Clean Sky	CS2 – ENG ITD DEVILS – 737972 D2.3– Benefits of Variable Flow Control	protom

Deliverable Title:

DEVILS – *Benefits of Variable Flow Control*

Project Acronym:	**DEVILS**
Project full Title:	**Development of Intelligent Oil System enablers for Large VHBR Oil Lubrication and Heat Management Systems**
Grant number:	**737972**
Dissemination level	Public
Document Number	26931/DEVILS/001
Issue Date	10/07/2018
Keywords	Engine Oil lubrication, Oil Pump, Variable Oil Flow

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This project has received funding from the Clean Sky 2 Joint Undertaking under the European Union's Horizon 2020 research and innovation programme under grant agreement No 737972

DEVILS - Benefit of Variable Flow Control

Short Description

This document outlines the benefits of the DEVILS variable flow rate pump.

Document n° : ° 26931/DEVILS/001	Rev: 00	Codice Commessa (CUP): *E62F17000250006*	Date: 10/07/2018

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REVISION NOTIFICATION		
Rev	**Date**	**Revision description**
00	10/07/2018	First issue

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1. Introduction

Scope

Scope of this document is to describes the benefit of variable flow control provided by the Innovative Oil Lubrication System Demonstrator based on Variable Oil Flow Pump to be developed by the DEVILS project within Clean Sky 2 Program.
The benefits of the variable flow achieved, in the task 2.3, are demonstrated using modelling approaches: lumped parameter and three-dimensional CFD. A lumped parameter model of the pump has been built up on a preliminary design of the Devils pump and the output of the simulations have been given, as input, to the model of the oil lubrication system, already presented in the Deliverable 2.2.
Simulations have demonstrated that a variable flow control can give improvement to the lubrification and, as consequence, to the entire Turbofan. All the benefits are deeply described in this document.
By the end, a three-dimensional CFD model of the first design of the pump has been built up and run for both full and partialized flow.

Background

Within European Research Program, Clean Sky 2, Rolls Royce issued a Call for Proposal looking for innovative Variable Oil Flow Pumps technologies suitable for their range of Aircraft Engines.
DEVILS Consortium presented a solution based on an innovative Variable Flow Rate Oil Pump concept. The final output of DEVILS project is an oil lubrication system with the innovative variable flow rate oil pump demonstrator to be tested in laboratory to TRL 5.
Turbofan components lubrication is very important to achieve high level of performances. The purpose of an oil lubrication system is to ensure that transmission components within a Gas Turbine are protected from metal to metal contacts that would lead to wear and degradation, and to manage the temperature of these components. The heat management is the parameter used to size a system, as oil flows required for lubrication are relatively small compared to the flow required for temperature control. Lubrication system has both lubrication function both cooling function of engine bearings and gearbox components. This second function is most important to control their temperature limits.
In the following, the main phenomena of thermal hydraulic system are described:

- The main losses due to flow of viscous fluid in ducts are "distributed" and "concentrated". These losses produce heat due to friction between the fluid and the ducts walls. Generated heat flows out into the environment because of the temperature difference between ducts and environment.

- In the bearing housing, the bearings support the engine shaft and reduce the frictions. Their design is developed in order to eliminate oil leakages that could be dangerous because they could generate flames and smoke in the engine. The figure below shows the typical bearing housing design.

The bearing housing provides an auxiliary external chamber that contains pressurized air; it is necessary to border the oil in the internal chamber. Air enters in the lubrication circuits and subsequently it is eliminated by the breather. Air modify the heat exchange and it can damage the components of the hydraulic circuit.

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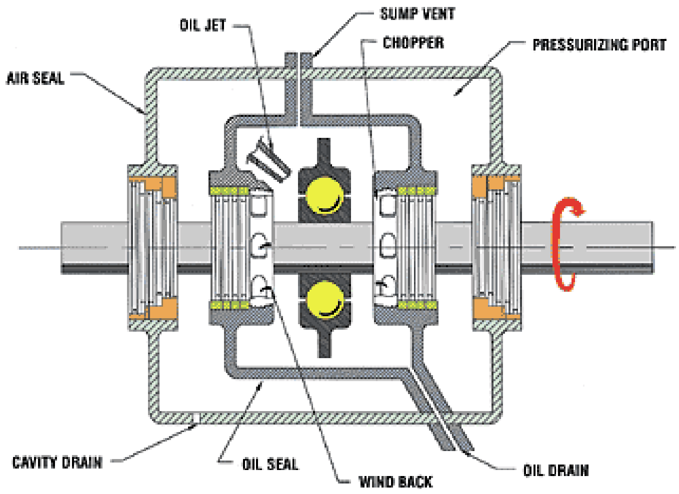

Figure 1 Bearing Housing.

Figure 1 shows a typical turbofan lubrication system and its main components.

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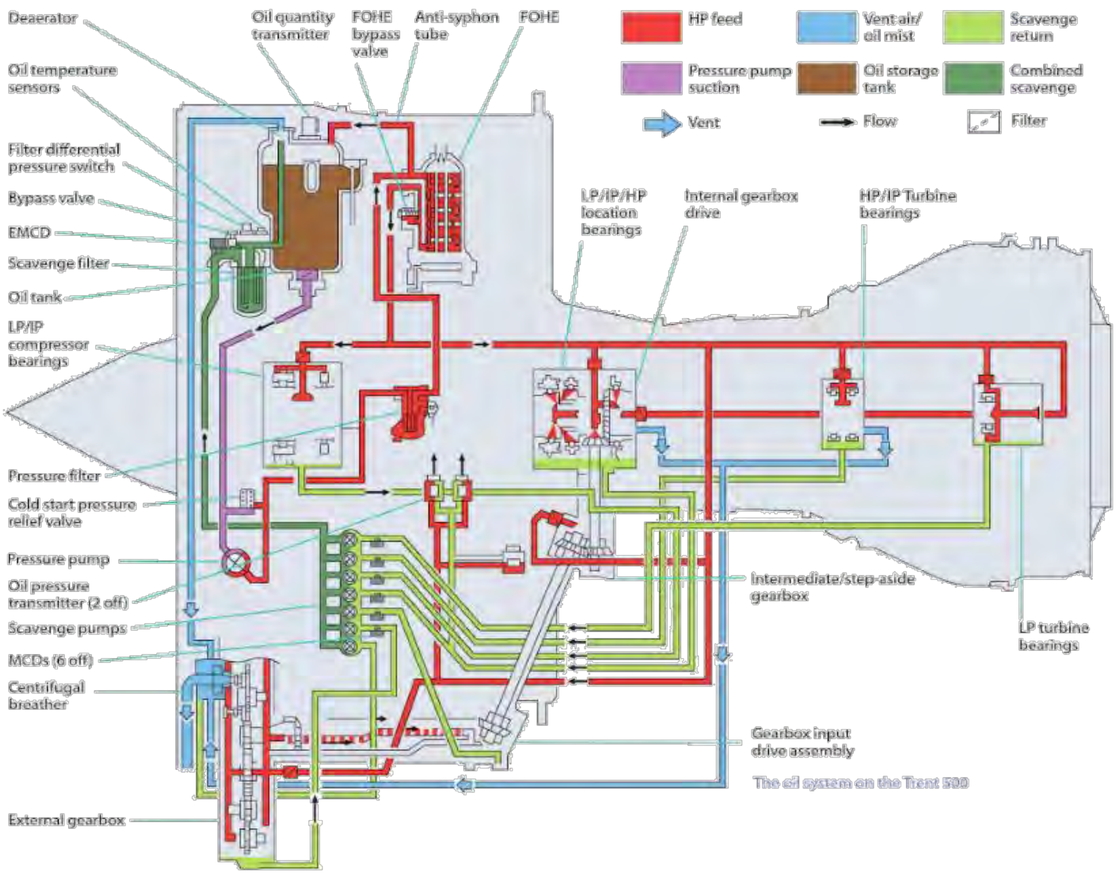

Figure 2 Trent 500 oil lubrication system *(The Jet Engine-Rolls-Royce).*

2. Glossary and Terminology

Definitions

Alternative: Compared with the original, a replacement item showing parametric changes or providing a limited parametric match, but which, after consultation with the person or organization responsible for the design, may still be acceptable (also “Substitute”).
Analysis: Analysis is defined as the verification that a specified requirement has been met through the technical evaluation of equations, charts, reduced data and/or representative data. Data derived from the performance of any test in the qualification program may be used to fulfil similar requirements in any other test through the application of analysis techniques.
Component: A self-contained part, combination of parts, sub-assemblies or units, which performs a distinct function of a system. The term “Component” identifies a general item covered by a Part Number or Drawing Number used for but not necessarily designed for the system.
Demonstration: Is defined as a non-instrumental test where verification is determined by observation alone. Included in this category are tests that require simple quantitative measurements such as dimensions, time to perform tasks, etc. Demonstration shall also be used to provide a reference to a more detailed test being used for verification of the same equipment element. In such cases, a reference to the detailed test shall be given.
Modelling / Simulation: The process of conducting experiments with a model. Simulation may include the use of analog or digital devices, laboratory models or “test bed” sites.
System: A collection of hardware and software components organised to accomplish a specific function or set of functions. The term “System” identifies the whole system including the equipment described in this specification and all other components envisaged for the fulfilment of the system function.
System Architecture: The structure of the hardware and the software selected to implement the system requirements.
Test: Test is defined as the verification that a specified requirement is met by a thorough exercising of the subject element in accordance with the applicable test method and test procedure. Testing shall be performed using the equipment specified, and the conditions and duration specified in the detailed test procedures.
ISA: The International Standard Atmosphere (ISA) is an atmospheric model of how the pressure, temperature, density, and viscosity of the Earth's atmosphere change over a wide range of altitudes or elevations.

