# POLITECNICO DI TORINO

Corso di Laurea Magistrale
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Modeling of External Gear Pumps for High Pressure Fluid Power Application


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# Contents

# Abstract

The external gear pumps are devices used in a wide range of applications that go from the industrial field where, for example, they can be used to transfer the fluid along a productive process system, or in mobile applications and off-road vehicles. They are valuable for their simplicity, robustness, the possibility of working with a wide range of fluid viscosity and the low cost.

Such a widespread usage underlines the advantage that a deeper control in the design process would lead.

The deeper control just mentioned, is brought through the model presented in this thesis. More precisely, in this thesis is first described HYGESim, an external gear pump lumped parameter model developed by the MAHA Fluid Power Research Center.

Additionally, are described the phases that characterize the design of the fluid dynamic aspect of an external gear pump, giving also an insight of the phenomena that characterize this kind of device.

Moreover, are presented other not closely fluid-dynamic linked but still essential aspects faced during the design phase of an external gear pump and improving of the model.

One of these aspects is the developing of a graphic user interface that extend the capability of using the model to every user. The other is the design of the pump in a 3D CAD ambient focusing on one of this specific project requirement: the integration of the hydraulic machine inside the electric machine that activate it.


# Intro

The internship I carried out and the thesis I developed were accomplished to the MAHA Fluid Power Research Center. The research center just mentioned is a lab that is part of the Agricultural and Biological Engineering department of Purdue University, located in West Lafayette (IN) United States. As the name suggests, the lab main research topic is the fluid power field. The staff of the lab consists of 27 students leaded by prof. Andrea Vacca with the help of a post-doctoral researcher Lizhi Shang. The whole team constitute the two main sections of the lab: the system section and the pump section.

The system section is in charge to model and develop or improve hydraulic system. This means make the system waste less energy and increase their efficiency and simplify them reducing the number of components and consequently the costs. Additionally, to improve the designed systems controllability are usually developed customized control systems and logics.

The pump section instead is divided in smaller teams and every teams operate in a specific field developing a model for a different type of pump. The pumps modelled are internal gear pumps, gerotors, piston pumps and external gear pumps. The aim of every team is developing a software lumped parameter model based that allow to predict the behavior of a specific pump. Also, every model is constituted of more modules with different utilities that allow to simulate a very wide range of features of a pump, from the geometric characteristics, to the noise emission. The skills and the knowledge behold by the members of the lab allow very important partnerships and research projects with many big companies as Bosh Rexroth, Parker, John Deere, Caterpillar, Danfoss, Casappa etc.

In the following pages I'm going to describe the activities that I carried out during my experience as a member of the external gear pump group.

# State of the art

In the first section will be described the state of the art of the hydraulic external gear machines EGM. This will allow the reader to better understand the results and the considerations that will be further analyzed. Below it's possible to see a section of an external gear machine.

The operating principle of an EGM is that through the rotation of the gears finite volumes of fluid are isolated and delivered from one port to the other port. The physiological structure of this kind of device allow them to operate as pumps or as motors. The delivery always happens through a rotation caused by the forces that act on the gears.

## External Gear Motors

If the external gear machine is a motor, as it's shown in the figure below, there is a first phase when the pump is first filled, and the pressure acting on the surface of the gears generate a resulting force that rotate the gears. After this phase the motion continue promoted by an inhomogeneous distribution of pressure. In the figure aside is possible to see how the high pressure in the inlet zone generates higher forces and torque on the gears respect to the low-pressure zone. For the sake of clarity in the just mentioned picture only some volumes are highlighted, but the same concept can be applied to all of them.

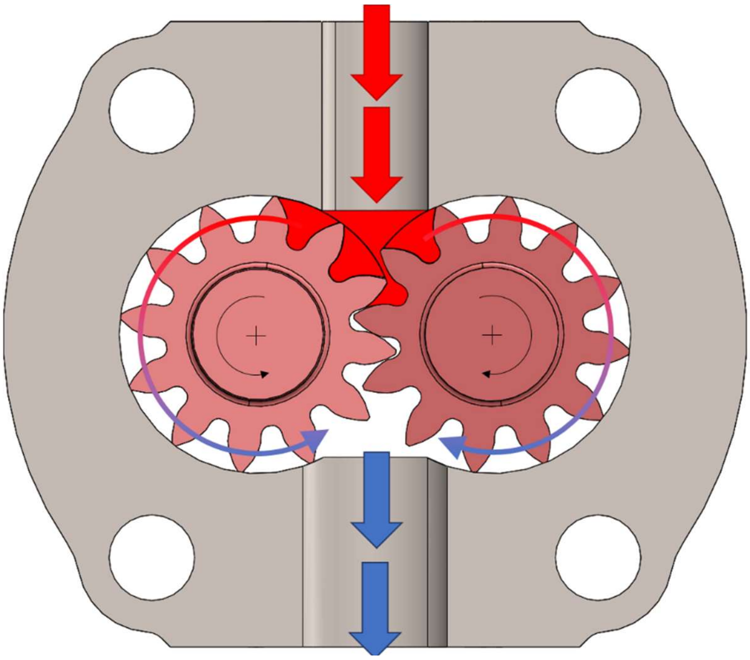


Figure 1 - External gear motor

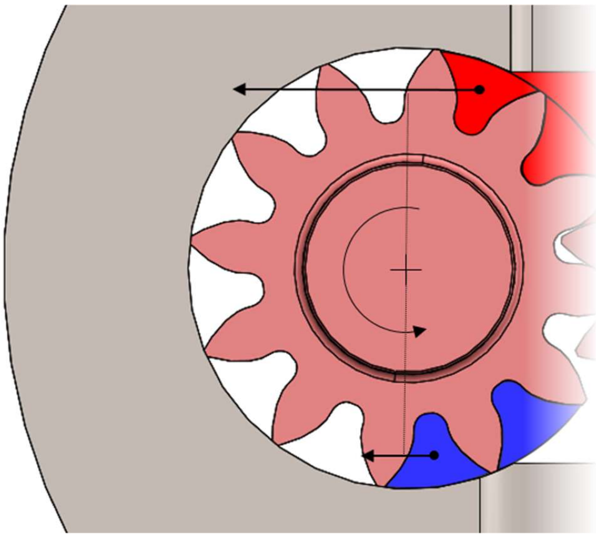


Figure 2 - Torque generated in external gear motor

The schemes related to the operating conditions of an external gear motor are shown below.

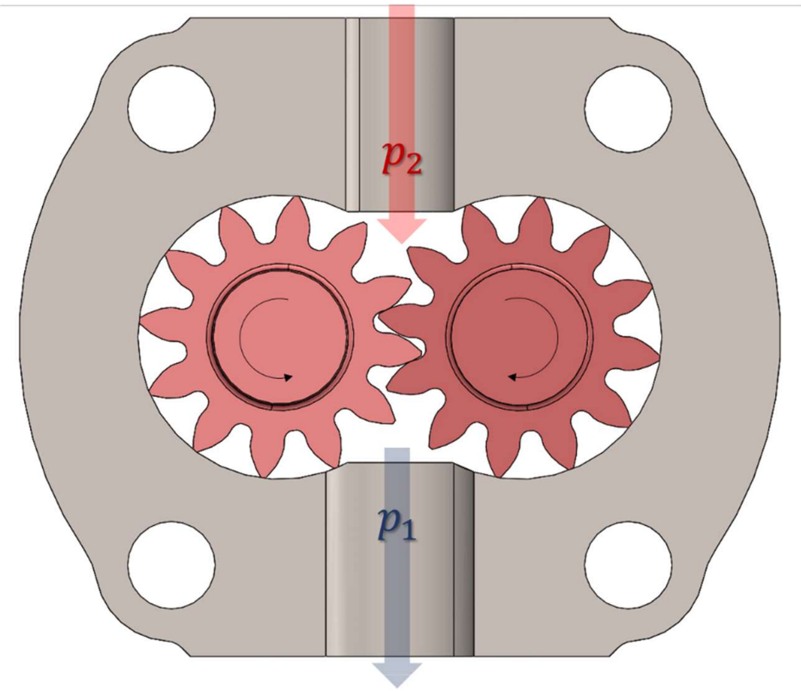



Figure 3 - Functional scheme of an external gear motor

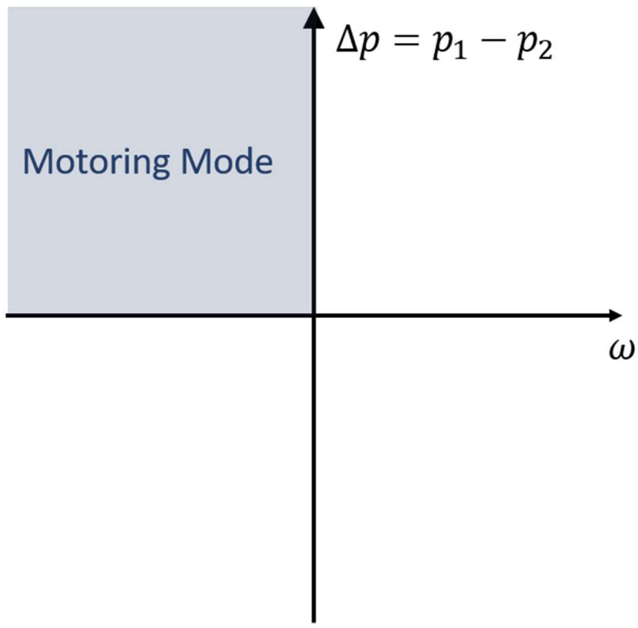



Figure 4 - Functional diagram of an external gear motor

## External Gear Pumps

If the external gear machine works as a pump the operating principle is different. Since in the further sections many considerations will be made about the pumping operations, before going deeper in the operating principle description some useful definitions are listed.

First, it’s possible to define a gear that receive the motion from a combustion engine or an electric motor and transmits it to the other gear. The gear that receive the motion from the motor is named Drive Gear, and on the other hand the gear that receive the motion is named Driven Gear.

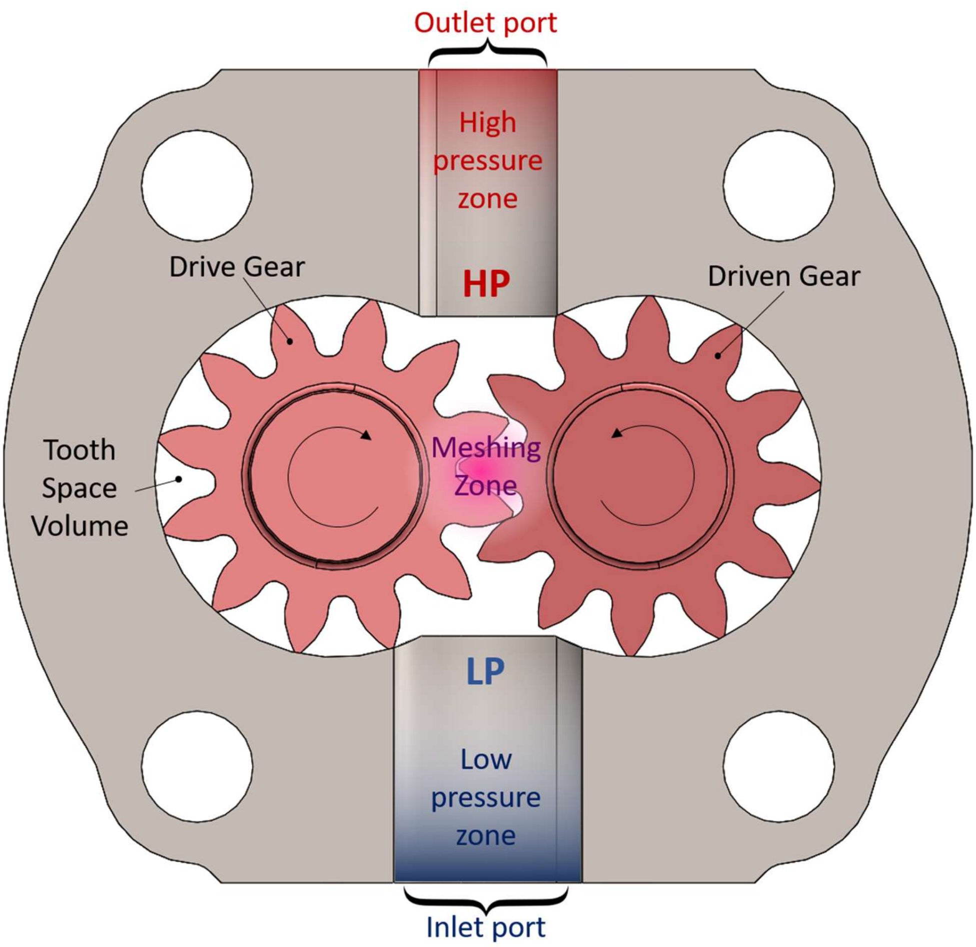



*Figure 5 - Fundamental definitions in an external gear pump*

Looking at the section in the figure above it’s possible to find a low-pressure zone and a high-pressure zone. The low-pressure zone is the zone of the pump connected through a channel to a low-pressure fluid source. The connection with the low-pressure source happen through a low-pressure port. In an actual case the low-pressure source can be a tank if the pump is mounted in an open center circuit, or an accumulator in case the pump is mounted in a closed center circuit.
The high-pressure zone is the zone of the pump where the fluid is delivered.

The delivery of the fluid usually faces some kind of resistance, and for this reason a high-pressure is built. In the actual case an actuator (cylinder or hydraulic motor) or sometimes an accumulator constitutes the just mentioned resistance. If an accumulator is connected to the outlet it’s more proper to consider the flow facing a high-pressure source than a resistance.

To a better understanding of the topic discussed further is important to define the Tooth Space Volume. The tooth space volume is the volume delimited by two adjacent teeth of a gear.

The gears mounted in external gear machines must be free of rotate incurring in the less friction as possible. For this reason, they are usually installed in the machine using roller bearings or journal bearings.

The operating principles of roller bearing is straight forward: rotating elements as spheres or cylinders are used to keep to rings concentrically allowing their relative rotation. The operating principle of the journal bearing instead is slightly different. There are no rotating elements, but just a bearing and a shaft separated by a fluid film. The relative rotation of the elements and the viscosity of the fluid avoid the contact between them. However, every time the pump faces a stop, the contact happens. For this reason, the journal bearing manufacture theory consists in the coupling of elements with different hardness. This allow a first phase of wearing that leads to a more stable phase that lengthen the component life.

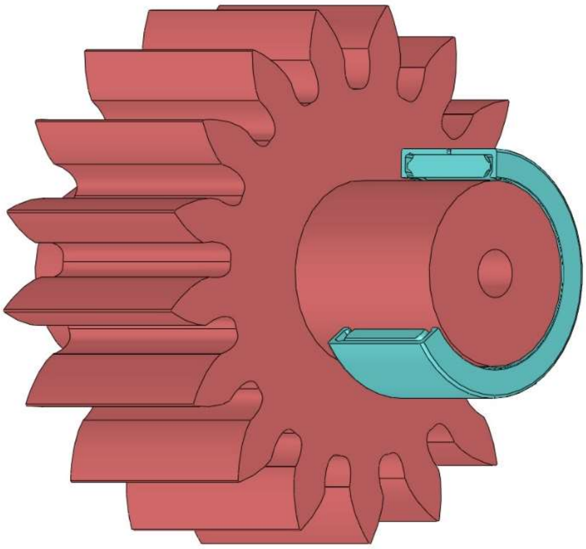


*Figure 6 - Example of a roller bearing*

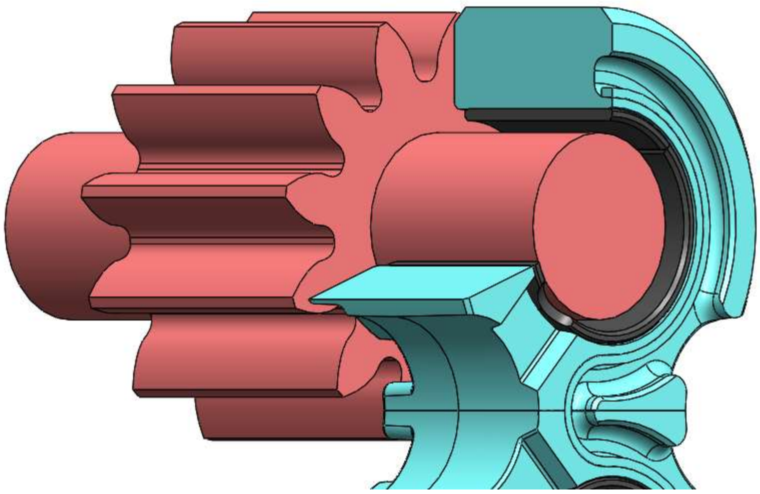


*Figure 7 - Example of a journal-bearing*

Another very important element for an external gear machine are the lateral elements. The first distinction is between plates or bushing. Substantially, the main differences between these two elements are the thickness and the capability of support the gears.

The plates are thin, and they can't bear the gears. Using the plates, the designer must design a casing and a lid of the pump in a fashion in which the bearing element can be mounted. This is usually accomplished through the machining of grooves in them.

The bushings are thicker, and they can host the bearing system. Is reminded to the reader that by bearing system is meant both journal bearing and roller bearing, depending on the application.

The lateral elements have other fundamental functions.

Looking at their faces, it's possible to see grooves machined on them. Depending on the face the grooves have a different purpose.

First, we can identify the pressure relieve face, as the surface in contact with the gears.

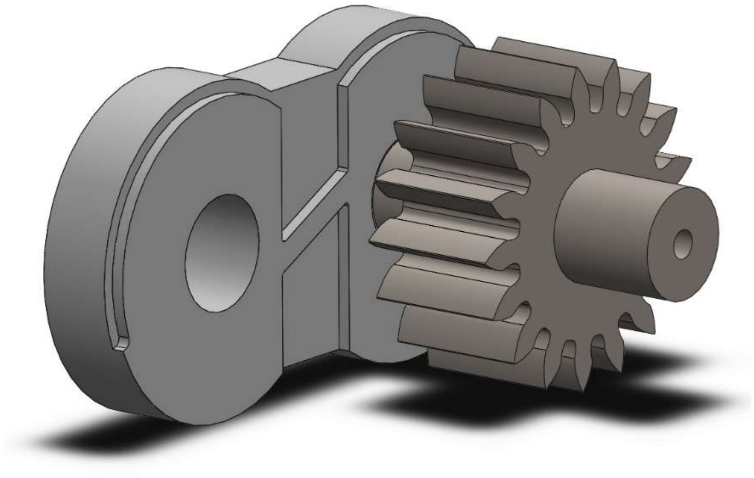


*Figure 8 - Coupling gear-bushing*

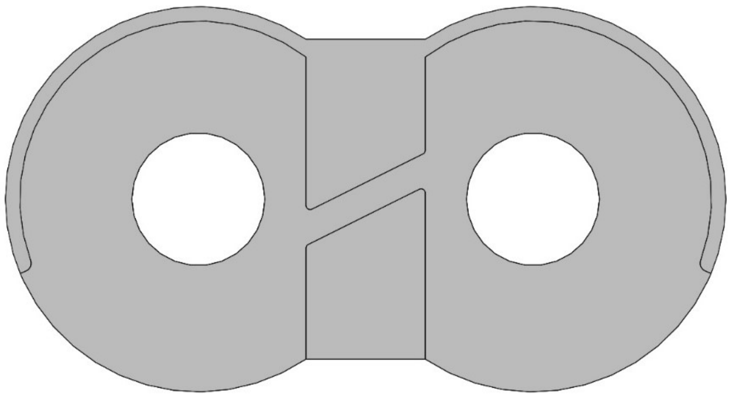


*Figure 9 - Pressure relieve face*

On the pressure relieve face there are two kind of grooves. The pressure relieves grooves, that give the name to this surface and the back-flow grooves.

The pressure relieve grooves are placed to avoid the generation of huge pressure in the meshing zone. This is possible because during the meshing, the tooth space volumes are not completely trapped between the teeth and the bushings but have a little connection with the inlet and/or outlet volume.

