expressed in Nm/bar for the third scenario

6.29 Auxiliary tables of the design of FWD showcase shifting lever

6.30 Auxiliary tables of the design of REV showcase shifting lever

6.31	Table 132 - Necessary force to actuate the cam
6.32	Table 133 – Applied load on the shifting lever
6.33	Table 134 – Auxiliary table for the calculation of the trendline of the kinematic viscosity vs temperature
6.34	Calculations data
6.35	Preliminary assembly feasibility study
6.36	Datasheet LOCTITE® 638™
6.37	Datasheet LOCTITE® 648™
6.38	Showcase model's BOM
6.39	Poster
6.40	Showcase photos
6.41	Technical drawings of individual parts of the showcase model
6.42	Technical drawings of assemblies of the showcase model

6 APPENDIX

6.1 Extra theoretical aspects

6.1.1 Clutches

6.1.1.1 Dry clutches

Dry clutches are typically equipped on manual gearbox for manual shifting.

As can be seen in Figure 264, engine torque is transmitted to the input shaft of the gearbox via clutch plate which is locked between the flywheel and pressure plate. The clutch system is fastened to the flywheel and as consequence the pressure plate is always rotating at engine speed. When the clutch is engaged, the force exerted by the diaphragm spring is applied to the pressure plate pulling pressing it against the clutch plate. Both sides of the clutch friction plate will be loaded with the same force and the

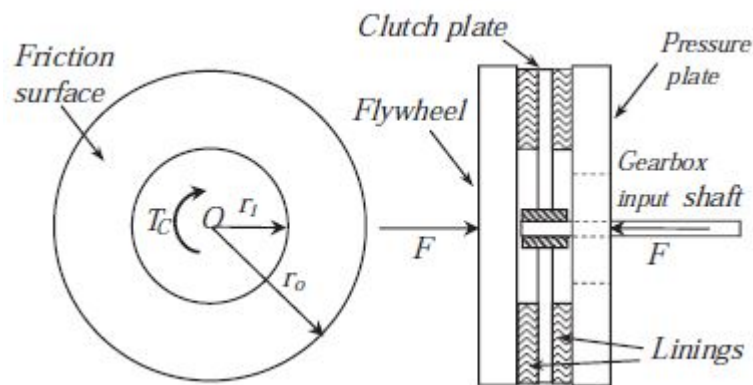


Figure 264 – Friction surfaces of a clutch plate (Mashadi & Crolla, 2012)

total produced torque is the summing up of the torque produced for each friction plate, in this case, two. Which means that each friction plate will produce half of the total torque (Mashadi & Crolla, 2012).

The phenomenon of torque generation in the clutch plate is the consequence of the existence of a normal load and slip that produce friction forces at distances from the

plate centre which results in friction torque. The amount of torque depends on the normal force and slip values (Mashadi & Crolla, 2012).

6.1.1.2 Wet clutches

In contrast to dry clutches, wet clutches are more complex due to the oil supply. Generally, wet clutches are applied in multi-plate design, immersed by the transmission oil in the housing. The energy dissipated during the clutch engagement by the frictional contact transforms itself in heating. The transmission oil cools down the clutch plates. The other function of the oil is to reduce wear. This makes possible the clutch to operate at higher loads when compared to a dry clutch (Gustafsson, 2014).

Wet clutches are applied mainly in AT used in automobiles and trucks. Wet clutches are also used in applications where control of the torque transfer process is demanded (Maki, 2005).

This type of clutches is known for the following advantages (Naunheimer *et al.*, 2010):

- small moment of inertia, depending on the type of damping system;
- small installation space requirement;
- high power/weight ratio;
- large torque capacity;
- high heat capacity as a result of higher thermal energy inputs;
- high durability at low wear;
- good controllability.

There are several important parameters in terms of durability of both dry and wet clutches such as (Naunheimer *et al.*, 2010):

- lining pressure;
- specific frictional power;
- type of friction lining;
- lining thickness;
- lining grooving;
- type and amount of oil;
- shifting frequencies;
- moving-off load profiles;
- temperature.

In Figure 265, the major components that are found in a typical wet clutch device may be found.

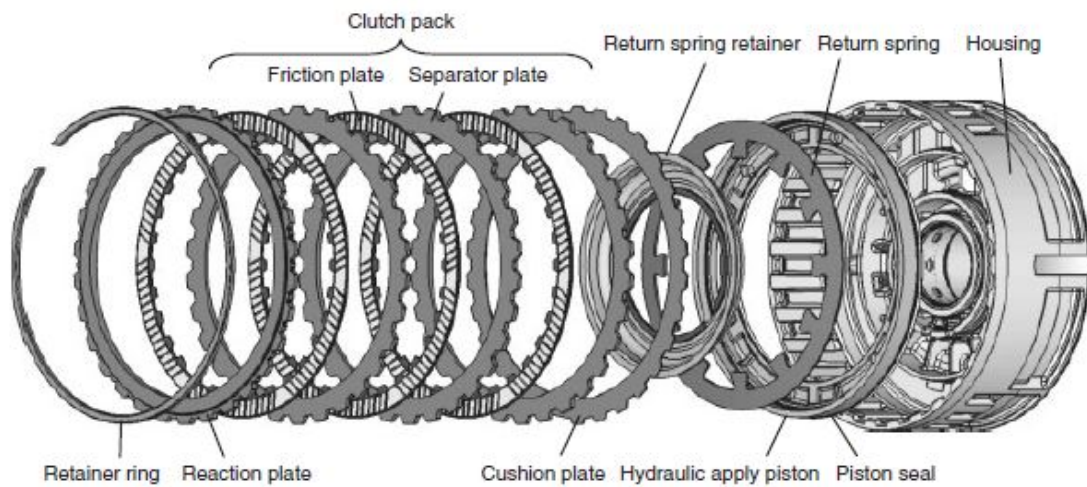


Figure 265 – Major components of wet clutch system (D. Crolla et al., 2014)

Friction plates and separator plates are alternatively placed. A clutch pack is composed by a set of friction and separator plates. The friction plates are assembled on a centre hub which is driven by a rotating drivetrain component. Separator plates are usually splined to the housing which is connected to another drivetrain component. The interface between the plates is lubricated by means of a transmission fluid. The plates are retained between the retaining ring on one side and the piston on the other end. When the clutch is open (disengaged), friction and separator plates can rotate at different speeds while spreading fluid at the interface. When the engagement of the clutch is activated, an actuator applies loading force in the piston. The piston is pulled against a return spring while squeezing oil film at the friction interface. Therefore, the torque is transmitted through viscous shear between sliding plates. A contact between the friction and separator plates takes place when the oil film is squeezed out. By this way torque is partially transmitted through mechanical friction. The clutch engagement is completed when both friction and separator plates are coupled (D. Crolla, Foster, Kobayashi, & Vaughan, 2014).

The primary function of the friction plate is to transmit torque from its spline (which is connected to a drivetrain element) to the separator plates through wet friction. The main function of the separator plate is to transit torque from its friction interface to another gear element. The other function of this component is to act as a heat sink to absorb heat generation during the engagement and releasing of the clutch. The reaction plate (or pressure plate) is located at the end of the clutch pack. The function of this component is to provide a reaction force against the clutch actuator on a uniform way supporting the friction and separator plates. Retainer ring main function is to retain clutch components such as the pressure plate and a return spring. The main purpose of

the (apply) piston is to convert the hydraulic pressure into axial force. The piston return spring pushes the piston back to its released position when the hydraulic pressure is lifted. The cushion plate normally has the shape of a wave plate or coil spring. Its main function is to absorb the initial engagement shock. The main purpose of the balance dam is to avoid or reduce the effect of the centrifugal force created on the piston (D. Crolla et al., 2014).

In Figure 266, it is presented three different geometries of the grooves of the friction plates.

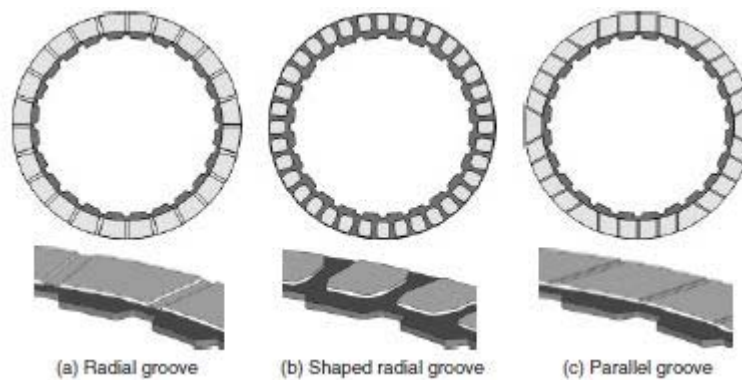


Figure 266 – Shapes of grooves of friction plates (D. Crolla et al., 2014)

The slip control of a wet clutch is the solution for a smooth launch without inducing undesirable driveline noise, vibration and harshness disturbance in the absence of torque converter (D. Crolla et al., 2014).

Due to the working principle of wet clutches (immersed on oil bath) it is inevitable the generation of drag torque. Drag torque depends strongly on the system temperature, oil viscosity. It is important to underline the fact that clutch lining do not get worn however the oil is worn and periodic oil changes are necessary to avoid clutch judder (Naunheimer *et al.*, 2010).

Torsional vibration damping is necessary for wet clutch as also for dry clutches. Torsional vibration damping may be achieved by means of a torsional vibration damper, controlled slip or by combination of torsional vibration damper and continuous slip operation. Damping is considerably high in idle speeds as well as at low torques and low speeds. On the other hand, damping is low at high torques and high speeds. The exemption of damping can be achieved depending on the type of combustion engine in use. By this way the unit becomes more compact and light. Due to the high controllability of wet clutches it is possible to go through low speed range where torsional vibration is critical with light and controlled slip (Naunheimer *et al.*, 2010).

Crawling or stopping on slopes is achievable with wet clutches. Power losses can be attained by the cooling oil (transmission oil). High feed flow rates are necessary for this operation. There are two different concepts for wet clutches in oil-tight housing (Naunheimer *et al.*, 2010):

- the 2-line principle: one channel for the clutch pressure oil, from which the cooling oil is channelled off through suitable throttles, as well as one channel for cooling oil recirculation,
- the 3-line principle: one channel each for the clutch pressure oil, cooling oil and cooling oil recirculation.

6.1.2 Torque converter

The working principle of a torque converter is based on the transfer of momentum from a fluid (oil) to a turbine. The impeller pump is attached to the crankshaft so when the engine rotates the fluid is accelerated, moving from the inner radius to the outer radius along the vanes. This phenomenon is shown in Figure 267 (Mashadi & Crolla, 2012).

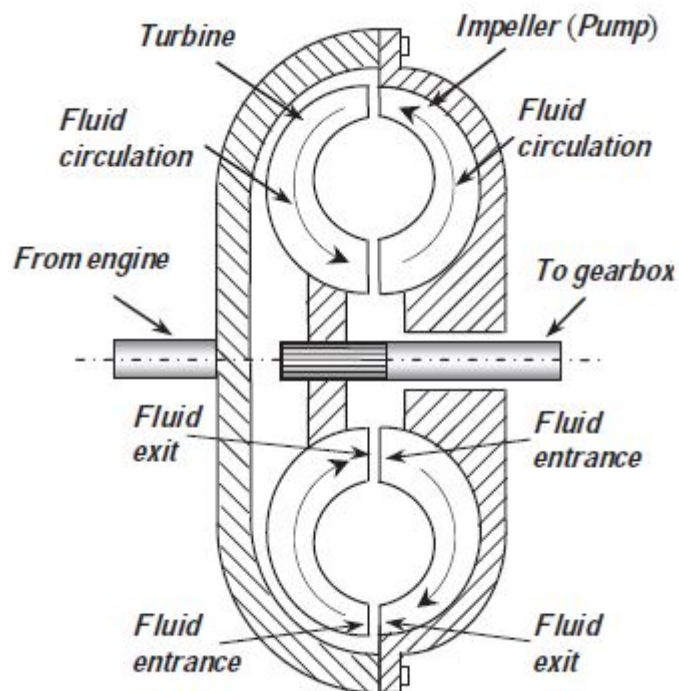


Figure 267 – Schematic of a fluid coupling (Mashadi & Crolla, 2012)

When the fluid is at the outlet of the vane, the fluid with high momentum hits the blade of the turbine. The turbine is fixed to the input shaft of the gearbox (as can be seen in Figure 267). The momentum is transferred to the turbine while the fluid returns to the

inner radius with low velocity and exits the turbine blade. After that a new cycle begins: the fluid enters to the impeller inlet channel and the fluid continuously circulates. When the input shaft of the transmission is rotating, but not the engine, the fluid is under tension but there is no mechanical connection between the engine and the gearbox. This phenomenon (slip in the fluid coupling) allows the vehicle to stop even when the engine is running. However, some torque is transmitted from the engine to the gearbox via the torque converter. When the engine is in idle state, a small amount of torque is transmitted and if the vehicle is in gear it tends to move. This motion is known as “creep”. In fact, this action is desirable in automatic transmissions: when the engine speed rises, the transferred torque to the gearbox increases too which translated in a smooth acceleration of the car (Mashadi & Crolla, 2012).

6.1.3 Bearings

6.1.3.1 Radial needle roller bearings

Needle bearings or specifically needle roller bearings are used where low section height is a benefit (D. Crolla et al., 2014).

The length of the rollers of this type of bearing is four times their diameter. Needle bearings are available with or without inner race. Full-complement is used for high loads, oscillating or slow speeds. Needle bearings cannot support thrust loads (Avallone, III, & Sadegh, 2007).

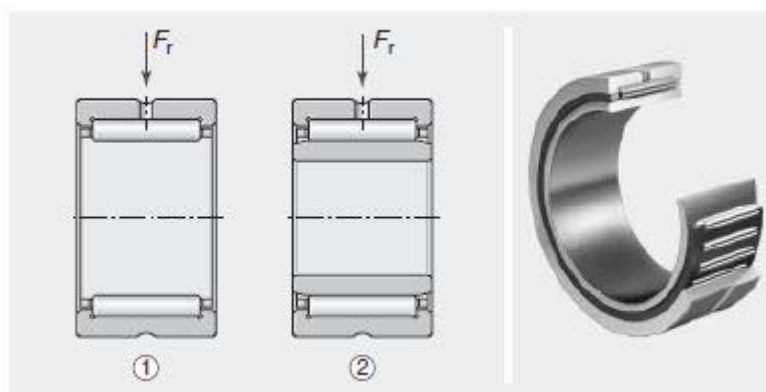


Figure 268 – Structure of a radial needle roller bearing (D. Crolla et al., 2014)

6.1.3.2 Axial needle roller bearings

Axial needle roller bearings or thrust bearings are widely applied in automatic transmissions. In fact, most of the bearings in an automotive transmission are of this type. The forces generated in planetary gear trains cancel each other in a large extent. This means that the bearing is responsible for supporting the axial forces that occur due to the helical gearing (D. Crolla et al., 2014).

This type of bearing should be used for moderate speeds and loads applications where other bearings carry the radial loads (Avallone *et al.*, 2007).

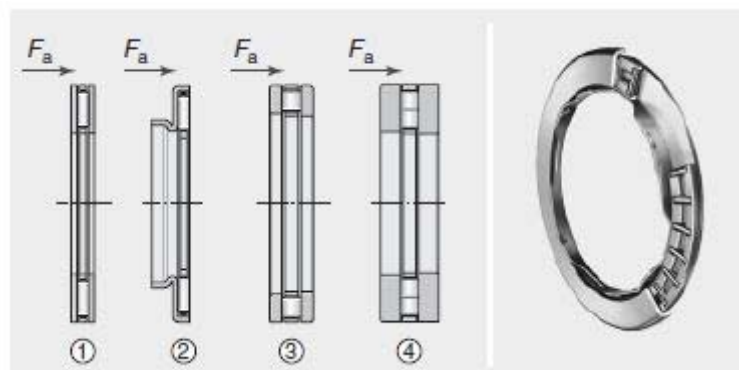


Figure 269 – Different design variants of axial needle roller bearings and cylindrical roller thrust bearings (D. Crolla et al., 2014)

6.2 Table 98 – Resulting centrifugal force of nominal and second scenario for FWD clutch

Table 98 – Resulting centrifugal force of nominal and second scenario for FWD clutch

NOMINAL CASE					SECOND SCENARIO		
Speed [rpm]	Speed [rad/s]	F_centrif_pi ston [N]	F_centrif_ bal [N]	F_centri f [N]	F_centrif _piston [N]	F_centri f_bal [N]	F_ce ntrif [N]
0	0.000	0.000	0.000	0.000	0.000	0.000	0.00 0
500	52.360	12.141	-9.732	2.409	11.910	-9.526	2.38 4

1000	104.72 0	48.565	-38.930	9.635	47.641	-38.105	9.53 6
1500	157.08 0	109.270	-87.592	21.678	107.192	-85.735	21.4 57
2000	209.44 0	194.258	-155.720	38.539	190.563	- 152.41 8	38.1 45
2500	261.79 9	303.529	-243.312	60.216	297.755	- 238.15 4	59.6 02
3000	314.15 9	437.081	-350.369	86.712	428.768	- 342.94 1	85.8 26
3500	366.51 9	594.916	-476.892	118.02 4	583.600	- 466.78 1	116. 819
4000	418.87 9	777.033	-622.879	154.15 4	762.254	- 609.67 4	152. 580
4500	471.23 9	983.433	-788.331	195.10 1	964.727	- 771.61 8	193. 109
5000	523.59 9	1214.114	-973.248	240.86 6	1191.021	- 952.61 5	238. 406
5500	575.95 9	1469.078	-1177.631	291.44 8	1441.136	- 1152.6 64	288. 471
6000	628.31 9	1748.325	-1401.478	346.84 7	1715.071	- 1371.7 66	343. 305
6500	680.67 8	2051.853	-1644.790	407.06 3	2012.826	- 1609.9 20	402. 906
7000	733.03 8	2379.664	-1907.567	472.09 7	2334.402	- 1867.1 26	467. 276
7500	785.39 8	2731.757	-2189.809	541.94 8	2679.798	- 2143.3 84	536. 414

8000	837.75 8	3108.133	-2491.516	616.61 7	3049.015	- 2438.6 95	610. 320
8500	890.11 8	3508.791	-2812.688	696.10 3	3442.052	- 2753.0 58	688. 994
9000	942.47 8	3933.731	-3153.325	780.40 6	3858.909	- 3086.4 73	772. 436
9500	994.83 8	4382.953	-3513.427	869.52 6	4299.587	- 3438.9 41	860. 646

6.3 Table 99 – Auxiliary table for the study of springs' association of FWD clutch

Table 99 – Auxiliary table for the study of springs' association of FWD clutch

Travelling	Spring ratio stiffness	Force
mm	N/mm	N
0	218.817	0.000
0.1	218.817	21.882
0.2	218.817	43.763
0.3	218.817	65.645
0.4	218.817	87.527
0.5	218.817	109.409
0.6	218.817	131.290
0.7	218.817	153.172
0.8	218.817	175.054
0.9	218.817	196.936
1	218.817	218.817
1.1	218.817	240.699
1.2	218.817	262.581
1.3	218.817	284.463
1.4	218.817	306.344
1.5	218.817	328.226
1.6	218.817	350.108
1.7	218.817	371.990
1.8	218.817	393.871
1.9	1218.817	2315.753
2	1218.817	2437.635
2.1	1218.817	2559.516
2.2	1218.817	2681.398
2.3	1218.817	2803.280
2.4	1218.817	2925.162
2.5	1218.817	3047.043
2.6	1218.817	3168.925

6.4 Table 100 - Auxiliary table for calculations of the load along the travelling of the FWD springs' association

Table 100 - Auxiliary table for calculations of the load along the travelling of the FWD springs' association

Pressure	Pressure	F_H	ω	F_centr	F at spring	Travelling	Travelling
bar	Mpa	N	rpm	N	N	mm	mm
0	0	0.000			154.154	0.000	0.000
0.5	0.05	382.705			536.859	0.000	0.000
1	0.1	765.410			919.564	0.000	0.000
1.5	0.15	1148.115			1302.269	1.705	1.705
2	0.2	1530.820			1684.974	2.097	2.097
2.5	0.25	1913.524			2067.679	2.411	2.411
3	0.3	2296.229			2450.384	2.725	2.600
3.5	0.35	2678.934			2833.088	3.039	2.600
4	0.4	3061.639			3215.793	3.353	2.600
4.5	0.45	3444.344	4000	154.154	3598.498	3.667	2.600
5	0.5	3827.049			3981.203	3.981	2.600
5.5	0.55	4209.754			4363.908	4.295	2.600
6	0.6	4592.459			4746.613	4.609	2.600
6.5	0.65	4975.164			5129.318	4.923	2.600
7	0.7	5357.868			5512.023	5.237	2.600
7.5	0.75	5740.573			5894.728	5.551	2.600
8	0.8	6123.278			6277.432	5.865	2.600
8.5	0.85	6505.983			6660.137	6.179	2.600

9	0.9	6888.688	7042.842	6.493	2.600
9.5	0.95	7271.393	7425.547	6.807	2.600
10	1	7654.098	7808.252	7.121	2.600
10.5	1.05	8036.803	8190.957	7.435	2.600
11	1.1	8419.508	8573.662	7.749	2.600
11.5	1.15	8802.212	8956.367	8.063	2.600
12	1.2	9184.917	9339.072	8.377	2.600
12.5	1.25	9567.622	9721.776	8.691	2.600
13	1.3	9950.327	10104.481	9.005	2.600
13.5	1.35	10333.032	10487.186	9.319	2.600
14	1.4	10715.737	10869.891	9.633	2.600
14.5	1.45	11098.442	11252.596	9.947	2.600
15	1.5	11481.147	11635.301	10.261	2.600
15.5	1.55	11863.852	12018.006	10.575	2.600
16	1.6	12246.556	12400.711	10.889	2.600
16.5	1.65	12629.261	12783.416	11.203	2.600
17	1.7	13011.966	13166.120	11.517	2.600
17.5	1.75	13394.671	13548.825	11.831	2.600
18	1.8	13777.376	13931.530	12.145	2.600