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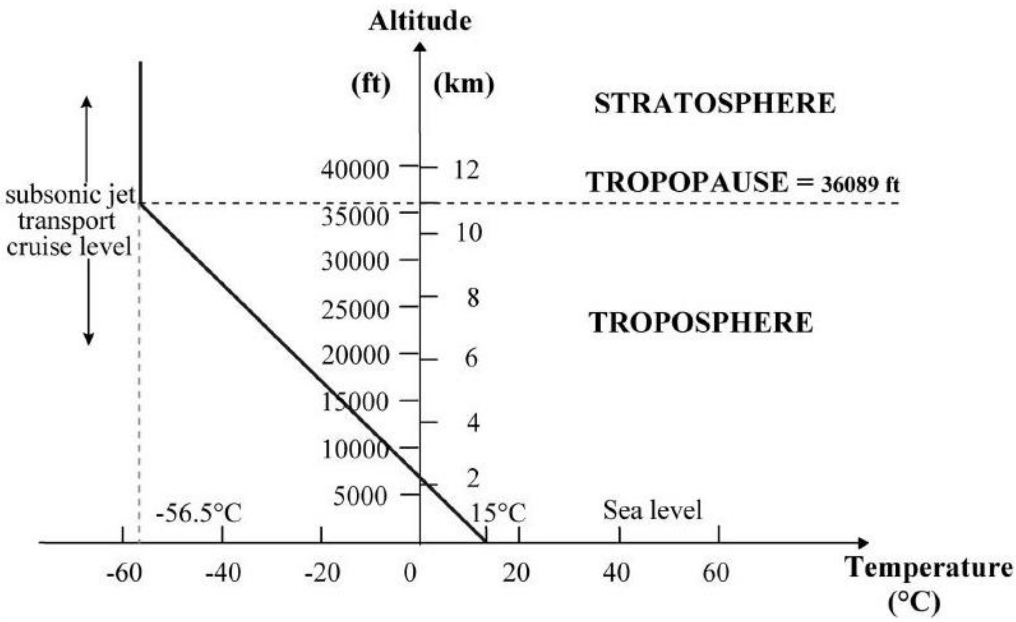

Figure 3 Standard ISA.

OIL_JET MOBIL_JET_387: Lubrication oil used in turbofan like as RR Trent 900
MIL_PRF_23699: This specification is approved for use by all Departments and Agencies of the Department of Defense. This specification covers four classes of gas turbine engine lubricating oils, primarily used for aircraft engines, which have a nominal viscosity of 5 centistokes at 100 °C and which are typically made with neopentyl polyol ester base stocks. This oil is identified by NATO Code Numbers O-152, O-154, O-156, and O-167.
JET_A1: It is a type of aviation fuel designed for use in aircraft powered by gas-turbine engines. It is colorless to straw-colored in appearance. The most commonly used fuels for commercial aviation are Jet A and Jet A-1, which are produced to a standardized international specification.
Mesh: In computational solutions of partial differential equations, meshing is a discrete representation of the geometry that is involved in the problem. Essentially, it partitions space into elements (or cells or zones) over which the equations can be approximated. Zone boundaries can be free to create computationally best shaped zones, or they can be fixed to represent internal or external boundaries within a model.

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Abbreviations

Flight Conditions

ISA = International Standard Atmosphere
CRZ = Cruise
TOC = Top Of Climb
EOC = Edge Of Coverage
MTO = Maximum Take-off weight
MCT = Maximum Continuous Thrust

Lubrication Components

ACOC = Air Cooler Oil Cooler
FCOC = Fuel Cooler Oil Cooler
FOHE = Fuel Oil Heat Exchanger
AOHE = Air Oil Heat Exchanger
FBH = Front Bearing Housing
PGB = Power GearBox
IGB = Internal GearBox
AGB = Auxiliary GearBox
RBC = Rear Bearing Chamber
TBH = Tail Bearing Housing
HMCU = Health Monitoring Control Unit
TM = Turbo Machinery
MCD = Magnetic Chip Detector
EMCD = Electro Magnetic Chip Detector
VHBR = Very High Bypass Ratio
LP = Low Pressure
HP = High Pressure
IP = Intermediate Pressure

General abbreviations

RR = Rolls Royce
CFD = Computational fluid dynamics
MBSE = Model Based Systems Engineering
VHBR = Variable High Bypass Ratio

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3. Applicable and Reference Documents

In this paragraph, the requirements about the lubricating circuit are reported. These specific requirements are extracted by the 16/PG/CT/00163/SYS/HLTR/0042 Devils System High Level Technical Requirements Document.

ISSUED BY	*NUMBER*	*TITLE*
SAE	ARP4754A	Guidelines for Development of Civil Aircraft and Systems
SAE	ARP4761	Guidelines and Methods for Conducting the Safety Assessment Process on Civil Airborne Systems and Equipment
SAE	AS4941	Aerospace-general requirements for commercial aircraft hydraulic components
SAE	AIR4543	Aerospace Hydraulics and actuation lesson learned
SAE	AIS1290	Graphic symbols for aircraft hydraulic and pneumatic system
RTCA	RTCA/DO16G	Environment conditions and test procedures for Airborn equipment
SAE	AS1300	Port-ring locked fluid connection type, standard…
SAAE	AS5202	Port or fitting and internal straight thread
EAS	CS-E	Certification specification & acceptable means of compliance for engines
EAS	CS-25	Certification specification for large airplanes

Table 1 Standards Aviation

CS-25 legal standards required

DEVILS-HLTR-OPT-1	The Variable Flow Oil Pump may comply with **CS.25-603** (Materials).
DEVILS-HLTR-OPT-2	The Variable Flow Oil Pump may comply with **CS.25-605** (Fabrication Methods).
DEVILS-HLTR-OPT-3	The Variable Flow Oil Pump may comply with **CS.25-613** (Material Strength Properties and Material Design Values).
DEVILS-HLTR-OPT-4	The Variable Flow Oil Pump may comply with **CS.25-621** (Casting Factors).
DEVILS-HLTR-OPT-5	The Variable Flow Oil Pump may comply with **CS.25-623** (Bearing Factor).
DEVILS-HLTR-OPT-6	The Variable Flow Oil Pump may comply with **CS.25-625** (Fitting Factor).
DEVILS-HLTR-OPT-7	The Variable Flow Oil Pump may comply with **CS.25-899** (Electrical Bonding and Protection against Static electricity).
DEVILS-HLTR-OPT-8	The Variable Flow Oil Pump may comply with **CS.25-1183** (Flammable fluid carrying components).

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DEVILS-HLTR-OPT-9	The Variable Flow Oil Pump may comply with **CS.25-1301** (Functional and Installation Equipments).
DEVILS-HLTR-OPT-10	The Variable Flow Oil Pump may comply with **CS.25-1309** (Equipments, Systems and Installation).
DEVILS-HLTR-OPT-11	The Variable Flow Oil Pump may comply with **CS.25-1353** (Electrical Equipments and Installations).
DEVILS-HLTR-OPT-12	The Variable Flow Oil Pump shall comply with **CS.25-1435** (Hydraulic Systems)**.**

Environmental Requirements

ESA CS-25 1435

Fire resistant

DEVILS – HLTR-OPT-30 / ESA 25.863; 25.1183; 25.1435.

Others reference standards Plugs

AS3121	
AS3131	
AS5169	Type plug with elastomeric seal
AS1300	Ring lock fluid connection type

Table 2 Standards Aviation

Fatigue Factors

DEVILS – HLTR-OPT-42 / ANSI B46.1

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4. MBSE: Detection of system benefits

Introduction

The main scope of the task 2.3 is the demonstration of the benefits of variable flow control applied to turbofan engine. Two different numerical approaches have been used to bring out all the benefits. The design flowchart of the DEVILS pump is shown in figure 4. It consists of a close loop specific sequence of steps that include two numerical approaches a lumped parameter and a three-dimensional CFD. First of all, the lumped parameter model of the pump has been built up using the commercial code AMESim®, developed by Siemens®. Simulation results obtained with the numerical model of the variable displacement Getoror pump have been given, as input, to the model of the whole engine lubrication circuit already discussed in the deliverable 2.2. The benefits of the variable flow control will be shown in this document. By the end, the flow behaviour of a preliminary design of the pump has been studied using a three-dimensional CFD numerical approach. The model has been built up using a commercial code PumpLinx®, developed by Simerics Inc.®. The results of the lumped parameter model of the entire oil lubrication circuit have been given as boundaries conditions to the 3D CFD model. The data exchanges between 1D and 3D models, therefore, allows individuating the optimal pump design and benefits forecasting of this solution before the manufacturing of the first prototype.