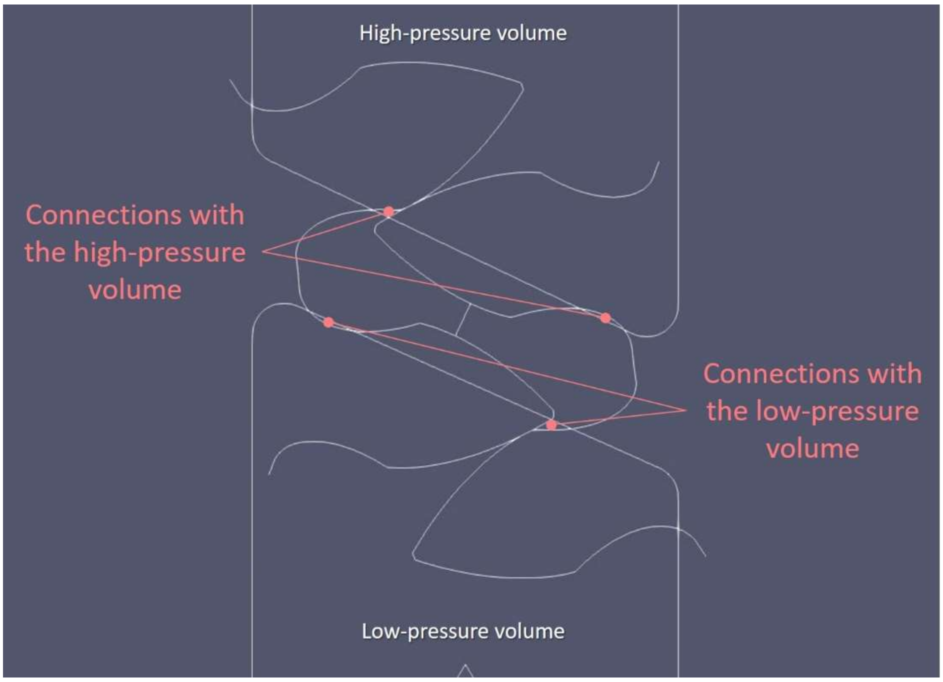



*Figure 10 - Pressure relieve grooves connections*

This small connection is shown in the figure above.

Even if the idea behind these grooves seems very simple, design a proper geometry is one of the most challenging phases. If the connection doesn't open enough the pressure in the meshing zone will reach too high values causing noise, vibrations, and the breaking down of the machine. If the connection areas are too high instead, the pump will accomplish very bad performances, and a phenomenon called cross-port flow will happen. This consists in the reversing of the flow direction due to the connection of the high-pressure volume with the low-pressure volume. This will also lead to an annoying behavior of the actuator that will cog.

The second kind of grooves noticeable on the pressure relieve side are the back-flow grooves.

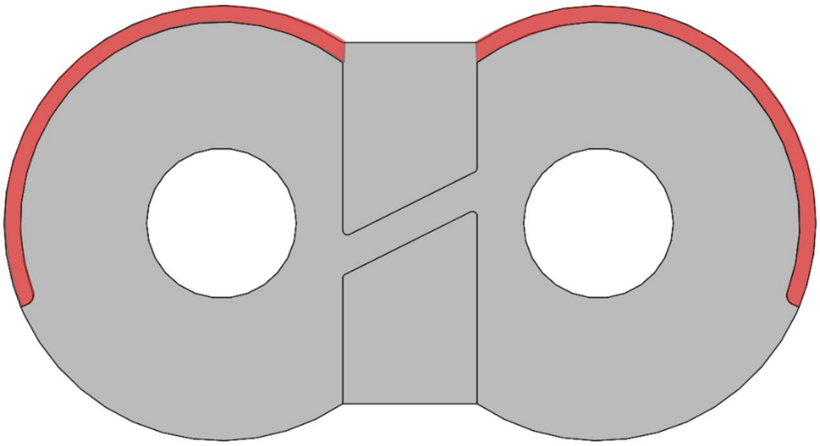


*Figure 11 - Back-flow grooves*

The function of these grooves is to allow a gradual pressurization of the Tooth Space Volumes before they actually reach the outlet volume. This is very important, and the positives led by them cannot be neglected. First, the gradual pressurization of the Tooth Space Volumes avoids the sudden generation of forces acting on the gears. This reduces mechanical stresses, vibration, and noise. Before introducing the second consequence of the Tooth Space Volume pressurization is reminded to the reader that the gears in an external gear machine are not perfectly fixed, but, because of the clearance between the bushing and the casing and the journal-bearing or roller bearing consequence, they can slightly move. This phenomenon is called micromotion. So, because of the clearances and the diffused pressurization the gears are pushed toward the low pressure zone, ensuring a better sealing effect of the transitioning Tooth Space Volume toward the inlet pressure and a worse sealing effect towards the outlet pressure. This guarantee a natural build up behavior of the Tooth Space Volume.

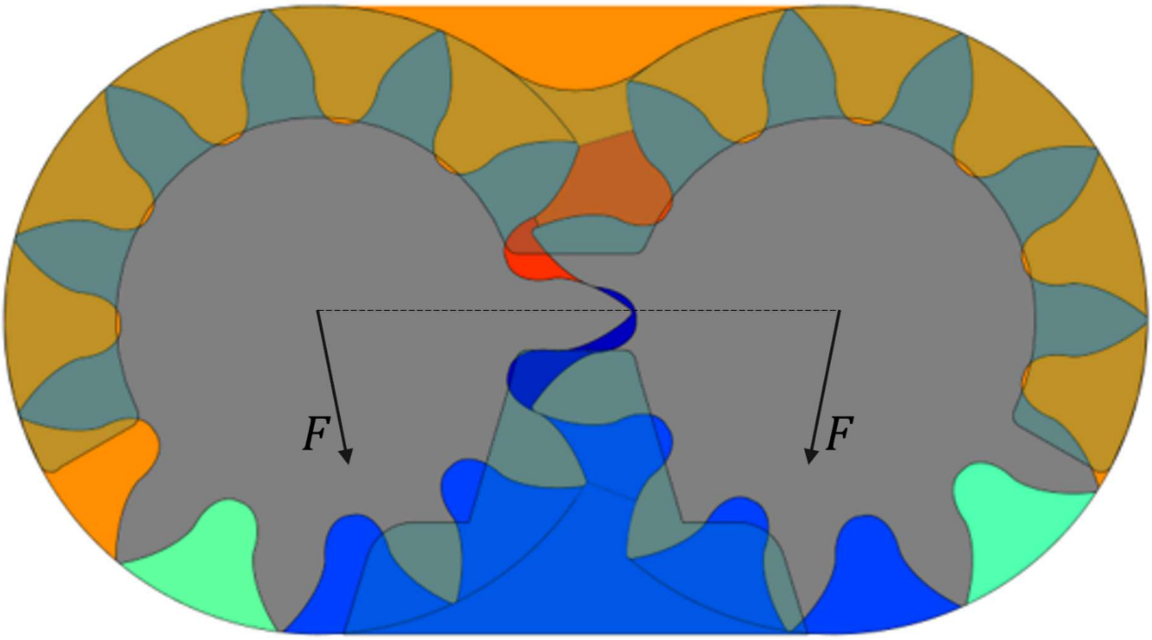



*Figure 12 - Effect of the unbalanced pressurization of the Tooth Space Volumes*

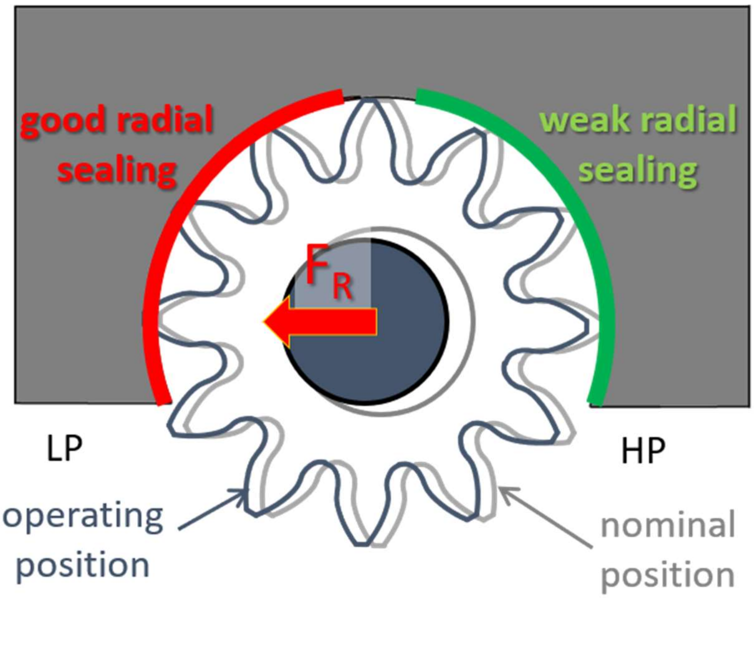



*Figure 13 - Consequences of the micromotion of the gears*

As last are shown the functional schematic and diagram of an external gear pump.

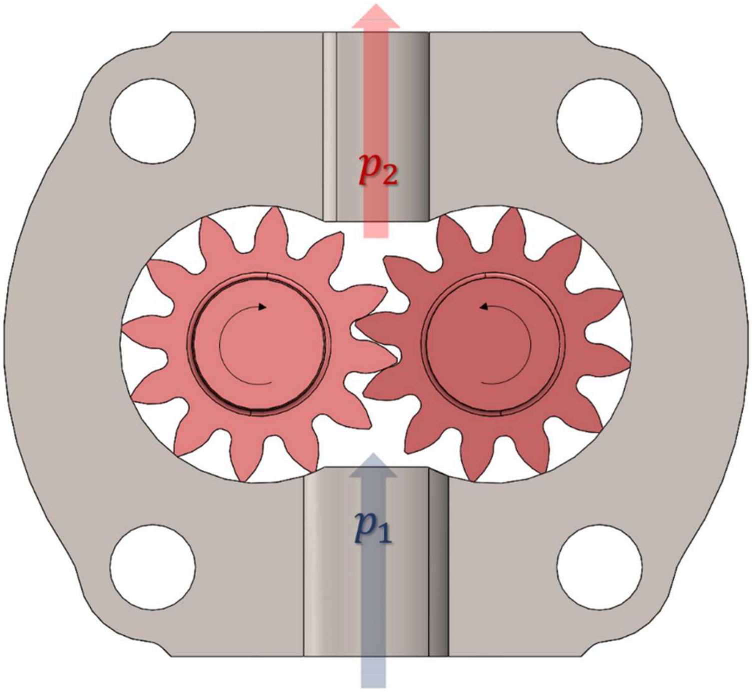



*Figure 14 -Functional schematic of an external gear pump*

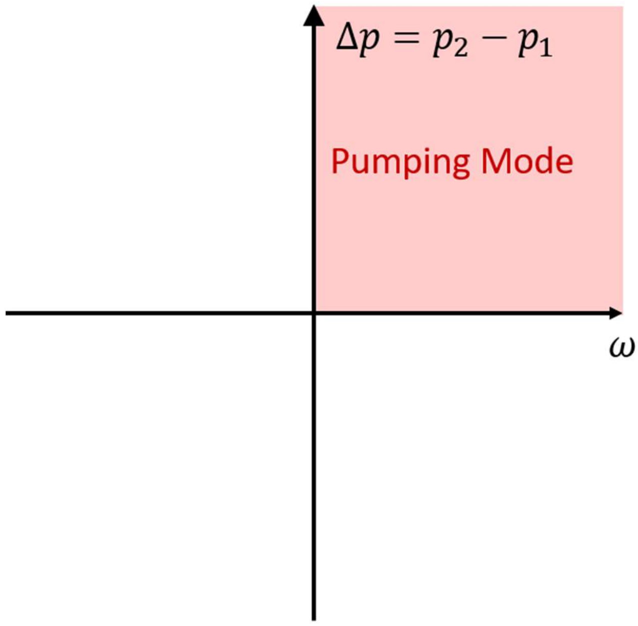



*Figure 15 - Functional diagram of an external gear pump*

### Two quadrants External Gear Machines

Since now the external gears machines has been described as motors or as pumps, but they can be designed to accomplish both functions. Essentially, they merge the functions of the two machines described above. Obviously, even if conceptually very simple, the design of such machines is way more complex than the design of a single function machine. Also, the capability of working in moth mode, lead to a trade-off of the performances.

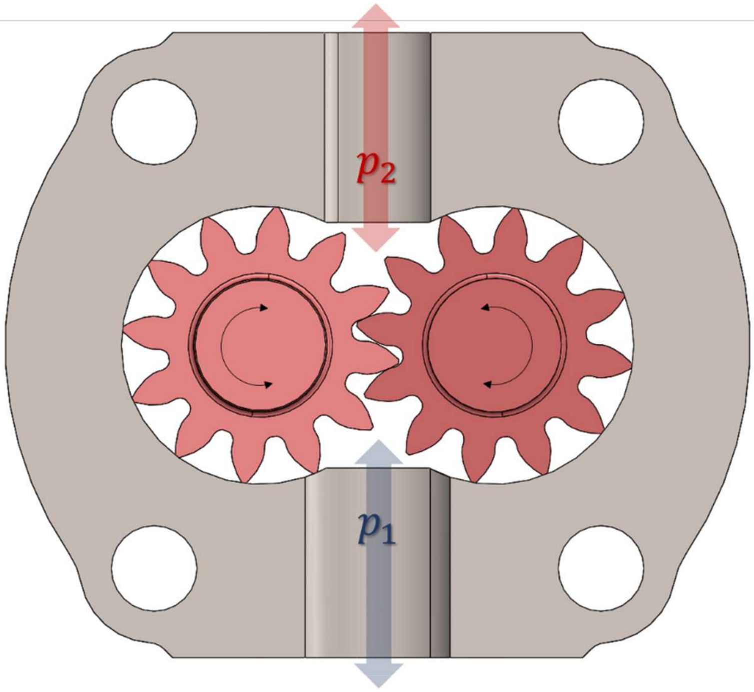



*Figure 16 - Functional schematic of a 2 quadrant gear machine*

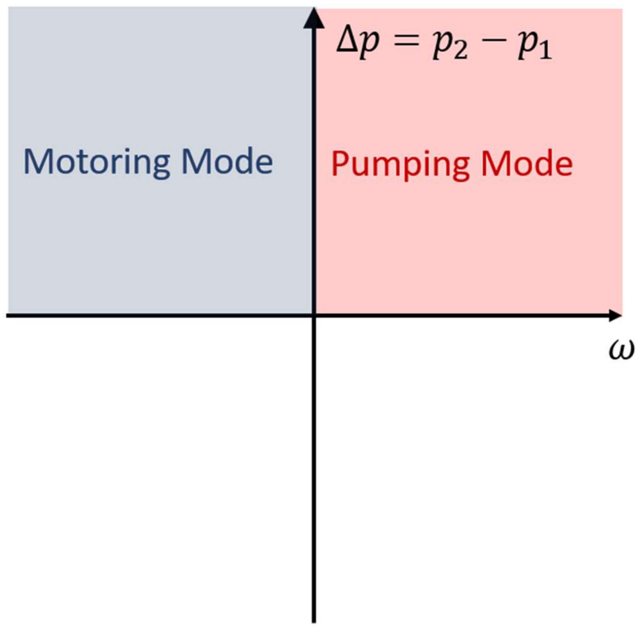



*Figure 17 - Functional diagram of a 2 quadrant gear machine*

### Four quadrants External Gear Machine

A four quadrant external gear machine can work as pump or as motor in both directions.
The machine presented in this thesis belongs to this category. Further will be also explained how the device exploits its features. Obviously to properly design this kind of machines a deep study of the gears profile and lateral elements geometries is needed, but there are other peculiarities that characterize it and clearly point out its nature.

First, in a normal pump the inlet port is larger than the outlet port in order to avoid cavitation. For the sake of clarity cavitation will be quickly described.
Even if it's not visible a small amount of air is soluble in oil. The amount depends on temperature and pressure. For example, higher pressure allows a higher percentage of air to dissolve in the oil, on the other hand lower pressure allows a lower percentage of air to dissolve in the oil. For this reason, when the oil sucked by the pump is subjected to severe depression it can release air that forms cavities that collapse right after their formation.
The formation and collapse of these air cavities in the liquid is called cavitation. Also, they can cavities generate vibration, noise and can damage mechanical components. Cavitation can be avoided or at least reduced increasing the inlet volume or increasing the inlet port area.

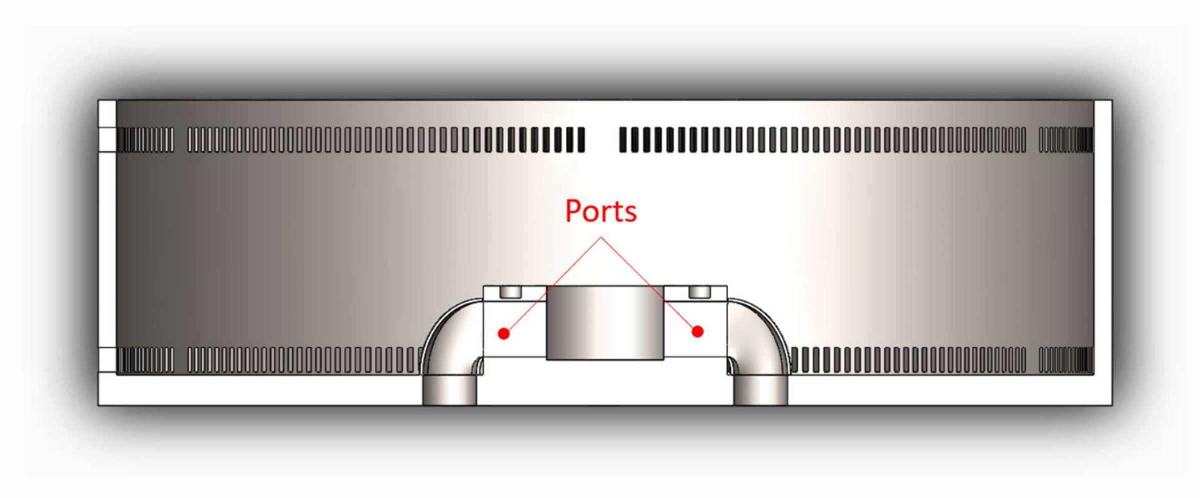



*Figure 18 - Example of ports of the same diameter in a four quadrants pump*

Second, the back-flow grooves aren't directly connected with any port.
The direct connection with a port would lead to a very good pressurization in the case in which the device is delivering toward that port, but it would lead to very bad performance in the case in which the fluid is delivered toward the other port. In fact, in the last case described the Tooth Space Volume would be very well connected to the low pressure volume until very few degrees before reaching the high pressure volume. This would completely disrupt the pumping mode in that direction.
Is also noticeable that in this case is not proper to speak about inlet or outlet port, because depending on the phase of the duty cycle the flow can enter the pump from a port or the other.
Below is shown an example of lateral element with grooves designed for the four quadrant mode.

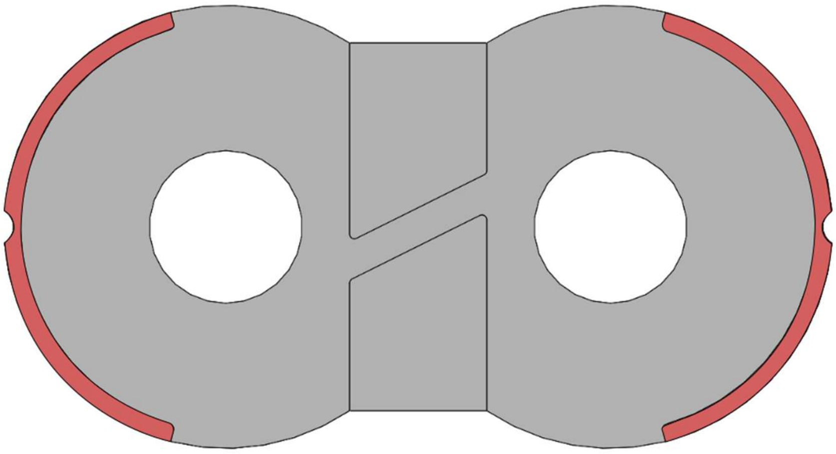


*Figure 19 - Four-quadrant machine lateral element*

As last is shown the functional schematic and functional diagram of the 4 quadrant device.