CONTRIBUTION TO THE VALIDATION AND VERIFICATION OF DRIVE-NEUTRAL-
REVERSE SUBSYSTEM OF A CVT TRANSMISSION

PEDRO NUNO DOS SANTOS ALMEIDA

6.5 Table 101 – Results of transmissible torque of the nominal case for FWD clutch

Table 101 – Results of transmissible torque of the nominal case for FWD clutch

NOMINAL CASE							
Pressure[bar]	Pressure [Mpa]	F_H [N]	ω [rpm]	F_centrl [N]	F_ax [N]	r_m [m]	Torque [N.m]
0	0	0,000	6500,000	407,063	-1891,062		-89,607
1	0,1	766,843			-1124,219		-53,271
2	0,2	1533,686			-357,375		-16,934
3	0,3	2300,529			409,468		19,403
4	0,4	3067,373			1176,311		55,739
5	0,5	3834,216			1943,154		92,076
6	0,6	4601,059			2709,997		128,412
7	0,7	5367,902			3476,840		164,749
8	0,8	6134,745			4243,683		201,086
9	0,9	6901,588			5010,526		237,422
10	1	7668,431			5777,370	0,062	273,759
11	1,1	8435,274			6544,213		310,096
12	1,2	9202,118			7311,056		346,432
13	1,3	9968,961			8077,899		382,769
14	1,4	10735,804			8844,742		419,106
15	1,5	11502,647			9611,585		455,442
16	1,6	12269,490			10378,428		491,779
17	1,7	13036,333			11145,272		528,116
18	1,8	13803,176			11912,115		564,452
19	1,9	14570,020			12678,958		600,789
20	2	15336,863			13445,801		637,126

6.6 Table 102 - Results of transmissible torque of the second scenario for FWD clutch

Table 102 - Results of transmissible torque of the second scenario for FWD clutch

SECOND SCENARIO									
Pressure [bar]	Pressure [Mpa]	F_H [N]	ω [rpm]	F_centr [N]	F_ax [N]	r_m [m]	Torque [N.m]	Torque [N.m]	Torque [N.m]
0	0	0.000	6500.000	402.906	-2015.619	0.058	-86.146	-104.873	-95.510
1	0.1	765.410			-1250.209		-53.433	-65.049	-59.241
2	0.2	1530.820			-484.799		-20.720	-25.224	-22.972
3	0.3	2296.229			280.611		11.993	14.600	13.297
4	0.4	3061.639			1046.020		44.706	54.425	49.565
5	0.5	3827.049			1811.430		77.419	94.249	85.834
6	0.6	4592.459			2576.840		110.132	134.074	122.103
7	0.7	5357.868			3342.250		142.845	173.898	158.372
8	0.8	6123.278			4107.659		175.558	213.723	194.640
9	0.9	6888.688			4873.069		208.271	253.547	230.909
10	1	7654.098			5638.479		240.984	293.372	267.178
11	1.1	8419.508			6403.889		273.697	333.196	303.447
12	1.2	9184.917			7169.299		306.410	373.021	339.715
13	1.3	9950.327			7934.708		339.123	412.845	375.984
14	1.4	10715.737			8700.118		371.836	452.670	412.253
15	1.5	11481.147			9465.528		404.549	492.494	448.521
16	1.6	12246.556			10230.938		437.262	532.319	484.790
17	1.7	13011.966			10996.347		469.975	572.143	521.059

18	1.8	13777.376		11761.757	502.688	611.968	557.328
19	1.9	14542.786		12527.167	535.401	651.792	593.596
20	2	15308.196		13292.577	568.114	691.617	629.865

6.7 Table 103 - Results of transmissible torque of the third and fourth scenarios for FWD clutch

Table 103 - Results of transmissible torque of the third and fourth scenarios for FWD clutch

3rd scenario									
Pressure [bar]	Pressure [Mpa]	F _H [N]	ω [rpm]	F _{centr} [N]	F _{ax} [N]	r _m [m]	Torque [N.m]	Torque [N.m]	Torque [N.m]
0	0	0.000	6500.000	422.743	-1488.444	0.062	-63.532	-77.343	-70.438
1	0.1	768.276			-720.168		-30.739	-37.422	-34.081
2	0.2	1536.553			48.109		2.053	2.500	2.277
3	0.3	2304.829			816.385		34.846	42.422	38.634
4	0.4	3073.106			1584.662		67.639	82.343	74.991
5	0.5	3841.382			2352.938		100.432	122.265	111.348
6	0.6	4609.659			3121.215		133.225	162.186	147.706
7	0.7	5377.935			3889.491		166.017	202.108	184.063
8	0.8	6146.212			4657.768		198.810	242.030	220.420
9	0.9	6914.488			5426.044		231.603	281.951	256.777
10	1	7682.765			6194.321		264.396	321.873	293.134
11	1.1	8451.041			6962.597		297.189	361.795	329.492
12	1.2	9219.318			7730.874		329.981	401.716	365.849
13	1.3	9987.594			8499.150		362.774	441.638	402.206
14	1.4	10755.871			9267.427		395.567	481.560	438.563
15	1.5	11524.147			10035.703		428.360	521.481	474.921
16	1.6	12292.424			10803.980		461.153	561.403	511.278
17	1.7	13060.700			11572.256		493.945	601.325	547.635

18	1.8	13828.977		12340.533	526.738	641.246	583.992
19	1.9	14597.253		13108.809	559.531	681.168	620.349
20	2	15365.530		13877.085	592.324	721.090	656.707

4th scenario							
F_H [N]	F_centra [N]	F_ax [N]	r_m [m]	Torque [N.m]	Torque [N.m]	Torque [N.m]	
0.000		-2360.047		-100.998	-122.954	-111.976	
765.410		-1594.637		-68.242	-83.078	-75.660	
1530.820		-829.227		-35.487	-43.201	-39.344	
2296.229		-63.817		-2.731	-3.325	-3.028	
3061.639		701.592		30.025	36.552	33.288	
3827.049		1467.002		62.780	76.428	69.604	
4592.459		2232.412		95.536	116.305	105.920	
5357.868		2997.822		128.292	156.181	142.236	
6123.278		3763.232		161.047	196.058	178.553	
6888.688	389.786	4528.641	0.062	193.803	235.934	214.869	
7654.098		5294.051		226.559	275.811	251.185	
8419.508		6059.461		259.314	315.687	287.501	
9184.917		6824.871		292.070	355.564	323.817	
9950.327		7590.280		324.826	395.440	360.133	
10715.737		8355.690		357.581	435.317	396.449	
11481.147		9121.100		390.337	475.193	432.765	
12246.556		9886.510		423.093	515.070	469.081	
13011.966		10651.920		455.848	554.946	505.397	

13777.376	11417.329	488.604	594.822	541.713
14542.786	12182.739	521.360	634.699	578.029
15308.196	12948.149	554.116	674.575	614.345

6.8 Table 104 - Resulting centrifugal force of third and fourth scenarios for FWD clutch

Table 104 - Resulting centrifugal force of third and fourth scenarios for FWD clutch

3rd scenario			4th scenario				
Speed [rpm]	Speed [rad/s]	F_centrif_piston [N]	F_centrif_bal [N]	F_centrif [N]	F_centrif_piston [N]	F_centrif_bal [N]	F_centrif [N]
0	0.000	0.000	0.000	0.000	0.000	0.000	0.000
500	52.360	12.170	-9.668	2.501	12.088	-9.781	2.306
1000	104.720	48.678	-38.673	10.006	48.351	-39.125	9.226
1500	157.080	109.526	-87.013	22.513	108.790	-88.032	20.758
2000	209.440	194.714	-154.691	40.023	193.404	-156.501	36.903
2500	261.799	304.240	-241.704	62.536	302.194	-244.533	57.661
3000	314.159	438.106	-348.054	90.052	435.159	-352.128	83.031
3500	366.519	596.310	-473.740	122.571	592.300	-479.285	113.015
4000	418.879	778.854	-618.762	160.092	773.616	-626.005	147.611
4500	471.239	985.737	-783.121	202.617	979.108	-792.288	186.820
5000	523.599	1216.960	-966.816	250.144	1208.776	-978.133	230.642
5500	575.959	1472.521	-1169.847	302.674	1462.619	-1183.541	279.077
6000	628.319	1752.422	-1392.215	360.207	1740.637	-1408.512	332.125
6500	680.678	2056.662	-1633.919	422.743	2042.831	-1653.045	389.786
7000	733.038	2385.241	-1894.959	490.282	2369.200	-1917.141	452.059
7500	785.398	2738.159	-2175.335	562.824	2719.745	-2200.800	518.945
8000	837.758	3115.417	-2475.048	640.369	3094.466	-2504.021	590.445
8500	890.118	3517.014	-2794.097	722.916	3493.362	-2826.805	666.557

9000	942.478	3942.950	-3132.483	810.467	3916.433	-3169.152	747.281
9500	994.838	4393.225	-3490.205	903.020	4363.680	-3531.061	832.619

6.9 Table 105 – Auxiliary table to plot the graph of Error! Reference source not found.

Table 105 – Auxiliary table to plot the graph of Error! Reference source not found.

3rd scenario					4th scenario									
Pressure bar	Pressure Mpa	F_H N	ω rpm	F_centr N	F at spring N	Travelling mm	Travelling mm	F_H N	F_centr N	F at spring N	Travelling mm	Travelling mm		
0	0	0.000	6500	422.743	422.743	0.000	0.000	0.000	389.786	389.786	0.000	0.000		
0.5	0.05	384.138			806.882	0.000	0.000	382.705		773.924	0.000			
1	0.1	768.276			1191.020	1.952	1.952	765.410		1158.062	0.473			
1.5	0.15	1152.415			1575.158	2.362	2.319	1148.115		1542.200	2.148			
2	0.2	1536.553			1959.296	2.772	2.319	1530.820		1926.339	1.427			
2.5	0.25	1920.691			2343.435	3.182	2.319	1913.524		2310.477	1.666			
3	0.3	2304.829			2727.573	3.592	2.319	2296.229		2694.615	1.905			
3.5	0.35	2688.968			3111.711	4.002	2.319	2678.934		3078.753	2.143			
4	0.4	3073.106			3495.849	4.412	2.319	3061.639		3462.892	2.382			
4.5	0.45	3457.244			3879.988	4.822	2.319	3444.344		3847.030	2.621			
5	0.5	3841.382			4264.126	5.232	2.319	3827.049		4231.168	2.859			
5.5	0.55	4225.521			4648.264	5.642	2.319	4209.754		4615.306	3.098			
6	0.6	4609.659			5032.402	6.052	2.319	4592.459		4999.445	3.337			
6.5	0.65	4993.797			5416.541	6.462	2.319	4975.164		5383.583	3.576			
7	0.7	5377.935			5800.679	6.872	2.319	5357.868		5767.721	3.814			
7.5	0.75	5762.074			6184.817	7.282	2.319	5740.573		6151.859	4.053			
8	0.8	6146.212			6568.955	7.692	2.319	6123.278		6535.998	4.292			
8.5	0.85	6530.350			6953.093	8.103	2.319	6505.983		6920.136	4.530			

9	0.9	6914.488		7337.232	8.513	2.319	6888.688		7304.274	4.769	2.901
9.5	0.95	7298.627		7721.370	8.923	2.319	7271.393		7688.412	5.008	2.901
10	1	7682.765		8105.508	9.333	2.319	7654.098		8072.550	5.246	2.901
10.5	1.05	8066.903		8489.646	9.743	2.319	8036.803		8456.689	5.485	2.901
11	1.1	8451.041		8873.785	10.153	2.319	8419.508		8840.827	5.724	2.901
11.5	1.15	8835.180		9257.923	10.563	2.319	8802.212		9224.965	5.963	2.901
12	1.2	9219.318		9642.061	10.973	2.319	9184.917		9609.103	6.201	2.901
12.5	1.25	9603.456		10026.199	11.383	2.319	9567.622		9993.242	6.440	2.901
13	1.3	9987.594		10410.338	11.793	2.319	9950.327		10377.380	6.679	2.901
13.5	1.35	10371.733		10794.476	12.203	2.319	10333.032		10761.518	6.917	2.901
14	1.4	10755.871		11178.614	12.613	2.319	10715.737		11145.656	7.156	2.901
14.5	1.45	11140.009		11562.752	13.023	2.319	11098.442		11529.795	7.395	2.901
15	1.5	11524.147		11946.891	13.433	2.319	11481.147		11913.933	7.633	2.901
15.5	1.55	11908.285		12331.029	13.843	2.319	11863.852		12298.071	7.872	2.901
16	1.6	12292.424		12715.167	14.253	2.319	12246.556		12682.209	8.111	2.901
16.5	1.65	12676.562		13099.305	14.663	2.319	12629.261		13066.348	8.350	2.901
17	1.7	13060.700		13483.444	15.073	2.319	13011.966		13450.486	8.588	2.901
17.5	1.75	13444.838		13867.582	15.483	2.319	13394.671		13834.624	8.827	2.901
18	1.8	13828.977		14251.720	15.893	2.319	13777.376		14218.762	9.066	2.901

CONTRIBUTION TO THE VALIDATION AND VERIFICATION OF THE
REVERSE SUBSYSTEM OF A CVT TRANSMISSION

PEDRO NUNO DOS SANTOS ALMEIDA

6.10 Table 106 – Auxiliary table to the calculations of pre-load conditions

Table 106 – Auxiliary table to the calculations of pre-load conditions

Nominal scenario						Second scenario					
ω [rpm]	ω [rad/s]	F_centrif [N]	Max_Fh [N]	Max_P [Mpa]	Max_P [bar]	F_centrif [N]	Max_Fh [N]	Max_P [Mpa]	Max_P [bar]		
0	0.000	0.000	808.800	0.105	1.055	0.000	808.800	0.106	1.057		
500	52.360	2.409	806.391	0.105	1.052	2.384	806.416	0.105	1.054		
1000	104.720	9.635	799.165	0.104	1.042	9.536	799.264	0.104	1.044		
1500	157.080	21.678	787.122	0.103	1.026	21.457	787.343	0.103	1.029		
2000	209.440	38.539	770.261	0.100	1.004	38.145	770.655	0.101	1.007		
2500	261.799	60.216	748.584	0.098	0.976	59.602	749.198	0.098	0.979		
3000	314.159	86.712	722.088	0.094	0.942	85.826	722.974	0.094	0.945		
3500	366.519	118.024	690.776	0.090	0.901	116.819	691.981	0.090	0.904		
4000	418.879	154.154	654.646	0.085	0.854	152.580	656.220	0.086	0.857		
4500	471.239	195.101	613.699	0.080	0.800	193.109	615.691	0.080	0.804		
5000	523.599	240.866	567.934	0.074	0.741	238.406	570.394	0.075	0.745		
5500	575.959	291.448	517.352	0.067	0.675	288.471	520.329	0.068	0.680		
6000	628.319	346.847	461.953	0.060	0.602	343.305	465.495	0.061	0.608		
6500	680.678	407.063	401.737	0.052	0.524	402.906	405.894	0.053	0.530		
7000	733.038	472.097	336.703	0.044	0.439	467.276	341.524	0.045	0.446		
7500	785.398	541.948	266.852	0.035	0.348	536.414	272.386	0.036	0.356		
8000	837.758	616.617	192.183	0.025	0.251	610.320	198.480	0.026	0.259		
8500	890.118	696.103	112.697	0.015	0.147	688.994	119.806	0.016	0.157		
9000	942.478	780.406	28.394	0.004	0.037	772.436	36.364	0.005	0.048		

6.11 Table 107 – Creeping effect on FWD clutch set

Table 107 – Creeping effect on FWD clutch set for slipping speed of 200 rpm

CREEPING EFFECT							
Speed [rpm]	Slipping [rpm]	speed [rpm]	Effective [rpm]	speed [rad/s]	F_centrif_piston [N]	F_centrif_bal [N]	F_centrif [N]
0	200		-200	-20.944	1.943	-1.557	0.385
500	200		300	31.416	4.371	-3.504	0.867
1000	200		800	83.776	31.081	-24.915	6.166
1500	200		1300	136.136	82.074	-65.792	16.283
2000	200		1800	188.496	157.349	-126.133	31.216
2500	200		2300	240.855	256.907	-205.939	50.967
3000	200		2800	293.215	380.746	-305.211	75.536
3500	200		3300	345.575	528.868	-423.947	104.921
4000	200		3800	397.935	701.272	-562.148	139.124
4500	200		4300	450.295	897.959	-719.815	178.144
5000	200		4800	502.655	1118.928	-896.946	221.982
5500	200		5300	555.015	1364.179	-1093.542	270.637
6000	200		5800	607.375	1633.712	-1309.603	324.109
6500	200		6300	659.734	1927.528	-1545.129	382.399
7000	200		6800	712.094	2245.626	-1800.120	445.506
7500	200		7300	764.454	2588.006	-2074.577	513.430
8000	200		7800	816.814	2954.669	-2368.498	586.171

8500	200	8300	869.174	3345.614	-2681.884	663.730
9000	200	8800	921.534	3760.841	-3014.735	746.106
9500	200	9300	973.894	4200.350	-3367.051	833.300

6.12 Table 108 – Resulting transmissible torque for a constant slipping speed of 200 rpm

Table 108 – Resulting transmissible torque for a constant slipping speed of 200 rpm

Pressure[bar]	Pressure [Mpa]	F_H [N]	ω [rpm]	F_centrl [N]	F_ax [N]	r_m [m]	Torque [N.m]	Torque [N.m]	Torque [N.m]
0	0	0.000	6500.000	382.399	-1915.726	0.062	-81.877	-99.676	-90.776
1	0.1	766.843			-1148.883		-49.102	-59.777	-54.440
2	0.2	1533.686			-382.040		-16.328	-19.878	-18.103
3	0.3	2300.529			384.803		16.446	20.021	18.234
4	0.4	3067.373			1151.646		49.220	59.920	54.570
5	0.5	3834.216			1918.489		81.995	99.820	90.907
6	0.6	4601.059			2685.332		114.769	139.719	127.244
7	0.7	5367.902			3452.176		147.543	179.618	163.580
8	0.8	6134.745			4219.019		180.317	219.517	199.917
9	0.9	6901.588			4985.862		213.092	259.416	236.254
10	1	7668.431			5752.705		245.866	299.315	272.590
11	1.1	8435.274			6519.548		278.640	339.214	308.927
12	1.2	9202.118			7286.391		311.414	379.113	345.264
13	1.3	9968.961			8053.234		344.189	419.012	381.600
14	1.4	10735.804			8820.077		376.963	458.911	417.937
15	1.5	11502.647			9586.921		409.737	498.810	454.274
16	1.6	12269.490			10353.764		442.511	538.709	490.610

17	1.7	13036.333		11120.607	475.285	578.608	526.947
18	1.8	13803.176		11887.450	508.060	618.507	563.284
19	1.9	14570.020		12654.293	540.834	658.407	599.620
20	2	15336.863		13421.136	573.608	698.306	635.957

6.13 Table 109 - Resulting transmissible torque for a constant slipping speed of 100 rpm

Table 109 - Resulting transmissible torque for a constant slipping speed of 100 rpm

Pressure [bar]	Pressure [Mpa]	F_H [N]	ω [rpm]	F_centr [N]	F_ax [N]	r_m [m]	Torque [N.m]	Torque [N.m]	Torque [N.m]
0	0	0.000	6500.000	394.635	-1903.490	0.062	-81.354	-99.039	-90.196
1	0.1	766.843			-1136.647		-48.579	-59.140	-53.860
2	0.2	1533.686			-369.804		-15.805	-19.241	-17.523
3	0.3	2300.529			397.039		16.969	20.658	18.814
4	0.4	3067.373			1163.882		49.743	60.557	55.150
5	0.5	3834.216			1930.725		82.518	100.456	91.487
6	0.6	4601.059			2697.568		115.292	140.355	127.824
7	0.7	5367.902			3464.412		148.066	180.254	164.160
8	0.8	6134.745			4231.255		180.840	220.153	200.497
9	0.9	6901.588			4998.098		213.615	260.052	236.833
10	1	7668.431			5764.941		246.389	299.952	273.170
11	1.1	8435.274			6531.784		279.163	339.851	309.507
12	1.2	9202.118			7298.627		311.937	379.750	345.843
13	1.3	9968.961			8065.470		344.711	419.649	382.180
14	1.4	10735.804			8832.313		377.486	459.548	418.517
15	1.5	11502.647			9599.157		410.260	499.447	454.853
16	1.6	12269.490			10366.000		443.034	539.346	491.190

17	1.7	13036.333		11132.843	475.808	579.245	527.527
18	1.8	13803.176		11899.686	508.583	619.144	563.863
19	1.9	14570.020		12666.529	541.357	659.043	600.200
20	2	15336.863		13433.372	574.131	698.942	636.537

6.14 Auxiliary tables for the calculation of the influence of fluid entry radius in the centrifugal force

Table 110 – Input data for the calculation of the influence of fluid entry radius in the centrifugal force of FWD piston