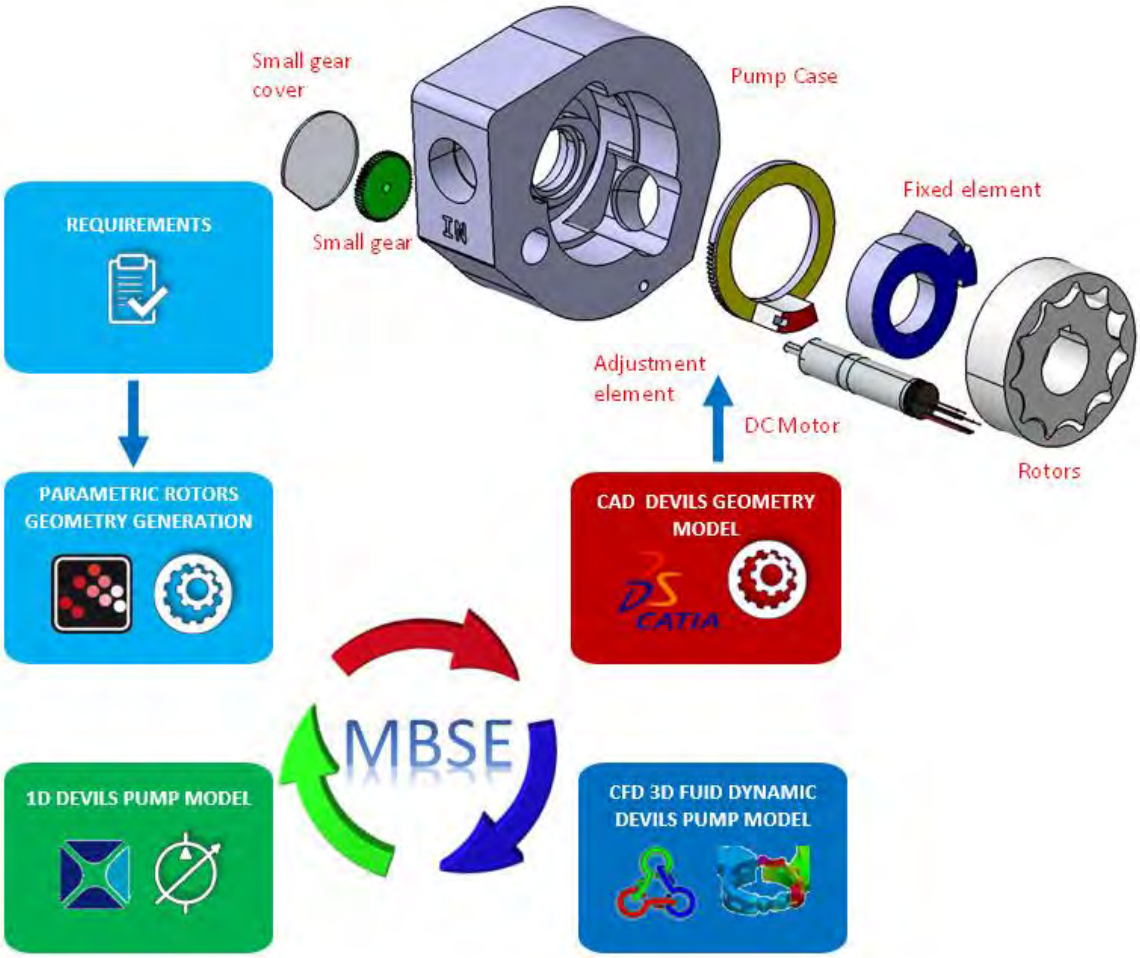

Figure 4 Devils pump design flowchart

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The defined numerical approach has given important information identifying the fluid-dynamic parameters that mainly effect on the pump performance. From this analysis, the CAD 3D drawn of the DEVILS pump can be optimized, in the task 2.4, including the regulation system.

UltraFan® Oil System Data

As known, the main requirement of the oil lubrication system is the heat management; all other parameters become secondary. For this reason, as said in the introduction, the oil lubrication circuit of a turbofan is sized to respect temperature requirements for a given heat load. Secondary capability of the oil lubrication system is that it must deliver the required flow contributing to the reliability and efficiency of the turbofan.
The system temperature limits are listed in Table 3.

PARAMETER	TEMPERATURE LIMIT (°C)
PGB Oil Inlet Temperature	120
TM Oil Inlet Temperature	140
PGB Scavenge Temperature	160
TM Scavenge Temperature	180
Max temp rise across TM/PGB	40
FCOC exit Temperature	120

Table 3 Temperature limits

Table 3 shows the oil temperature limits for the candidate system architecture (16PGCT00163-SYS-HLTR-0042_DEVILS High Level Technical Requirements). As previously said, the oil flows and heat rejection breakdowns values are indicative only, meeting temperature limits is the cardinal requirement.

Lumped parameter model

In this section the developed numerical model is described. The lumped parameter numerical model has been used for demonstrating the benefits of variable flow control.
The *MTO* flight condition has been selected for the simulations at the following operating conditions:

- Pump rotation speed of *5000 rpm*;
- Inlet pressure: *1.1 barA*;
- Outlet pressure of points 1, 2, and 3: *17 barA*;
- Outlet pressure of point 4: *6 barA*;
- Outlet pressure of point 5: *7.8barA*;
- Different position of the adjustment element.

The simulated points are clearly shown in figure 5 where the line in light blue represents the deliver flow at the pump maximum displacement (no rotation of the adjustment element). The pump would be sized to achieve a delivered flow of almost 350 L/min at 5000rpm (max displacement trend) for an outlet pressure of 17barA. In this way, the $Q_{Operation}$, in the graph, is guaranteed.

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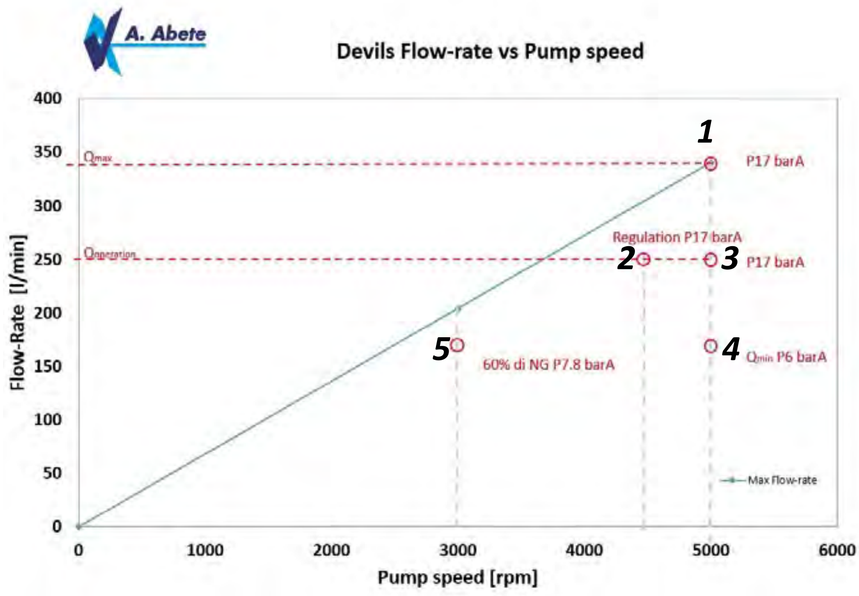
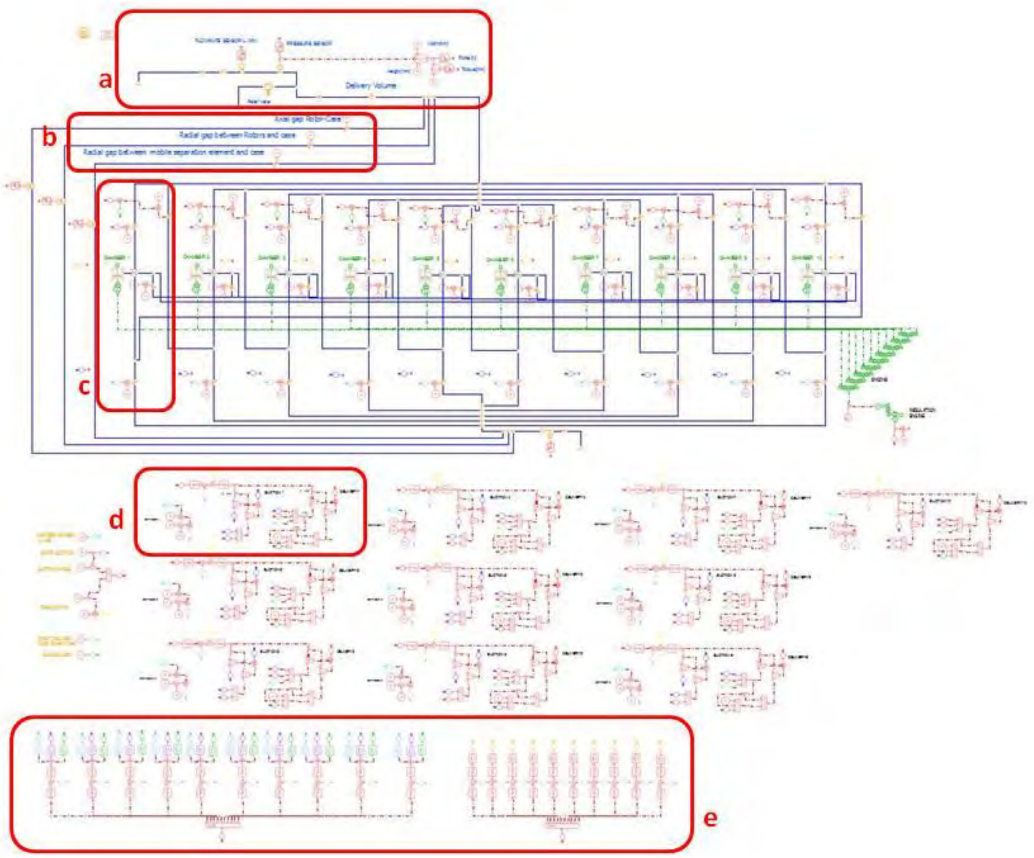


Figure 5 Investigation operation points

The lumped parameter model of the entire pump and zoom of each unit are reported in figure 6.