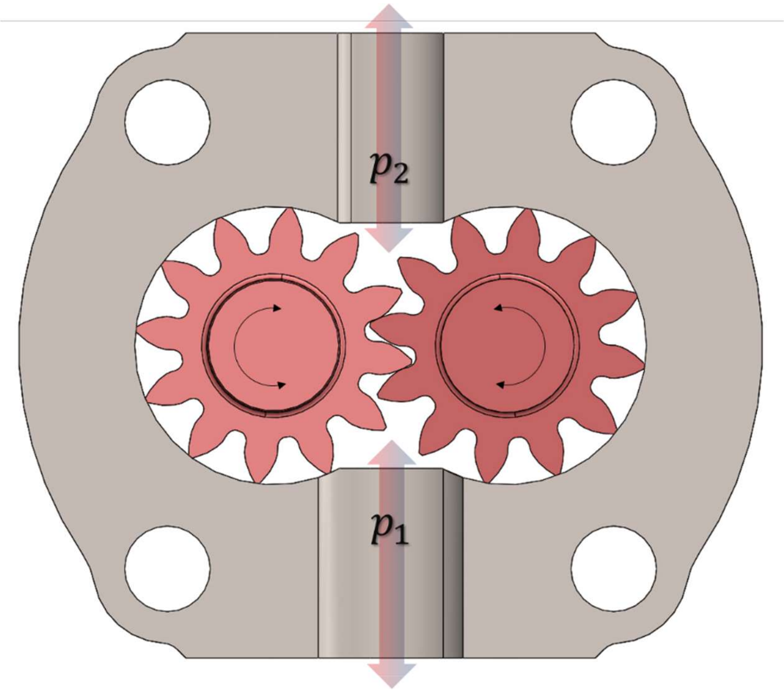



*Figure 20 - Functional schematic of a 4 quadrant external gear machine*

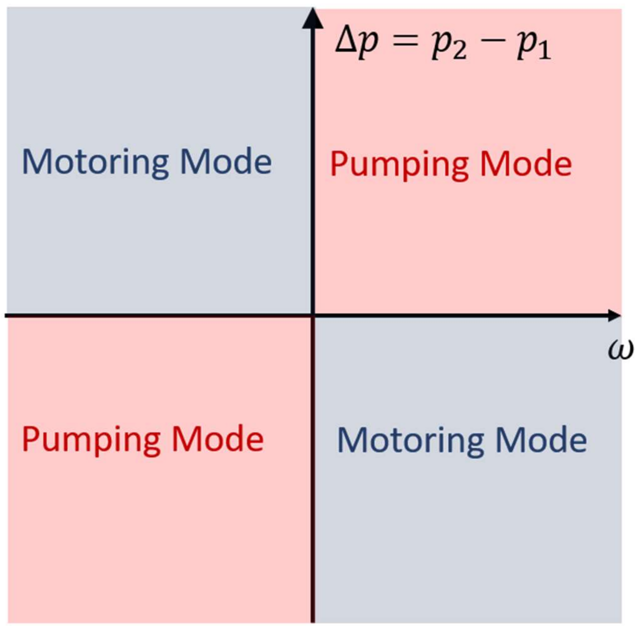



*Figure 21 - Functional diagram of a 4 quadrant external gear machine*

# HYGESim: HYdraulic Gear machine Simulator

To model the external gear machine, I used HYGESim. HYGESim is a lumped parameter model first developed by prof. Andrea Vacca himself, and then by his students during the past years. Even if the CFD models provide information for local fluid flow phenomena, a successful simulation of an external gear pump requires high mesh density and consequently a high computational cost. Also, some important dynamic behaviors, such us micromotion and deformation, are not easily captured by the CFD models. Additionally, these aspects imply a non-uniform distribution of the leakage gap geometry, which are sometimes crucial for recognize the gear pump performance. In contrast, lumped-parameter models. Are order of magnitude faster than CFD models and are suitable to capture those dynamic behavior of external gear pump.

The model is a structure in modules. Every module allows the calculation of a category of characteristics of the pump. The submodules are schematized below.

Also, to understand the work done, it's important to remember that the model was developed in AMESim ambient, and now the team is transitioning toward C++ coding language.

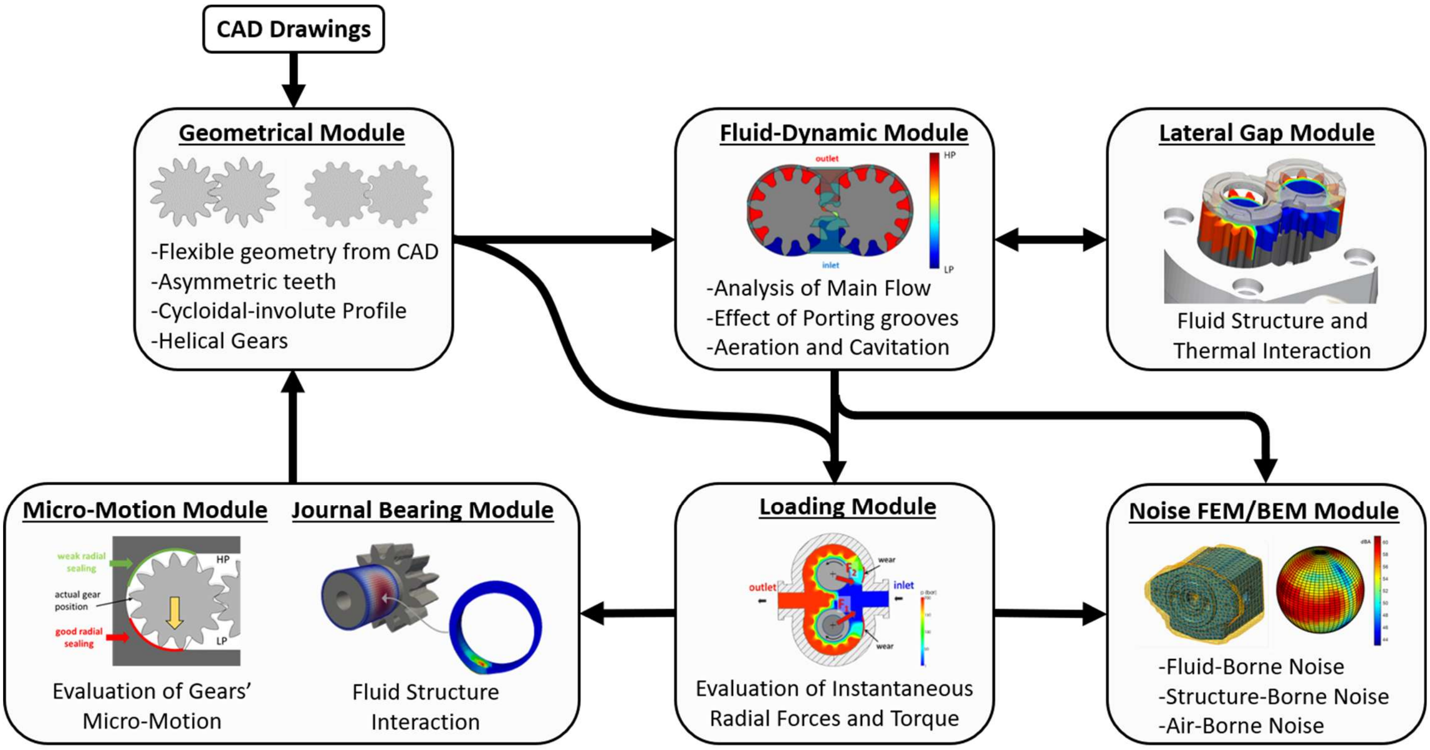



*Figure 22 - HYGESim schematic*

## Preliminary processing

The first step to face is defining the geometry features of a specific pump.
The CAD drawings can be elaborated to extrapolate the geometrical module needs. The data needed are the profile of the tooth and the location of the grooves and the inlet and outlet volumes respect to the gear.
Additionally, for simplicity, for designing purpose, or for optimization purpose a gear generator can be used. The Gear Generator is a software developed in Mathworks Matlab ambient. The operating principle is that with the input parameters the user defines the geometry of the cutter rack, and through a rotation-transfer motion of the cutter rank the tooth geometry is defined.

The parameters needed by the gear generator are:

- $n$: Number of teeth
- $m$: Module
- $x$: Correction factor
- $\alpha_d$: Drive pressure angle
- $\alpha_c$: Coast pressure angle
- $h_a$: Addendum
- $h_d$: Dedendum
- $r_d$: Drive fillet radius
- $r_c$: Coast fillet radius

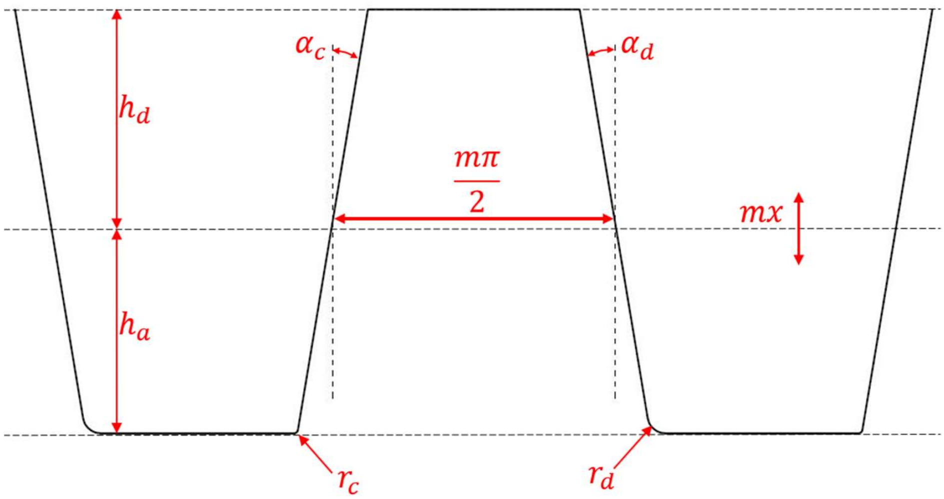



*Figure 23 - Cutter rack inputs*

Furthermore, it allows to calculate some parameter specific for the studied gear set and depending on the interaxis.  Examples are the pressure angle, the contact ratio, the root radius, outer radius, or the dual-flank interaxis.
Is reminded to the reader that the dual flank interaxis is the distance between the meshing gears when the contact between them happen through both flank of the tooth.

Generally, is preferred to avoid the dual flank contact because such a condition isolate the Tooth Space Volume promoting the generation of very high pressure.

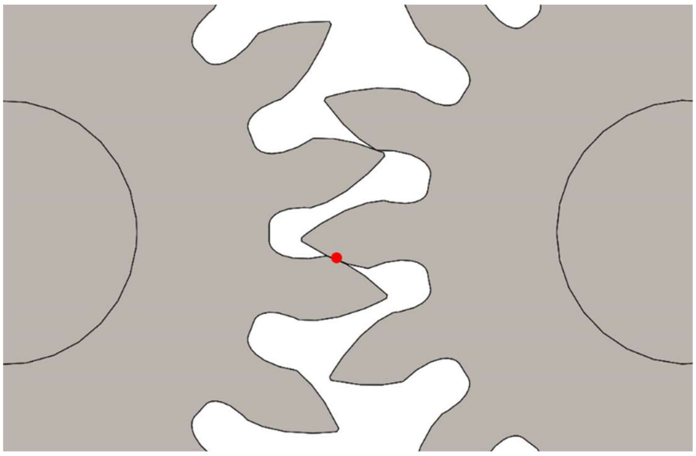


Figure 24 - Single flank meshing

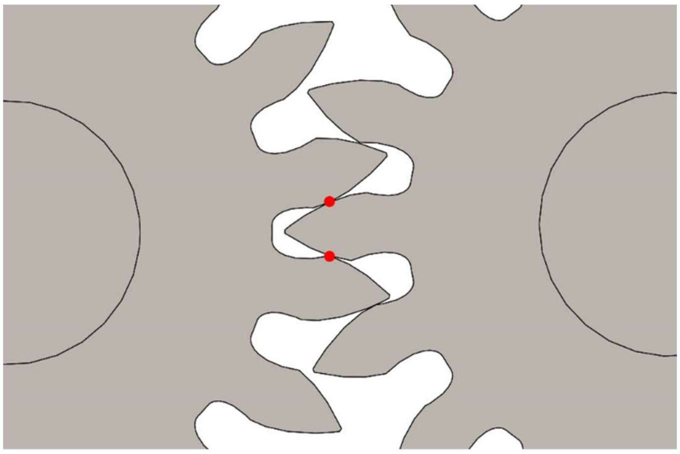


Figure 25 - Dual flank meshing

Below is shown the interface of the gear generator.

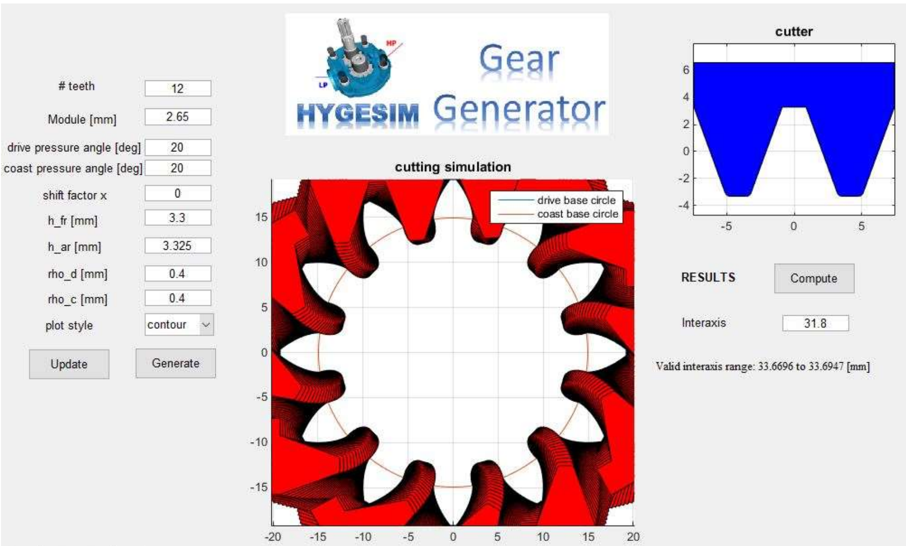



Figure 26 - GUI Gear Generator

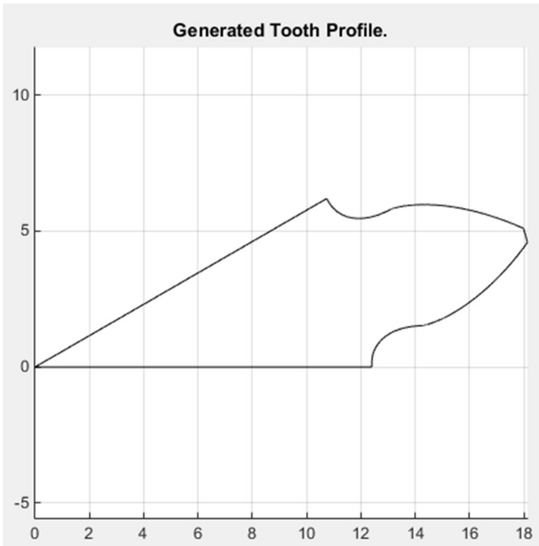



Figure 27 - Tooth profile generated by the Gear generator

## Geometrical Module

The geometrical module uses the tooth profile and the geometries of the casing, the bushing, the relieve grooves and the back-flow groove to calculate all the parameters that will be used in the fluid-dynamic module.

In the further section will be shown examples of the parameters calculated by the Geometry module. To fully understand how the model works it’s important to highlight that the following features are not constant during the operating conditions. The variation of these values depends on the rotation angle of the gears and on the interaxis.

For this reason, the geometry module generates a ".txt" file where the parameter described below are expressed as a function of the rotation angle. Then, this procedure must be repeated for more interaxis. It's suggested to generate them for 4 interaxis. In the chart below is explained the logic behind the four interaxis. In such a way the fluid-dynamic module can interpolate for a specific rotation angle between the two closer values to the calculated interaxis.

Every interaxis calculated is the result of the linear combination between interaxis and journal-bearing clearance. In this way it's possible to reach the minimum and maximum allowed interaxis, passing through a couple of intermediate point to increase the interpolation precision.

| **Interaxis 1** | **Interaxis 2** | **Interaxis 3** | **Interaxis 4** |
|---|---|---|---|
| Nom. Interaxis – 2*c | Nom. Interaxis – 1*c | Nom. Interaxis + 1*c | Nom. Interaxis + 2*c |

To better understand the plots shown further, in the figure below is shown the reference system.

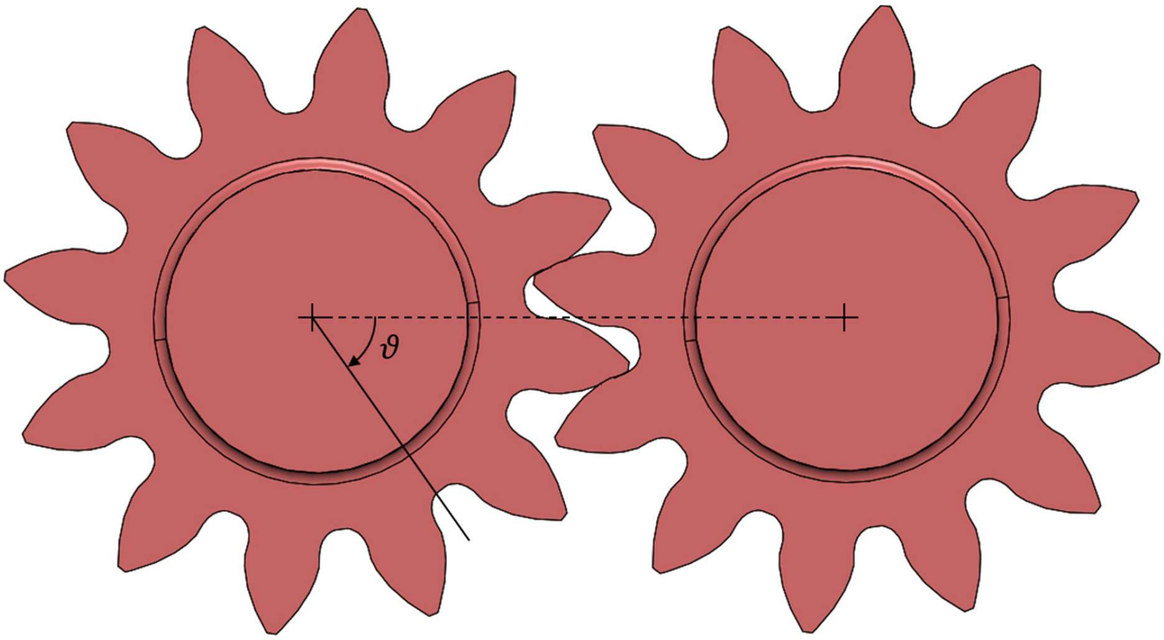



*Figure 28 - Reference system*

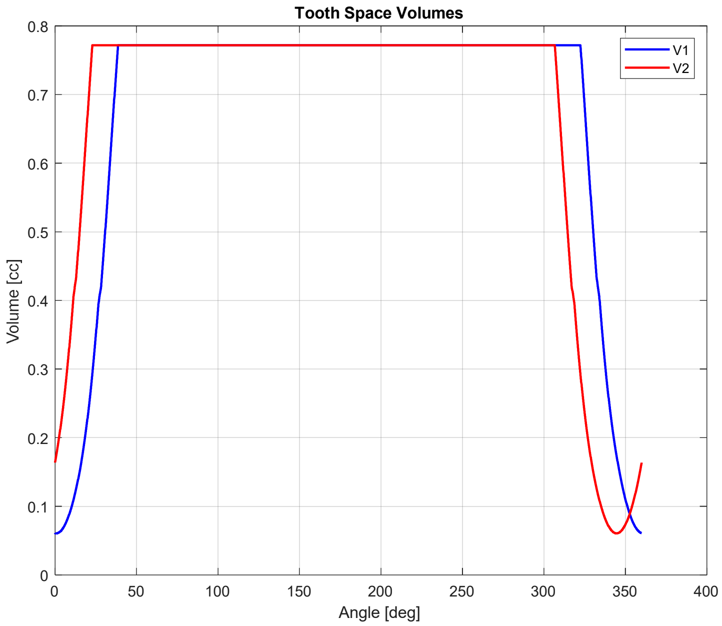



*Figure 29 - Tooth Space Volumes*

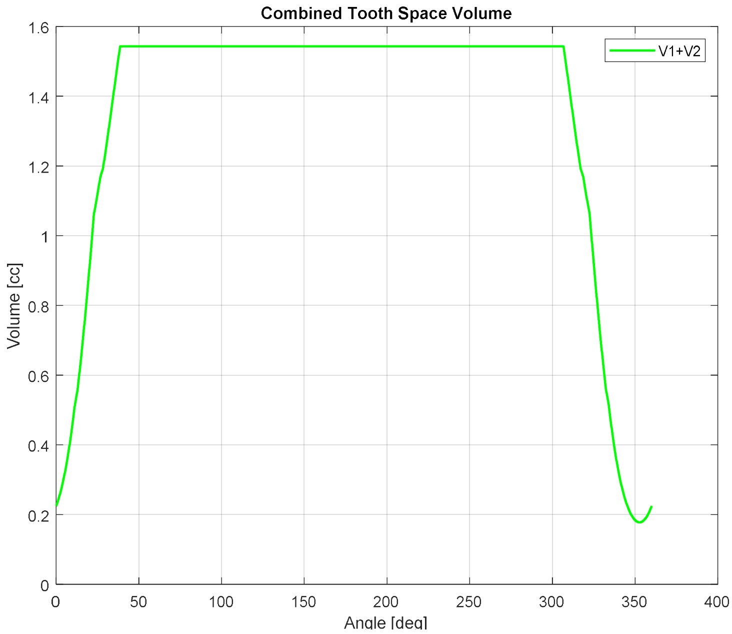



*Figure 30 - Summation of two single volume of the drive and driven gear*

More precisely, the Geometry Module it calculates the Tooth Space Volume in terms of rotation angle of the gears and the connection areas between the volumes. The model considers several connections between the inner volumes of the pump. The connection can be divided in two categories: the face-wide connections, and the depth-wise connections. Above, are presented the plots that represent the Tooth Space Volume, below instead are presented the connection areas.