FWD piston			
Outer Hydraulic diameter of FWD piston	D_H	mm	118.050
Outer Hydraulic radius of FWD piston	R_H	m	0.059
Inner hydraulic diameter of FWD piston	d_H	mm	64.450
Inner hydraulic radius of FWD piston	r_H	m	0.032

Table 111 - Input data for the calculation of the influence of fluid entry radius in the centrifugal force of BD

Balance dam			
Outer Hydraulic diameter of balance dam	D_H	mm	113.12
Outer Hydraulic radius of balance dam	R_H	m	0.05656
Inner hydraulic diameter of balance dam	d_H	mm	63.5
Inner hydraulic radius of balance dam	r_H	m	0.03175

Table 112 – Results for plotting the graphs of Figure 154 and Figure 155

Speed [rpm]	Speed [rad/s]	F_centrif_piston [N]	F_centrif_bal [N]
6500	680.678	3.495	-2.830
		3.386	-2.732
		3.273	-2.631
		3.156	-2.525
		3.034	-2.416
		2.908	-2.303
		2.778	-2.187
		2.643	-2.066
		2.504	-1.942

6.15 Table 113 – Excerpt of first series of tables for the calculation of FWD drag losses

Table 113 – Excerpt of first series of tables for the calculation of FWD drag losses

Rotating speed		Feeding flow		Outer diameter pressure (R2)				Feeding pressure (R1)			
Speed [rpm]	Speed [rad/s]	Qi [l/min]	Qi [m^3/s]	Eq.1	Eq.2	p [Pa]	p [Mpa]	Eq.1	Eq.2	p0 [Pa]	p0 [Mpa]
0	0.000	0.3	0.000005	2.31E+00	0.00E+00	3.001E+04	0.030	2.961	0.000E+00	3.136E+04	0.031
50	5.236			Eq.3	1.45E+01	3.002E+04	0.030	Eq.3	1.132E+01	3.138E+04	0.031
100	10.472			3.00E+04	5.80E+01	3.006E+04	0.030	-31367.907	4.529E+01	3.141E+04	0.031
150	15.708				1.30E+02	3.014E+04	0.030		1.019E+02	3.147E+04	0.031
200	20.944				2.32E+02	3.024E+04	0.030		1.811E+02	3.155E+04	0.032
250	26.180				3.62E+02	3.037E+04	0.030		2.830E+02	3.165E+04	0.032
300	31.416				5.22E+02	3.053E+04	0.031		4.076E+02	3.177E+04	0.032
350	36.652				7.10E+02	3.072E+04	0.031		5.548E+02	3.192E+04	0.032
400	41.888				9.27E+02	3.093E+04	0.031		7.246E+02	3.209E+04	0.032
450	47.124				1.17E+03	3.118E+04	0.031		9.171E+02	3.228E+04	0.032
500	52.360				1.45E+03	3.145E+04	0.031		1.132E+03	3.250E+04	0.032
550	57.596				1.75E+03	3.176E+04	0.032		1.370E+03	3.273E+04	0.033
600	62.832				2.09E+03	3.209E+04	0.032		1.630E+03	3.300E+04	0.033
650	68.068				2.45E+03	3.245E+04	0.032		1.913E+03	3.328E+04	0.033
700	73.304				2.84E+03	3.285E+04	0.033		2.219E+03	3.358E+04	0.034
750	78.540				3.26E+03	3.327E+04	0.033		2.547E+03	3.391E+04	0.034
800	83.776				3.71E+03	3.372E+04	0.034		2.898E+03	3.426E+04	0.034

850	89.012	4.19E+03	3.419E+04	0.034	3.272E+03	3.464E+04	0.035
900	94.248	4.70E+03	3.470E+04	0.035	3.668E+03	3.503E+04	0.035
950	99.484	5.23E+03	3.524E+04	0.035	4.087E+03	3.545E+04	0.035
1000	104.720	5.80E+03	3.580E+04	0.036	4.529E+03	3.589E+04	0.036
1050	109.956	6.39E+03	3.640E+04	0.036	4.993E+03	3.636E+04	0.036
1100	115.192	7.01E+03	3.702E+04	0.037	5.480E+03	3.684E+04	0.037
1150	120.428	7.67E+03	3.767E+04	0.038	5.989E+03	3.735E+04	0.037
1200	125.664	8.35E+03	3.835E+04	0.038	6.521E+03	3.789E+04	0.038

Pressure variation		Flow rate for full film				Torque	
Δp [Pa]	Δp [MPa]	Eq.1	Eq.4	Eq.5	Q [m ³ /s]	Q [l/min]	T(single plate) T (all plates)
1.359E+03	0.001	-	-	-	4.998E-12	0.000	0.000
1.356E+03	0.001	Eq.2	9.076E+07	3.884E+04	1.166E-08	0.001	0.017
1.347E+03	0.001	-	5.181E+10	3.631E+08	4.662E-08	0.003	0.033
1.331E+03	0.001	Eq.3	8.169E+08	3.811E+04	1.049E-07	0.006	0.050
1.309E+03	0.001	7.399E+16	1.452E+09	3.747E+04	1.865E-07	0.011	0.066
1.280E+03	0.001	Eq.6	2.269E+09	3.666E+04	2.913E-07	0.017	0.083

1.245E+03	0.001	2.763E-04	3.268E+09	-	4.195E-07	0.025	0.066	0.017	0.099
1.204E+03	0.001	Eq.7	4.447E+09	-	5.710E-07	0.034	0.066	0.019	0.116
1.157E+03	0.001	-	5.809E+09	-	7.458E-07	0.045	0.066	0.022	0.132
1.103E+03	0.001	5.181E+10	7.352E+09	-	9.440E-07	0.057	0.066	0.025	0.149
1.042E+03	0.001		9.076E+09	-	1.165E-06	0.070	0.066	0.028	0.165
9.758E+02	0.001		1.098E+10	-	1.410E-06	0.085	0.066	0.030	0.182
9.029E+02	0.001		1.307E+10	-	1.678E-06	0.101	0.066	0.033	0.198
8.237E+02	0.001		1.534E+10	-	1.970E-06	0.118	0.066	0.036	0.215
7.381E+02	0.001		1.779E+10	-	2.284E-06	0.137	0.066	0.039	0.231
6.462E+02	0.001		2.042E+10	-	2.623E-06	0.157	0.066	0.041	0.248
5.479E+02	0.001		2.324E+10	-	2.984E-06	0.179	0.066	0.044	0.264
4.433E+02	0.000		2.623E+10	-	3.369E-06	0.202	0.066	0.047	0.281
3.324E+02	0.000		2.941E+10	-	3.777E-06	0.227	0.066	0.050	0.297

CONTRIBUTION TO THE VALIDATION AND VERIFICATION OF DRIVE-NEUTRAL-
REVERSE SUBSYSTEM OF A CVT TRANSMISSION

PEDRO NUNO DOS SANTOS ALMEIDA

2.151E+02	0.000	-	4.208E-06	0.253	0.066	0.052	0.314
		3.277E+10	6.161E+03				
9.152E+01	0.000	-	4.663E-06	0.280	0.066	0.055	0.330
		3.631E+10	2.621E+03				
-			5.141E-06	0.308	0.065	0.056	0.336
3.843E+01	0.000	4.003E+10	1.101E+03	0.339	0.065	0.053	0.317
		4.393E+10	5.003E+03	0.370	0.064	0.050	0.301
1.747E+02	0.000	4.801E+10	9.088E+03	0.403	0.064	0.048	0.286
		5.228E+10	1.335E+04				
3.174E+02	0.000		6.168E-06				
			6.716E-06				
4.663E+02	0.000		06				

6.16 Table 114 – Excerpt of second series of tables of FWD drag losses

Table 114 – Excerpt of second series of tables of FWD drag losses

Feeding flow		Pressure variation	Flow rate for full film				Torque		
Qi [l/min]	Qi [m ³ /s]	Δp [MPa]	Eq.1	Eq.4	Eq.5	Q [m ³ /s]	Q [l/min]	T (single plate)	T (all plates)
0.3	0.000005	0.000	-	2.720E+08	0.000E+00	0.000E+00	0.000	0.066	0.000
		0.000	Eq.2	9.076E+07	0.000E+00	1.165E-08	0.001	0.066	0.017
		0.000	-	5.181E+10	3.631E+08	0.000E+00	0.003	0.066	0.033
		0.000	Eq.3	8.169E+08	0.000E+00	1.049E-07	0.006	0.066	0.050
		0.000	7.399E+16	1.452E+09	0.000E+00	1.864E-07	0.011	0.066	0.066
		0.000	Eq.6	2.269E+09	0.000E+00	2.913E-07	0.017	0.066	0.083
		0.000	2.763E-04	3.268E+09	0.000E+00	4.195E-07	0.025	0.066	0.099
		0.000	Eq.7	4.447E+09	0.000E+00	5.710E-07	0.034	0.066	0.116
		0.000	-	5.181E+10	5.809E+09	0.000E+00	0.045	0.066	0.132
		0.000		7.352E+09	0.000E+00	9.440E-07	0.057	0.066	0.149
		0.000		9.076E+09	0.000E+00	1.165E-06	0.070	0.066	0.165
		0.000		1.098E+10	0.000E+00	1.410E-06	0.085	0.066	0.182
		0.000		1.307E+10	0.000E+00	1.678E-06	0.101	0.066	0.198
		0.000		1.534E+10	0.000E+00	1.970E-06	0.118	0.066	0.215

0.000	1.779E+10	0.000E+00	2.284E-06	0.137	0.066	0.039	0.231
0.000	2.042E+10	0.000E+00	2.623E-06	0.157	0.066	0.041	0.248
0.000	2.324E+10	0.000E+00	2.984E-06	0.179	0.066	0.044	0.264
0.000	2.623E+10	0.000E+00	3.369E-06	0.202	0.066	0.047	0.281
0.000	2.941E+10	0.000E+00	3.777E-06	0.227	0.066	0.050	0.297
0.000	3.277E+10	0.000E+00	4.208E-06	0.253	0.066	0.052	0.314
0.000	3.631E+10	0.000E+00	4.663E-06	0.280	0.066	0.055	0.330
0.000	4.003E+10	0.000E+00	5.141E-06	0.308	0.065	0.056	0.336
0.000	4.393E+10	0.000E+00	5.643E-06	0.339	0.065	0.053	0.317
0.000	4.801E+10	0.000E+00	6.168E-06	0.370	0.064	0.050	0.301
0.000	5.228E+10	0.000E+00	6.716E-06	0.403	0.064	0.048	0.286
0.000	5.673E+10	0.000E+00	7.288E-06	0.437	0.063	0.045	0.272

6.17 Table 115 - Excerpt of third series of tables of FWD drag losses

Table 115 - Excerpt of third series of tables of FWD drag losses

Oil temperature T [°C]	Oil properties		Feeding flow		Outer diameter pressure (R2)			
	Dynamic viscosity [Pa.s]	Density [kg/m^3]	Qi [l/min]	Qj [m^3/s]	Eq.1	Eq.2	p [Pa]	p [Mpa]
-20	0.81096	872	0.3	0.000005	2.47E+00	0.00E+00	2.636E+06	2.636
					Eq.3	1.55E+01	2.636E+06	2.636
					-2.64E+06	6.19E+01	2.636E+06	2.636
						1.39E+02	2.636E+06	2.636
						2.47E+02	2.636E+06	2.636
						3.87E+02	2.636E+06	2.636
						5.57E+02	2.636E+06	2.636
						7.58E+02	2.637E+06	2.637
						9.90E+02	2.637E+06	2.637
						1.25E+03	2.637E+06	2.637
						1.55E+03	2.637E+06	2.637
						1.87E+03	2.638E+06	2.638
						2.23E+03	2.638E+06	2.638
						2.61E+03	2.639E+06	2.639
						3.03E+03	2.639E+06	2.639
						3.48E+03	2.639E+06	2.639

3.96E+03	2.640E+06	2.640
4.47E+03	2.640E+06	2.640
5.01E+03	2.641E+06	2.641
5.58E+03	2.642E+06	2.642
6.19E+03	2.642E+06	2.642
6.82E+03	2.643E+06	2.643
7.49E+03	2.643E+06	2.643
8.18E+03	2.644E+06	2.644
8.91E+03	2.645E+06	2.645
9.67E+03	2.646E+06	2.646
1.05E+04	2.646E+06	2.646
1.13E+04	2.647E+06	2.647

Feeding pressure (R1)			
Eq.1	Eq.2	p0 [Pa]	p0 [Mpa]
3.160	0	2.755E+06	2.755
Eq.3	12.08396439	2.755E+06	2.755
-			
2755398.865	48.33585756	2.755E+06	2.755
	108.7556795	2.756E+06	2.756
	193.3434302	2.756E+06	2.756
	302.0991097	2.756E+06	2.756

CONTRIBUTION TO THE VALIDATION AND VERIFICATION OF DRIVE-NEUTRAL-REVERSE SUBSYSTEM OF A CVT TRANSMISSION

PEDRO NUNO DOS SANTOS ALMEIDA

435.022718	2.756E+06	2.756
592.1142551	2.756E+06	2.756
773.3737209	2.756E+06	2.756
978.8011155	2.756E+06	2.756
1208.396439	2.757E+06	2.757
1462.159691	2.757E+06	2.757
1740.090872	2.757E+06	2.757
2042.189982	2.757E+06	2.757
2368.45702	2.758E+06	2.758
2718.891988	2.758E+06	2.758
3093.494884	2.758E+06	2.758
3492.265708	2.759E+06	2.759
3915.204462	2.759E+06	2.759
4362.311144	2.760E+06	2.760
4833.585756	2.760E+06	2.760
5329.028296	2.761E+06	2.761
5848.638764	2.761E+06	2.761
6392.417162	2.762E+06	2.762
6960.363488	2.762E+06	2.762
7552.477743	2.763E+06	2.763
8168.759927	2.764E+06	2.764
8809.21004	2.764E+06	2.764

Pressure variation		Flow rate for full film			Torque		P			
Δp [Pa]	Δp [MPa]	Eq.1	Eq.4	Eq.5	Q [m^3/s]	Q [l/min]	Rs	T(single plate)	T (all plates)	Power [W]

CONTRIBUTION TO THE VALIDATION AND VERIFICATION OF DRIVE-NEUTRAL-
REVERSE SUBSYSTEM OF A CVT TRANSMISSION

PEDRO NUNO DOS SANTOS ALMEIDA

1.195E+05	0.119	-2.389E+10	0.000E+00	-3.651E+06	5.000E-12	0.000	0.066	0.000	0.000
1.195E+05	0.119	Eq.2	1.034E+08	-3.651E+06	1.466E-10	0.000	0.066	0.242	1.450
1.195E+05	0.119	-5.530E+10	4.136E+08	-3.651E+06	5.714E-10	0.000	0.066	0.483	2.901
1.194E+05	0.119	Eq.3	9.306E+08	-3.650E+06	1.279E-09	0.000	0.066	0.725	4.351
1.194E+05	0.119	5.709E+20	1.654E+09	-3.650E+06	2.270E-09	0.000	0.066	0.967	5.801
1.194E+05	0.119	Eq.6	2.585E+09	-3.649E+06	3.545E-09	0.000	0.066	1.209	7.252
1.193E+05	0.119	2.763E-04	3.722E+09	-3.648E+06	5.102E-09	0.000	0.066	1.450	8.702
1.193E+05	0.119	Eq.7	5.066E+09	-3.646E+06	6.943E-09	0.000	0.066	1.692	10.152
1.192E+05	0.119	-5.530E+10	6.617E+09	-3.645E+06	9.067E-09	0.001	0.066	1.934	11.603
1.192E+05	0.119		8.375E+09	-3.643E+06	1.147E-08	0.001	0.066	2.175	13.053
1.191E+05	0.119		1.034E+10	-3.641E+06	1.416E-08	0.001	0.066	2.417	14.503
1.191E+05	0.119		1.251E+10	-3.639E+06	1.714E-08	0.001	0.066	2.659	15.954
1.190E+05	0.119		1.489E+10	-3.636E+06	2.039E-08	0.001	0.066	2.901	17.404
1.189E+05	0.119		1.747E+10	-3.634E+06	2.393E-08	0.001	0.066	3.142	18.854
1.188E+05	0.119		2.027E+10	-3.631E+06	2.776E-08	0.002	0.066	3.384	20.305
1.187E+05	0.119		2.326E+10	-3.628E+06	3.186E-08	0.002	0.066	3.626	21.755
1.186E+05	0.119		2.647E+10	-3.625E+06	3.625E-08	0.002	0.066	3.868	23.205
1.185E+05	0.118		2.988E+10	-3.621E+06	4.092E-08	0.002	0.066	4.109	24.656
1.184E+05	0.118		3.350E+10	-3.618E+06	4.588E-08	0.003	0.066	4.351	26.106
1.182E+05	0.118		3.733E+10	-3.614E+06	5.112E-08	0.003	0.066	4.593	27.556
1.181E+05	0.118		4.136E+10	-3.610E+06	5.664E-08	0.003	0.066	4.834	29.007
1.180E+05	0.118		4.560E+10	-3.606E+06	6.245E-08	0.004	0.066	5.076	30.457
1.178E+05	0.118		5.004E+10	-3.601E+06	6.853E-08	0.004	0.066	5.318	31.907
1.177E+05	0.118		5.470E+10	-3.597E+06	7.491E-08	0.004	0.066	5.560	33.358
1.175E+05	0.118		5.956E+10	-3.592E+06	8.156E-08	0.005	0.066	5.801	34.808
1.174E+05	0.117		6.462E+10	-3.587E+06	8.850E-08	0.005	0.066	6.043	36.258

CONTRIBUTION TO THE VALIDATION AND VERIFICATION OF DRIVE-NEUTRAL-
REVERSE SUBSYSTEM OF A CVT TRANSMISSION

PEDRO NUNO DOS SANTOS ALMEIDA

1.172E+05	0.117	6.990E+10	-3.581E+06	9.572E-08	0.006	0.066	6.285	37.709	5133.495
1.170E+05	0.117	7.538E+10	-3.576E+06	1.032E-07	0.006	0.066	6.526	39.159	5535.973

6.19 Table 116 - Drive clutch zero-slip current-to-torque sensitivity expressed in Nm/bar for the nominal scenario

Table 116 - Drive clutch zero-slip current-to-torque sensitivity expressed in Nm/bar for the nominal scenario

Min. torque	Max. torque	Nominal torque
32.774	39.899	36.337
32.774	39.899	36.337
32.774	39.899	36.337
32.774	39.899	36.337
32.774	39.899	36.337
32.774	39.899	36.337
32.774	39.899	36.337
32.774	39.899	36.337
32.774	39.899	36.337
32.774	39.899	36.337
32.774	39.899	36.337
32.774	39.899	36.337
32.774	39.899	36.337
32.774	39.899	36.337
32.774	39.899	36.337
32.774	39.899	36.337
32.774	39.899	36.337
32.774	39.899	36.337
32.774	39.899	36.337
32.774	39.899	36.337
32.774	39.899	36.337

6.20 Table 117 - Drive clutch zero-slip current-to-torque sensitivity expressed in Nm/bar for the second scenario

Table 117 - Drive clutch zero-slip current-to-torque sensitivity expressed in Nm/bar for the second scenario

Min. torque	Max. torque	Nominal torque
32.713	39.824	36.269
32.713	39.824	36.269
32.713	39.824	36.269
32.713	39.824	36.269
32.713	39.824	36.269
32.713	39.824	36.269
32.713	39.824	36.269
32.713	39.824	36.269
32.713	39.824	36.269
32.713	39.824	36.269
32.713	39.824	36.269
32.713	39.824	36.269
32.713	39.824	36.269
32.713	39.824	36.269
32.713	39.824	36.269
32.713	39.824	36.269
32.713	39.824	36.269
32.713	39.824	36.269
32.713	39.824	36.269
32.713	39.824	36.269
32.713	39.824	36.269
32.713	39.824	36.269

6.21 Table 118 - Auxiliary tables of graphs of Figure 180 and Figure 181

Table 118 - Auxiliary tables of graphs of Figure 180 and Figure 181

Travelling	Spring ratio stiffness	Force
mm	N/mm	N
0	116.5178571	0
0.1	116.5178571	11.65178571
0.2	116.5178571	23.30357143
0.3	116.5178571	34.95535714
0.4	116.5178571	46.60714286
0.5	116.5178571	58.25892857
0.6	116.5178571	69.91071429
0.7	116.5178571	81.5625
0.8	116.5178571	93.21428571
0.9	116.5178571	104.8660714
1	116.5178571	116.5178571
1.1	116.5178571	128.1696429
1.2	1548.517857	1858.221429
1.3	1548.517857	2013.073214
1.4	1548.517857	2167.925
1.5	1548.517857	2322.776786
1.6	1548.517857	2477.628571
1.7	1548.517857	2632.480357
1.8	1548.517857	2787.332143
1.9	1548.517857	2942.183929
2	1548.517857	3097.035714
2.1	1548.517857	3251.8875
2.2	1548.517857	3406.739286
2.3	1548.517857	3561.591071
2.4	1548.517857	3716.442857
2.5	1548.517857	3871.294643
2.6	1548.517857	4026.146429

6.22 Table 119 – Results of the transmissible torque for REV clutch set (nominal scenario)

Table 119 – Results of the transmissible torque for REV clutch set (nominal scenario)