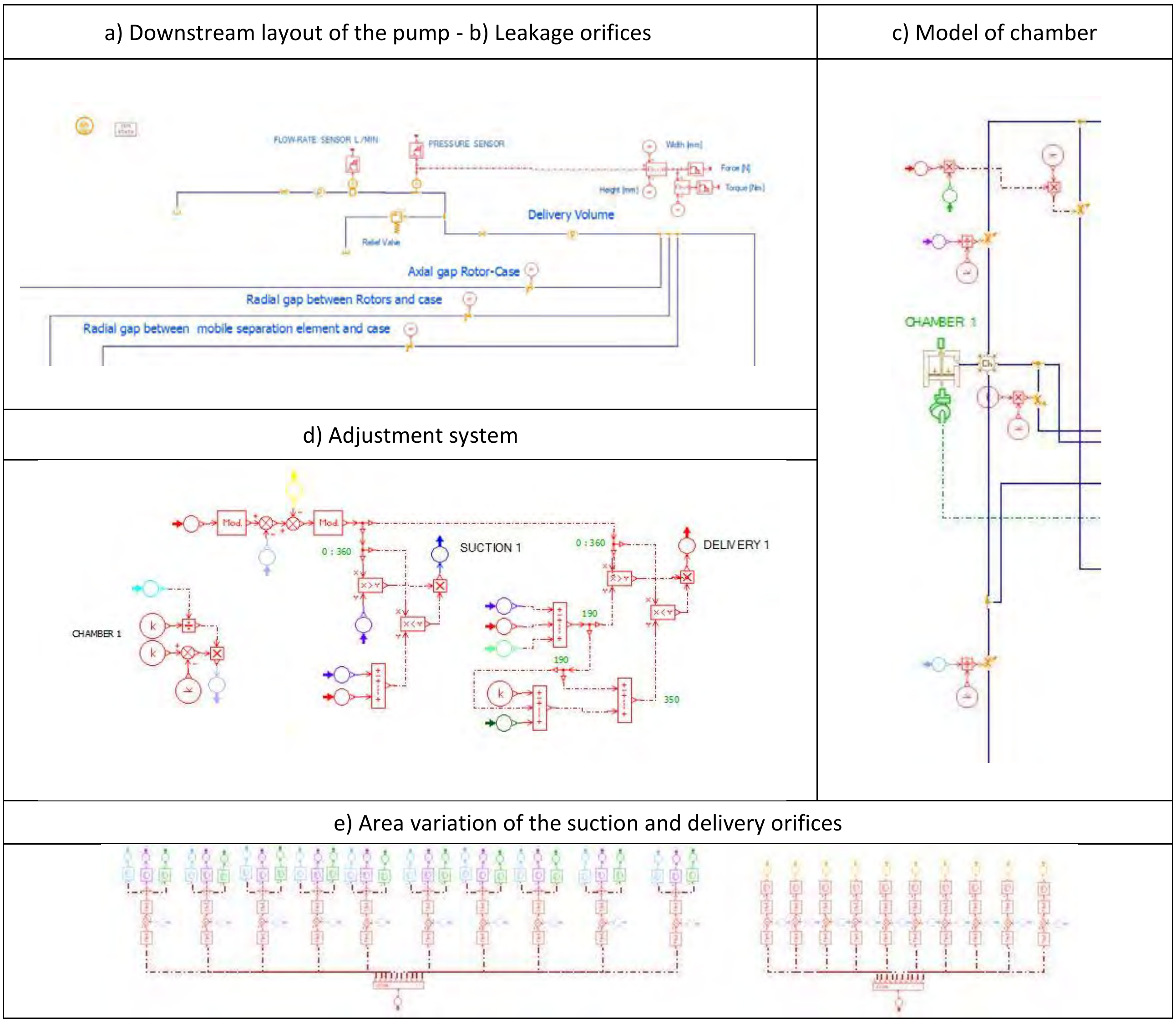


Figure 6 Lumped parameter model of the variable displacement Gerotor pump.

In figure 6, each pumping volume, obtained from the inner and the outer rotors, has been modelled as a piston. The timing and of the pump is controlled by signals that link each chamber respectively with inlet and outlet ports volumes. The model includes signals to rotate the adjustment element allowing, as consequence, the variation of the delivered flow.

The leakages, axial, radial and tooth, have been modelled with calibrated orifices reproducing the real geometry gap due to manufactory tolerance and the coupling among components of the pump.

Simulations have been run also for other working conditions in order to fully characterize the pump. The obtained numerical results have been given as input to the model of the oil lubrication circuit of the turbofan. Figure 7 shows the entire model of the oil lubrication circuit. The relief valve has been included in the model to compare a typical “fixed displacement gerotor pump” with the “Devils pump”.

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Two configurations have been simulated:

[1] Configuration 1: The pump works at maximum flow-rate (as a common fixed gerotor pump). In this configuration the relief valve is activated and, as consequence, the flow-rate surplus is recirculated. The relief valve has a cracking pressure of 17 barA (limit of the circuit pressure).

[2] Configuration 2: the relief is closed, and the pump delivers the necessary flow rate by adjusting the position of the regulating element.

Numerical results of the model in figure 7 are shown in the next paragraph.

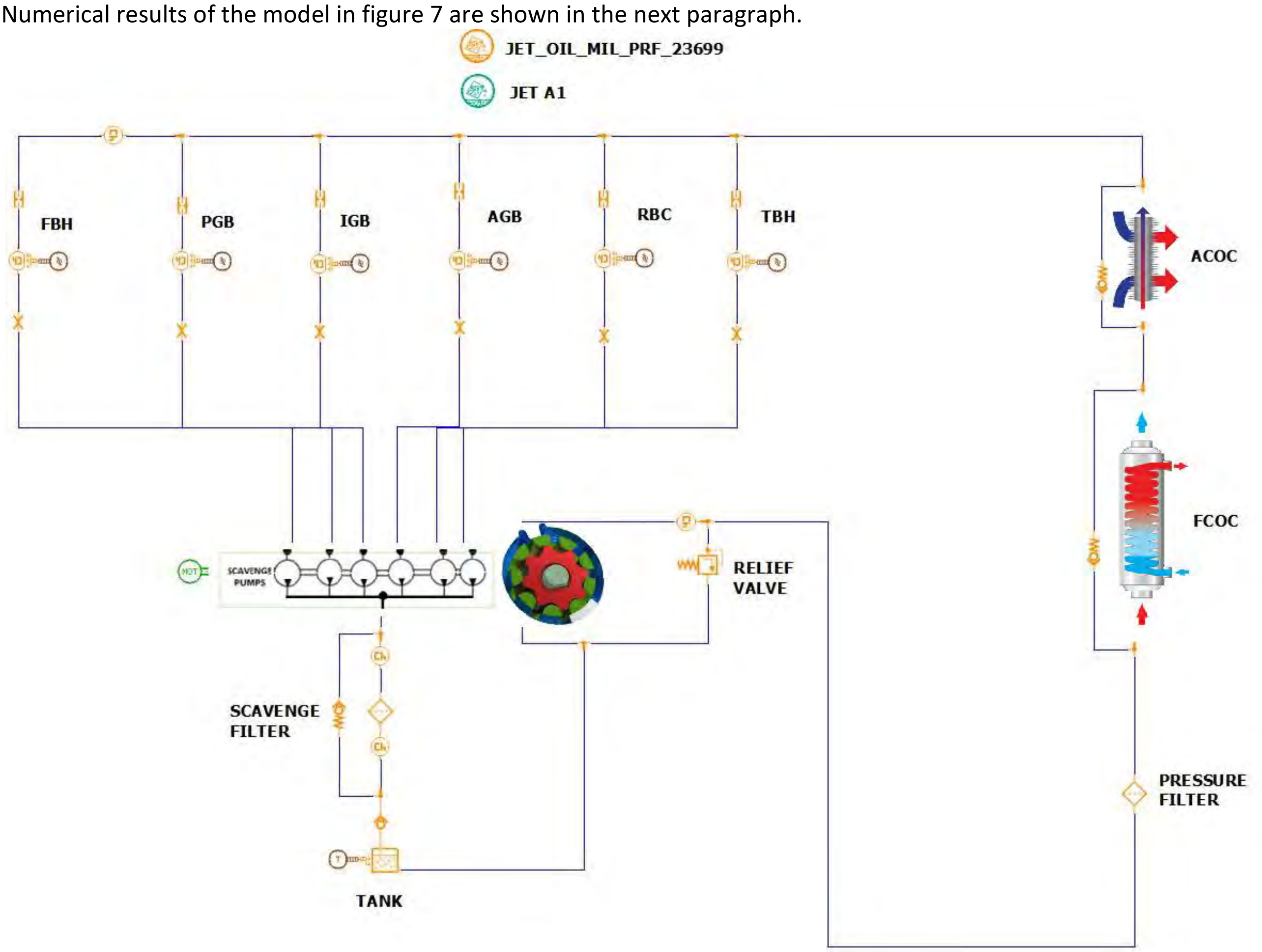

Figure 7 Lumped parameter model of the lubrication circuit model.

5. Benefit of variable flow control

As said in the introduction, the aim of this document is describing the benefits of variable flow control provided by the Innovative Oil Lubrication System Demonstrator based on Variable Oil Flow Pump to be developed by the DEVILS project within Clean Sky Program.

The benefits of the proposed variable displacement Gerotor pump are better clarified in this section of the document; where, using the lumped parameter in figure 7, an analysis of the model output has been performed with comparing a fixed geometry pump with a variable one.

Simulations have provided interesting benefits that are listed below:

- Adjustment of Circuit Pressure

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- Variable Flow Rate and Pressure with Fixed Pump Speed
- Fixed Flow Rate and Pressure with Variable Pump Speed
- Compensation of Viscosity Variation
- Compensation of loss due to Components Wear
- Compensation of loss due to physical-chemical properties of the oil

Those benefits are common to all the variable displacement pumps whoever are new for the Gerotor pumps.

Adjustment of Circuit Pressure

The Adjustment of Circuit Pressure is, as well know, common to all the variable displacement pumps. In fact, by varying the delivered flow-rate the pressure of the lubrication circuit can be varied with respect temperature limits. This has been demonstrated with the adopted numerical approaches. The main results are summarized in table 4.

PERFORMANCE			TEMPERATURE [°C]					
Adj. angle [degree]	Pressure [barA]	Flow-rate [l/min]	FBH	PGB	IGB	AGB	RBC	TBH
0°	17.2	322.3	117.1	130.7	128.3	128.1	126.0	115.4
80°	6.2	158.9	113.8	135.3	131.8	131.5	130.17	114.3
Temperature limits			180	160	180	180	180	180

Table 4 Benefit of variable flow control: adjustment of Circuit Pressure

Table 4 shows that in case of pressure reduction the pump can supply the necessary flow respecting the temperature limits.

Variable Flow Rate and Pressure with Fixed Pump Speed

As for the above point, as expected, one advantage of using a variable displacement pump, for a fixed the pump rotation speed, is that it is possible to vary the delivered flow-rate and the pressure as well. An example of this benefit is shown in table 5.