## Face-wise connections

The face-wise connections are the connections between the Tooth Space Volume and the other volumes through the lateral face.

Below are shown the just mentioned connections.

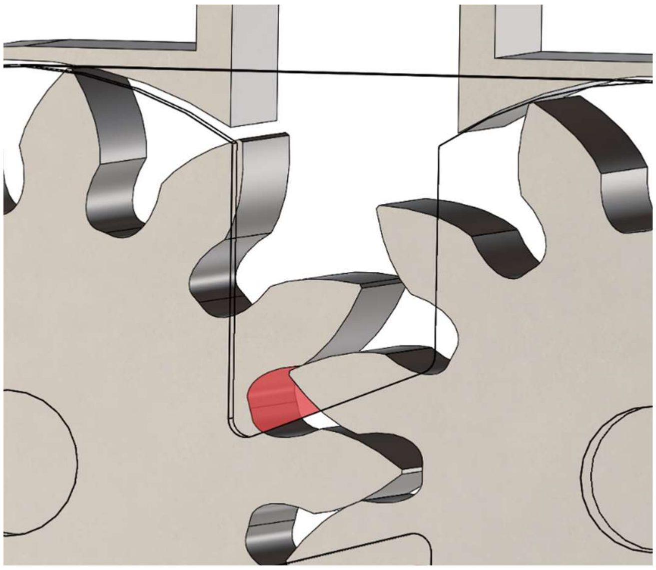


Figure 31 - Connection through the relieve grooves

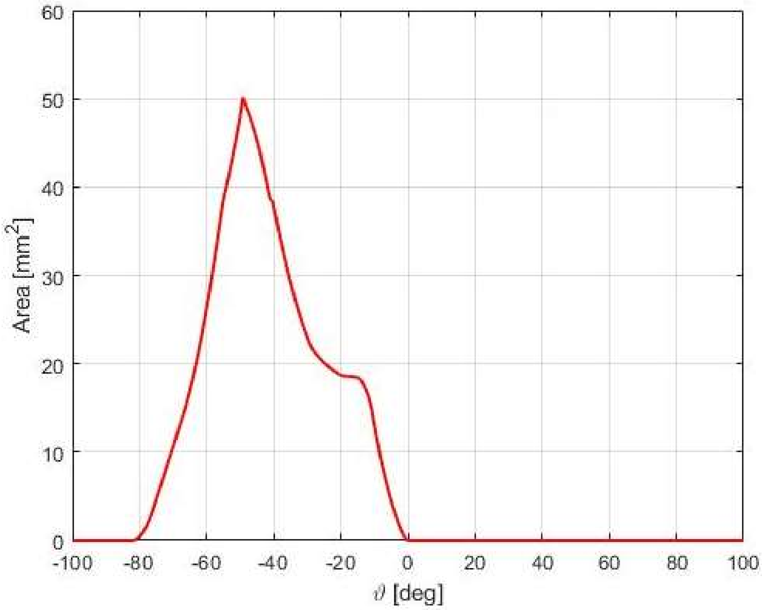



Figure 32 - Plot of the connection area through the relieve groove

Above is shown the connection of the Tooth Space Volume and the outlet volume through the relieve groove. The same connection happens towards the inlet volume, but for sake of brevity is not reported.

Below is shown the connection between a Tooth Space Volume and the back-flow groove.

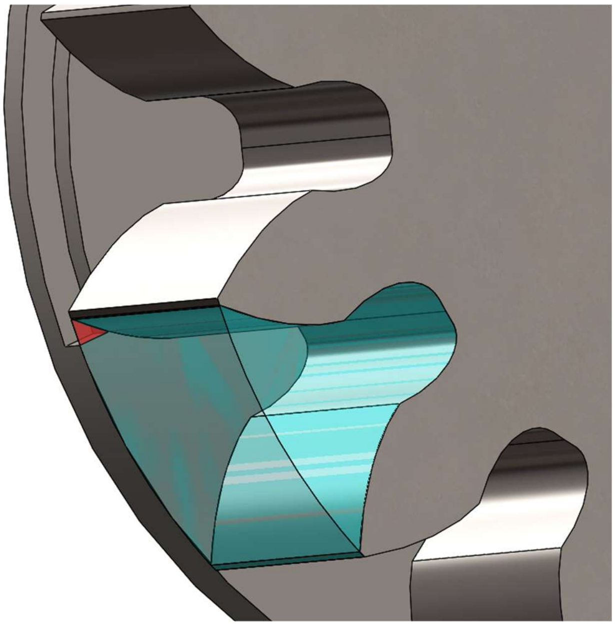


Figure 33 - Connection through the back-flow grooves

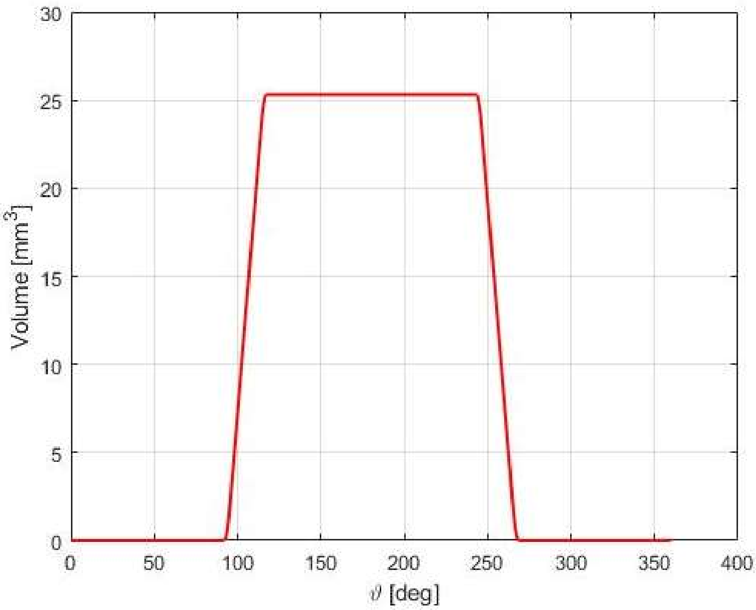



Figure 34 - Plot of the connection area through the relieve groove

### Depth-wise connections

The depth wise connections are the connections between the Tooth Space Volume and the other volume through the depth of the gear.

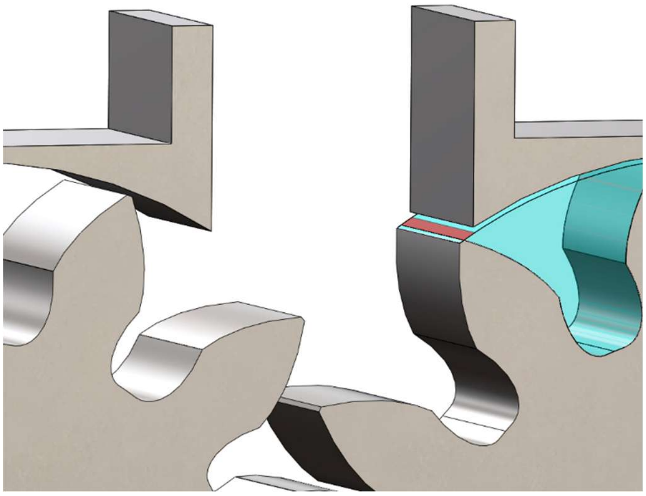


*Figure 35 – Connection through the depth wise area*

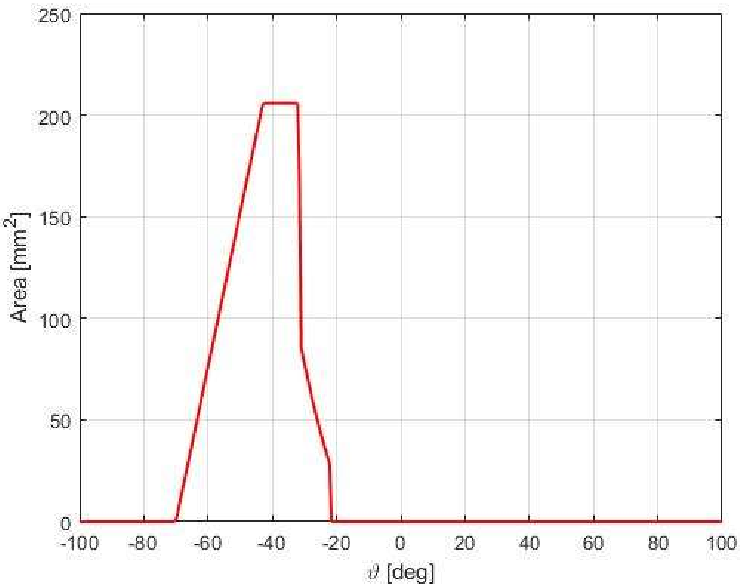



*Figure 36 - Plot of the connection area through the depth wise area*

The figures above show the connection of the Tooth Space Volume with the outlet volume. The correspondent connection with the inlet volume is omitted for the sake of brevity.
Below are shown the connection generate in the meshing zone. They are conceptually similar but while the first one is the connection between two corresponding Tooth Space Volume and the second one is the connection between a Tooth Space Volume and the adjacent to the correspondent of the other gear.

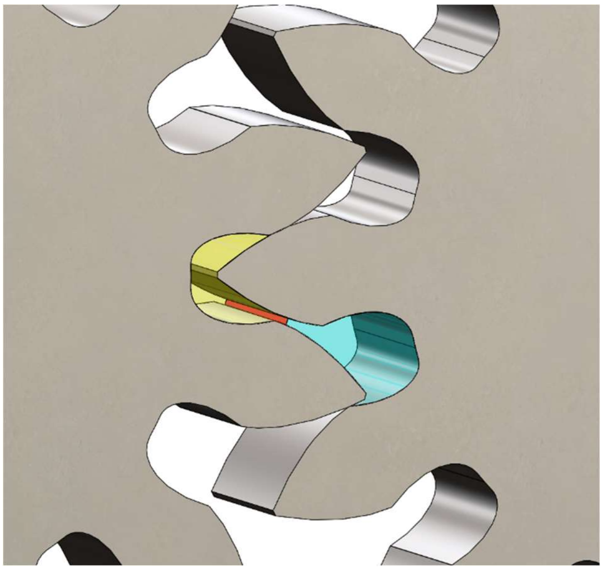


*Figure 37 - Connection between two corresponding Tooth Space Volumes*

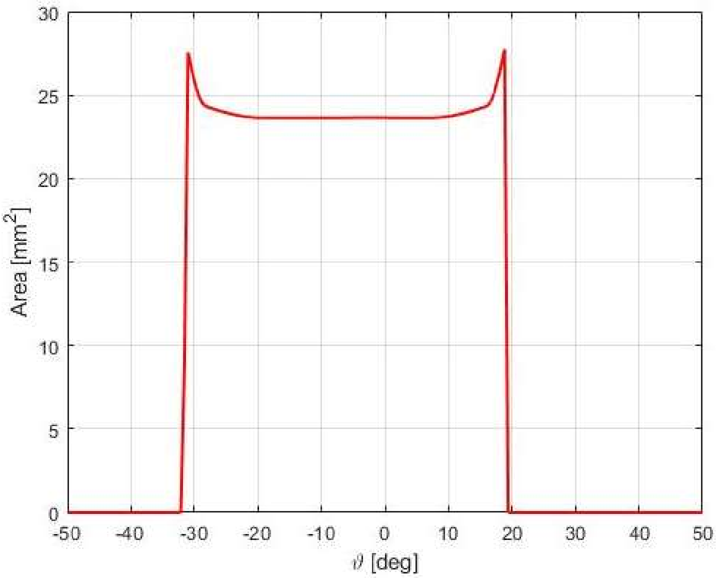



*Figure 38 - Plot of the connection area between two corresponding Tooth Space Volumes*

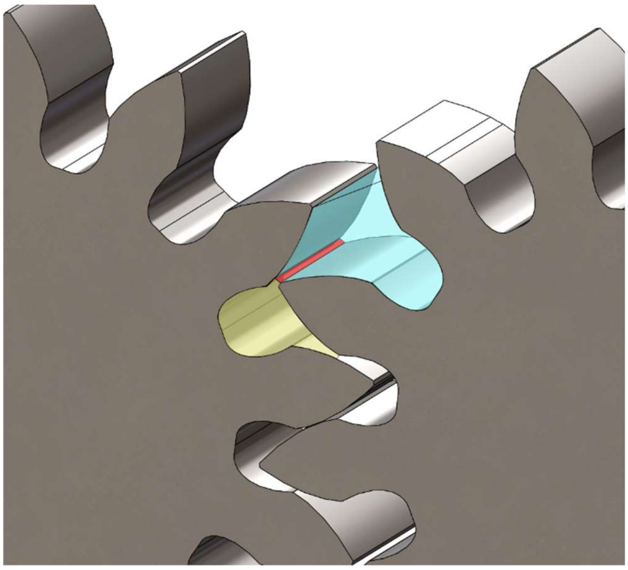


*Figure 39 - Connection in the meshing zone*

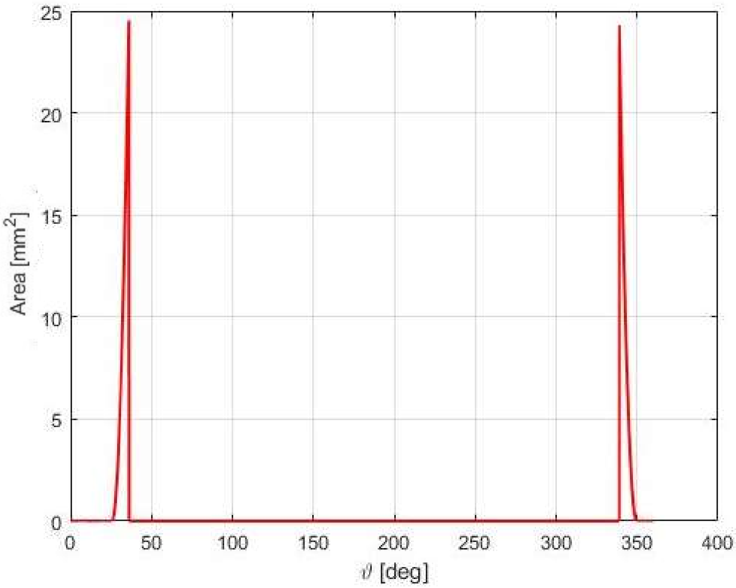



*Figure 40 - Plot of the connection area in the meshing zone*

### *Projections*

Another fundamental feature of the Geometry Module is that it calculates the areas projection of every volumes of the pump in the X, Y and Z directions. Furthermore, the distance of the projections from the center of the gears are calculated. These will be needed in the Loading Module to calculate the forces and moments generated by the pressure acting on the gears.

## Fluid-dynamic Module

The Fluid-dynamic Module takes advantage of the features calculated by the Geometrical Module to calculate the pressure of every volume in the gear machine and the flows that connect them.

More precisely the volumes are modelled with the pressure build-up equation, that is reported below.

$$\frac{dp_j}{dt} = \frac{1}{V_j}\frac{dp}{d\rho}\bigg|_{p=p_j} \cdot \left[\sum \dot{m}_{in,j} - \sum \dot{m}_{out,j} - \rho|_{p=p_j}\left(\frac{dV_j}{dt} - \frac{dV_{j,var}}{dt}\right)\right]$$

The flow through the connection areas are modelled with the orifice equation.

$$\dot{m}_{i,j} = \frac{(p_i - p_j)}{|(p_i - p_j)|} \cdot \rho|_{P=\overline{P_{i,j}}} \cdot \alpha \cdot \Omega_{i,j} \cdot \sqrt{\frac{2 \cdot (p_i - p_j)}{\rho|_{P=\overline{P_{i,j}}}}}$$

Focusing on the areas reported in the previous section it's possible to see that they are not always modellable as orifice, for example the connection of the Tooth Space Volume with the inlet and outlet volumes can be too wide to fit the previous description, but it's also true that the exchange of fluid happen only when the area is very small, and for this reason the connection fit the definition of the orifice equation.

However, there are connections that are never modellable as orifices. These connections will be named leakages. More precisely they are the connection between two Tooth Space Volumes through the gap between the tooth tip and the casing, between the lateral surface of the tooth and the bushing, and through the journal-bearing.
The leakages are modelled with the Couette-Poiseuille equation.

$$\dot{m}_{i,j} = \rho\frac{h^3 b}{12\,\mu}\frac{p_i - p_j}{L} + \rho h b\frac{u}{2}$$

The equation just mentioned describe the flow between two parallel plates promoted both by a differential pressure and the relative motion of the two plates. More specifically the first term, named Poiseuille term, describes the flow promoted by a differential pressure, the second term, named Couette term, describes the flow promoted by the relative motion of the surfaces.

In the figure below the leakages are shown.