Nominal scenario									
Pressure [bar]	Pressure [Mpa]	F_H [N]	F_R1	F_R2	F_ax [N]	r_m [m]	Torque [N.m]	Torque [N.m]	Torque [N.m]
0	0	0.000	-656.420	-1817.808	-1817.808		-73.761	-89.796	-81.778
1	0.1	1068.770	412.350	-749.038	-749.038		-30.394	-37.001	-33.697
2	0.2	2137.540	1481.120	319.732	319.732		12.974	15.794	14.384
3	0.3	3206.309	2549.890	1388.501	1388.501		56.341	68.589	62.465
4	0.4	4275.079	3618.660	2457.271	2457.271		99.708	121.384	110.546
5	0.5	5343.849	4687.429	3526.041	3526.041		143.076	174.179	158.627
6	0.6	6412.619	5756.199	4594.811	4594.811		186.443	226.974	206.708
7	0.7	7481.389	6824.969	5663.581	5663.581		229.810	279.769	254.790
8	0.8	8550.159	7893.739	6732.351	6732.351	0,088	273.177	332.564	302.871
9	0.9	9618.928	8962.509	7801.120	7801.120		316.545	385.359	350.952
10	1	10687.698	10031.279	8869.890	8869.890		359.912	438.154	399.033
11	1.1	11756.468	11100.048	9938.660	9938.660		403.279	490.949	447.114
12	1.2	12825.238	12168.818	11007.430	11007.430		446.647	543.744	495.195
13	1.3	13894.008	13237.588	12076.200	12076.200		490.014	596.539	543.276
14	1.4	14962.777	14306.358	13144.969	13144.969		533.381	649.334	591.357
15	1.5	16031.547	15375.128	14213.739	14213.739		576.749	702.129	639.439
16	1.6	17100.317	16443.897	15282.509	15282.509		620.116	754.924	687.520
17	1.7	18169.087	17512.667	16351.279	16351.279		663.483	807.719	735.601

18	1.8	19237.857	18581.437	17420.049	17420.049	706.850	860.514	783.682
19	1.9	20306.627	19650.207	18488.819	18488.819	750.218	913.308	831.763
20	2	21375.396	20718.977	19557.588	19557.588	793.585	966.103	879.844

6.23 Table 120 - Results of the transmissible torque for REV clutch set (second scenario)

Table 120 - Results of the transmissible torque for REV clutch set (second scenario)

Second scenario

Pressure [bar]	Pressure [Mpa]	F_H [N]	F_R1	F_R2	F_ax [N]	r_m [m]	Min. Torque [N.m]	Max. Torque [N.m]	Torque [N.m]
0	0	0.000	-588.389	-1615.661	-1615.661		-65.558	-79.810	-72.684
1	0.1	1072.332	483.943	-543.329	-543.329		-22.047	-26.839	-24.443
2	0.2	2144.665	1556.275	529.004	529.004		21.465	26.132	23.798
3	0.3	3216.997	2628.608	1601.336	1601.336		64.977	79.103	72.040
4	0.4	4289.330	3700.940	2673.668	2673.668		108.489	132.074	120.281
5	0.5	5361.662	4773.273	3746.001	3746.001		152.001	185.045	168.523
6	0.6	6433.994	5845.605	4818.333	4818.333	0.088	195.513	238.015	216.764
7	0.7	7506.327	6917.937	5890.666	5890.666		239.025	290.986	265.005
8	0.8	8578.659	7990.270	6962.998	6962.998		282.536	343.957	313.247
9	0.9	9650.991	9062.602	8035.330	8035.330		326.048	396.928	361.488
10	1	10723.324	10134.935	9107.663	9107.663		369.560	449.899	409.730
11	1.1	11795.656	11207.267	10179.995	10179.995		413.072	502.870	457.971
12	1.2	12867.989	12279.599	11252.328	11252.328		456.584	555.841	506.212
13	1.3	13940.321	13351.932	12324.660	12324.660		500.096	608.812	554.454

14	1.4	15012.653	14424.264	13396.992	13396.992	543.607	661.783	602.695
15	1.5	16084.986	15496.597	14469.325	14469.325	587.119	714.754	650.937
16	1.6	17157.318	16568.929	15541.657	15541.657	630.631	767.725	699.178
17	1.7	18229.651	17641.261	16613.990	16613.990	674.143	820.696	747.419
18	1.8	19301.983	18713.594	17686.322	17686.322	717.655	873.667	795.661
19	1.9	20374.315	19785.926	18758.654	18758.654	761.167	926.638	843.902
20	2	21446.648	20858.258	19830.987	19830.987	804.679	979.609	892.144

CONTRIBUTION TO THE VALIDATION AND VERIFICATION OF DRIVE-NEUTRAL-
REVERSE SUBSYSTEM OF A CVT TRANSMISSION

PEDRO NUNO DOS SANTOS ALMEIDA

6.24 Table 121 - Results of the transmissible torque for REV clutch set (third scenario)

Table 121 - Results of the transmissible torque for REV clutch set (third scenario)

Third scenario										
Pressure [bar]	Pressure [Mpa]	F_H [N]	F_R1	F_R2	F_ax [N]	r_m [m]	Min. Torque [N.m]	Max. Torque [N.m]	Torque [N.m]	
0	0	0.000	-724.450	-2019.955	-2019.955		-81.963	-99.781	-90.872	
1	0.1	1065.207	340.757	-954.748	-954.748		-38.741	-47.163	-42.952	
2	0.2	2130.415	1405.965	110.460	110.460		4.482	5.456	4.969	
3	0.3	3195.622	2471.172	1175.667	1175.667		47.705	58.075	52.890	
4	0.4	4260.829	3536.379	2240.874	2240.874		90.928	110.694	100.811	
5	0.5	5326.036	4601.586	3306.081	3306.081		134.150	163.313	148.732	
6	0.6	6391.244	5666.794	4371.289	4371.289		177.373	215.932	196.653	
7	0.7	7456.451	6732.001	5436.496	5436.496		220.596	268.551	244.574	
8	0.8	8521.658	7797.208	6501.703	6501.703	0.088	263.819	321.170	292.494	
9	0.9	9586.865	8862.415	7566.910	7566.910		307.041	373.789	340.415	
10	1	10652.073	9927.623	8632.118	8632.118		350.264	426.408	388.336	
11	1.1	11717.280	10992.830	9697.325	9697.325		393.487	479.027	436.257	
12	1.2	12782.487	12058.037	10762.532	10762.532		436.709	531.646	484.178	
13	1.3	13847.694	13123.244	11827.739	11827.739		479.932	584.265	532.099	
14	1.4	14912.902	14188.452	12892.947	12892.947		523.155	636.884	580.020	
15	1.5	15978.109	15253.659	13958.154	13958.154		566.378	689.503	627.940	
16	1.6	17043.316	16318.866	15023.361	15023.361		609.600	742.122	675.861	

17	1.7	18108.523	17384.073	16088.568	16088.568	652.823	794.741	723.782
18	1.8	19173.731	18449.281	17153.776	17153.776	696.046	847.360	771.703
19	1.9	20238.938	19514.488	18218.983	18218.983	739.269	899.979	819.624
20	2	21304.145	20579.695	19284.190	19284.190	782.491	952.598	867.545

6.25 Table 122 – Excerpt of first series of tables of drag losses calculations of REV clutch

Table 122 – Excerpt of first series of tables of drag losses calculations of REV clutch

Rotating speed		Feeding flow		Outer diameter pressure (R2)				Feeding pressure (R1)			
Speed [rpm]	Speed [rad/s]	Qi [l/min]	Qi [m ³ /s]	Eq.1	Eq.2	p [Pa]	p [Mpa]	Eq.1	Eq.2	p0 [Pa]	p0 [Mpa]
0	0.000	0.45	0.0000075	2.67E+00	0.00E+00	5.136E+04	0.051	3.19755107	0	5.331E+040.053	5.331E+040.053
50	5.236			Eq.3	2.88E+01	5.139E+04	0.051	Eq.3	23.9951769	5.334E+040.053	5.334E+040.053
100	10.472			-5.14E+04	1.15E+02	5.147E+04	0.051	-53315.68699	95.98070759	5.341E+040.053	5.341E+040.053
150	15.708				2.59E+02	5.162E+04	0.052		215.9565921	5.353E+040.054	5.353E+040.054
200	20.944				4.60E+02	5.182E+04	0.052		383.9228304	5.370E+040.054	5.370E+040.054
250	26.180				7.19E+02	5.208E+04	0.052		599.8794224	5.391E+040.054	5.391E+040.054
300	31.416				1.04E+03	5.239E+04	0.052		863.8263683	5.418E+040.054	5.418E+040.054
350	36.652				1.41E+03	5.277E+04	0.053		1175.763668	5.449E+040.054	5.449E+040.054
400	41.888				1.84E+03	5.320E+04	0.053		1535.691321	5.485E+040.055	5.485E+040.055
450	47.124				2.33E+03	5.369E+04	0.054		1943.609329	5.526E+040.055	5.526E+040.055
500	52.360				2.88E+03	5.423E+04	0.054		2399.51769	5.571E+040.056	5.571E+040.056
550	57.596				3.48E+03	5.484E+04	0.055		2903.416405	5.622E+040.056	5.622E+040.056
600	62.832				4.14E+03	5.550E+04	0.056		3455.305473	5.677E+040.057	5.677E+040.057
650	68.068				4.86E+03	5.622E+04	0.056		4055.184896	5.737E+040.057	5.737E+040.057
700	73.304				5.64E+03	5.700E+04	0.057		4703.054672	5.802E+040.058	5.802E+040.058
750	78.540				6.47E+03	5.783E+04	0.058		5398.914802	5.871E+040.059	5.871E+040.059
800	83.776				7.37E+03	5.872E+04	0.059		6142.765286	5.946E+040.059	5.946E+040.059
850	89.012				8.32E+03	5.967E+04	0.060		6934.606123	6.025E+040.060	6.025E+040.060

900	94.248	9.32E+03	6.068E+04	0.061	7774.437315	6.109E+040.061
950	99.484	1.04E+04	6.174E+04	0.062	8662.25886	6.197E+040.062
1000	104.720	1.15E+04	6.287E+04	0.063	9598.070759	6.291E+040.063
1050	109.956	1.27E+04	6.405E+04	0.064	10581.87301	6.389E+040.064
1100	115.192	1.39E+04	6.528E+04	0.065	11613.66562	6.493E+040.065
1150	120.428	1.52E+04	6.658E+04	0.067	12693.44858	6.601E+040.066
1200	125.664	1.66E+04	6.793E+04	0.068	13821.22189	6.713E+040.067

Pressure variation	
Δp [Pa]	Δp [MPa]
1.956E+03	0.002
1.951E+03	0.002
1.937E+03	0.002
1.913E+03	0.002
1.879E+03	0.002
1.836E+03	0.002
1.784E+03	0.002
1.722E+03	0.002
1.650E+03	0.002
1.569E+03	0.002
1.478E+03	0.001
1.377E+03	0.001
1.267E+03	0.001
1.148E+03	0.001

1.019E+03 0.001
8.804E+02 0.001
7.322E+02 0.001
5.745E+02 0.001
4.072E+02 0.000
2.304E+02 0.000
4.400E+01 0.000
-1.519E+02 0.000
-3.575E+02 0.000
-5.725E+02 -0.001
-7.971E+02 -0.001

Flow rate for full film				Torque				Power	
Eq.1	Eq.4	Eq.5	Q [m^3/s]	Q [l/min]	T (single plate)	T (all plates)	T (single plate) on primary shaft on primary shaft	T (all plates) on primary shaft	Drag power
-2.608E+08	0.000E+00	-2.041E+04	7.498E-12	0.000	0.0920.000	0.000	0.000	0.000	0.000
Eq.2	4.988E+07	-2.036E+04	1.833E-08	0.001	0.0920.011	0.043	0.026	0.106	0.553
-1.888E+10	1.995E+08	-2.021E+04	7.330E-08	0.004	0.0920.022	0.087	0.053	0.211	2.211
Eq.3	4.489E+08	-1.996E+04	1.649E-07	0.010	0.0920.033	0.130	0.079	0.317	4.974
6.803E+16	7.981E+08	-1.961E+04	2.932E-07	0.018	0.0920.043	0.174	0.106	0.422	8.843
Eq.6	1.247E+09	-1.916E+04	4.581E-07	0.027	0.0920.054	0.217	0.132	0.528	13.817
2.763E-04	1.796E+09	-1.862E+04	6.597E-07	0.040	0.0920.065	0.261	0.158	0.633	19.897
Eq.7	2.444E+09	-1.797E+04	8.979E-07	0.054	0.0920.076	0.304	0.185	0.739	27.082
-1.888E+10	3.192E+09	-1.722E+04	1.173E-06	0.070	0.0920.087	0.348	0.211	0.844	35.373
	4.040E+09	-1.637E+04	1.484E-06	0.089	0.0920.098	0.391	0.238	0.950	44.769

CONTRIBUTION TO THE VALIDATION AND VERIFICATION OF DRIVE-NEUTRAL-
REVERSE SUBSYSTEM OF A CVT TRANSMISSION

PEDRO NUNO DOS SANTOS ALMEIDA

4.988E+09	-1.542E+04	1.832E-06	0.110	0.0920.109	0.435	0.264	1.056	55.270
6.035E+09	-1.438E+04	2.217E-06	0.133	0.0920.120	0.478	0.290	1.161	66.877
7.183E+09	-1.323E+04	2.639E-06	0.158	0.0920.130	0.522	0.317	1.267	79.589
8.430E+09	-1.198E+04	3.097E-06	0.186	0.0920.141	0.565	0.343	1.372	93.406
9.776E+09	-1.063E+04	3.592E-06	0.216	0.0920.152	0.609	0.369	1.478	108.329
1.122E+10	-9.188E+03	4.123E-06	0.247	0.0920.163	0.652	0.396	1.583	124.357
1.277E+10	-7.642E+03	4.692E-06	0.281	0.0920.174	0.695	0.422	1.689	141.491
1.442E+10	-5.996E+03	5.296E-06	0.318	0.0920.185	0.739	0.449	1.794	159.730
1.616E+10	-4.250E+03	5.938E-06	0.356	0.0920.196	0.782	0.475	1.900	179.074
1.801E+10	-2.405E+03	6.616E-06	0.397	0.0920.206	0.826	0.501	2.006	199.524
1.995E+10	-4.592E+02	7.331E-06	0.440	0.0920.217	0.869	0.528	2.111	221.079
2.200E+10	1.586E+03	8.083E-06	0.485	0.0920.210	0.841	0.511	2.043	224.686
2.414E+10	3.731E+03	8.871E-06	0.532	0.0910.199	0.797	0.484	1.936	222.991
2.639E+10	5.975E+03	9.696E-06	0.582	0.0900.189	0.757	0.460	1.839	221.511
2.873E+10	8.320E+03	1.056E-05	0.633	0.0900.180	0.722	0.438	1.752	220.211

6.26 Table 123 - Reverse clutch zero-slip current-to-torque sensitivity expressed in Nm/bar for the nominal scenario

Table 123 - Reverse clutch zero-slip current-to-torque sensitivity expressed in Nm/bar for the nominal scenario

Min. torque	Max. torque	Nominal torque
43.367	52.795	48.081
43.367	52.795	48.081
43.367	52.795	48.081
43.367	52.795	48.081
43.367	52.795	48.081
43.367	52.795	48.081
43.367	52.795	48.081
43.367	52.795	48.081
43.367	52.795	48.081
43.367	52.795	48.081
43.367	52.795	48.081
43.367	52.795	48.081
43.367	52.795	48.081
43.367	52.795	48.081
43.367	52.795	48.081
43.367	52.795	48.081
43.367	52.795	48.081
43.367	52.795	48.081
43.367	52.795	48.081
43.367	52.795	48.081
43.367	52.795	48.081

6.27 Table 124 - Reverse clutch zero-slip current-to-torque sensitivity expressed in Nm/bar for the second scenario

Table 124 - Reverse clutch zero-slip current-to-torque sensitivity expressed in Nm/bar for the second scenario

Min. torque	Max. torque	Nominal torque
43.512	52.971	48.241
43.512	52.971	48.241
43.512	52.971	48.241
43.512	52.971	48.241
43.512	52.971	48.241
43.512	52.971	48.241
43.512	52.971	48.241
43.512	52.971	48.241
43.512	52.971	48.241
43.512	52.971	48.241
43.512	52.971	48.241
43.512	52.971	48.241
43.512	52.971	48.241
43.512	52.971	48.241
43.512	52.971	48.241
43.512	52.971	48.241
43.512	52.971	48.241
43.512	52.971	48.241
43.512	52.971	48.241
43.512	52.971	48.241

6.28 Table 125 - Reverse clutch zero-slip current-to-torque sensitivity expressed in Nm/bar for the third scenario

Table 125 - Reverse clutch zero-slip current-to-torque sensitivity expressed in Nm/bar for the third scenario

Min. torque	Max. torque	Nominal torque
43.223	52.619	47.921
43.223	52.619	47.921
43.223	52.619	47.921
43.223	52.619	47.921
43.223	52.619	47.921
43.223	52.619	47.921
43.223	52.619	47.921
43.223	52.619	47.921
43.223	52.619	47.921
43.223	52.619	47.921
43.223	52.619	47.921
43.223	52.619	47.921
43.223	52.619	47.921
43.223	52.619	47.921
43.223	52.619	47.921
43.223	52.619	47.921
43.223	52.619	47.921
43.223	52.619	47.921
43.223	52.619	47.921
43.223	52.619	47.921
43.223	52.619	47.921

6.29 Auxiliary tables of the design of FWD showcase shifting lever

In Table 126, on a first instance, it is calculated the value of Δp for different values of L_1 . On the second instance, it is calculated the value of Δp for different values of $\Delta\alpha$.

Table 126 – Auxiliary table to the design of FWD showcase shifting lever (lower lever)

Constant $\Delta\alpha$ and variable L_1				Constant L_1 and variable $\Delta\alpha$			
L_1 mm	$\Delta\alpha$ °	β °	Δp mm	$\Delta\alpha$ °	β °	L_1 mm	Δp mm
0			0.000	5	85		6.101
10			6.428	10	80		12.155
20			12.856	15	75		18.117
30			19.284	20	70		23.941
40			25.712	25	65		29.583
50			32.139	30	60		35.000
60			38.567	35	55		40.150
70			44.995	40	50		44.995
80	40	50	51.423	45	45	70	49.497
90			57.851	50	40		53.623
100			64.279	55	35		57.341
110			70.707	60	30		60.622
120			77.135	65	25		63.442
130			83.562	70	20		65.778
140			89.990	75	15		67.615
150			96.418	80	10		68.937
				85	5		69.734

Table 127, is an auxiliary table to calculate, firstly, the value of Δs for different values of L_2 and secondly to calculate the value of Δs for different values of $\Delta\alpha$.

Table 127 - Auxiliary table to the design of FWD showcase shifting lever (upper lever)

Constant $\Delta\alpha$ and variable L_2				Constant L_2 and variable $\Delta\alpha$			
L_2 mm	$\Delta\alpha$ °	β °	Δs mm	$\Delta\alpha$ °	β °	L_2 mm	Δs mm
100			64.279	5	85		6.101
110	40	50	70.707	10	80	70	12.155

120	77.135	15	75	18.117
130	83.562	20	70	23.941
140	89.990	25	65	29.583
150	96.418	30	60	35.000
160	102.846	35	55	40.150
170	109.274	40	50	44.995
180	115.702	45	45	49.497
190	122.130	50	40	53.623
200	128.558	55	35	57.341
210	134.985	60	30	60.622
220	141.413	65	25	63.442
230	147.841	70	20	65.778
240	154.269	75	15	67.615
250	160.697	80	10	68.937
		85	5	69.734

Finally, during the design in PTC Creo® it was necessary to develop Table 128 where the value of Δp and Δs are quickly returned when the value of L_1 , $\Delta\alpha$ and L_2 are introduced.

Table 128 – Resume auxiliary table to the design of FWD showcase shifting lever

L_1	$\Delta\alpha$	β	Δp	PN	L_2	Δs
mm	°	°	mm	mm	mm	mm
35	25	65	14.792	3.27923	300	271.892

6.30 Auxiliary tables of the design of REV showcase shifting lever

In Table 129, on a first instance, it is calculated the value of Δp for different values of L_1 . On the second instance, it is calculated the value of Δp for different values of $\Delta\alpha$.

Table 129 - Auxiliary table to the design of REV showcase shifting lever (lower lever)

Constant $\Delta\alpha$ and variable L_1				Constant L_1 and variable $\Delta\alpha$			
L_1 mm	$\Delta\alpha$ °	β °	Δp mm	$\Delta\alpha$ °	β °	L_1 mm	Δp mm
0			0.000	5	85		6.101
10			6.428	10	80		12.155
20			12.856	15	75		18.117
30			19.284	20	70		23.941
40			25.712	25	65		29.583
50			32.139	30	60		35.000
60			38.567	35	55		40.150
70			44.995	40	50		44.995
80	40	50	51.423	45	45	70	49.497
90			57.851	50	40		53.623
100			64.279	55	35		57.341
110			70.707	60	30		60.622
120			77.135	65	25		63.442
130			83.562	70	20		65.778
140			89.990	75	15		67.615
150			96.418	80	10		68.937
				85	5		69.734

Table 130, is an auxiliary table to calculate, firstly, the value of Δs for different values of L_2 and secondly to calculate the value of Δs for different values of $\Delta\alpha$.