PERFOMANCES			TEMPERATURE [°C]					
Adj. angle [degree]	Pressure [barA]	Flowrate [l/min]	FBH	PGB	IGB	AGB	RBC	TBH
0°	17.2	322.3	117.1	130.7	128.3	128.1	126.0	115.4
40°	17.2	239.9	115.0	129.9	127.7	127.4	125.4	114.7

Table 5 Benefit of variable flow control: Variable Flow Rate and Pressure with Fixed Pump Speed

In table 5 have been presented two conditions. In the first one, the pump works by delivering the maximum flow-rate as a fixed gerotor pump. The flow surplus is recirculated by the relief valve, which, as said, has a cracking pressure of 17 barA. In the second configuration, the relief is closed and the pump adjusts the delivered flow to the only necessary flow-rate. This latter condition could ensure an energy saving.
The numerical results show an important feature of the pump; indeed, it allows new strategy to engine oil management. In the MTO condition the pump can, in fact, vary the flow-rate and decrease the pressure of lubrication circuit up to minimum value of 6 barA.

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Fixed Flow Rate and Pressure with Variable Pump Speed

Several tests have been performed at variable pump speed in order to prove the capability of the Devils pump to supply a fixed flow-rate before and then a fixed delivery pressure of the oil lubrication circuit.
The main results are summarized in the following table.

PERFOMANCE				TEMPERATURE [°C]					
Pump speed [rpm]	Adj. angle [degree]	Pressure [barA]	Flowrate [l/min]	FBH	PGB	IGB	AGB	RBC	TBH
5000	40°	17.15	239.9	115	129.9	127.66	127.39	125.38	114.69
4000	33°	17.12	237.8	114.99	129.9	127.64	127.37	125.36	114.67

Table 6 Benefit of variable flow control: Fixed Flow Rate and Pressure with Variable Pump Speed

In table 6 there have been compared the results evaluated for two pump rotation speed of 4000 rpm and 5000 rpm. As shown in figure 5, at 5000 rpm the delivery flow of a fixed displacement pump (at 17 barA of delivery pressure) is of almost 350 L/min and becomes almost 300 L/min at 4000 rpm. In this case, with different rotation of the adjustment element (of 33° for the case at 4000 rpm and of 40° at 5000 rpm) the delivery flow-rate is the same. This could ensure an energy saving. The regulation system is able to move the adjustment element in 0.01s with a consequent lag of a second on the system's pressure response.

Compensation of loss due to Components Wear and physical-chemical properties of the oil

Other simulations have been run on the model in figure 7, in order to prove the capability of the Devils pump to compensate the components wear and then to provide both: fixed flow-rate and pressure in the circuit. This analysis has been performed since the numerical model, in figure 6, already includes the leakages. Therefore, to simulate the components wear those leakages have been increased. The main results are summarized in the following table.

PERFOMANCE				TEMPERATURE [°C]					
WORKING CONDITION	Adj. angle [degree]	Pressure [barA]	Flowrate [l/min]	FBH	PGB	IGB	AGB	RBC	TBH
Nominal	40°	17.15	239.9	115	129.9	127.66	127.39	125.38	114.69
Components wear	40°	13.7	203	114.27	131.16	128.54	128.27	126.29	114.08
Compensation	35°	17.11	241	115.02	129.9	127.67	127.4	125.39	114.7

Table 7 Benefit of variable flow control: Compensation of loss due to Components Wear, at 5000rpm

Numerical model has given also interesting results showing the capability of compensating the effect on the circuit of the viscosity increment due to the reduction of the temperature. In the following table, there are results at the Idle condition (engine cold warm up) for three different air temperatures.

IDLE CONDITION		AIR CONDITION AT SEA LEVEL [°C]	TYPE OF PUMP	PRESSURE [barA]	FLOW-RATE [L/min]
		ISA - 45	Conventional pump	13.8	208.3

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3000 RPM	**Pressure target 8barA**		Devils Pump (42°) *	8.02	143
		ISA -15	Conventional pump	13.25	200.3
			Devils Pump (40°) *	8.3	149
		ISA + 25	Conventional pump	12.99	195.5
			Devils Pump (40°) *	8.16	145.5

Table 8 Benefit of variable flow control: Compensation of physical-chemical properties of the oil – 3000rpm

Furthermore, since in the tank 2.7 the CNR – IM will perform a research in cavitating conditions, some simulations have been run by forcing the pump to cavitate. The cavitation phenomenon has been forced by inducing calibrated orifices at the suction port. Results are shown in table 9.

	FLOW-RATE [L/MIN]	**% VOID CONTENT**
Conventional pump	275	30%
Adjustment 40°	240	20%
Adjustment 50°	210	8%

Table 9 Preliminary study on cavitation; calibrated orifice inlet line = 28 mm

* adjustment angle of Devils Pump

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6. The 3D CFD modelling analysis

The adopted Three-Dimensional Methodology

As said in the chapter introduction, on a preliminary geometry models have been built up using a commercial code. After an analysis on codes available, the software PumpLinx® has be chosen because it has been developed to study many of the common problem of hydraulic components. PumpLinx® (developed by Simerics Inc.®) solves numerically the fundamental conservation equations of mass, momentum and energy and includes accurate physical models for turbulence and cavitation.

PumpLinx® grids use a body-fitted binary tree approach. This type of grids is accurate and efficient because:

- The parent-child tree architecture allows for an expandable data structure with reduced memory storage;
- Binary refinement is optimal for transitioning between different length scales and resolutions within the model;
- The majority of cells are cubes, which is the optimum cell type in terms of orthogonally, aspect ratio, and skew ness thereby reducing the influence of numerical errors and improving speed and accuracy;
- It can be automated, greatly reducing the set-up time;
- Since the grid is created from a volume, it can tolerate "dirty" CAD surfaces with small cracks and overlaps.

In the boundary layer, the binary tree approach can easily increase the grid density on the surface without excessively increasing the total cell count. In the regions of high curvature and small details, the grid has been subdivided and cut to conform to the surface.

As already said, the code, during the simulations, solves the conservation equations for mass, momentum and energy:

Mass Conservation:

$$\frac{\vartheta}{\vartheta t}\int_{\Omega(t)}\rho d\Omega + \int_{\sigma}\rho(v - v_{\sigma})nd\sigma = 0 \qquad (1.1)$$

Momentum Conservation:

$$\frac{\vartheta}{\vartheta t}\int_{\Omega(t)}\rho v d\Omega + \int_{\sigma}\rho\big((v - v_{\sigma})n\big)v d\sigma = \int_{\sigma}\tilde{\tau} n d\sigma - \int_{\sigma} p n d\sigma + \int_{\Omega} f d\Omega \qquad (1.2)$$

Energy Conservation:

$$\frac{\vartheta}{\vartheta t}\left[\rho\left(u + \frac{v^2}{2} + gz\right)\right] + \nabla\left[\rho v\left(h + \frac{v^2}{2} + gz\right)\right] + \nabla Q - \nabla(T_d v) = 0 \qquad (1.3)$$

in which $\Omega(t)$ is the control volume, σ is the control volume surface, n is the surface normal pointed outwards, ρ is the fluid density, p is the pressure, f is the body force, v is the fluid velocity, v_{σ} is the surface motion velocity. The shear stress tensor $\tilde{\tau}$ is a function of the fluid viscosity μ and of the velocity gradient; for a Newtonian fluid this is given by the following equation:

$$\tau_{ij} = \mu\left(\frac{\vartheta u_i}{\vartheta x_j} + \frac{\vartheta u_j}{\vartheta x_i}\right) - \frac{2}{3}\mu\frac{\vartheta u_k}{\vartheta x_k}\delta_{ij} \qquad (1.4)$$

where u_i (i = 1, 2, 3) is the velocity component and δ_{ij} is the Kronecker delta function.

The code allows for the simultaneous treatment of moving and stationary fluid volumes. Each volume connects to the others via an implicit interface. The PumpLinx® Mismatched Grid Interface is a very efficient implicit algorithm that identifies the overlap areas and matches them without interpolation. After that, this area is treated as the common face connecting cells on both sides of the interface. During the simulation process, this face is treated no differently than an internal face between two neighbouring cells in the same grid domain. Thanks to this approach, the solution becomes very robust, quick and accurate.

Turbulence models for computing effective liquid viscosities are important at high Reynolds numbers. Mature turbulence models, such as the standard $k - \varepsilon$ is implemented. This model has been available for more than a decade and has been widely demonstrated to provide good engineering results. In the CFD 3D modelling there are other resolution method more accurate than the $k - \varepsilon$ model and RNG $k - \varepsilon$ model. As a matter of fact, for the specific problem, losses due to the viscous stresses are negligible compared to pressure forces. The adoption of higher order

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turbulence models would have increased the computational time with no relevant improvement of the results. Therefore, the K - Epsilon model was used as it is numerically robust, computationally efficient and it provides good accuracy. This strategy was used by the Hydraulic Research Group of the University of Naples for many analyses confirming the solution accuracy. Thus, the adopted model for simulations is the standard $k - \varepsilon$ (and not others RNG, LES, or DES etc.) because, in this field, results are faithful at the real phenomenon.
The standard k − ε model is based on the following two equations:

$$\frac{\vartheta}{\vartheta t}\int_{\Omega(t)}\rho k d\Omega+\int_{\sigma}\rho\big((v-v_{\sigma})n\big)k d\sigma=\int_{\sigma}\left(\mu+\frac{\mu_t}{\sigma_k}\right)(\nabla k n)n d\sigma+\int_{\Omega}(G_t-\rho\varepsilon)d\Omega \tag{1.5}$$

$$\frac{\vartheta}{\vartheta t}\int_{\Omega(t)}\rho \varepsilon d\Omega+\int_{\sigma}\rho\big((v-v_{\sigma})n\big)\varepsilon d\sigma=\int_{\sigma}\left(\mu+\frac{\mu_t}{\sigma_\varepsilon}\right)(\nabla \varepsilon n)n d\sigma+\int_{\Omega}\left(c_1 G_t\frac{\varepsilon}{k}-c_2\rho\frac{\varepsilon^2}{k}\right)d\Omega \tag{1.6}$$

with $c_1 = 1.44$, $c_1 = 1.92$, $\sigma_k = 1$, $\sigma_\varepsilon = 1.3$; where σ_k e σ_ε are the turbulent kinetic energy and the turbulent kinetic energy dissipation rate Prandtl numbers.
The turbulent kinetic energy, k, is defined as:

$$k=\frac{1}{2}(v'\cdot v') \tag{1.7}$$

with v' being the turbulent fluctuation velocity, and the dissipation rate, ε, of the turbulent kinetic energy is defined as:

$$\varepsilon=2\frac{\mu}{\rho}\left(\overline{S'_{ij}S'_{ij}}\right) \tag{1.8}$$

in which the strain tensor is:

$$S'_{ij}=\frac{1}{2}\left(\frac{\vartheta u'_i}{\vartheta x_j}+\frac{\vartheta u'_j}{\vartheta x_i}\right) \tag{1.9}$$

with u'_i (i = 1, 2, 3) being components of v'.
The turbulent viscosity μ_t is calculated by:

$$\mu_t=\rho C_\mu\frac{k^2}{\varepsilon} \tag{1.10}$$

with $C_\mu = 0.09$.
The turbulent generation term G_t can be expressed as a function of velocity and the shear stress tensor as:

$$G_t=-\rho\overline{u'_i u'_j}\frac{\vartheta u'_i}{\vartheta x_j} \tag{1.11}$$

where $\tau'_{ij} = \rho\overline{u'_i u'_j}$ is the turbulent Reynolds stress, which can be modelled by the Boussinesq hypothesis:

$$\tau'_{ij}=\mu_t\left(\frac{\vartheta u_i}{\vartheta x_j}+\frac{\vartheta u_j}{\vartheta x_i}\right)-\frac{2}{3}\left(\rho k+\frac{\vartheta u_k}{\vartheta x_k}\right)\delta_{ij} \tag{1.12}$$

A real fluid model based on the work of Singhal is implemented allowing the calculation of cavitation effects; when pressure in a specified zone of the fluid domain falls below the saturation pressure. Vapor bubbles form and then collapse as the pressure rise again.
Many physical models for the formation and transport of vapor bubbles in the liquid are available in literature, but only few computational codes offer robust cavitation models. This is due to the difficulty to handle gas/liquid mixtures with very different densities. Even small pressure variations may cause numerical instability if they are not optimally treated.
The cavitation model implemented in PumpLinx® uses the following equations (Rayleigh – Plesset):

$$\frac{\vartheta}{\vartheta t}\int_{\Omega(t)}\rho f d\Omega+\int_{\sigma}\rho\big((v-v_{\sigma})n\big)f d\sigma=\int_{\sigma}\left(D_f+\frac{\mu_t}{\sigma_f}\right)(\nabla f n)\,d\sigma+\int_{\Omega}(R_e-R_c)d\Omega \tag{1.13}$$

where D_f is the diffusivity of the vapor mass fraction and σ_f is the turbulent Schmidt number. In the present study, these two numbers are set equal to the mixture viscosity and unity, respectively. The vapor generation term, R_e, and the condensation rate, R_c, are modelled as:

$$R_e=C_e\frac{\sqrt{k}}{\sigma_l}\rho_l\rho_v\left[\frac{2}{3}\frac{(p-p_v)}{\rho_l}\right]^{\frac{1}{2}}(1-f_v-f_g) \tag{1.14}$$

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$$R_c = C_c \frac{\sqrt{k}}{\sigma_l} \rho_l \rho_v \left[\frac{2}{3}\frac{(p - p_v)}{\rho_l}\right]^{\frac{1}{2}} f_v \tag{1.15}$$

in which the model constants are $C_e = 0.02$ and $C_c = 0.01$.
The final density calculation for the mixture is done by:

$$\frac{1}{\rho} = \frac{f_v}{\rho_v} + \frac{f_g}{\rho_g} + \frac{\left(1 - f_v - f_g\right)}{\rho_l} \tag{1.16}$$

The fluid model accounts for liquid compressibility. This is critical to accurately model pressure wave propagation in liquid. The liquid compressibility is found to be very important for a high-pressure system and the systems in which water hammer effects are relevant.

Designed Geometry

The innovative designed gerotor pump is shown in Figure 8, where the adjustment and the fixed elements are clearly shown. The pump geometry has been fixed after an optimization progress looking at the timing of both ports and the interfaces with the contact points between the inner and the outer rotors. However, the optimization progress will be completed in the task 2.4 where the final geometry will be designed.

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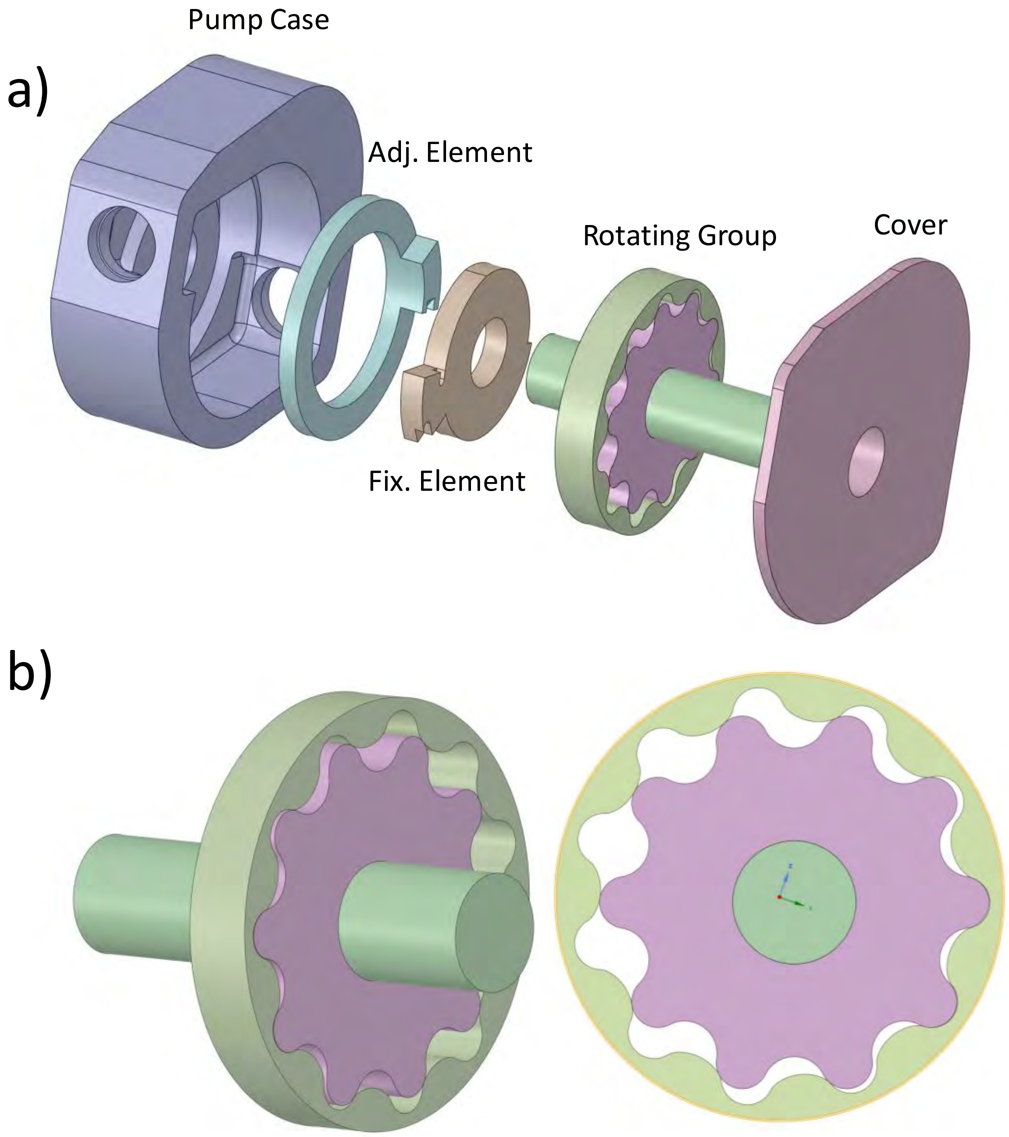

Figure 8 – Main gerotor pump, first geometry: (a) overall view; (b) rotors.

- *Grooves optimization*

Considering the fact that the operating oils have a higher bulk modulus, the distance between the inlet of the delivery port and the outlet of the suction ones should be equal to the overall length of the pocket in the maximum displacement configuration. If the distance is smaller than the pocket length (so-called overlapped condition), the delivery and suction port are passed through each other so the volumetric efficiency of the gerotor pump is degraded due to the delivery flow loss from the delivery port to the suction one. On the other hand, if the distance becomes longer (so-called underlapped condition), the pressure of the chamber can be seriously increased.

This condition should be respected also in delivery-to-suction side; when the pocket is in the minimum displacement configuration (Figure 9a).

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In a gerotor pump relief grooves have been included to increase the opening area of the ports. Without grooves, towards the end of the suction port, the rotor pocket is connected to the suction through a very small area (Figure 9a), as well as at the beginning of the delivery port, when the fluid suddenly passes from low to high pressure. Therefore, grooves are installed in order to extend partial outlet of the suction port, to increase the filling of the pocket (when its volume is still expanding), and inlet of the delivery port, to smoothly variate the pressure level and reduce outlet pressure ripples.
To avoid overlapping and underlapping of the chambers, the grooves final section should be positioned on the edge of the pocket when it is in the maximum displacement configuration (Figure 9b).