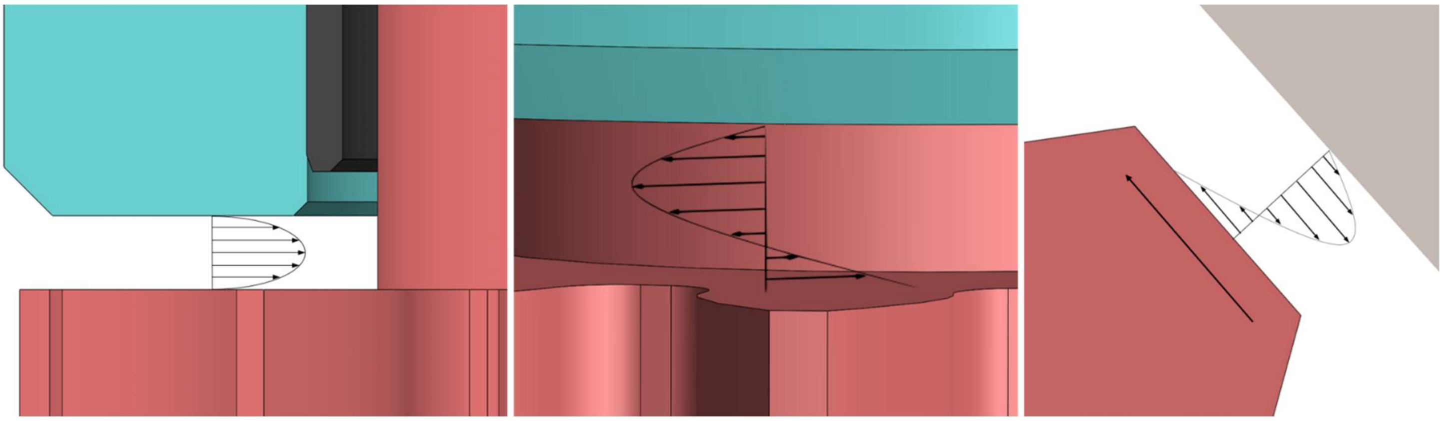


*Figure 41 – Leakages in external gear machines*

Is noticeable that referring to the pumping mode, while the Poiseuille term counteracts the delivering of the fluid, the Couette term and the relative motion push the fluid toward the outlet port. The results given by the fluid-dynamic module will be detailed later, in the section concerning the study case.

## Loading Module

The loading module is the model part in charge of calculate the forces applied to the gears. Using the projection areas calculated by the Geometric Module, and the pressures calculated by the Fluid-dynamic module, the calculation of the forces and moments acting on the gears is possible.

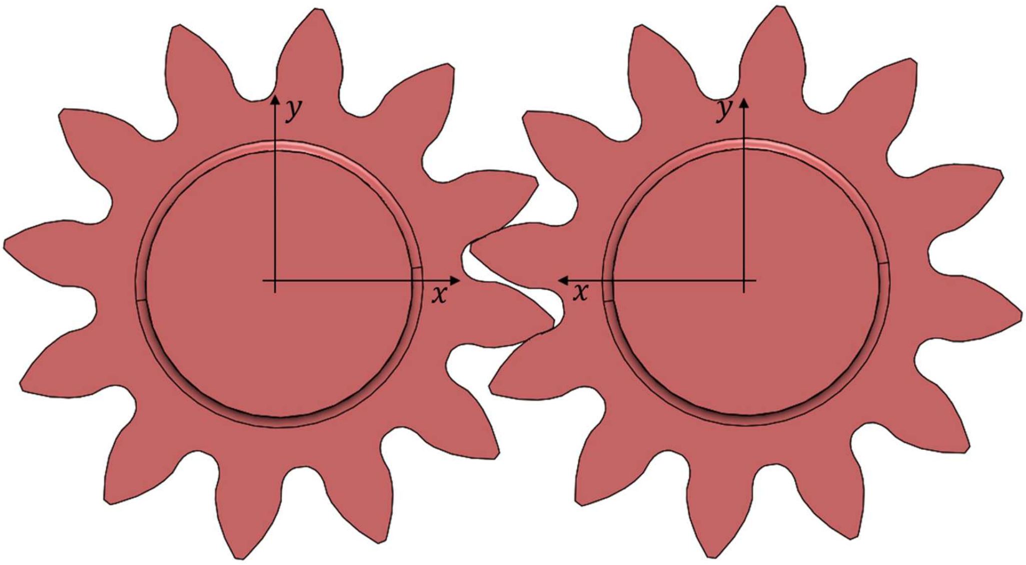



*Figure 42 - Loading Module reference system*

Above is shown the reference system used by the gear module.

## Journal-Bearing module

The Journal-Bearing Module is the module in charge to calculate the micromotion of the gears. In the module are implemented two models to calculate the micromotions of the gears, one analytical method and a CFD model. The analytical method is the Mobility Method, the CFD method instead is based on the numerical solution of the Reynolds equation. In the further section a brief description of these two methods will be given.

### The Mobility Method

The Mobility Method is detailed in a paper dated 1965 and written by J. F. Booker.
This method is based on the analytical solution of the Reynolds equation. This is possible only making some assumption: neglecting the flow in the circumferential direction or neglecting the flow in the axial direction. Depending on which assumption is made the solution is named Ocvirk solution, or Summerfield solution. It's clear that the first assumption is theoretically true only for journal-bearing whit an infinite small length. On the other hand, the second is theoretically true only for journal-bearing with an infinite length. So, the practical cases will always be a sort of compromise, and to define which is the best analytical approximation what can be done is to use the length-diameter ratio. More precisely, a length diameter ratio lower than 1 suggests the usage of the Ocvirk solution, otherwise Summerfield solution.

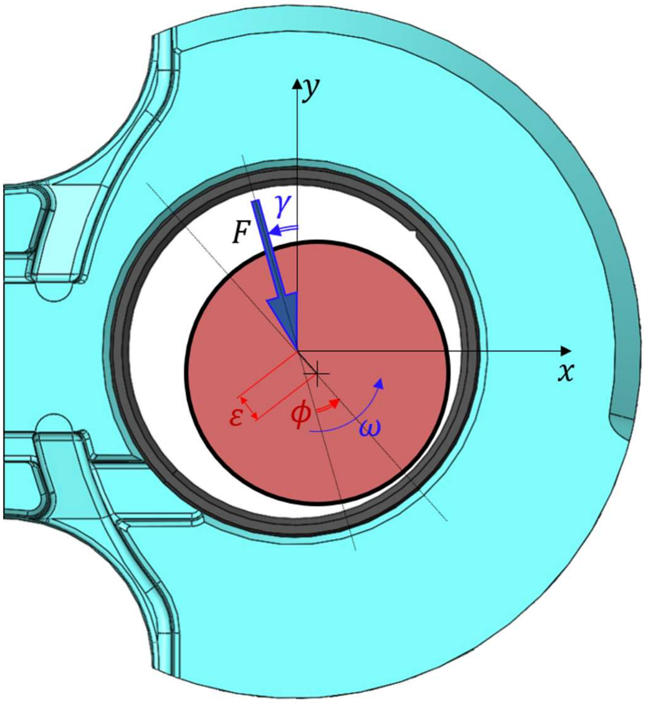



*Figure 43 - Journal-bearing mobility method symbols*

In both cases the procedure is solving the Reynolds equation making the proper assumption, to find the pressure distribution in the zone of the journal-bearing where the pressure is positive. After that, through an integration it's possible to find the force generated by the fluid film, and its direction, as a function of the rotation velocity, the eccentricity and the attitude angle. If the force and the attitude angle are known, we can elaborate the equation to obtain the eccentricity derivative and the attitude angle as a function of rotation velocity, force and force direction.

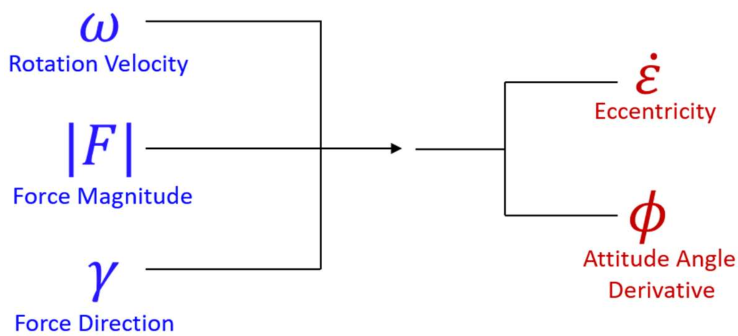
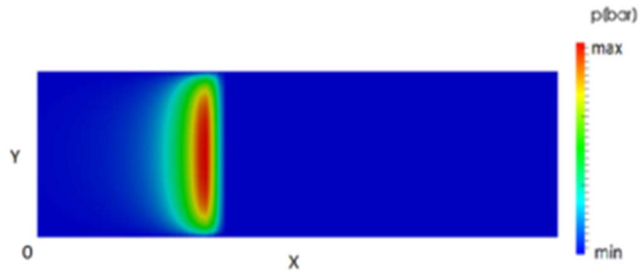


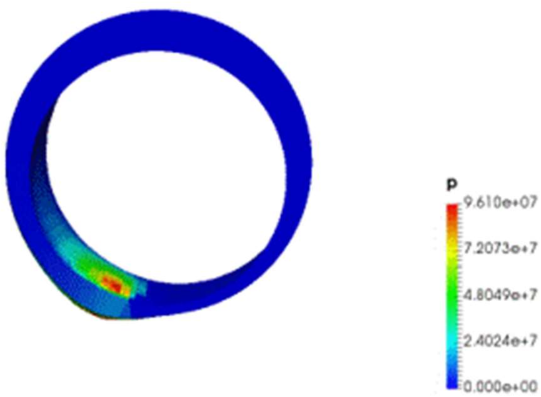

*Figure 44 - Journal-bearing mobility method schematic*

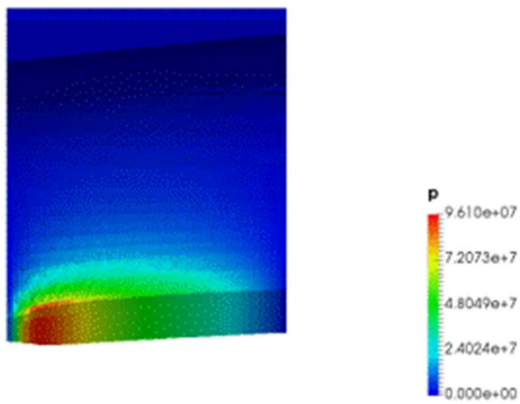
As mentioned before this method focus only on the journal-bearing section where the pressure built is positive neglecting the part in which it's negative and where cavitation happens. This is a big assumption, but tests made in the past demonstrate that the results are very similar when there are no bending or deformation.

### CFD method

In the CFD approach the procedure is the following. The fluid film that carries the load is considered. Than is unwrapped and meshed. The aim is numerically solving the Reynolds equation for every node. This allow the calculation of the thickness of the fluid film and the pressure distribution built in it. From the pressure it's possible to calculate the force. With this method is possible to model also the zone with pressure below the reference pressure, and for this reason can be modelled also the aeration and cavitation phenomena. Below are shown some images that illustrate the process of unwrapping and then rewrapping of the fluid film. It's also shown a section of the fluid film to illustrate how it deform accordingly to the deformation of the shaft and because of this the pressure distribution inside it change.



## Gear micro-motion module

Because of the constructive clearances in the external gear machines, and the presence of differential pressure inside them the gears are not perfectly steady, they move. More precisely the movement of the gears inside the machine casing is the result of two positioning. The positioning of journal-bearing system, so, more precisely, the positioning of the gear shaft respect to the bushing, and the positioning of the bushing inside the casing. They are calculated separately and then summed together.

## Circuit termination

To understand the results presented in the second part of the thesis, it's important to know that the model simulate the pump as if it was installed in a ideal circuit. Usually, two kind of circuit are used. The first one is named Volume Termination (VT), the second one is named Restriction Termination (RT).
Below are shown the two circuit.

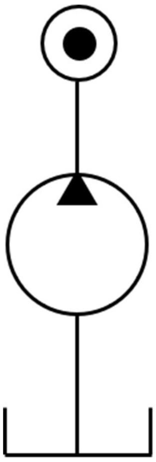


*Figure 45 - Schematic of the Volume Termination circuit*

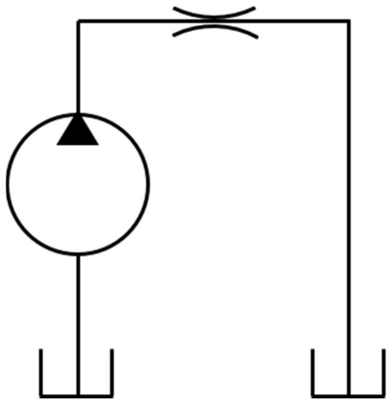


*Figure 46 - Schematic of the Restriction Termination circuit*

The volume termination circuit simulate the presence of an infinite capacity at constant pressure connected to the outlet of the hydraulic machine. The infinite capacity can be seen as an infinite big accumulator. The Restriction Termination circuit simulate the presence of an orifice between the hydraulic machine outlet pressure port and a low pressure source. This kind of circuit simulate the presence of a load. The first configuration allows the user to see only the flow ripple due to the kinematic displacement of the pump. The pressure ripple is almost totally mitigated by the presence of the high pressure source. In the other case, this mitigation doesn't happen and it's also noticeable the flow ripple.

## Graphical User Interface

HYGESim has been developed in AMESim first, and now the team is moving to a C++ model. The reason of this choice is that even though AMESim interface seems more user friendly, the sub-model customization is more difficult, there is no control on the integrator, and the license is very expensive. C++ is way faster and customizable, but without a deep knowledge of the code or an interface is almost impossible to use it. So, the first goal that I accomplished is to build a Graphical User Interface (GUI) to control it. The GUI that I made is dynamically built based on some ".yaml" files that define it's structure. This solution, as opposite of a hardcoding procedure, give flexibility to the GUI and ease its modification. Also, as shown in the figure below it is structured like a tree.

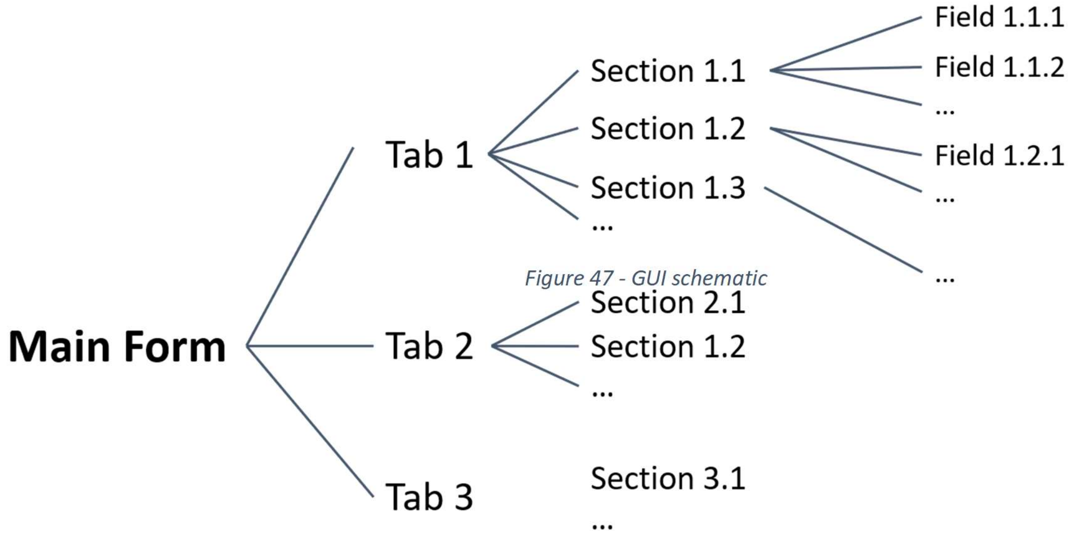



*Figure 47 - GUI schematic*

In the following pages the GUI will be shown in all its features.

## Option tab

In the option tab is possible to select a working folder. The working folder is necessary because it will contain the data needed to run the fluid dynamic module. Also, in the selected folder the inputs can be saved and lead later. This function is very useful when the user work with more pumps, or the same pumps but with different geometry features.

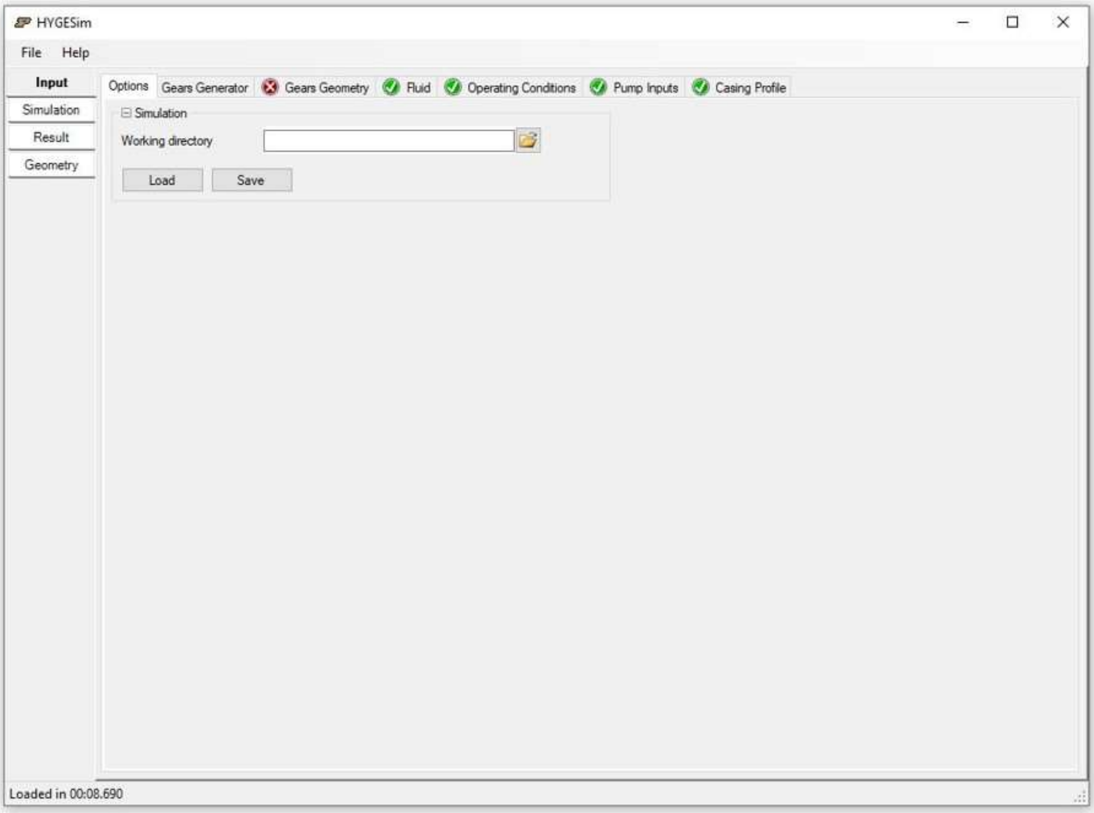



*Figure 48 - Option tab*

## Gear generator tab

In the gear generator tab, the user can run the gear generator software. Some images of the gear generator software are shown in the section dedicated to the model description.