Table 130 - Auxiliary table to the design of REV showcase shifting lever (upper lever)

Constant $\Delta\alpha$ and variable L_2				Constant L_2 and variable $\Delta\alpha$			
L_2 mm	$\Delta\alpha$ °	β °	Δs mm	$\Delta\alpha$ °	β °	L_2 mm	Δs mm
100			64.279	5	85		6.101
110	40	50	70.707	10	80	70	12.155

120	77.135	15	75	18.117
130	83.562	20	70	23.941
140	89.990	25	65	29.583
150	96.418	30	60	35.000
160	102.846	35	55	40.150
170	109.274	40	50	44.995
180	115.702	45	45	49.497
190	122.130	50	40	53.623
200	128.558	55	35	57.341
210	134.985	60	30	60.622
220	141.413	65	25	63.442
230	147.841	70	20	65.778
240	154.269	75	15	67.615
250	160.697	80	10	68.937
		85	5	69.734

Finally, during the design in PTC Creo® it was necessary to develop Table 131 where the value of Δp and Δs are quickly returned when the value of L_1 , $\Delta\alpha$ and L_2 are introduced.

Table 131 - Resume auxiliary table to the design of REV showcase shifting lever

L_1	$\Delta\alpha$	β	Δp	PN	L_2	Δs
mm	°	°	mm	mm	mm	mm
35	20	70	11.971	2.11076	300	281.908

6.31 Table 132 - Necessary force to actuate the cam

Table 132 - Necessary force to actuate the cam

Ω	H	F	Fmax
°	mm	N	N
0	40	82.3	88.44086
3.814075	40.3326	82.89867	
7.594643	40.66082	83.48947	
11.30993	40.98058	84.06505	
14.93142	41.28831	84.61896	
18.43495	41.58114	85.14605	
21.80141	41.85695	85.64252	
25.01689	42.11443	86.10597	
28.07249	42.35294	86.53529	
30.96376	42.57248	86.93046	
33.69007	42.7735	87.2923	
36.25384	42.95682	87.62227	
38.65981	43.12348	87.92226	
40.91438	43.27465	88.19437	
43.02507	43.41159	88.44086	
45	43.53553	88.66396	
46.84761	43.64769	88.86583	
48.57633	43.74919	89.04854	
50.19443	43.84111	89.21399	
51.70984	43.92441	89.36394	
53.1301	44	89.5	
54.46232	44.06867	89.6236	
55.71312	44.13114	89.73605	
56.88866	44.18805	89.8385	
57.99462	44.23999	89.93198	
59.03624	44.28746	90.01744	

6.32 Table 133 – Applied load on the shifting lever

Table 133 – Applied load on the shifting lever

Ω	β	δ	H	F	σ	Rx2	Ry2	Rx3	Ry3	Δp	$\Delta\alpha$	U	Perpen- dicular	Umax	U perp max
$^{\circ}$	$^{\circ}$	$^{\circ}$	mm	N	$^{\circ}$	N	N	N	N	mm	$^{\circ}$	N	N	N	N
0.000	0.000	90.000	40.000	82.300	0.000	82.300	82.300	82.300		0.000	0.000	-9.602	-9.602	9.602	9.602
3.814	3.814	86.186	40.333	82.899	3.814	82.715		82.715		2.000	3.276	-9.650	-9.634		
7.595	7.595	82.405	40.661	83.489	7.595	82.757		82.757		4.000	6.562	-9.655	-9.592		
11.310	11.310	78.690	40.981	84.065	11.310	82.433		82.433		6.000	9.871	-9.617	-9.475		
14.931	14.931	75.069	41.288	84.619	14.931	81.762		81.762		8.000	13.213	-9.539	-9.286		
18.435	18.435	71.565	41.581	85.146	18.435	80.777		80.777		10.000	16.602	-9.424	-9.031		
21.801	21.801	68.199	41.857	85.643	21.801	79.517		79.517		12.000	20.051	-9.277	-8.715		
25.017	25.017	64.983	42.114	86.106	25.017	78.028		78.028		14.000	23.578	-9.103	-8.343		
28.072	28.072	61.928	42.353	86.535	28.072	76.355		76.355		16.000	27.203	-8.908	-7.923		
30.964	30.964	59.036	42.572	86.930	30.964	74.542		74.542		18.000	30.950	-8.697	-7.458		
33.690	33.690	56.310	42.774	87.292	33.690	72.632		72.632		20.000	34.850	-8.474	-6.954		
36.254	36.254	53.746	42.957	87.622	36.254	70.659		70.659		22.000	38.945	-8.244	-6.411		
38.660	38.660	51.340	43.123	87.922	38.660	68.656		68.656		24.000	43.292	-8.010	-5.830		
40.914	40.914	49.086	43.275	88.194	40.914	66.648		66.648		26.000	47.975	-7.776	-5.205		
43.025	43.025	46.975	43.412	88.441	43.025	64.655		64.655		28.000	53.130	-7.543	-4.526		
45.000	45.000	45.000	43.536	88.664	45.000	62.695		62.695		30.000	58.997	-7.314	-3.767		

46.848	46.848	43.152	43.648	88.866	46.848	60.779	60.779	32.000	66.104	-7.091	-2.872
48.576	48.576	41.424	43.749	89.049	48.576	58.916	58.916	34.000	76.271	-6.874	-1.631
50.194	50.194	39.806	43.841	89.214	50.194	57.113	57.113	36.000	#NÚM!	#NÚM!	#NÚM!
51.710	51.710	38.290	43.924	89.364	51.710	55.374	55.374	38.000	#NÚM!	#NÚM!	#NÚM!
53.130	53.130	36.870	44.000	89.500	53.130	53.700	53.700	40.000	#NÚM!	#NÚM!	#NÚM!
54.462	54.462	35.538	44.069	89.624	54.462	52.093	52.093	42.000	#NÚM!	#NÚM!	#NÚM!
55.713	55.713	34.287	44.131	89.736	55.713	50.552	50.552	44.000	#NÚM!	#NÚM!	#NÚM!
56.889	56.889	33.111	44.188	89.838	56.889	49.076	49.076	46.000	#NÚM!	#NÚM!	#NÚM!
57.995	57.995	32.005	44.240	89.932	57.995	47.664	47.664	48.000	#NÚM!	#NÚM!	#NÚM!
59.036	59.036	30.964	44.287	90.017	59.036	46.314	46.314	50.000	#NÚM!	#NÚM!	#NÚM!

6.33 Table 134 – Auxiliary table for the calculation of the trendline of the kinematic viscosity vs temperature

Table 134 – Auxiliary table for the calculation of the trendline of the kinematic viscosity vs temperature

Temperature	Kinematic viscosity	Density	viscosidade calculada	Erro
[°C]	[cst] = [mm ² /s]	[kg/m ³]		
-40	11200	884	5122,037	0,339775269
-30	2770	878	1752,318	0,198866864
-20	930	872	710,383	0,11699013
-10	386	866	330,381	0,067572664
0	188	860	171,842	0,039027639
10	103	854	97,967	0,021755482
20	61,7	848	60,238	0,010411432
30	39,8	842	39,435	0,003998989
40	27,3	836	27,200	0,001590407
50	19,6	830	19,600	5,96335E-08
60	14,6	824	14,653	0,001585347
70	11,3	817	11,302	9,19165E-05
80	8,94	811	8,953	0,000607344
90	7,26	805	7,255	0,000327556
100	6,02	799	5,995	0,001799677
110	5,08	792	5,039	0,003489961
120	4,35	786	4,299	0,005095692
130	3,78	780	3,716	0,007428828
140	3,32	773	3,249	0,009424444
150	2,95	767	2,869	0,012042384

6.34 Calculations data

In this section will be presented all the known input data of all the components which will have direct or indirect influence in the calculations.

The presentation of the input data of the components will be organized according to the type of component.

This makes possible to see the main differences between FWD and REV components not only in terms of dimensions but also in terms of materials and so on.

6.34.1 Transmission oil

The values presented on the table below were taken from the datasheet of the transmission oil in study.

Table 135 - Transmission oil properties (Courtesy of Punch Powertrain NV)

T [°C]	λ [cst]	ρ [kg/m ³]
-40	11200	884
-30	2770	878
-20	930	872
-10	386	866
0	188	860
10	103	854
20	61.7	848
30	39.8	842
40	27.3	836
50	19.6	830
60	14.6	824
70	11.3	817
80	8.94	811
90	7.26	805
100	6.02	799
110	5.08	792
120	4.35	786
130	3.78	780
140	3.32	773
150	2.95	767

6.34.2 Springs

6.34.2.1 FWD springs

All the data available on Table 136 was presented by the supplier.

Later these values will be verified.

Table 136 – FWD springs data

Designation	Value	Unit
Load at installed height	37.54	N
Installed height	15.63	mm
Total nominal coils	7.90	---
Active coils (nominal)	5.90	---
Spring OD	8.95 ± 0.13	mm
Wire diameter	1.232 ± 0.025	mm
Spring rate	8.42	N/mm
Ends closed not ground		
Material	ASTM A401	---

6.34.2.2 FWD spring pack plates

Table 137 – FWD spring pack plates data

Designation	Value	Unit
Thickness	0.75 ± 0.08	mm
Material	SAE 1008/1010	---

6.34.2.3 FWD spring pack

Table 138 – FWD spring pack data

Designation	Abbreviation	Unit	Value
Free length	FL	mm	21.59

Max. load at max. working height	$F_{\max (h_{\max})}$	N	1049.6
Min. load at max. working height	$F_{\min (h_{\max})}$	N	808.8
Max. working height	$h_{\max}$	mm	17.13
Max. load at min. working height	$F_{\max (h_{\min})}$	N	1815.5
Min. load at min. working height	$F_{\min (h_{\min})}$	N	1504.6
Min. working height	$h_{\min}$	mm	13.79
Nr. of springs	N	---	26
Coils material	---	---	ASTM A401
Wire diameter	d_c	mm	1.232
Number of active coils	N_A	---	5.9
Spring outer diameter	OD_C	mm	8.95
Stiffness	SR	N/mm	8.42

6.34.2.4 Wavy washer

Table 139 - Wavy washer data

Designation	Abbreviation	Unit	Value
Free length	FL	mm	2.35
Free length tolerance	FL_{tol}	$\pm$ mm	0.1
Load at $h=1.75$ mm	$F_{(h_{\max})}$	N	600
Load at $h=1.75$ mm tolerance	$F_{(h_{\max}) tol}$	$\pm$ N	90
Thickness	b	mm	1.55
Thickness tolerance	b_{tol}	$\pm$ mm	0.05
Outer diameter	OD	mm	135.4
Inner diameter	ID	mm	126.6
Nr. Of waves per turn	N_W	---	6
Material	---	---	51CrV4 acc. To DIN 10083

6.34.2.5 REV springs

6.34.2.6 REV spring pack plate

Table 140 – REV spring pack plate data

Designation	Value	Unit
Thickness	1.50 ± 0.10	mm
Material	SAE 1008/1010	---

6.34.2.7 REV spring pack

Table 141 – REV spring pack data

Designation	Abbreviation	Unit	Value
Free length	FL	mm	15.96
Max. load at max. working height	$F_{\max} (h_{\max})$	N	588.6
Min. load at max. working height	$F_{\min} (h_{\max})$	N	467.9
Max. working height	$h_{\max}$	mm	11.2
Max. load at min. working height	$F_{\max} (h_{\min})$	N	934.4
Min. load at min. working height	$F_{\min} (h_{\min})$	N	774.6
Min. working height	$h_{\min}$	mm	8.4
Nr. of springs	N	---	32
Coils material	---	---	ASTM A401
Wire diameter	d_c	mm	0.889
Number of active coils	N_A	---	3.5
Spring outer diameter	OD_c	mm	8.75
Stiffness	SR	N/mm	3.64

6.34.2.8 Belleville washer

Table 142 - Disc spring data

Designation	Abbreviation	Unit	Value
Free length	FL	mm	2.9
Max. working height	$h_{\max}$	mm	2.4
Load at working height	$F (h=2.4 \text{ mm})$	N	716
Thickness	b	mm	2.15
Outer diameter	OD	mm	186
Inner diameter	ID	mm	165
Material	---	---	51CrV4 acc. To DIN 10083

6.34.3 Pistons

6.34.3.1 FWD piston

Table 143 - FWD piston data

Designation	Abbreviation	Unit	Value
Outer hydraulic diameter	OD _h	mm	118
Outer hydraulic diameter tolerance	OD _{h tol}	± mm	0.05
Outer hydraulic radius	OR _h	mm	59
Inner hydraulic diameter	ID _h	mm	64.5
Inner hydraulic diameter tolerance	ID _{h tol}	± mm	0.05
Inner hydraulic radius	IR _h	mm	32.25
Fluid entry diameter	D _p	mm	58.2
Fluid entry diameter tolerance	D _{p tol}	+ mm	0.1
Fluid entry radius	r _p	mm	29.1
Material	---	---	HC 380
			acc. To
			EN 10
			268

6.34.3.2 Balance dam

Table 144 - Balance dam data

Designation	Abbreviation	Unit	Values
Outer hydraulic diameter	OD _h	mm	113.12
Outer hydraulic diameter tolerance	OD _{h tol}	± mm	0.1
Outer hydraulic radius	OR _h	mm	56.56
Inner hydraulic diameter	ID _h	mm	63.5
Inner hydraulic diameter tolerance	ID _{h tol}	± mm	0.1
Inner hydraulic radius	IR _h	mm	31.75
Fluid entry diameter	D _p	mm	58
Fluid entry diameter tolerance	D _{p tol}	+ mm	0.7
Fluid entry radius	r _p	mm	
Material	---	---	DC01
			acc. To

DIN
10130

6.34.3.3 REV piston

Table 145 – REV piston data

Designation	Abbreviation	Unit	Value
Outer hydraulic diameter	OD _h	mm	183
Outer hydraulic diameter tolerance	OD _{h tol}	± mm	0.07
Outer hydraulic radius	OR _h	mm	91.5
Inner hydraulic diameter	ID _h	mm	141
Inner hydraulic diameter tolerance	ID _{h tol}	± mm	0.07
Inner hydraulic radius	IR _h	mm	70.5
Material	---	---	AlSi9Cu3

6.34.4 Clutch plates

6.34.4.1 FWD clutch plates

Table 146 - FWD clutch plates data

Designation	Abbreviation	Unit	Value
Friction area per pad	A _F ^{pad}	mm ²	56.800
Number of pads	N _P	---	32
Outer friction surface diameter	OD _F	mm	131.350
Outer friction surface diameter tolerance	OD _{F tol}	± mm	0.150
Outer friction surface radius	OR _F	mm	65.675
Inner friction surface diameter	ID _F	mm	116.100
Inner friction surface diameter tolerance	ID _{F tol}	± mm	0.500
Inner friction surface radius	IR _F	mm	58.050
Min. friction coefficient	ζ _{min}	---	0.115
Max. friction coefficient	ζ _{max}	---	0.140

6.34.4.2 REV clutch plates

Table 147 - REV clutch plates data

Designation	Abbreviation	Unit	Value
Friction area per pad	A_F^{pad}	mm ²	63.318
Number of pads	N_P	---	40
Outer friction surface diameter	OD_F	mm	184.300
Outer friction surface diameter tolerance	$OD_{F \text{ tol}}$	± mm	0.15
Outer friction surface radius	OR_F	mm	92.150
Inner friction surface diameter	ID_F	mm	168.300
Inner friction surface diameter tolerance	$ID_{F \text{ tol}}$	± mm	0.5
Inner friction surface radius	IR_F	mm	84.150
Min. friction coefficient	$\zeta_{\min}$	---	0.3
Max. friction coefficient	$\zeta_{\max}$	---	0.3

6.34.5 Clutch sets

6.34.5.1 FWD clutch set

Table 148 - FWD clutch set data

Designation	Abbreviation	Unit	Value
Number of friction surfaces	N_S	---	6
Clearance between disc plates	h_P	mm	0.2

6.34.5.2 REV clutch set

Table 149 – REV clutch set data

Designation	Abbreviation	Unit	Value
Number of friction surfaces	N_S	---	4
Clearance between disc plates	h_P	mm	0.2

6.34.6 Gears

6.34.6.1 Ring gear

Table 150 – Ring gear data

Designation	Abbreviation	Unit	Value
Pitch inner circle diameter	ID	mm	122.516
Nr. Of teeth	Z	---	113
Material	---	---	QT900-2 (GB1348) / EN-GJS- 900-2 (EN1563)

6.34.6.2 Sun gear

Table 151 – Sun gear data

Designation	Abbreviation	Unit	Value
Pitch circle diameter	OD	mm	83.887
Nr. Of teeth	Z	---	79
Material	---	---	SCM420H acc. To JIS G4052

6.34.6.3 Planetary gear

Table 152 – Planetary gear data

Designation	Abbreviation	Unit	Value
Pitch circle diameter	OD	mm	19.113
Nr. Of teeth	Z	---	18
Material	---	---	SCM420H acc. To JIS G4052

6.34.7 External components

6.34.7.1 Transmission housing

Table 153 – Transmission housing for REV clutch set housing data

Designation	Abbreviation	Unit	Value
Outer diameter	OD	mm	183±0.07
Inner diameter	ID	mm	141±0.07

6.34.8 Materials

In this chapter are presented the mechanical properties that will be used for the analytical calculations and FEA.

It is important to underline the fact that these values are not exactly the same presented on the datasheets of each material. This means that when a range of values was presented or when there were maximum and minimum values, it was made a choice of values within the limits. Most of the times were considered the less favourable values.

All the material datasheets or extracts from catalogues are presented in the APPENDIX where more detailed information can be found.

On this chapter will be only presented the properties which are necessary for the calculations and simulations. In the case of the value of Young's Modulus and Poisson's ratio not being mentioned in the technical datasheet it will be considered the "standard" values of 210 GPa and 0.3 for steel and 70 GPa and 0.33 for aluminium alloys.

6.34.8.1 20CrMoH (acc. to GB/T5216)

Due to the lack of information regarding to the mechanical properties of the Chinese grade of the material it was necessary to find the equivalent grades as it can be seen on Table 154.

Table 154 – 20CrMoH equivalent grades (adapted from (National Standard of the People's Republic of China, 2004) and (Kyocera, 2017))

China (GB/T5216)	Japan (JIS G4052-1979)	Germany (DIN)
20CrMoH	SCM420H	18CrMo4

The mechanical properties of the material are presented on Table 155.

Table 155 – 18CrMo4 (DIN) mechanical properties (adapted from (Datenblatt Saerstahl, n.d.))

Property	Value	Unit
E	210	GPa
ν	0.3	---
$Rp_{0.2\%}$	685	MPa
R_m	930	MPa
A_5	9	%

6.34.8.2 SAPH 310 (acc. to JIS G 3113)

The mechanical properties of the material SAPH310 depend on the thickness of the metal sheet used to manufacture the metallic part. For this material it was considered a thickness of 4.2 mm that is the maximum thickness of the part which is made of this material. There is only one part that is made from this material (Clutch Carrier).

Table 156 – SAPH310 (JIS G 3113) mechanical properties (adapted from (Nippon Steel & Sumitomo Metal, 2012))

Property	Value	Unit
E	210	GPa
ν	0.3	---
R_p	185	MPa
R_m	310	MPa
A	38	%

6.34.8.3 HC 380 (acc. to EN 10 268)

Table 157 – HC 380 (EN 10 268) mechanical properties (adapted from (Baosteel, n.d.))

Property	Value	Unit
E	210	GPa
ν	0.3	---
$Rp_{0.2\%}$	380	MPa
R_m	590	MPa
A	26	%

6.34.8.4 DC 01 (acc. to EN 10130)

Table 158 – DC 01 (EN 10130) mechanical properties (adapted from (Terms, n.d.))

Property	Value	Unit
E	210	GPa
ν	0.3	---
$Rp_{0.2\%}$	140	MPa
R_m	270	MPa
A_{80}	28	%

6.34.8.5 SCM 420H (JIS)

SCM 420H is an equivalent grade of the material 18CrMo4 (DIN) (Table 154) which means that the mechanical properties considered for this material are presented on Table 155.

6.34.8.6 EN-GJS-900-2 (acc. to EN 1563:2012)

Table 159 – EN-GJS-900-2 (EN 1563:2012) mechanical properties (adapted from (Founderies de Brousseval et Montreuil, n.d.))

Property	Value	Unit
E	210	GPa
ν	0.3	---
$Rp_{0.2\%}$	600	MPa
R_m	900	MPa
A	2	%

6.34.8.7 S45C (acc. to JIS G 4051)

The mechanical properties of this material depend on the raw material form. There are only two parts that are made from this material which are the FWD and REV separator plates. As previously explained these components are manufactured by cold forming (metal sheet) so it is expected that the raw material is hot rolled.

In conclusion, for this material were considered the mechanical properties for the hot rolled condition whose values are not as well defined as for normalised condition.

Table 160 – S45C (JIS G 4051) mechanical properties (adapted from (Rods *et al.*, 1991))

Property	Value	Unit
E	210	GPa
ν	0.3	---
R _p	300	MPa
R _m	570	MPa
A	14	%

6.34.8.8 51CrV4 (*acc. to DIN EN 10083*)

The mechanical properties of the material 51CrV4 depend on the thickness of the sheet metal. Both wavy washer and disc spring are made of this material and the maximum thickness is 2.15 mm. So it was considered the lowest thickness range (<8 mm).

Table 161 – 51CrV4 (DIN EN 10083) mechanical properties (adapted from (Saarstahl, 2011a))

Property	Value	Unit
E	210	GPa
ν	0.3	---
R _{p0.2%}	900	MPa
R _m	1100	MPa
A ₅	9	%

6.34.8.9 SAE 1060-10 (*acc. to DIN EN 10083*)

Once again, the mechanical properties of the material SAE 1060-10 depend on the thickness of the raw material (sheet metal). There are three parts that are made of this material. All of them are snap rings. None of them has a thickness greater or equal to 8 mm. So, the mechanical properties of the material are based on the first category (thickness < 8 mm).