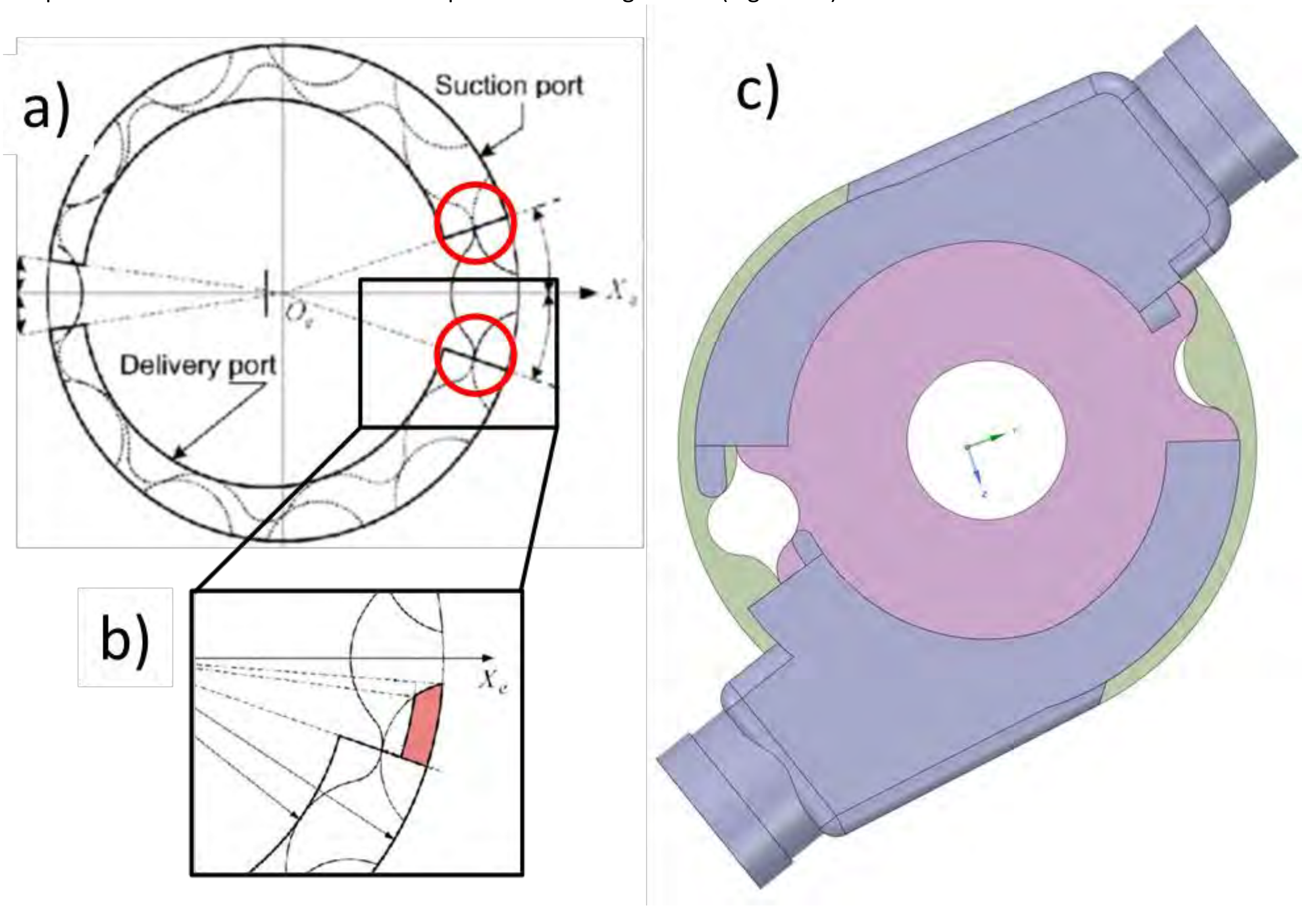

Figure 9 – Gerotor pump: (a) optimum position of suction and delivery chambers; (b) relief groove should be tangent to the edge of the pocket

In figure 10, the grooves are also results of an accurate analysis on the contact points between the inner and the outer rotors. Those points, in the condition of maximum displacement, define the area underlined with the red circles in Figure 9 a).
In the designed geometry the points are in fact located in correspondence of the end of the suction port and the beginning of the delivery one (Figure 9).

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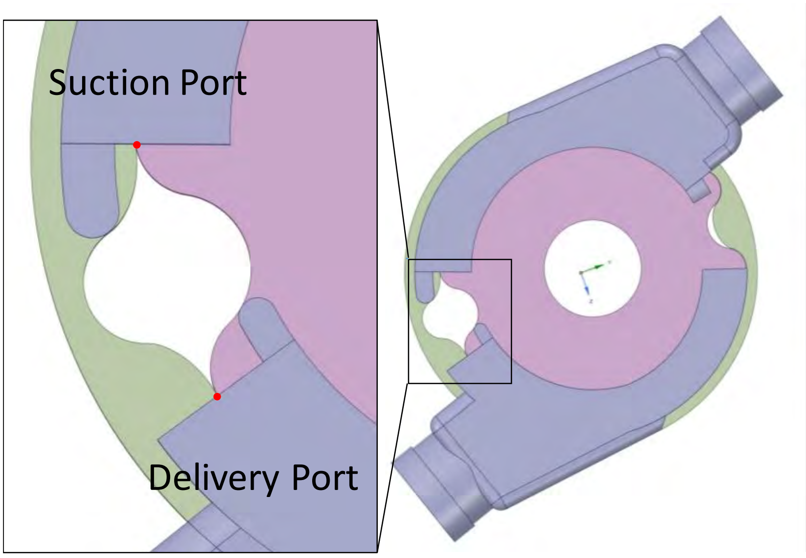

Figure 10 – Gerotor pump: Position of the contact points

The three-dimensional CFD Model

The geometry already presented in the previous section has been post processed with a dedicated code called ANSYS - SpaceClaim® to extract the fluid volume to be meshed (figure 11) . The fluid volume, shown below, consists of both ports and the pumping volume between rotors.

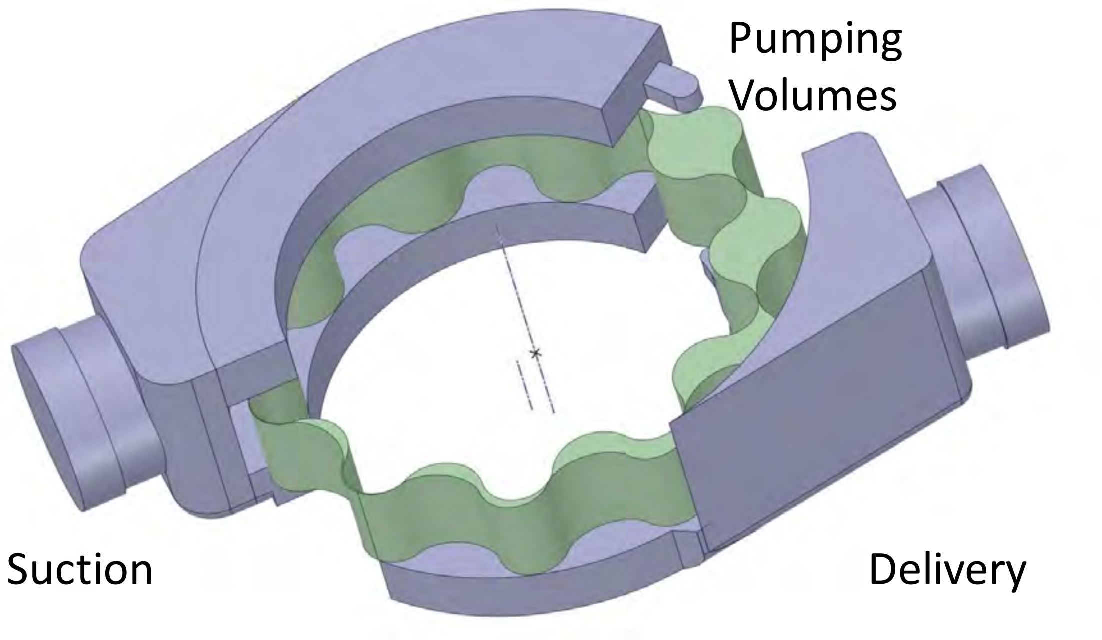

Figure 11 – Extracted fluid volume of the Devils Pump

The geometry obtained has then been meshed with PumpLinx® grid generator using the already described body-fitted binary tree approach.

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Figure 9 shows the binary tree mesh of the pump. In the boundary layer, the binary tree approach can easily increase the grid density on the surface without excessively increasing the total cell count. In the regions of high curvature and small details, the grid has been subdivided and cut to conform to the surface. Each part of the pump has been meshed separately.
The fluid volume of the ports (both suction and delivery ports) have been meshed with the PumpLinx® general grid mesher while the pumping fluid volume has been meshed using the pump template mesher also available in PumpLinx®.
A maximum cell size of 0.01 has been chosen, where no cell in the volume can have a cell side larger than the maximum cell size. The minimum cell size has been fixed at 0.0004. The minimum cell size is a parameter used to limit how small cells can be in attempting to resolve the geometry using the general mesher. No cell in the volume can have a cell side smaller than the minimum cell size. The cell size on surfaces has also been fixed at 0.005. This parameter is used to control the size of the cells for all surfaces of a mesh volume.
PumpLinx® uses a pure Eulerian approach and deform the mesh by squeezing and expanding the cells in the valve region. In order to obtain the pressure ripple two volumes have been add and connected to the ports.
The model consists of 1.1 M 3D cells and includes all the leakages.

Figure 12 – Three-dimensional mesh of the Devils pump

Before running the simulation, boundary conditions have been fixed.
The BCs are listed by following:

- Pressure at the inlet port: *1.1 barA*,
- Pressure at the outlet port: *17 barA*,
- Oil temperature: *127°C*,
- Pump rotating speed: *5000rpm*,
- Position of the adjustment element:
    - At the maximum displacement of the pump [0 deg],
    - At 40 deg,
- Working fluid is the *Oil MIL 23699*.

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The working fluid, as said, is Oil MIL 23699; whose properties change with temperature as described in table 10. These values are implemented in the CFD model, so the simulation can change the fluid properties with operating temperature.