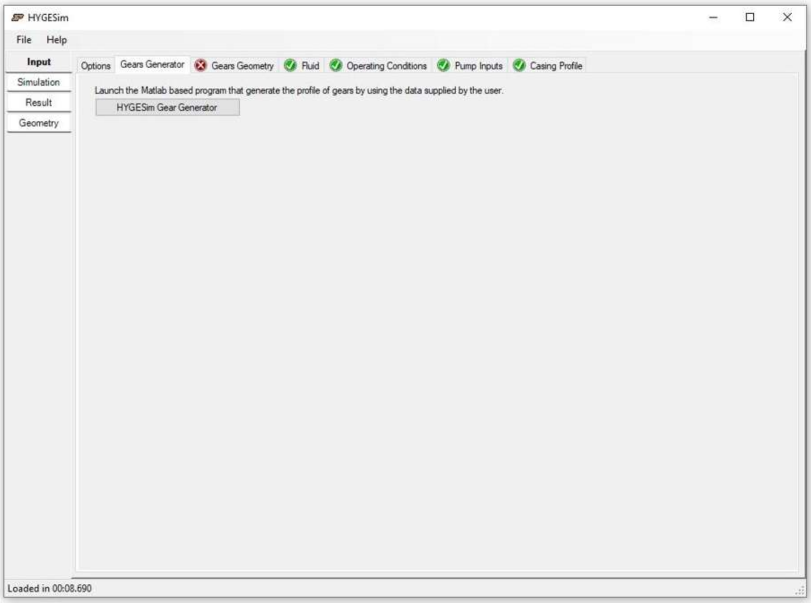



*Figure 49 - Gear generator tab*

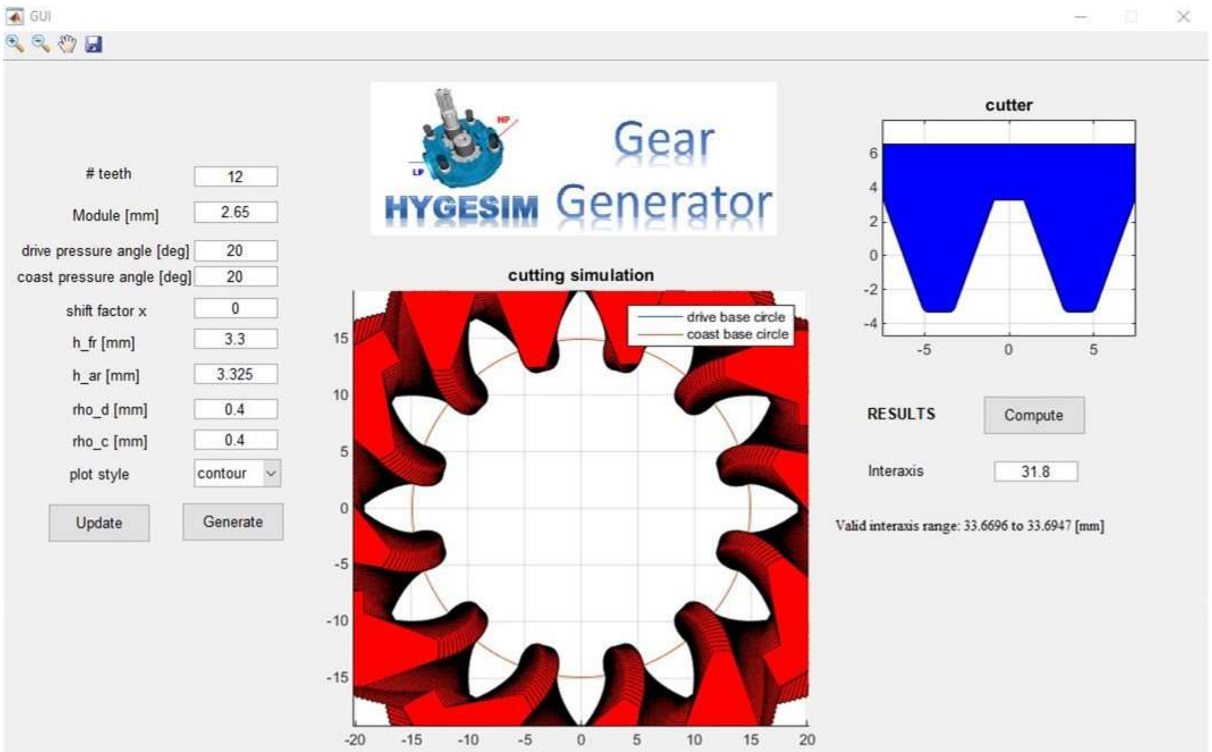



*Figure 50 - Gear generator interface*

## Geometry module tab

The geometry module tab must be filled with a compound of inputs. First, the user has to define how to input the geometry features such as casing, bearing block, relieve grooves. These can be input as ".stl" file in case the user wants to simulate an existing pump, otherwise the "parametric-full" and parametric-rectangular" options give the opportunity to define predetermined relieve grooves geometries. Also, the user must insert data related to the meshing of the gears as the nominal interaxis, the "delta angle", and the contact ratio. For the sake of clarity for the reader is necessary to highlight the fact that the delta angle is a virtual angle calculated by the geometry module and represent how much a symmetric to the drive gear, has to be rotated to mesh with the drive gear itself.

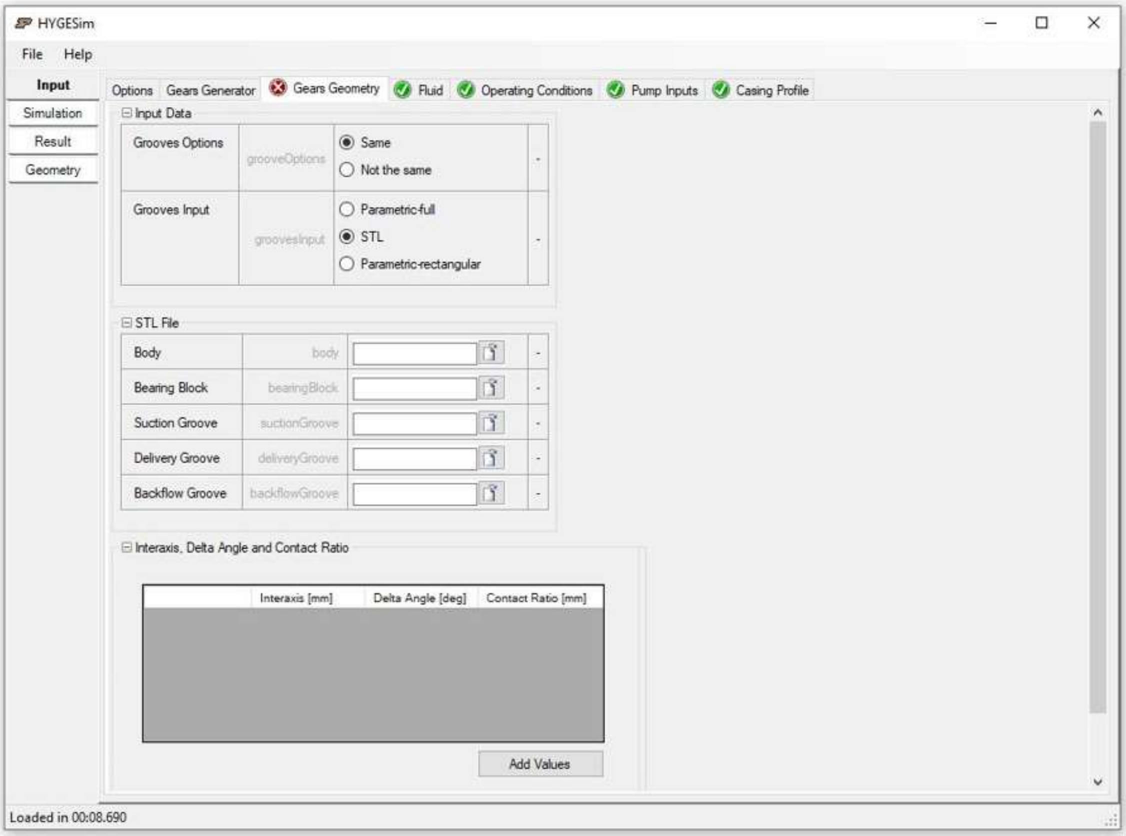



*Figure 51 - Geometry module tab*

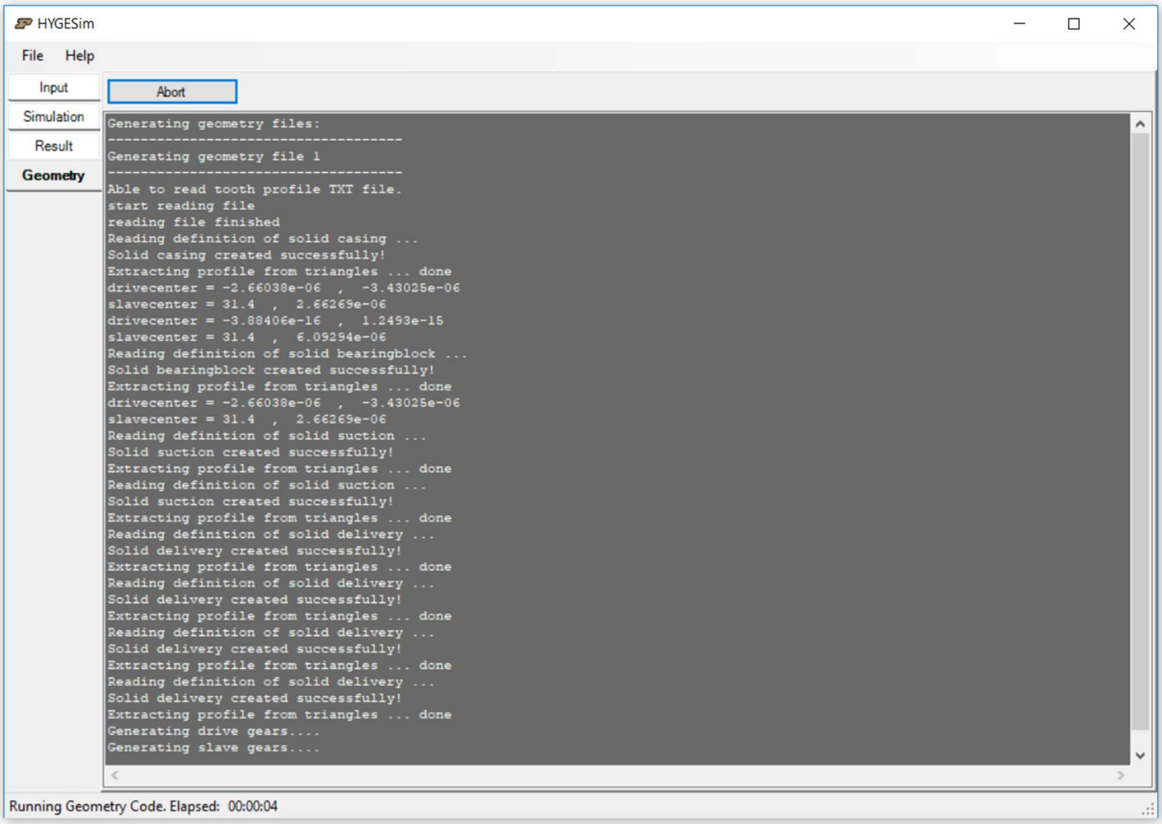



*Figure 52 - Geometry tab results*

## Fluid tab

The fluid tab allows the user to select or generate the fluid property for his simulation. The necessity of doing this comes from the fact that the fluid property change depending on the used oil of course, but also depends on temperature and pressure. For this reason, every user needs to properly select the right fluid property. It can browse in its computer or select it from an inside library. Other options are, inserting Bulk modulus, density, and dynamic viscosity, and considering them constant or insert an equation that express how the just mentioned value changes as relation of pressure and temperature. In this section is also possible to select which of the cavitation model use.

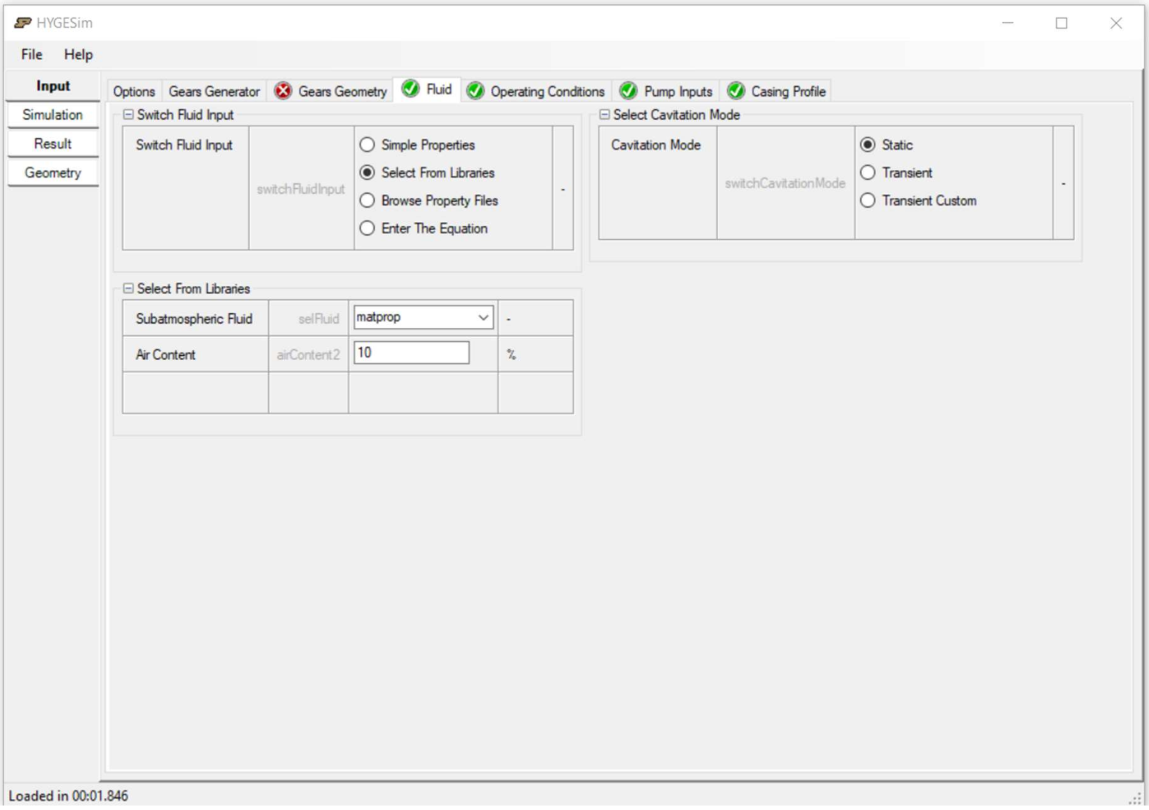



*Figure 53 - Fluid tab*

## Operating condition tab

The tab presented below allow the selection of the operating conditions in which the machine works, but also some parameter related to the integrator and the numerical approach.

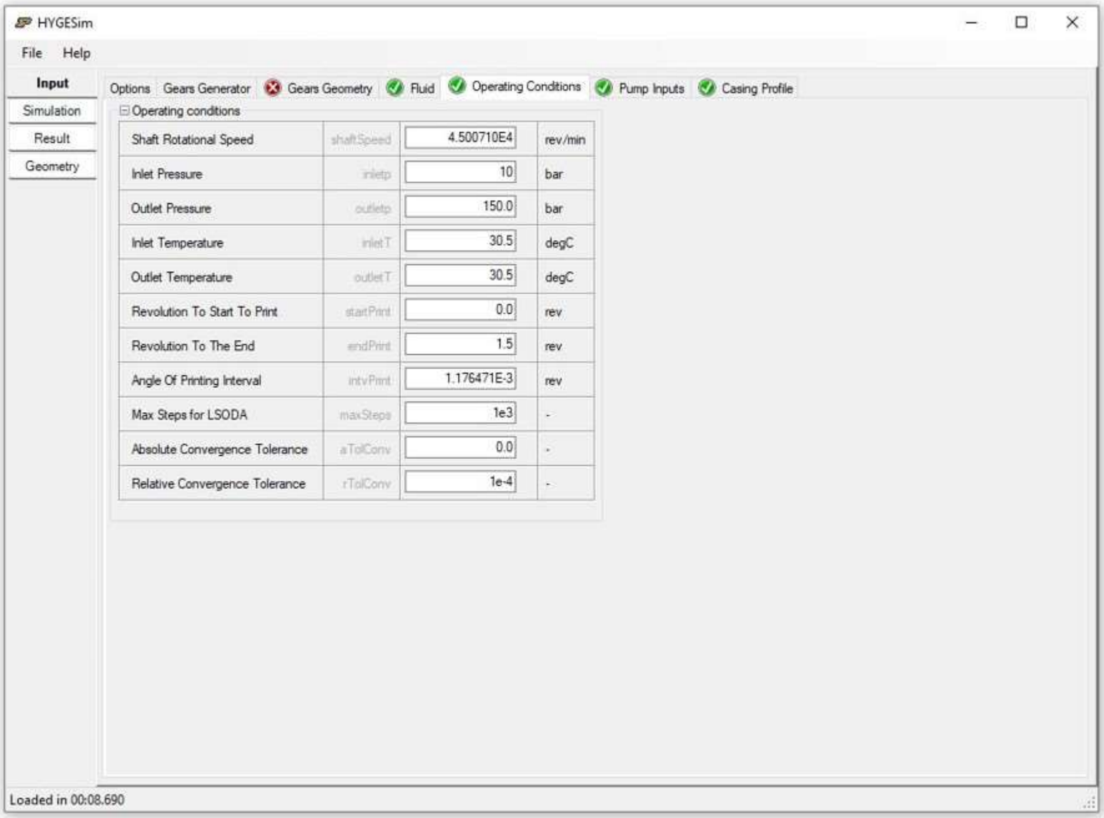



*Figure 54 - Operating condition tab*

## Pump inputs tab

In the following tab are defined all the geometric features of the pump. All the geometric features are used in the fluid-dynamic model. Sometimes the name of the parameter can be confusing, mostly for new user so, clicking on the label of the parameter is possible to see an imagine that clearly explain what the parameter refers about.

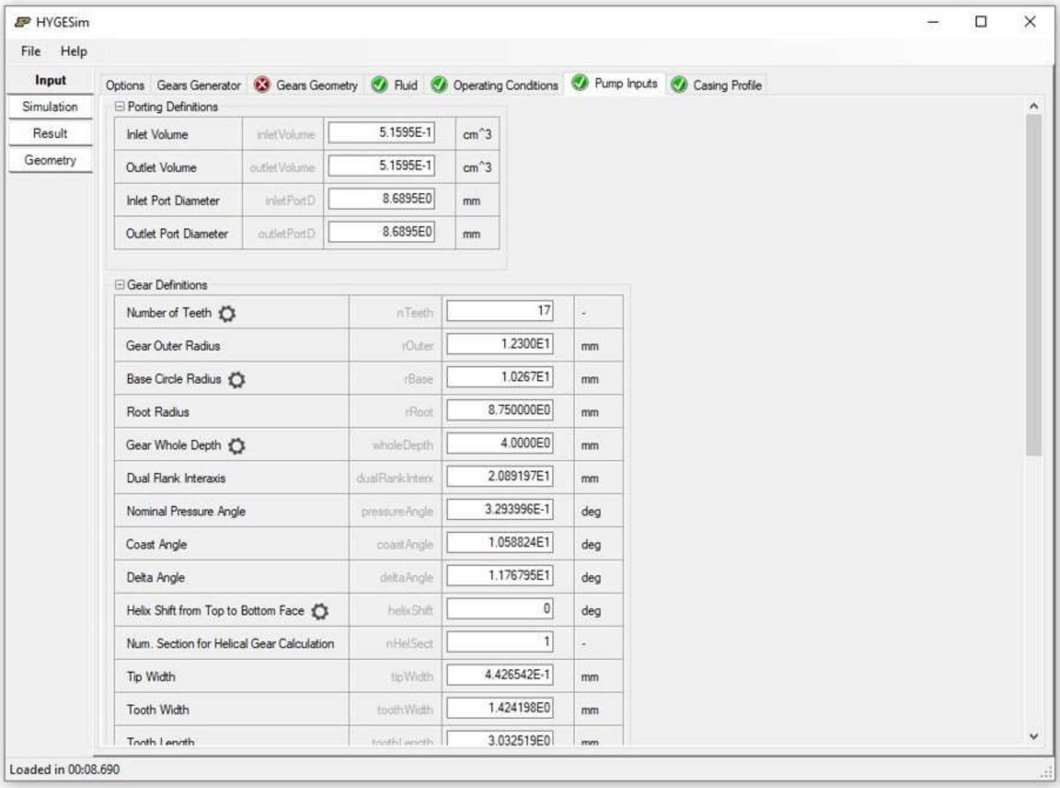



*Figure 55 - Pump features tab*

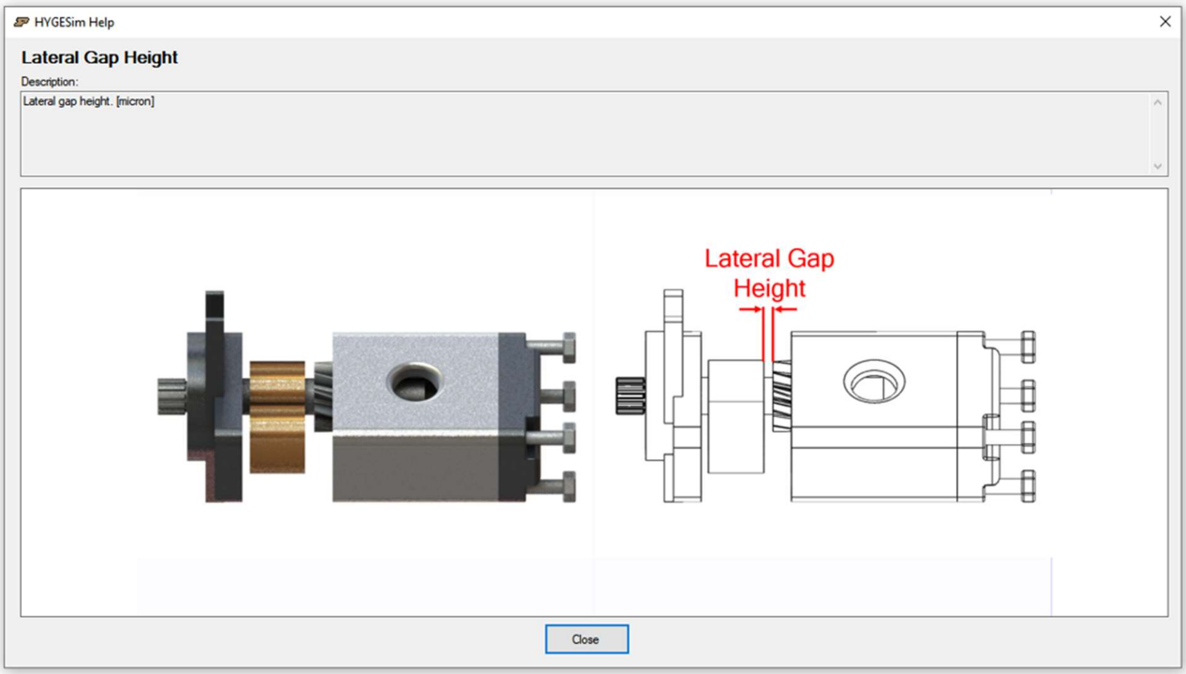



*Figure 56 - Explaining image related to a value*

## Casing tab

The casing tab allow the definition of the casing parameter or the browsing to reach premade ones. In the image below, it's possible to see how the casing cavity is divided in two part, the part that host the drive gear, and the part that host the driven gear.