Table 162 – SAE 1060-10 (DIN EN 10083) (adapted from (Saarstahl, 2011b))

Property	Value	Unit
E	210	GPa
ν	0.3	---
$Rp_{0.2\%}$	580	MPa
R_m	850	MPa
A_5	11	%

6.34.8.10 100Cr6 (acc. to DIN standard)

Table 163 – 100Cr6 (DIN) mechanical properties (adapted from (Sheet, 2006))

Property	Value	Unit
ρ	7.5	g/cm^3
E	210	GPa
ν	0.3	---
$Rp_{0.2\%}$	500	MPa
R_m	900	MPa
A	5	%

6.34.8.11 SAPH 440 (acc. to JIS G 3113-1990)

The mechanical properties of this material depend on the thickness of the metal sheet which is used to manufacture the parts. The parts that are made of this material are both FWD and REV endplates and the maximum thickness is 5 mm.

Table 164 – SAPH 440 (JIS G 3113-1990) mechanical properties (adapted from (Corporation, n.d.))

Property	Value	Unit
E		GPa
ν		---
R_p	305	MPa
R_m	440	MPa
A	35	%

6.34.8.12 *AlSi9Cu3 (acc. to EN 1706)*

Table 165 – AlSi9Cu3 (EN 1706) mechanical properties (adapted from (Stena Aluminium, 2010))

Property	Value	Unit
ρ	2.75	kg/dm ³
E	70	GPa
ν	0.33	---
R _{p0.2%}	140	MPa
R _m	240	MPa
A ₅₀	1	%

6.34.8.13 *S355MC (acc. to EN 10149)*

The thickness of metal sheet has influence in mechanical properties of the material. Planet carrier welded assy. Is made of this material and the maximum thickness is 3.5 mm. This value allows choosing the category in terms of thickness of the material (between 3 and 5 mm). In this case thickness has only influence in elongation.

Table 166 – S355MC (EN 10149) mechanical properties (adapted from (Tata Steel, 2010))

Property	Value	Unit
E	210	GPa
ν	0.3	---
R _p	355	MPa
R _m	430	MPa
A	23	%

6.34.8.14 *SAE 1008 (ASTM A568M)*

It was necessary to find the equivalent grades of this material due to the lack of trustworthy information regarding to the ASTM grade. The equivalent grades of the material SAE 1008 can be consulted on Table 167.

Table 167 – Equivalent grades of SAE 1008 (ASTM A568M) (adapted from (Products, n.d.))

ASTM A568M	EN 10130	EN 10130:1994	JIS G 3141
SAE 1008	DC 03	FeP03	SPCD

It was chosen the equivalent grade DC 03 (EN 10130) due to the amount of available information. Thickness has an influence in the elongation and Yield strength of the material. The plates of the FWD and REV spring pack are made of this material. FWD and REV plates' thickness are 0.75 and 1.50 mm, respectively. Both values are within the same range (between 0.7 and 2 mm). The mechanical properties of this material are available on Table 168.

Table 168 – DC 03 (EN 10130) mechanical properties (adapted from (Products, n.d.))

Property	Value	Unit
E	210	GPa
ν	0.3	---
R _p	240	MPa
R _m	270	MPa
A ₈₀	34	%

6.34.8.15 SAE 1010 (ASTM A568M)

Resembling the previous material, for the same reason, it was necessary to obtain the equivalent grade of this material. The corresponding grades of the material are presented on Table 169

Table 169 – SAE 1010 (ASTM A568M) equivalent grades (adapted from (Products, n.d.))

ASTM A568M	EN 10130	EN 10130:1994	JIS G 3141
SAE 1010	DC01	FeP01	SPCC

It was selected the grade DC01 (EN 10130). Mechanical properties of the material can be met on Table 158.

6.34.8.16 *ASTM A401*

Table 170 – ASTM A401 mechanical properties (adapted from (Dendoff Springs, n.d.))

Property	Value	Unit
E	207	GPa
ν	0.3	---
Design stress	760	MPa
R _m	1690	MPa

6.35 Preliminary assembly feasibility study

The aim of this study is to verify the feasibility of the assembly process of the showcase model. This is an important study since the manufacturing and assembly of the model will be made in an external supplier.

The assembly process must be the following:

1. To assemble the spring pins (6x) in the respective holes of the main plate. The aim of the spring pins is to ensure the correct position and alignment of the next components to be assembled.

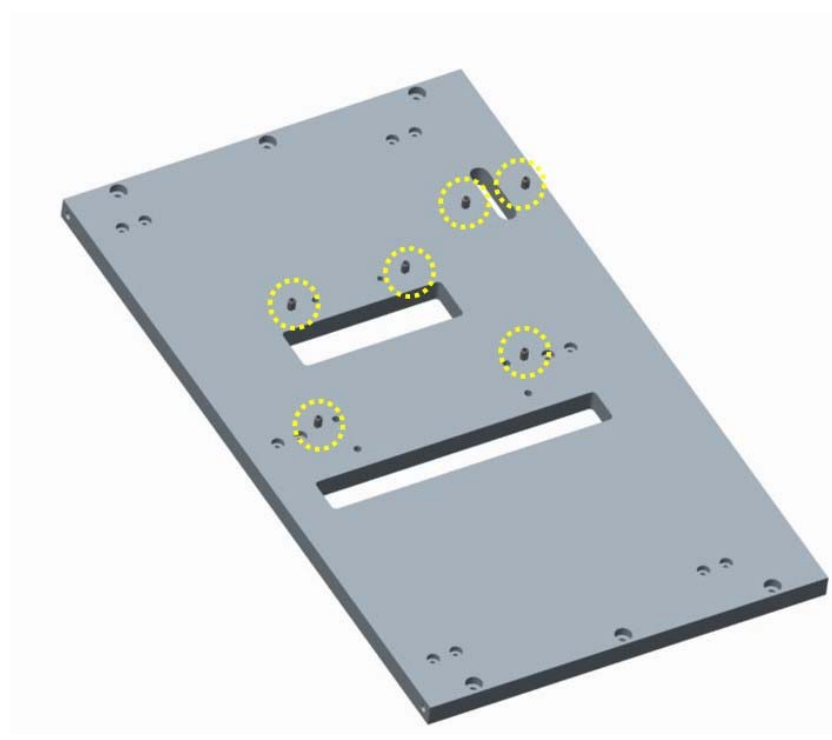


Figure 270 – Showcase assembly process: step 1

2. To assemble the guiding blocks and respective plates of FWD and REV guiding systems. Each block must be screwed with 4x M5.

3. To assemble FWD guiding system, shaft and shaft supports together. Two shaft supports must be left behind in order to allow the assemble of the next guiding system. To move for and backwards the guiding system and verify if it runs smoothly. When this condition is fulfilled, the shaft supports must be screwed to the main plate (2x M4 each unit);
4. To assemble the remaining guiding system and the two missing shaft supports. To verify if the

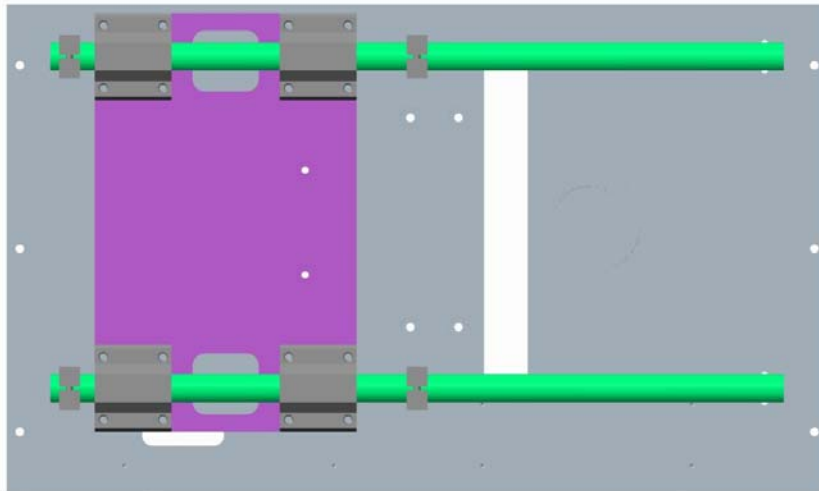


Figure 271 – Showcase assembly process: step 3

guiding system runs smoothly and then screw the shaft supports to the main plate. At this point it is supposed that the entire guidance system is perfectly aligned and running smoothly and all the shaft supports are correctly fastened to the main plate.

5. To remove the following components:
 - a. FWD system plate;
 - b. REV system plate;
 - c. 8x guiding blocks;
 - d. 2x shafts

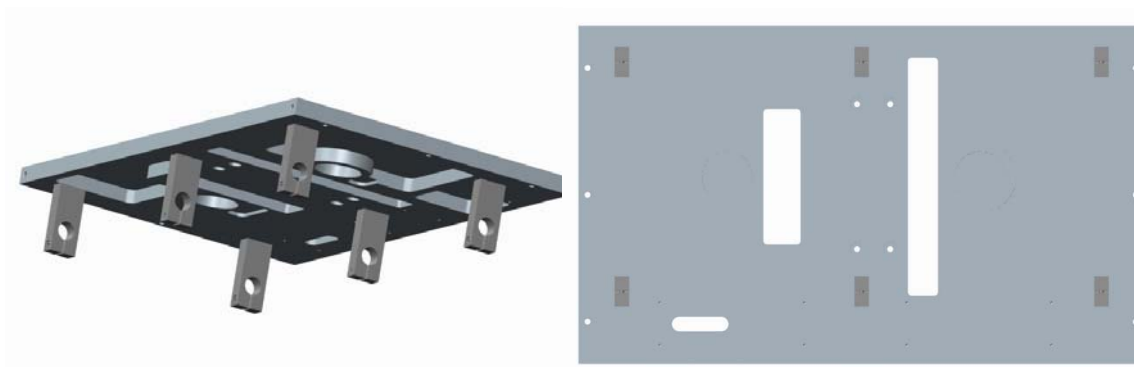


Figure 272 – Showcase assembly process: Step 5

6. To insert the spring pins in one side of the DNR housing, align both parts and screw them together with 4x M6;

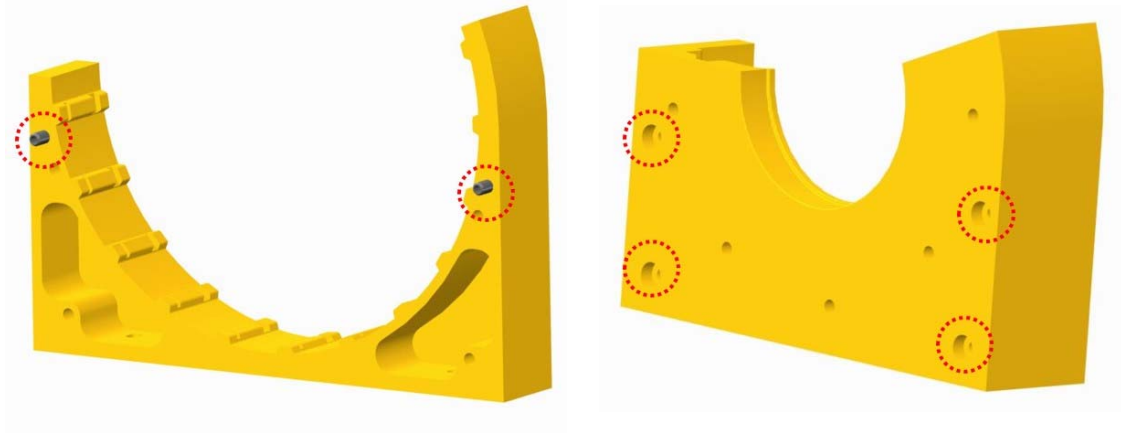


Figure 273 – Showcase assembly process: Step 6

7. To assemble the screwed pins in the REV pressing plate. By using the nuts, adjust the pins to ensure their alignment.
8. To assemble the stopper and the shims in the respective concavity. The main purpose of the

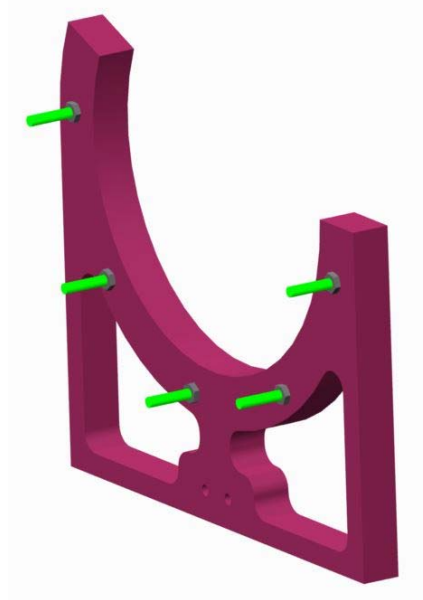


Figure 274 - Showcase assembly process:
Step 7

shims is to provide adjustment to the stroke of the pressing plate. The stopper will be in contact with the cam. It is supposed to use only 4x shims but it will depend on the dimensional accuracy of the parts. After assembling the shims and the stopper, fasten them with 2x M5.

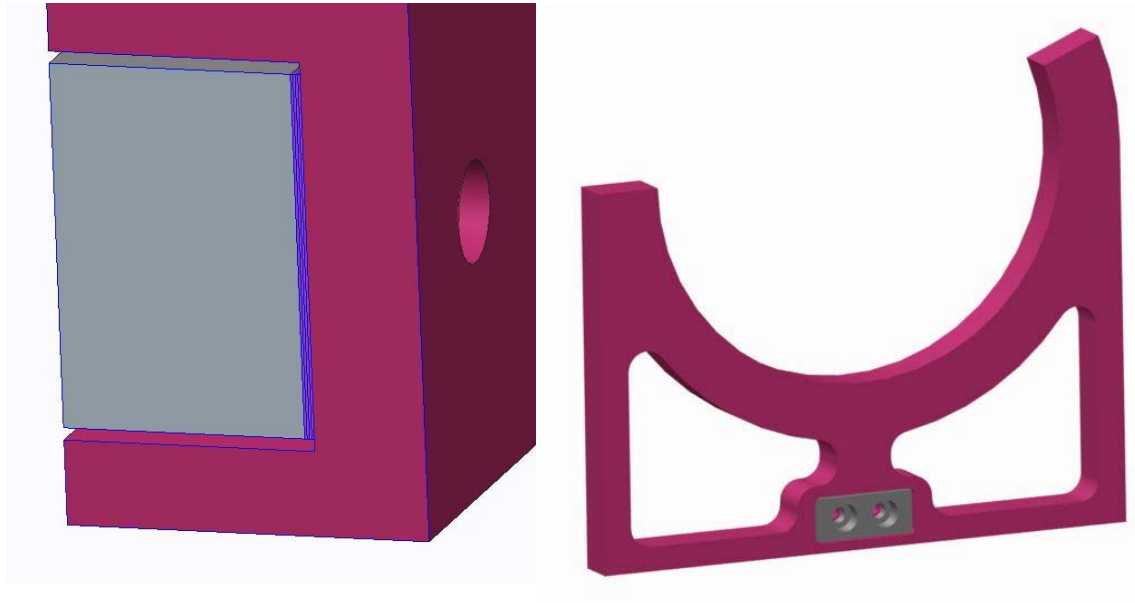


Figure 275 - Showcase assembly process: Step 8

9. To introduce the pins of the reverse pressing plate in the DNR housing in order to hold it and clamp them together. If necessary insert some spacers between the DNR housing and the pressing plate. In this operation, it is necessary to pay attention to avoid scratches on the parts since they are black anodized/oxidized.

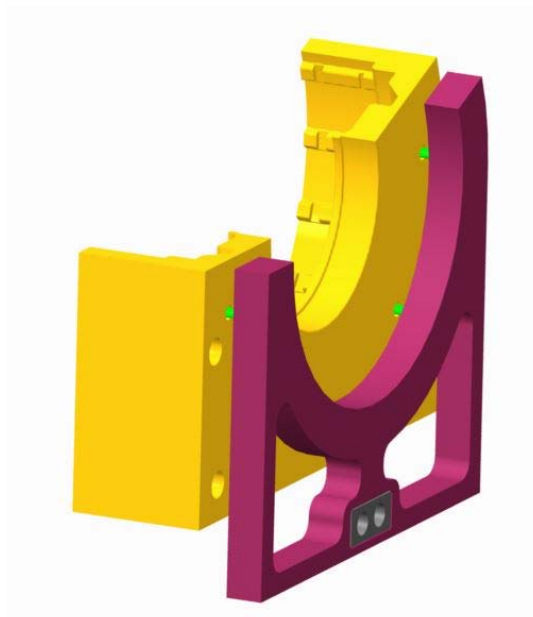


Figure 276 - Showcase assembly process: Step 9

10. To assemble both parts at the same time on the main plate. Use the spring pins (that were inserted before on the main plate) to guarantee the alignment of the DNR housing.

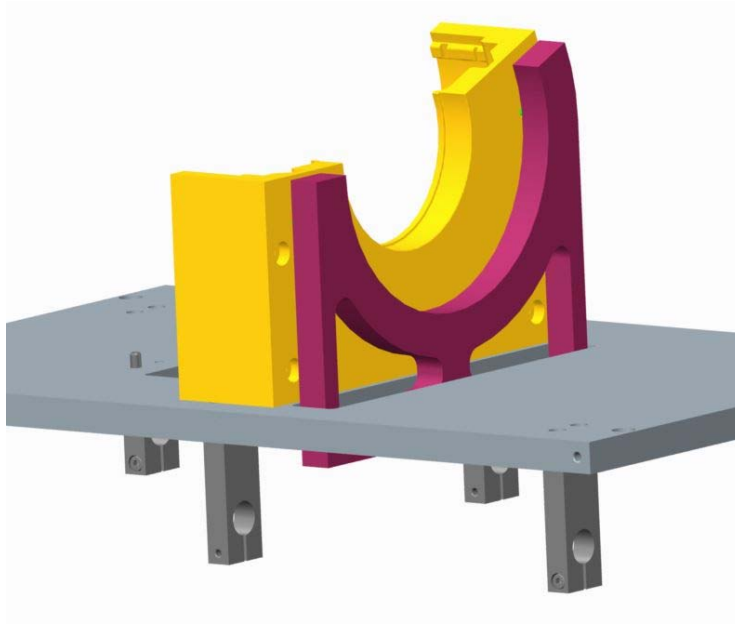


Figure 277 - Showcase assembly process: Step 10

11. To glue the 6 ball transfer units supports (contact face) to the FWD piston.

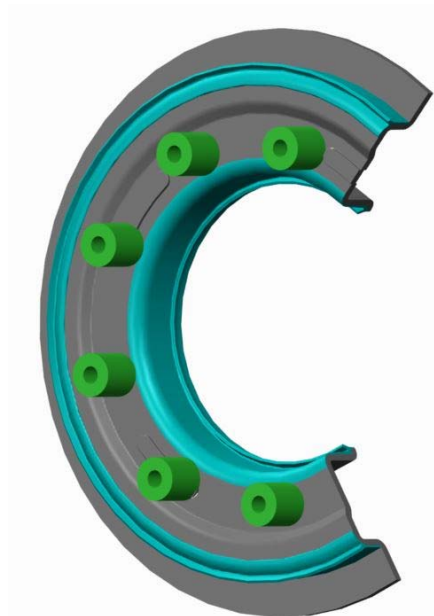


Figure 278 - Showcase assembly process:
Step 11

12. To assemble the FWD piston inside the clutch carrier by aligning the ball transfer unit supports with the respective holes and the cutting sections of the CC and FWD piston.

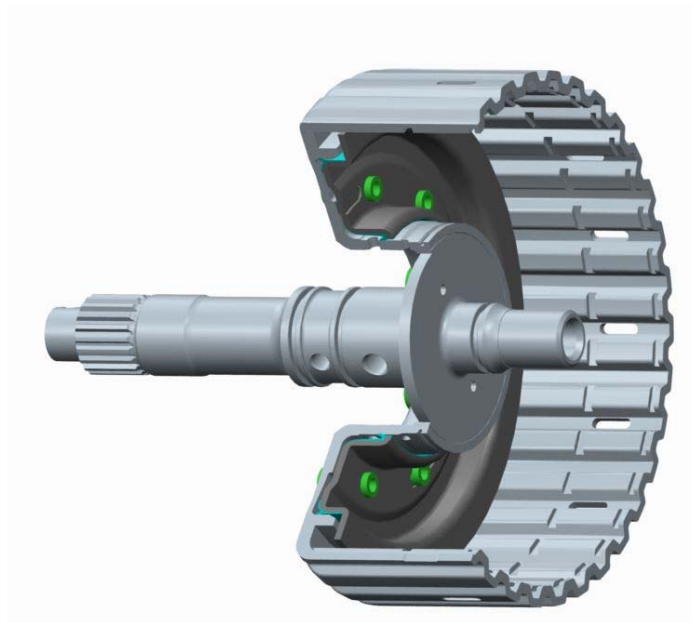
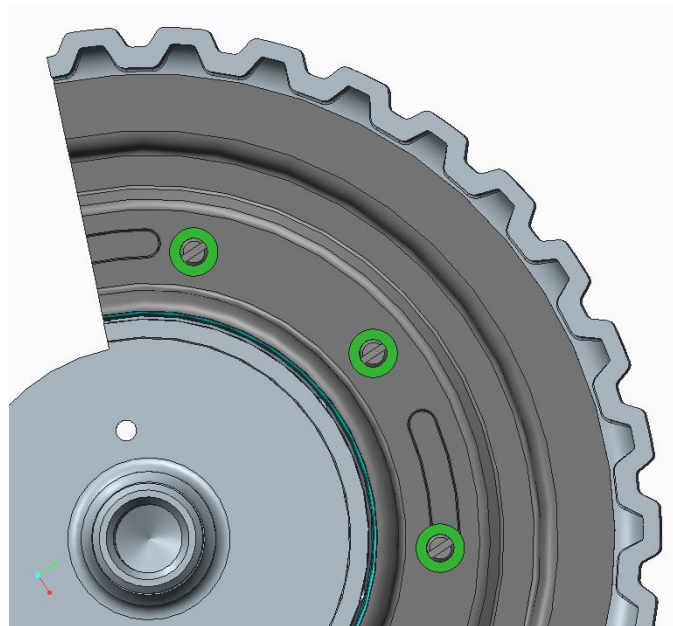


Figure 279 - Showcase assembly process: Step 12

13. To fasten the ball transfer units to their supports and verify the correct alignment.



14. To insert the plate of the spring pack and deform plastically the ball transfer unit adapter (swaging process) in order to guarantee a fixed position of the spring pack plate. In the image the adapter is not deformed.