	Temp.	Density	Cinematic Viscosity	Dynamic Viscosity	Thermal Conductivity	Specific heat
	[°C]	[kg/m3]	[cSt]	[Pa.s]	[W/mK]	[J/kgK]
	-50	1044.895	32553	34.036	0.497	1469.835
	-40	1037.776	10132	10.522	0.497	1517.368
	-30	1030.657	3503.61	3.613	0.496	1564.901
	-20	1023.538	1339.28	1.372	0.496	1612.434
	-10	1016.419	563.091	0.573	0.495	1659.967
	0	1009.3	259.09	0.262	0.494	1707.5
	10	1002.181	128.81	0.13	0.493	1755.033
	20	995.062	70.46	0.07	0.491	1802.566
	30	987.943	41.23	0.041	0.49	1850.099
	40	980.824	25.88	0.025	0.488	1897.632
	50	973.705	17.33	0.0169	0.486	1945.165
	60	966.586	12.32	0.0119	0.484	1992.698
	70	959.467	9.26	0.00889	0.4815	2040.231
	80	952.348	7.31	0.00697	0.479	2087.764
	90	945.229	6.04	0.00572	0.476	2135.297
	100	938.11	5.19	0.00488	0.474	2182.83
	110	930.991	4.62	0.0043	0.47	2230.363
	120	923.872	4.24	0.00392	0.467	2277.896
EOC	120.3	923.6584	4.2325	0.003912	0.46691	2279.322
CRZ	124.1	920.9532	4.1375	0.003813	0.46577	2297.385
MTC	127.9	918.248	4.0425	0.003715	0.46463	2315.447
MTO	128.8	917.6073	4.02	0.003691	0.46436	2319.725
	130	916.753	3.99	0.00366	0.464	2325.429
TOC	136.2	912.3392	3.8846	0.003548	0.46214	2354.899
	140	909.634	3.82	0.00348	0.46	2372.962
	150	902.515	3.72	0.00336	0.457	2420.495
	160	895.396	3.65	0.00328	0.453	2468.028

Table 10 Fluid Property

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The simulations have been run, as said, by setting the inlet and the outlet pressure with values of 1.1barA and 17barA respectability and with at the maximum displacement of the pump (zero rotation of the adjustment element).
Results are listed below:

- Average flow rate of 260 L/min;
- Average torque of 24.5 Nm

Three-dimensional visualization of the velocity distributions in the fluid volume are in line with the typical values available in literature (Figure 9).

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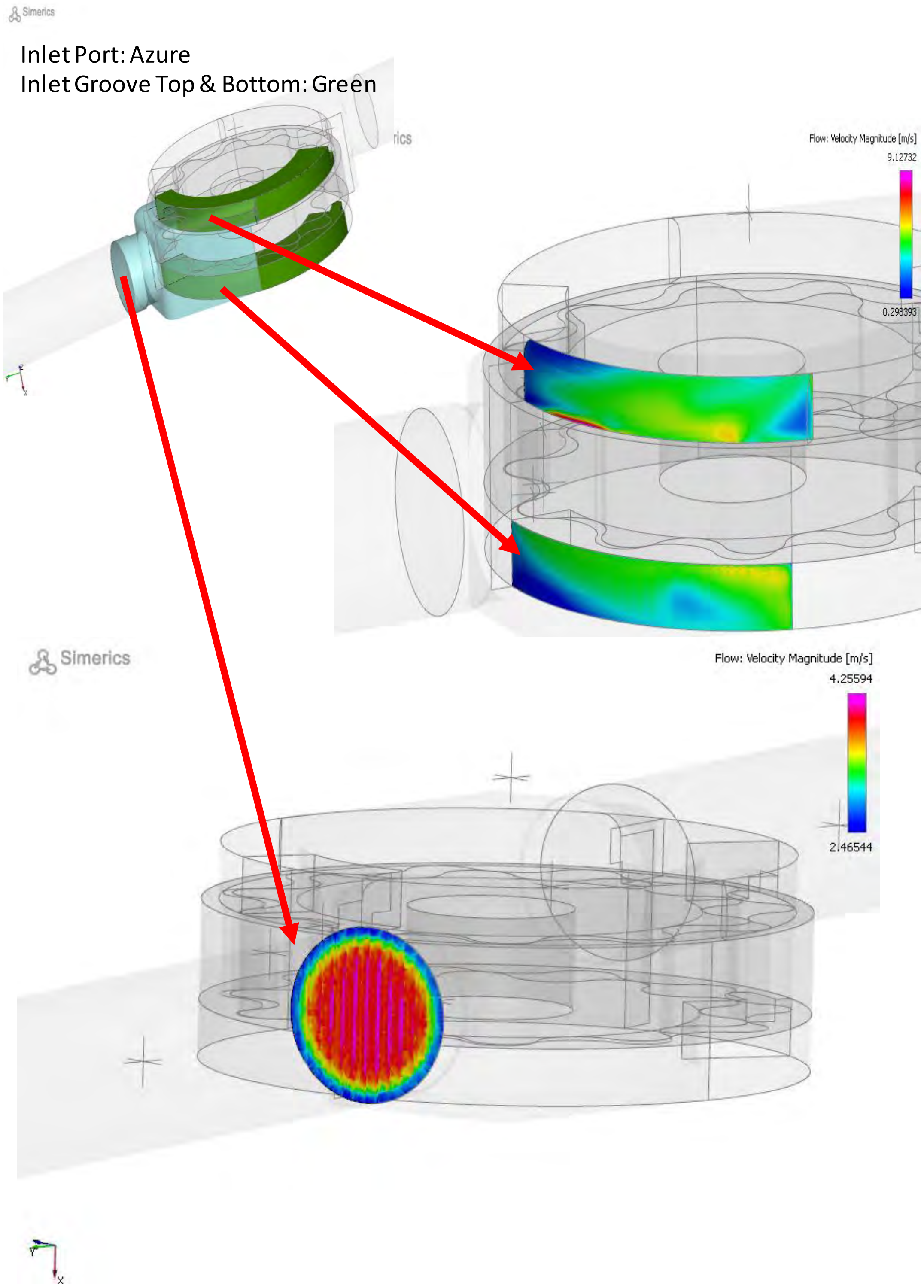

Figure 13 - Velocity magnitude at the inlet side

- *Rotation of the adjusted element of 40 degrees*

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Some simulations have been done on the pump by rotating the adjusted element of 40 degrees. The fluid domain in this working condition is shown in figure 14.

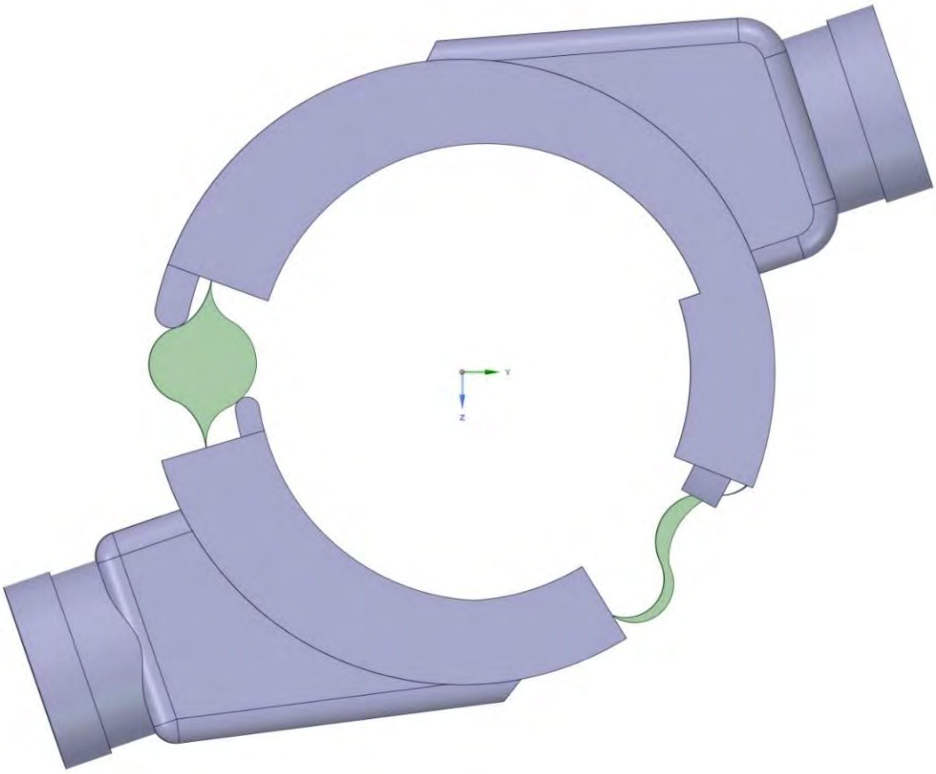

Figure 14 - Extracted fluid volume of the Devils pump - Rotation of the adjustment element of 40 degrees

The simulations have been run by setting the same working condition of the case at 0 degrees of rotation; results are listed below:

- Average flow rate of 210 L/min;
- Average torque of 28.0 Nm

This simulation has demonstrated that the innovative concept of variable displacement Gerotor pump works because the rotation of the adjustment element gives a reduction of the flow-rate with low influence on the absorbed torque. Since the research is still in progress, we are working to optimize the regulation system with the final target to reduce or maintain constant the absorbed torque with the rotation of the element itself. This activity is the topic of the task 2.4.

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7. Conclusions

The benefits of the variable flow control have been analysed and proved in this document through modelling approaches. It has been demonstrated that a variable flow control can give many benefits to the efficiency of the overall oil lubrication circuit of a turbofan. In fact, the capability of the system to adapt itself independently from the pump speed allows different strategies to optimize and control the complete lubrication circuit according to the engine request and with respect of the temperature limits. Simulations has demonstrated that the innovative concept of variable displacement Gerotor pump works because the rotation of the adjustment element gives a full control on the reduction of both flow-rate and pressure without any energy losses due to the "typical" recirculation of the flow through a relief valve. Therefore, in the deliverable 2.3, it has been defined a robust modelling methodology to design and optimize the Devils pump. The methodology consists in a close loop flowchart which include lumped parameter and three-dimensional CFD modelling techniques. As shown, the 3D CFD approach is useful to fully understanding the internal fluid dynamic behaviour of the component; the lumped parameter model of both the pump and the oil lubrication circuit is instead important to analyse the entire system. The sequence of steps define in this task will be used also in the task 2.4 to design the final geometry.

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