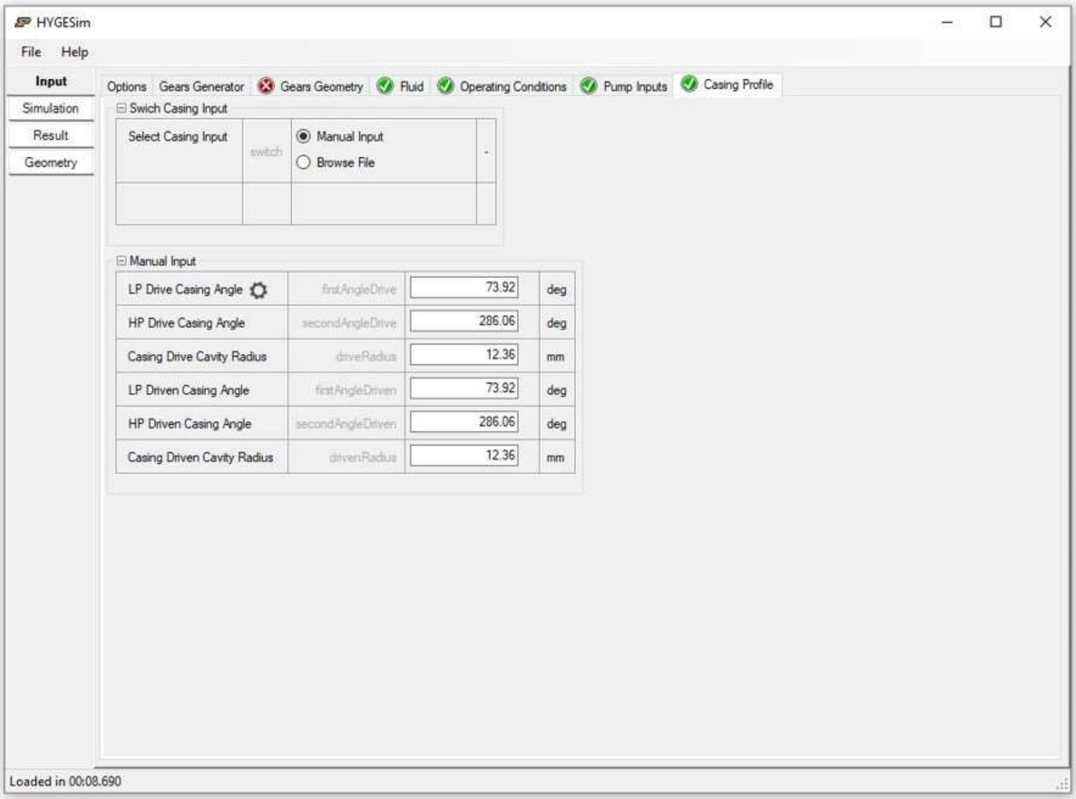



*Figure 57 - Casing tab*

## Result tab

As last is shown the tab where the results are shown. It's possible to plot almost every value calculated in HYGESim. Below is shown the tab that show the results. In this tab all the results can be visualized, but not elaborated and neither zoom. For this reason, for a post process operation is highly suggested the use of external software like Matlab.

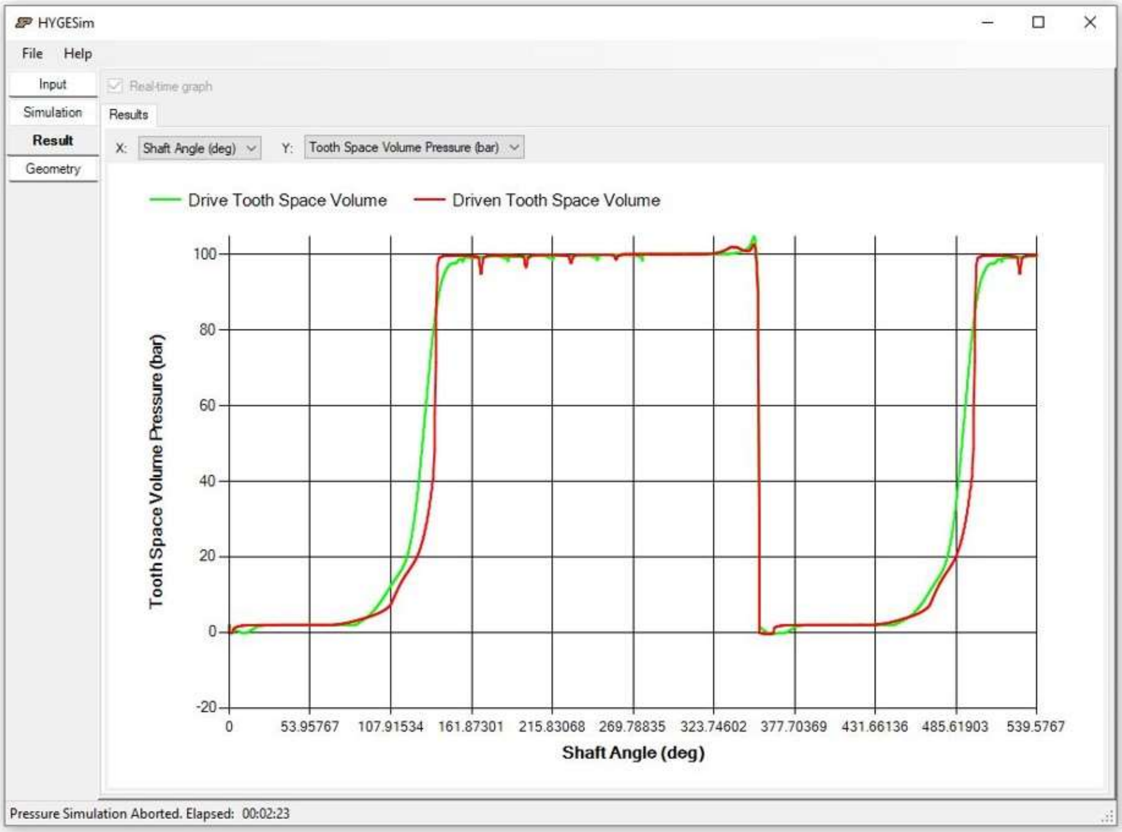



*Figure 58 - Results*

## Department of energy project

After having developed the GUI the model is ready to be used to run some simulations. The simulations are run in a wider contest, a project described below, in this section.

*Figure 59 - CASE TV380 scaled model*

The United States Department of Energy is founding a project that focus on the design and implementation of a hydraulic machine on an off-road vehicle capable of displacing energy to move the actuators, but also recover it from them to charge batteries. The aim of such a system is to increase the overall efficiency of the machine and reduce the fuel consumption. Other features needed by the device are the compactness, and the simplicity in term of number of components and assembly. While the MAHA lab is taking care of the hydraulic aspect, all the electronic parts, so the electric motor/ generator and its drivers, are designed by an Electronic and Computer Engineering research team led by prof. Scott Sudhoff from Purdue University.

Aside the Department of Energy the project is founded and supported by other two sponsor, CNH and Bosch Rexroth. CNH support the system design part, Bosch Rexroth support the pump design part. Both have offered their production facilities and components for the realization of the prototype. The machine in which the system and the device prototype will be implemented is a CASE TV 380.

For such machine was originally designed an open center system. This means that a pump constantly sucks fluid from a tank and displace fluid. If there are no moving actuator the fluid is displaced in the tank, on the other hand, whenever an actuator needs it, the circuit connect the actuator itself to the pump. The only efficiency of the circuit comes from the fact that displacing the fluid in the tank is an operation relatively low cost from an energy point of view, but still costly. However, this kind of circuit is that is very simple and cheap to produce and install.

In order to recover potential energy from the raised load. The circuit needs some modifications.

The solution is the deployment of a closed center circuit. This in this case the system layout is very different from the original one. Also, it uses different components. For example, it takes advantage the presence of an accumulator that replace the tank. The accumulator keeps the pressure in the circuit higher than the ambient pressure, reducing the cavitation issue. Also, another very deep change involves the hydraulic machine selection. In this case, the device is a four quadrant hydraulic machine.

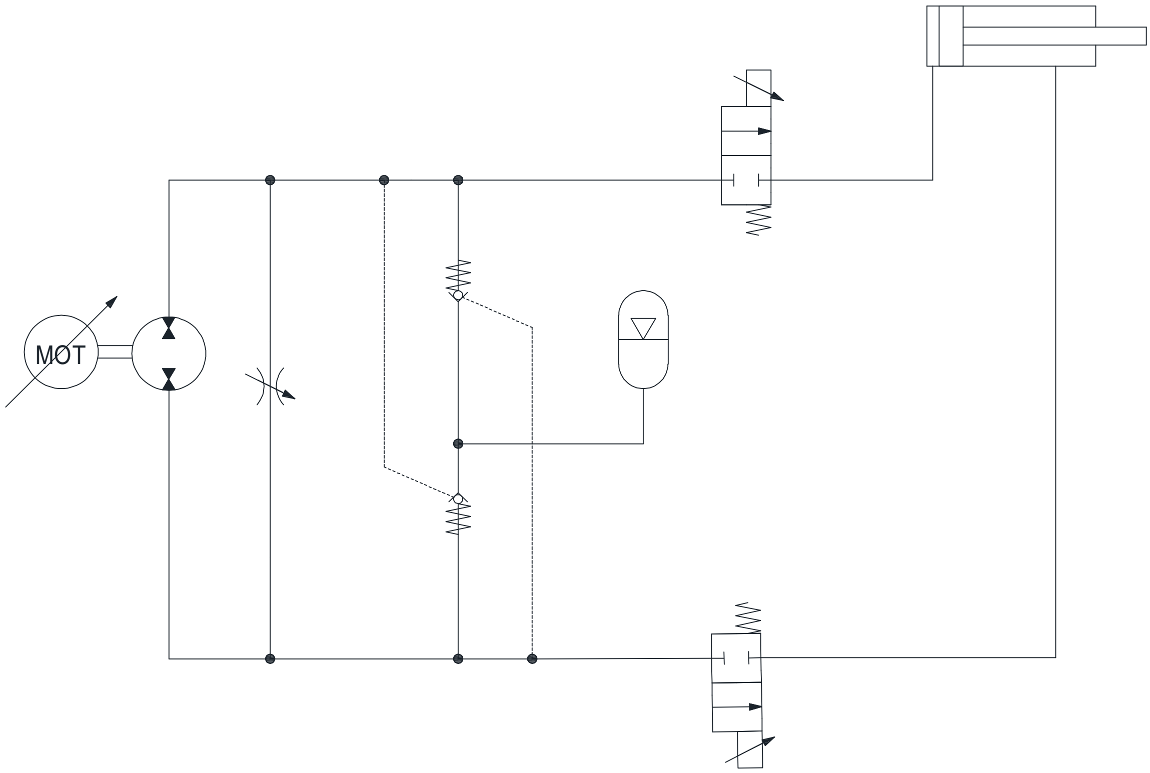



Figure 60 - Schematic of the closed center circuit

As shown in the schematic, the pump must be capable of pumping and receiving the fluid from both directions. Also, it’s possible to see valves that are used only as on-off to carry the load when the actuators are not moving. These valves are simple, robust and not expensive. Moreover, limiting the valve role to the one just described also allow to reduce the throttling losses. The biggest challenge lead by this solution is the design of the hydraulic machine. Also, such a system is groundbreaking and for this reason risky, both from a technical and economical point of view.

Below are highlighted the actuator in which the device will be implemented.


*Figure 61 - Track loader actuators*

The actuators in which the device will be implemented are the boom cylinders and the bucket cylinders. To allow the highest possible modulation every actuator is supplied with one device.

The new device must keep the same performances of the previous system so, it must be able to provide or handle the following conditions in pumping or motoring mode.

| Flow Rate | | Pressure | |
|---|---|---|---|
| Q raise max = | 46.17 [L/min] | Δp raise max = | 210 [bar] |
| Q lower max = | 77.60 [L/min] | Δp lower max = | 50 [bar] |

In the following sections will be explained in phases the work done until now on this project.

Other than the possibility of evaluating a wide range of parameter, one of the biggest advantages of using a lumped parameter model is the fast computation time.

It’s possible to take advantage of this feature implementing an optimization algorithm.

Later will be briefly described the features of the optimization algorithm and how it works.

Through the algorithm it's possible to obtain a tooth geometry and a relieve grooves geometry that lead to good performances.

However, the results must be manually tested and validated to be sure that every step of the algorithm was implemented properly. Also, to fully understand the importance of this step it's important to remember that the model is in a transitioning phase, and it is moving from AMESim ambient, where it's noticeable a perfect and validated implementation, to a C++ coding language where it's not so stable.

Also, the algorithm qualifies the gear set and bushing set based on some parameters obtained through a simulation. However, the simulation is run only for one or few operating conditions even though in the actual case the device must work properly in almost every operating condition. For this reason, after defining the gear set and bushing set through the just mentioned process, the device must be tested in more operating conditions, and eventually be properly modified.

While the optimization algorithm was developed by Thomas Ransegnola, a graduate student working at MAHA the first step of my job consisted in taking care of the validation and eventually correction of the model, considering both the four and the two quadrant hydraulic machine.

The second step focused on the lateral gap element. Substantially, the lateral elements float in the hydraulic machine. Depending on the pressures acting on their surfaces and their geometrical features they position differently. The aim of this second step consists in modifying their geometrical features to ensure the best overall positioning in all the operating conditions.

The third step focused on the developing of the design of the device. For the lack of time all the mechanical verification has still to be done. However, the special features of the device brought us to develop a brand new configuration unseen before. For this reason, before showing it in conferences and sharing it with our sponsor we decided to patent it. The features will be better described in the last part of the thesis.

## Optimization Algorithm

Since I didn't take care of this process, I will just explain briefly how the algorithm works. Every simulation is characterized by a very large number of parameters. Some of them are more effective on the results than others. For this reason, it's convenient to make the optimization algorithm change only some of them and keep the other constant. The selection of which value will be parametrized is made during the problem parametrization.

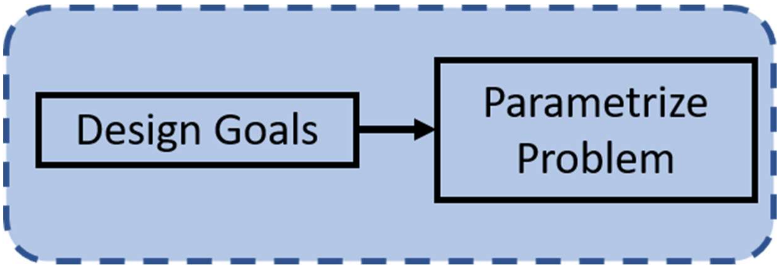



*Figure 62 - Optimization algorithm schematic*

It's reminded to the reader that in the following section the expression "parameters set" is equal to device design, because every specific device design is characterized by a specific parameter set.

To run the optimization were selected as variable values some parameters related to the gears, and some to the bushings. To set a proper optimization process is also needed the definitions of two set of constrains. The first set is applied to the variable parameters that generate the design, limiting them to a defined and feasible range.

The second constrain set is applied on the results that express the performances and determine if a values set is rejected or accepted. To do this, is needed the definition of criteria to evaluate the parameters set. Defining evaluation criteria means postprocess the result and extract numerical value that objectively express the performances.

Some example of criteria established to evaluate the set performance or feasibility are the following. During the meshing of the two gears the Tooth Space Volumes can be squeezed, and their pressure can reach very high value. Lower is the reached pressure, better is the parameter set, moreover, if the value reached is higher than 110% of the outlet pressure the value set is rejected. Another example to evaluate them is considering the cavitation phenomenon.

For the sake of clarity will be explained how the cavitation phenomenon is measured. When the Tooth Space Volume suddenly decrease after the meshing zone, its pressure drops. Also, in the filling phase, characterized by a sudden increasing of the volume, it tends to go below the inlet pressure. For this reason, the air that before was in a sort of equilibrium in the fluid is released generating bubble that imploding can damage the device. So, it's possible to indirectly measure the cavitation phenomenon measuring the area between the inlet pressure profile and TSV pressure profile when the TSV pressure profile is below the inlet pressure profile. If this value overcome a pre-defined limit, the variables set is rejected.