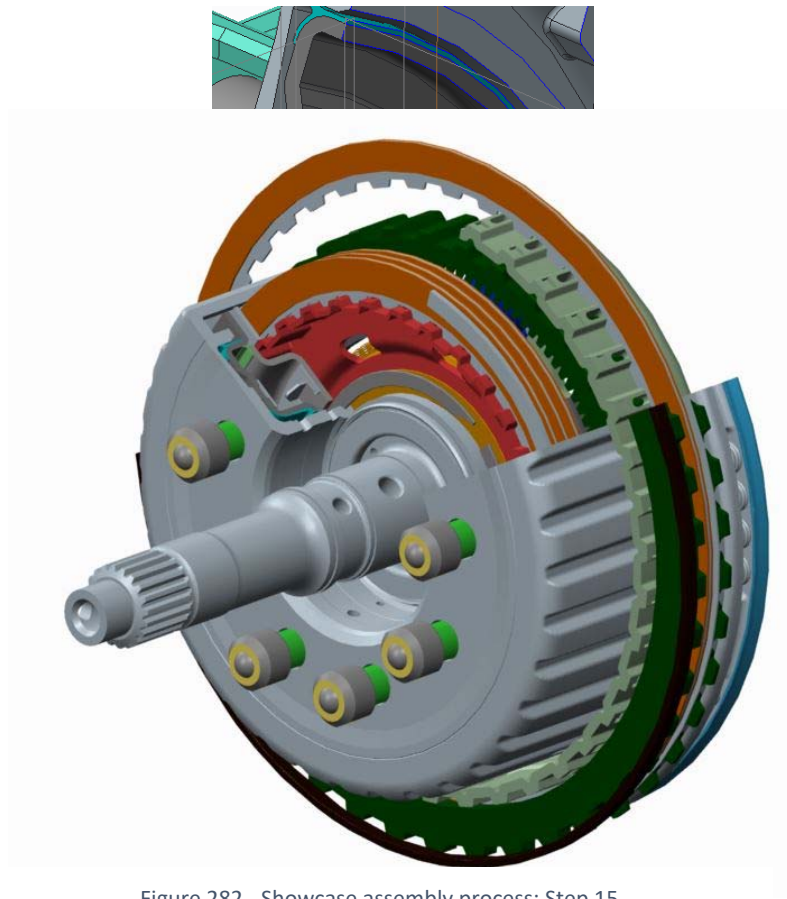


Figure 282 - Showcase assembly process: Step 15

15. To assemble the complete DNR and align the cutting section of the components.

16. To insert the DNR subsystem inside the DNR housing.

17. To glue the ring to the FWD pressing plate.

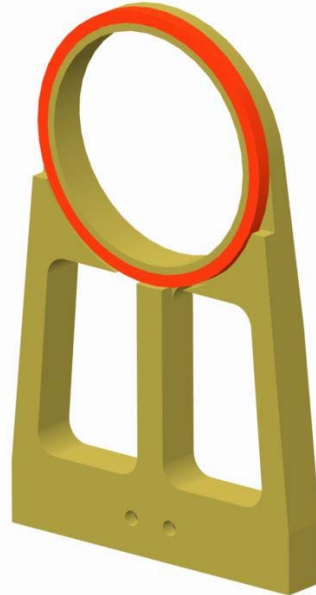
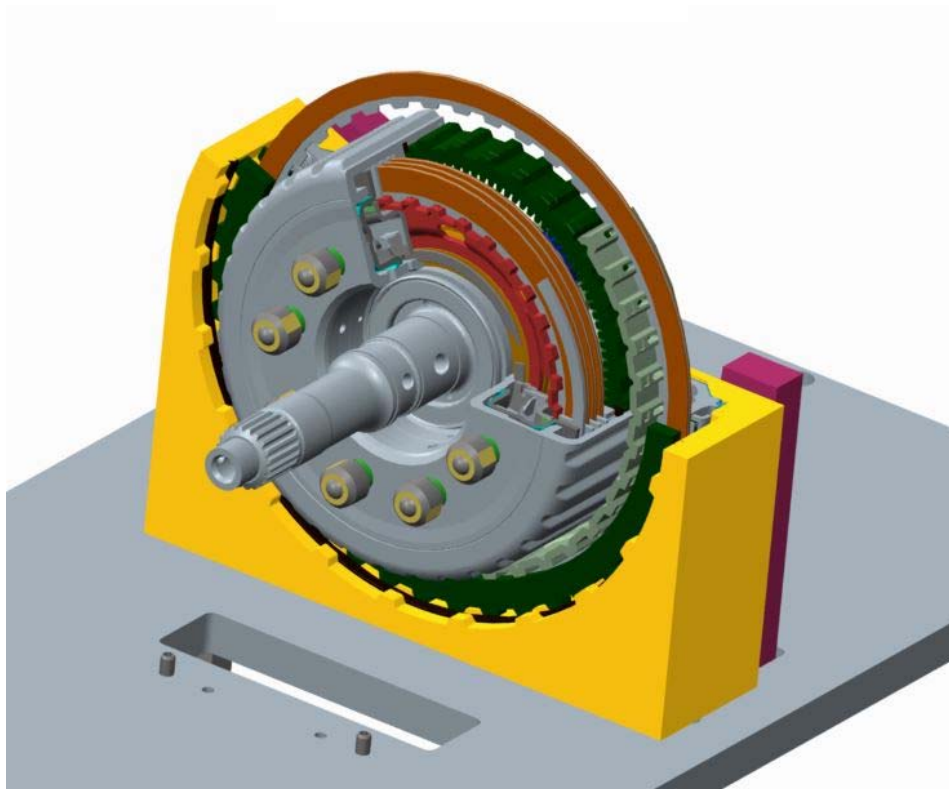


Figure 284 - Showcase assembly process:
Step 17



18. To assemble the shims and the stopper in the cavity. The aim of the shims was already explained before. After assembling the shims and the stopper fasten them with 2x M5.

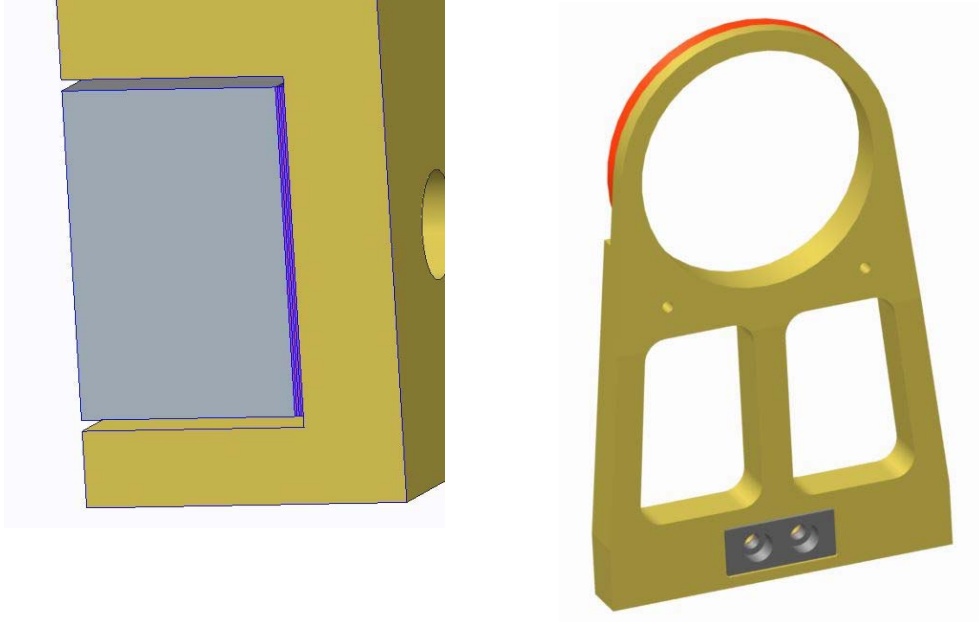


Figure 285 - Showcase assembly process: Step 18

19. To screw bushing support and FWD pressing plate together with 2x M4. These screws are auxiliary. By this way we can prevent the pressing plate to be free.

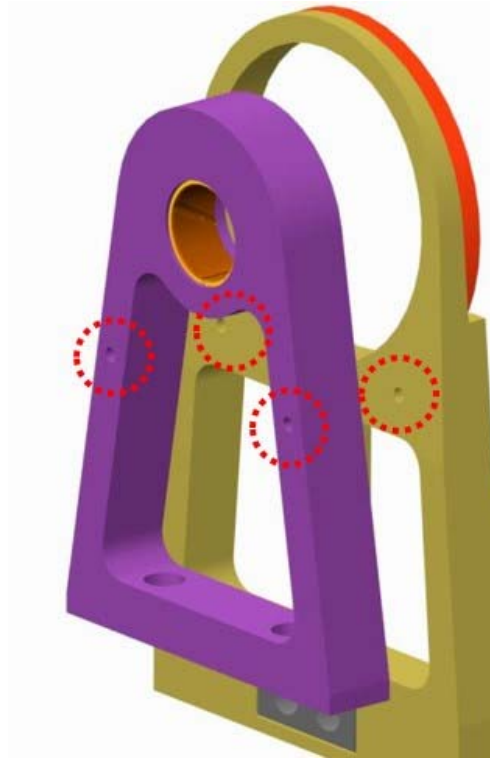
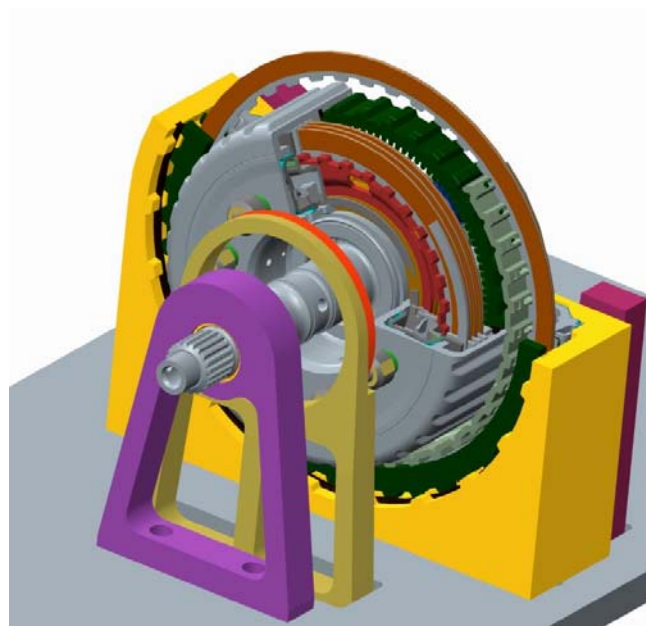


Figure 286 - Showcase assembly process: Step 19

20. The bushing support must be perfectly aligned with the auxiliary spring pins (of the main plate) and then screwed to the main plate. To check if everything is aligned and if the DNR rotates smoothly.



21. To screw the supports of the sheet metal (which transmits the movement from the lever to the cams) to the main plate by means of 2x M3 (each support). To verify if the sheet metal moves with low friction to ensure there will not be extra loads.
22. To insert the cam into the bearing by means of a press.

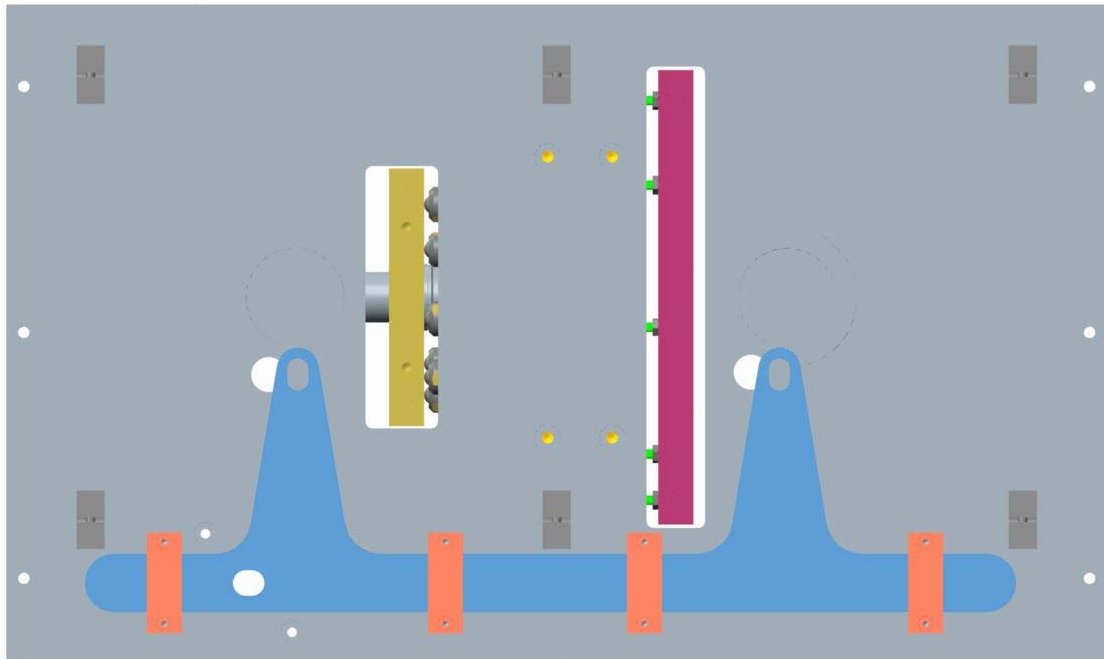
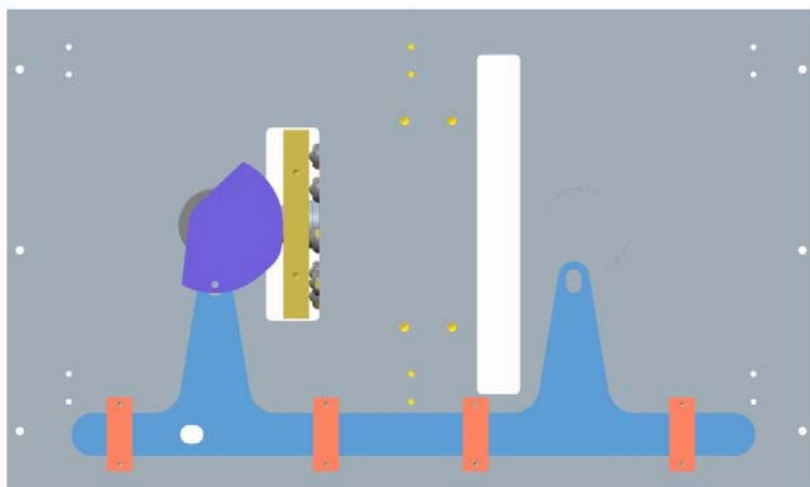


Figure 288 - Showcase assembly process: Step 21

23. After that, to glue the outer shell of the bearing to the cavity of the main plate. Guarantee that the bearing rotates freely and smoothly. At this point the cam and the sheet metal are not



connected, yet. They will be screwed together forward. It is advisable to add some lubricant to the contact surfaces of the cam and stopper.

24. To screw both shafts to originate a lever.



Figure 290 - Showcase assembly process: Step 24

25. To insert the shoulder screw MSB6-40 in the block and verify if the lever moves smoothly.
26. To screw the spring plungers inside the square nuts. Adjust the position of the 4 square nuts in

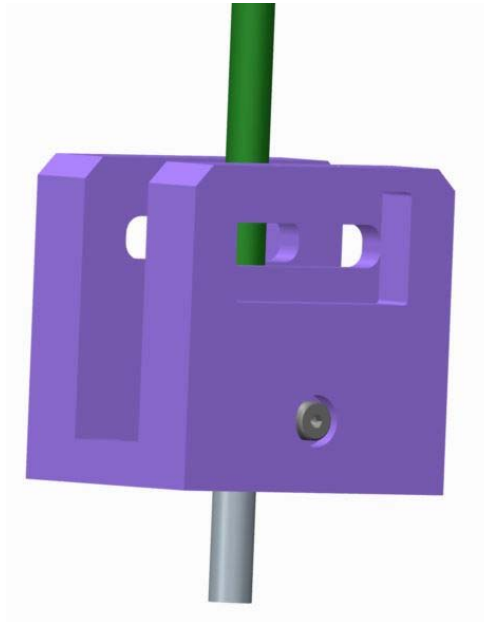


Figure 291 - Showcase assembly process: Step 25

order to ensure that the user feels a “click” when it is about to leave a position (D, N or R) or about to reach one. The square nuts will be adjusted again before the assembly of the cover.

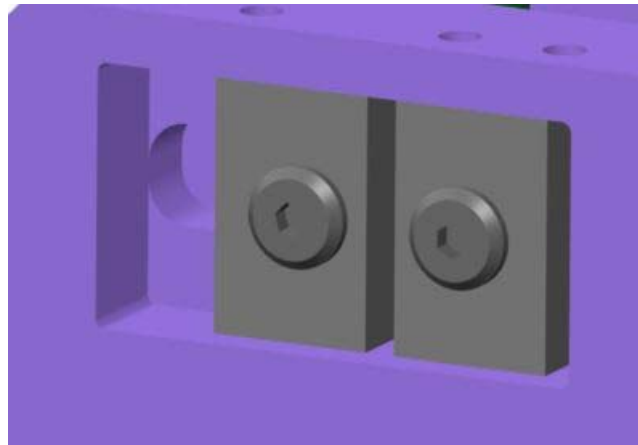


Figure 293 - Showcase assembly process: Step 26

27. To fasten the square nuts by means of the set screws after adjusting the position;

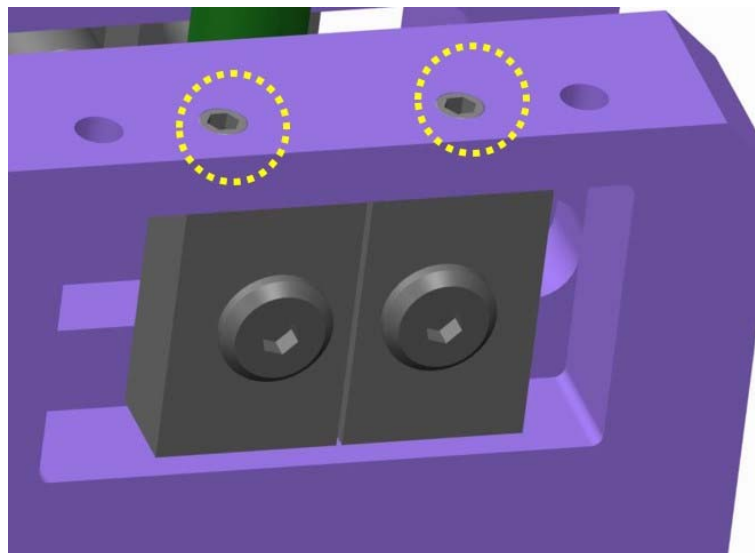


Figure 292 - Showcase assembly process: Step 27

28. It is necessary to assemble the levers system at this point to help in the fastening of the cam to the guiding sheet metal. To easily assemble the cams, it is necessary to guarantee that the cam is not being actuated by the pressing plate. By moving the lever to the REV position, we ensure that the FWD cam is not being actuated and vice-versa;

29. To move the lever to N gear.

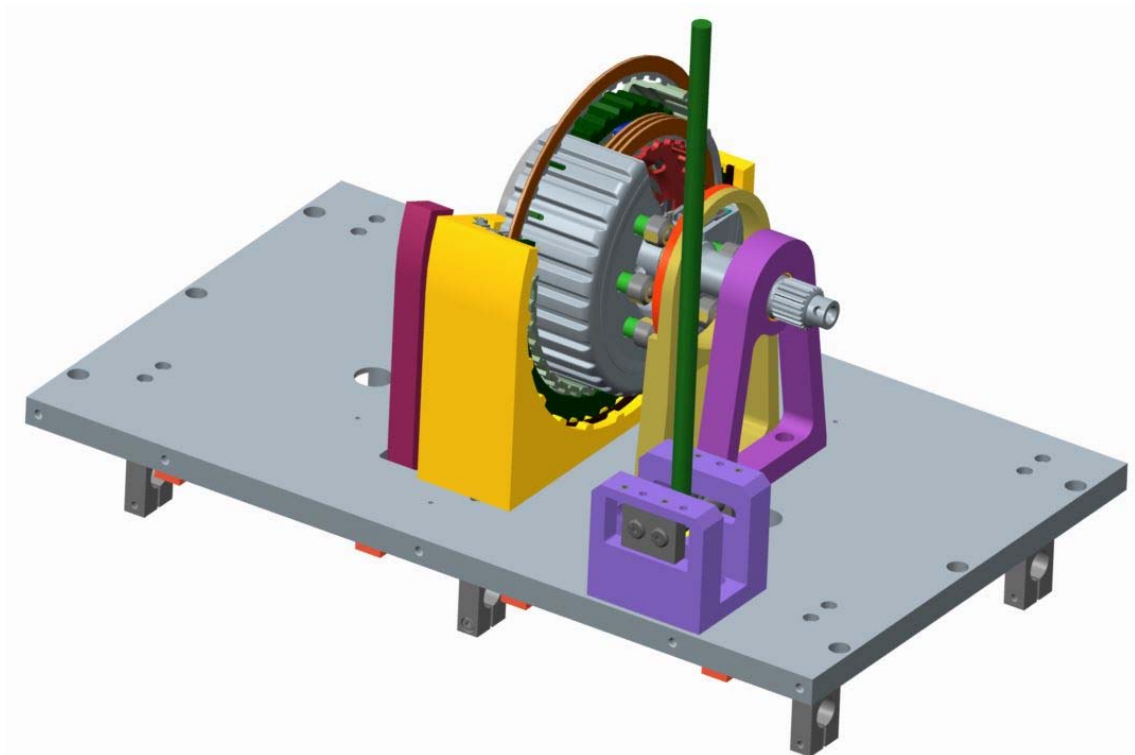


Figure 294 - Showcase assembly process: Step 29

30. To move the lever to the REV position and rotate the cam by hand, too.

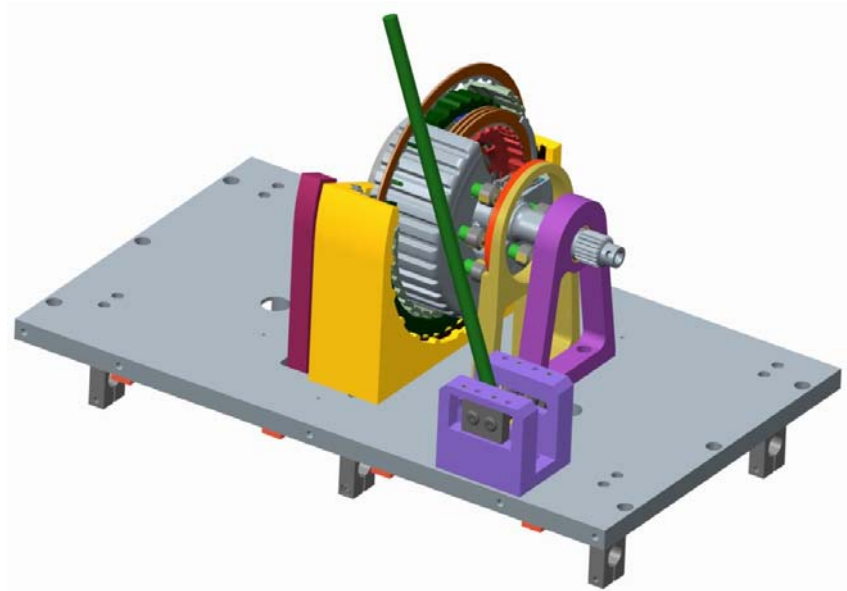


Figure 295 - Showcase assembly process: Step 30

31. In this figure, it is possible to see that in this position the FWD cam is not being actuated. There is contact but there is no transmission of force.

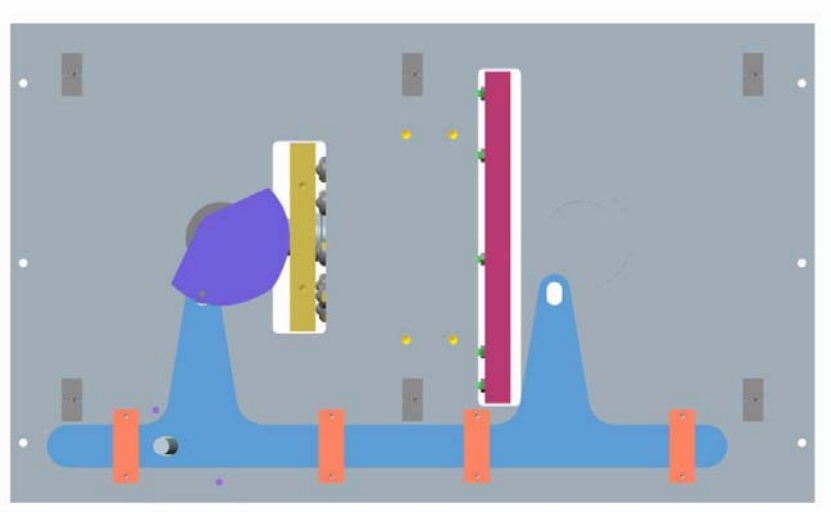


Figure 296 - Showcase assembly process: Step 31

32. To align the cam hole and the guiding plate hole. To fasten the cam to the guiding sheet (PNSA360) by mean of a shoulder screw DBGB5-8. An auxiliary hole in the main plate was created to allow the fastening of the screw. If you used a pin to help the alignment of the cam please remove it.

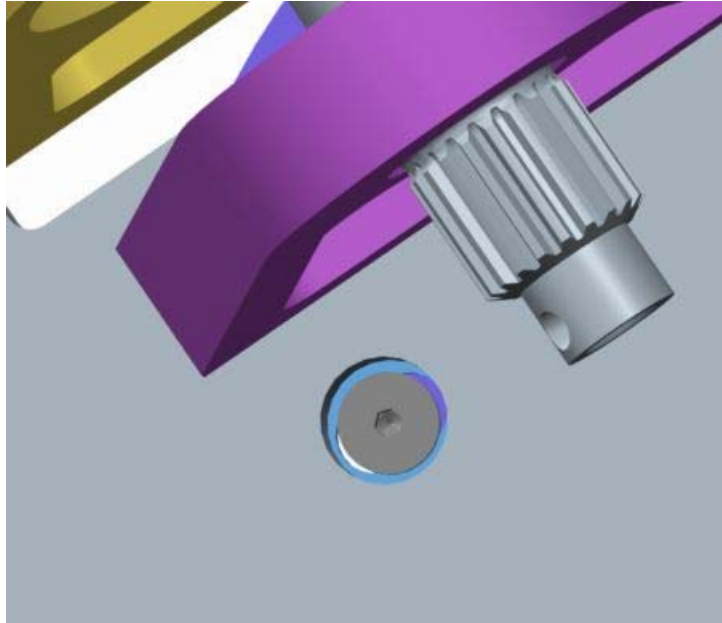


Figure 297 - Showcase assembly process: Step 32

33. To move the lever to Drive position and check if the travelling of the plate is enough to press the clutch plates and to ensure a correct working of the DNR. If necessary add or remove shims (the instructions of where these are located was previously explained). To verify at the end of the adjustment if the clutch plates are being pressed and that the DNR works correctly.

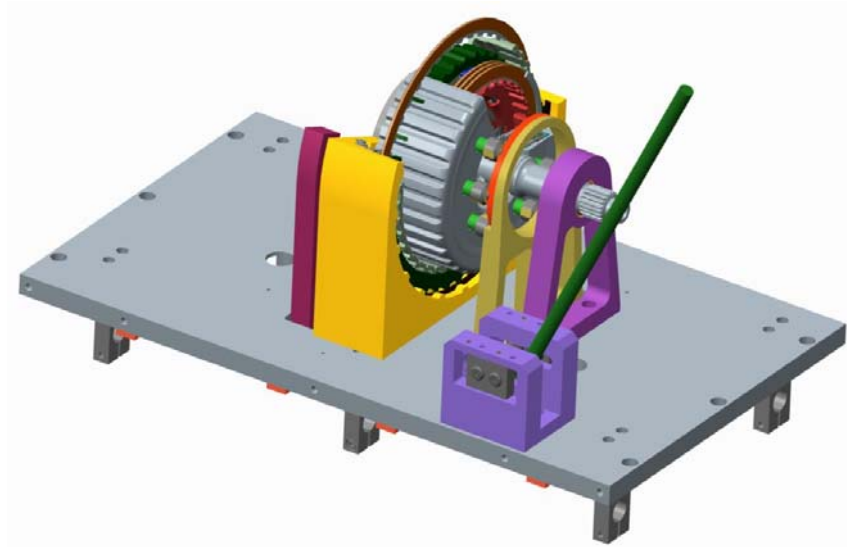


Figure 298 - Showcase assembly process: Step 33

34. To fasten the cam to the guiding sheet by mean of a shoulder screw DBGB5-8. An auxiliary hole in the base was created to allow the fastening of the screw.

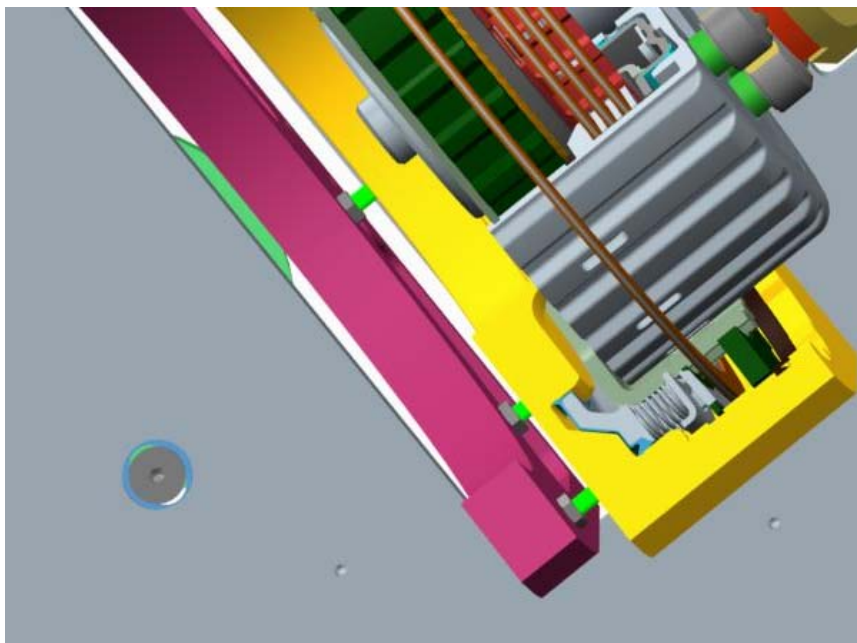


Figure 299 - Showcase assembly process: Step 34

35. To move the lever to Reverse position and check if the travelling of the plate is enough to press the clutch plates and to ensure a correct working of the DNR. If necessary add or remove shims (the instructions of where these are located was previously explained). To verify at the end of the adjustment if the clutch plates are being pressed and that the DNR works correctly. To move the lever to Drive position and verify if the forward clutches are being well pressed.
36. At this point the guiding trails need to be assembled. Remove the screws which are joining the forward pressing plate and the bushing support. Assemble the forward pressing plate with the guiding plate: align the pressing plate by the spring pins. Place the guiding bars trough the guiding blocks.
37. To assemble the reverse pressing plate with the guiding plate: align the pressing plate by the

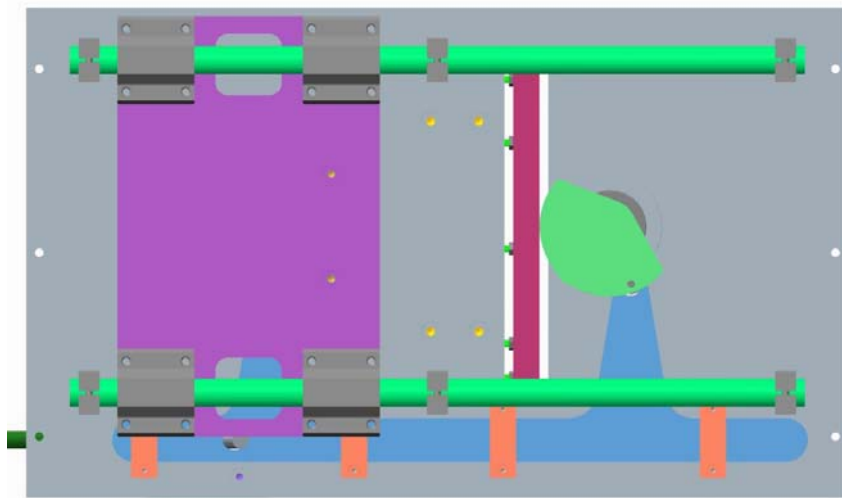


Figure 301 - Showcase assembly process: Step 36

spring pins. Place the guiding bars trough the guiding blocks. Fasten the 6 shaft supports. To check that the shafts do not move.

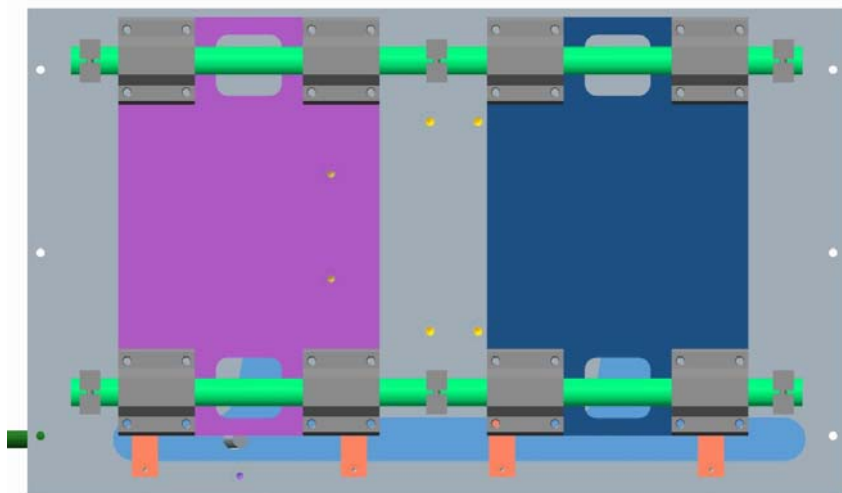


Figure 300 - Showcase assembly process. Step 37

38. To measure the play of the cam. If necessary adjust them by adding or removing shims.
39. To assemble the legs and screw them.
40. To verify that there is no interference between the cams and the guiding systems.

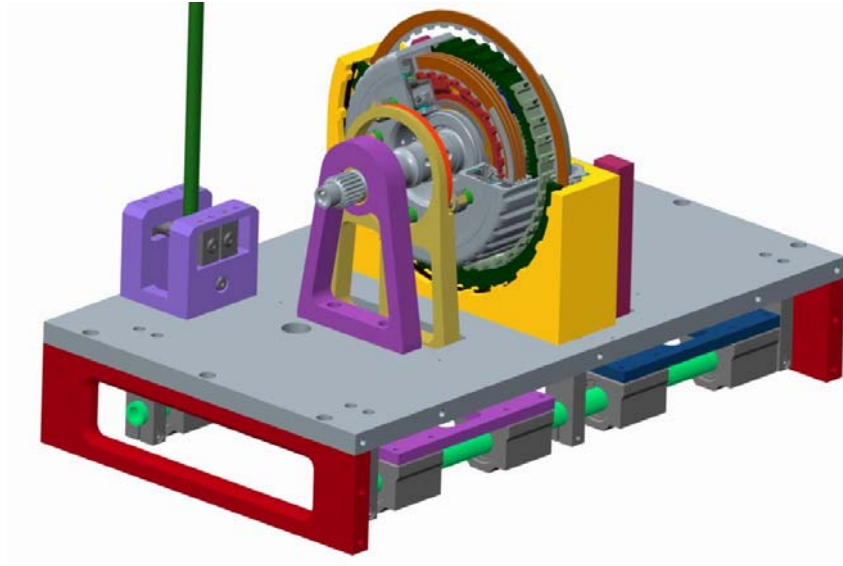


Figure 302 - Showcase assembly process: Step 39

41. To assemble and screw the side plates.

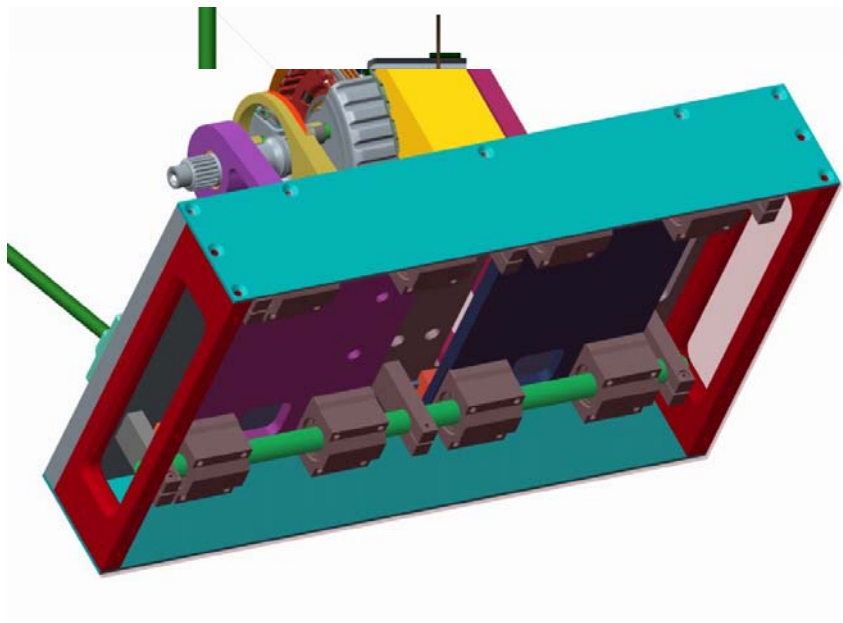


Figure 305 - Showcase assembly process: Step 42

42. To assemble and screw the transparent bottom cover.

43. To assemble the hand wheel by mean of 1x M5.

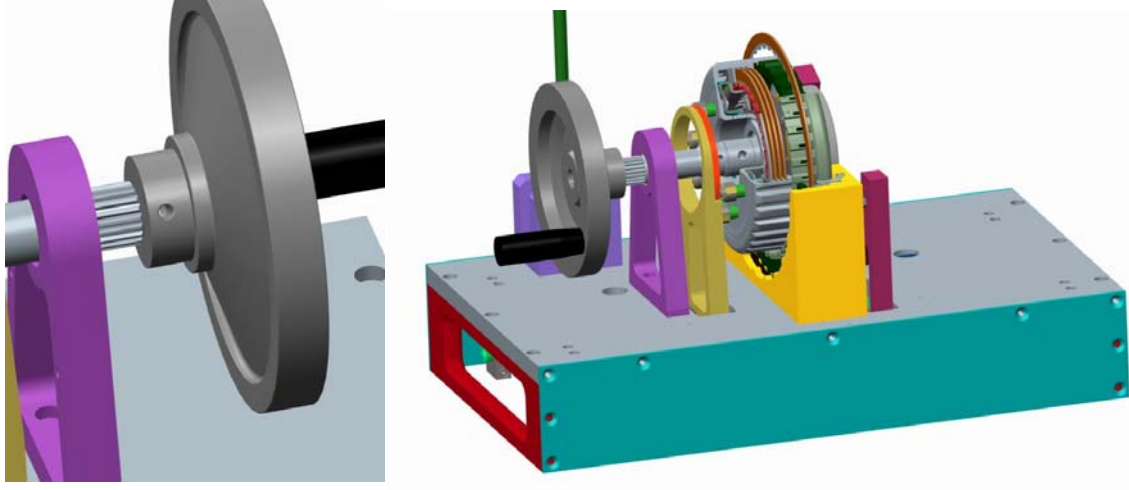


Figure 306 - Showcase assembly process: Step 43

44. To check if the travelling of both reverse and forward is enough to activate the clutches. To verify if the position of the lever is well defined and apply thread glue on the spring plungers. Assemble both covers: Lever system's cover and DNR transparent cover.

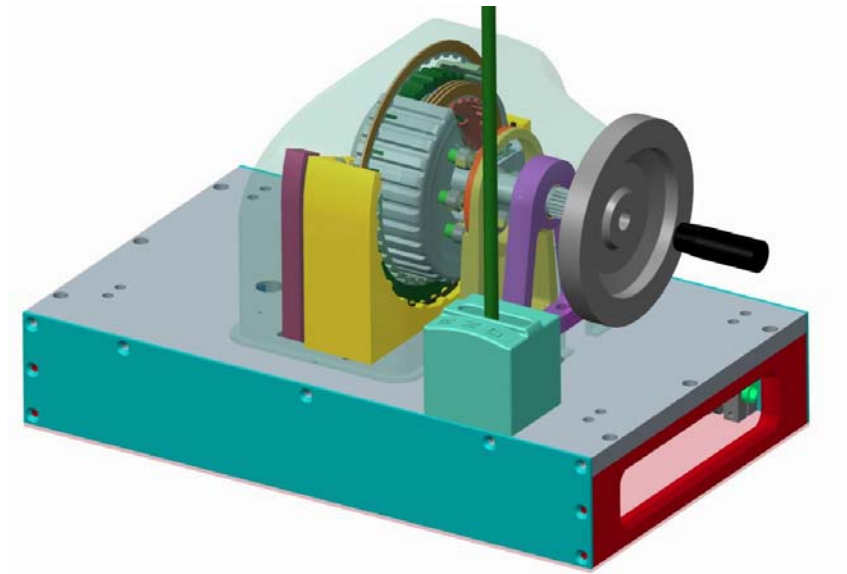


Figure 307 – Showcase assembly process: final step

6.36 Datasheet LOCTITE® 638™

6.37 Datasheet LOCTITE® 648™

6.38 Showcase model's BOM

6.39 Poster

6.40 Showcase photos

6.41 Technical drawings of individual parts of the showcase model

6.42 Technical drawings of assemblies of the showcase model