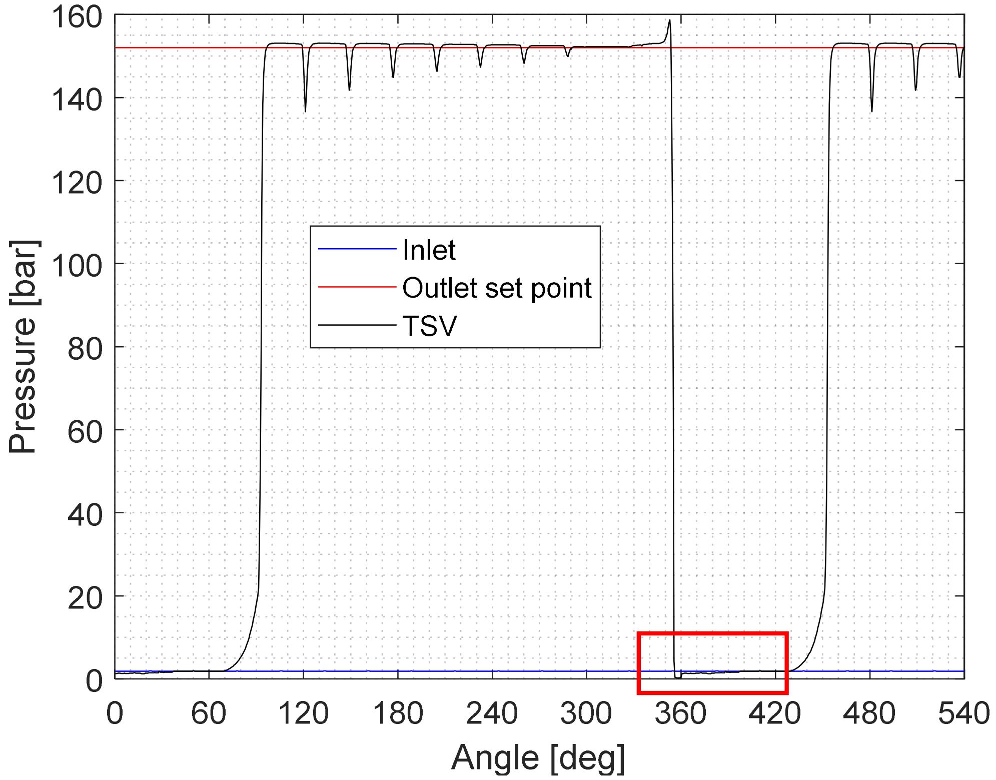


*Figure 63 - Cavitation constrain integral*

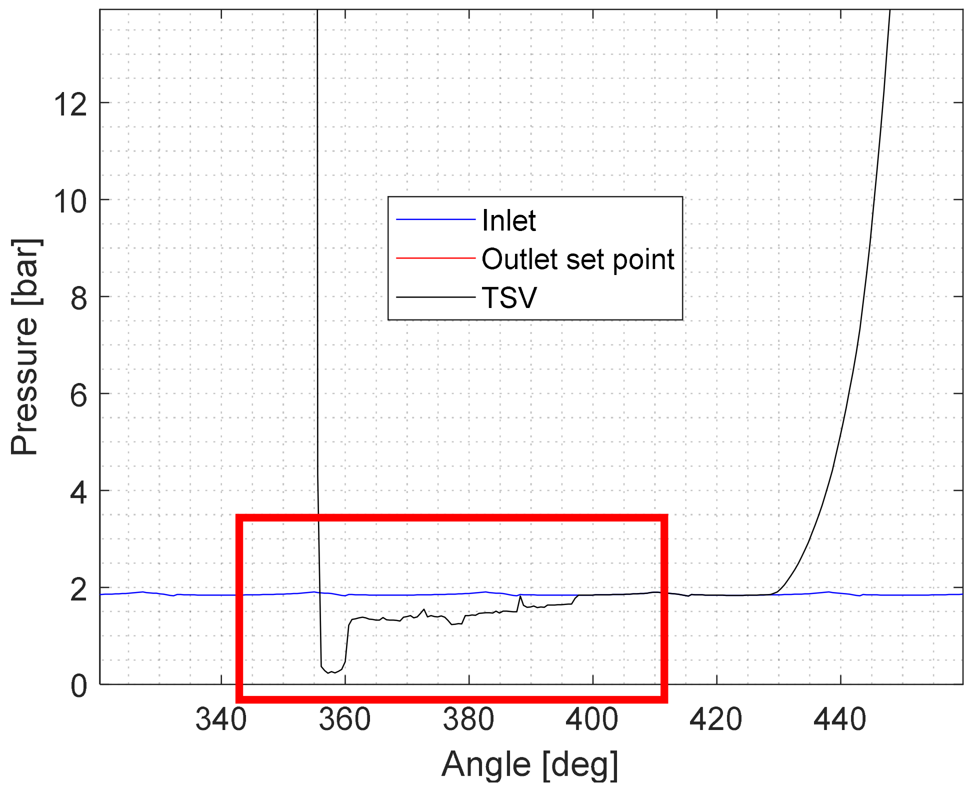



*Figure 64 - Cavitation constrains integral zoom*

While some parameters were used to select the feasible designs, other parameters were defined to evaluate the quality of a gear set. The just mentioned parameters are the overall efficiency of the cycle, the flow ripple, torque ripple and minimize the dimension of the pump. The energy of the cycle is calculated by integrating the instantaneous power spent by the device (torque times angular velocity). The flow ripple and torque ripple are calculated as the differential value over the mean value. The dimension of the unit is defined as the interaxis plus two times the gear outer radius. Every line of the plot shown below represent a variable parameter set. In this optimization was evaluated around 4000 parameter sets.

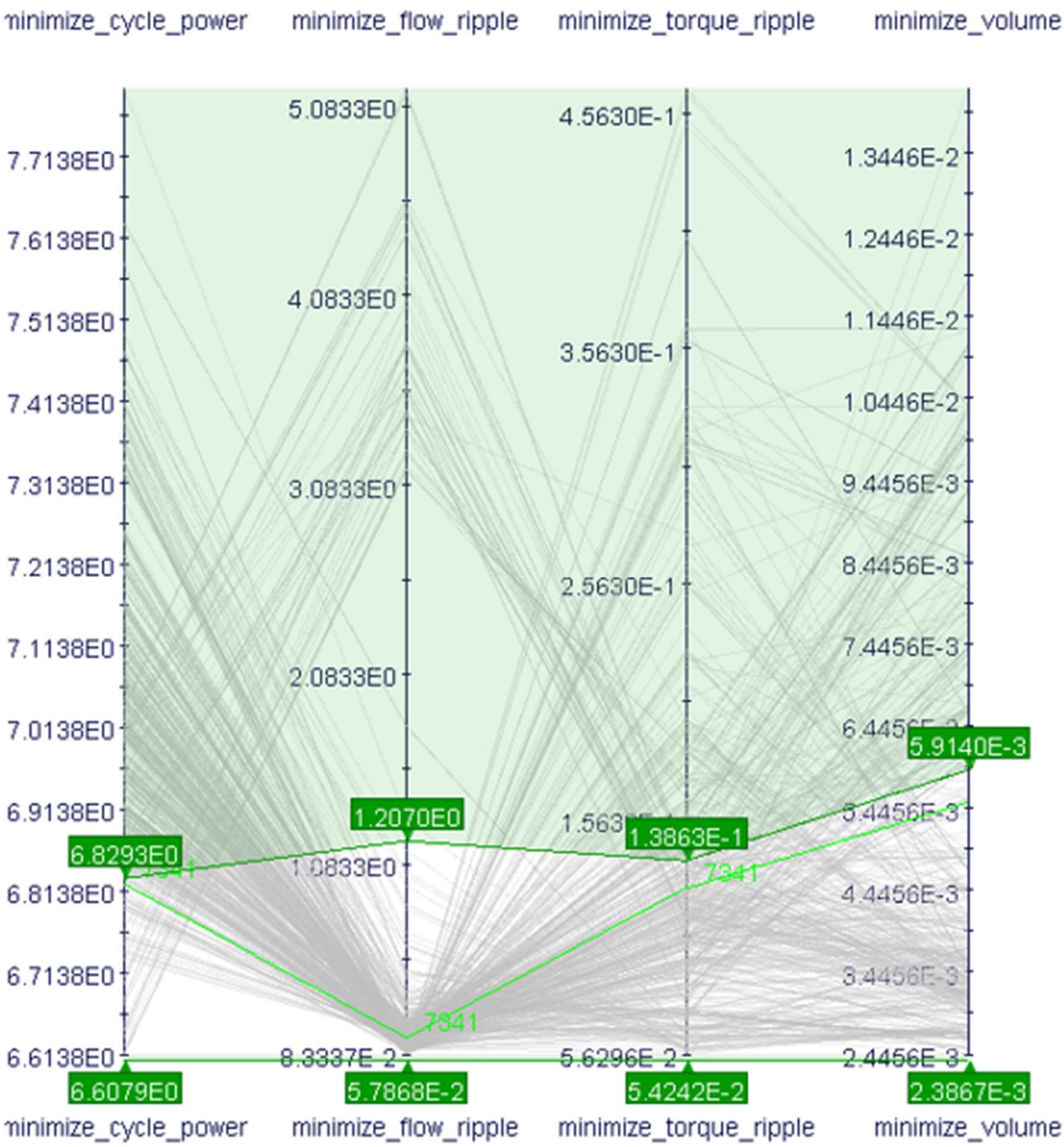



*Figure 65 - Designs performance*

Inserting all the designs in a plot that evaluate their performance we can also see the so called Pareto front. The Pareto front is a virtual line defined by all the layout with similar performances. More precisely all the design that define the pareto front are characterized by a peculiarity: they cannot improve one aspect without impair another one.

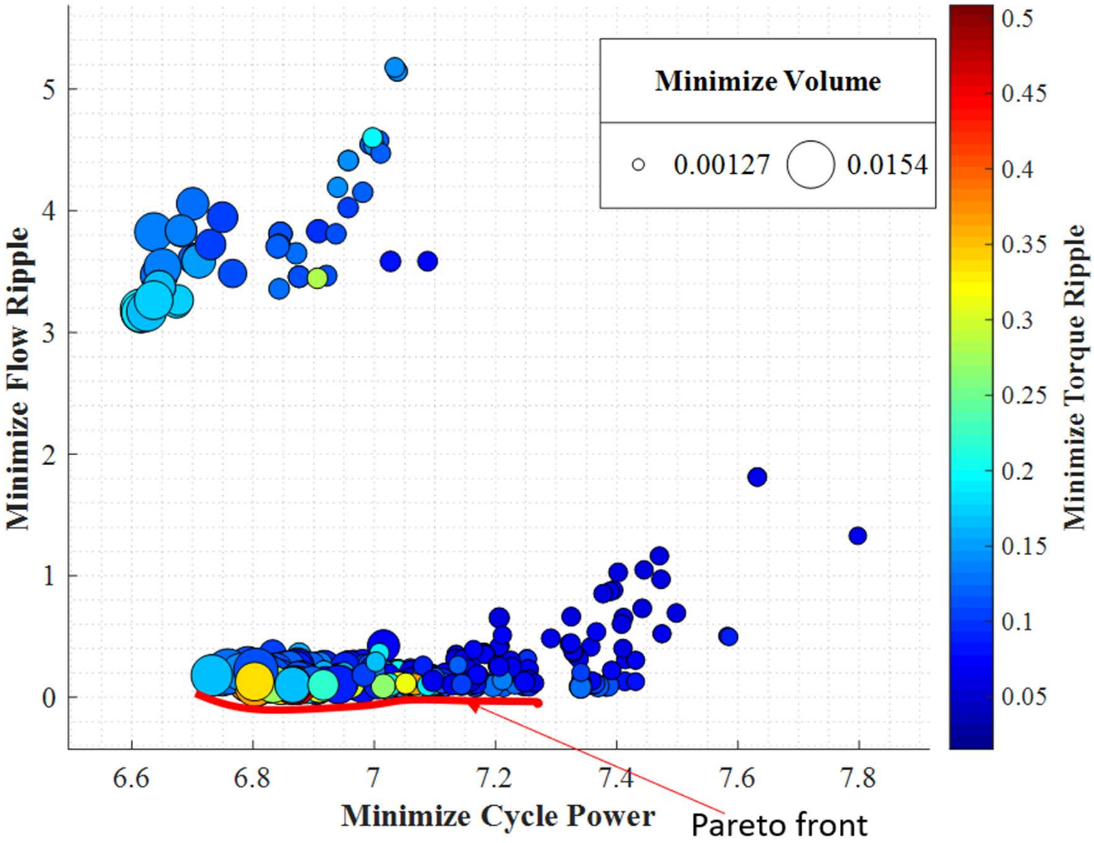



*Figure 66 - Optimization Pareto front*

After all this procedure started my evaluation process.

## Evaluation process

The work done on the four quadrant external gear machine can be divided in several step. The first step concerned the evaluation of two parameter sets. One characterized by the presence of 13 teeth and the second by 14 teeth. The 13 teeth gear set has a kinematic displacement of 14,62 cc/rev, and the 14 teeth gear set has a kinematic displacement of 13.45 cc/rev. After having evaluated the most interesting hydraulic features of the two gear set, the one with the behavior that better satisfy the requirements of the project will be select and deeply studied to validate every step taken to design it.

Below are shown the gear profiles. Hereafter, to refer to a specific gear set will be used its number of teeth.

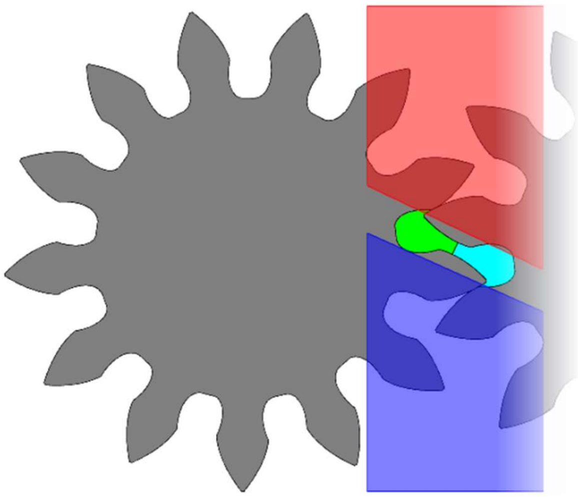


Figure 67 - 13 teeth gear set

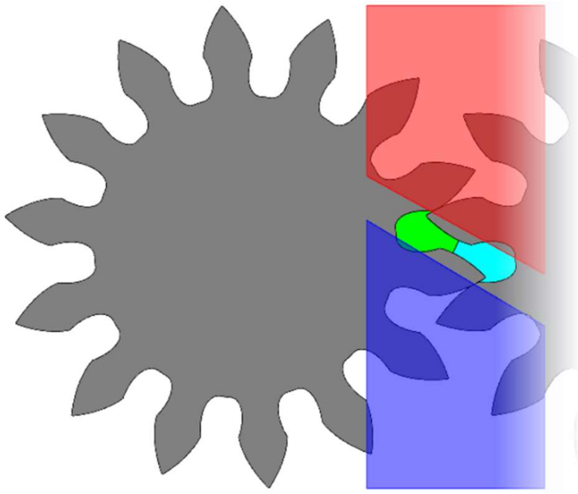


Figure 68 - 14 teeth gear set

As mentioned before the criteria analyzed in the optimization are the power consumption, the flow ripple, the torque ripple and the volume. In the following section will be shown the qualitative evaluation of these parameters. The simulations were run in three operating conditions:

| | |
|---|---|
| Max Δp, Max mean flow | 210 bar, 46.17 L/min |
| Medium Δp, Medium mean flow | 105 bar, 23.08 L/min |
| Max Δp, Low flow | 210 bar, 4.61 L/min |

The first two cases are needed to evaluate the performances of the machine in conditions that can be normally found in the every day usage of the machine. The third case instead, represent the most critical operating condition for a hydraulic machine low rotation velocity, high pressure. As a low rotation velocity was chosen the rotation velocity that should allow to send the 10% of the maximum needed flow. It can be considered the most critical because in such condition the device must operate against the highest pressure without the help of the high rotation velocity. For the sake of clarity is reminded to the reader that the high velocity increases the Couette term that reduce the leakages and limit the cross port flow phenomenon. For the sake of completeness are shown in the chart below the velocities set in the three operating conditions listed above for every gear set.

| | **Z13 (14.62 [cc/rev])** | **Z14 (13.45 [cc/rev])** |
|---|---|---|
| **46.17 [L/min]** | 340 [rpm] | 369 [rpm] |
| **23.08 [L/min]** | 1698 [rpm] | 1846 [rpm] |
| **4.61 [L/min]** | 3396 [rpm] | 3693 [rpm] |

Also, the simulations were run considering a pressure of the accumulator of 2 bar, and a volumetric efficiency of 0.93 was considered during the calculation of the rotation velocity. The assumption of the volumetric efficiency was used just to select a rotation velocity for the simulations. In the matter of fact, the flow, will be slightly different and depending on the real volumetric efficiency.

$$n = \frac{Q}{\eta_v \cdot d}$$

Finally, to properly read the following plots it's important to know that the X axis represent the rotation of the drive gear starting from the position in which a virtual line that connects the center of the gear and the reference Tooth Space Volume is perfectly horizontal, and the reference Tooth Space Volume is in the meshing zone. To better clarify the concept, below is shown the reference system.

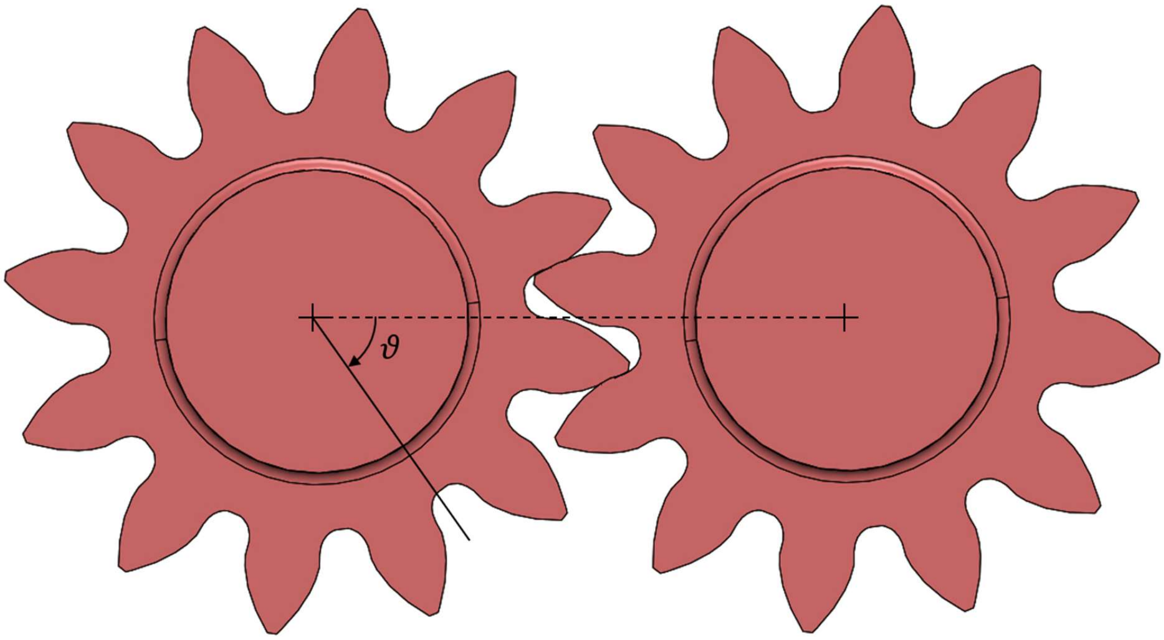



*Figure 69 - Reference system*

## Power consumption

To evaluate the power consumption was multiplied the torque and the rotation velocity. The rotation velocity is a constant, and the torque is a periodic function, so the resulting function is a periodic function. To complete the evaluation was calculated the mean value.

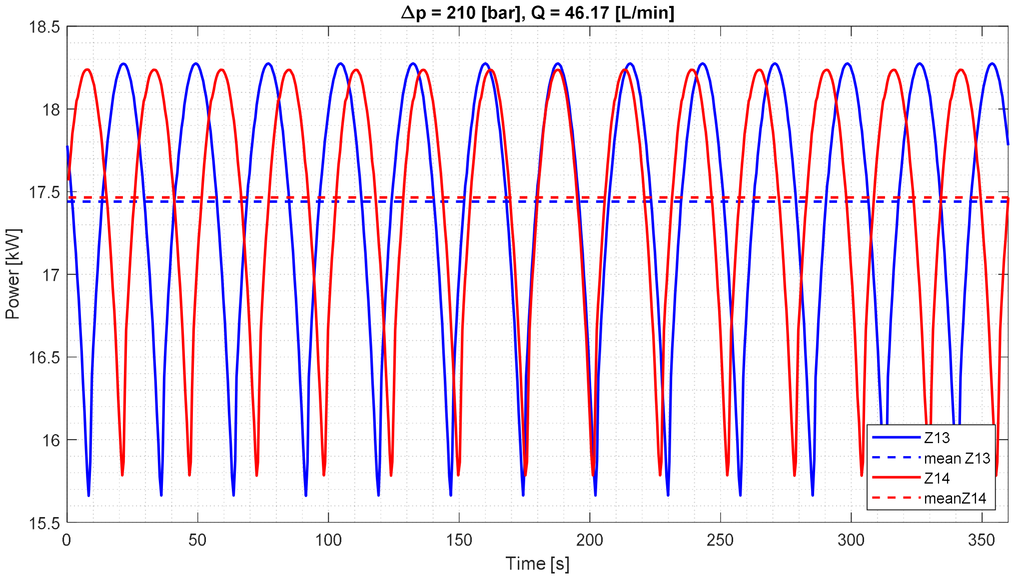



*Figure 70 - Power consumption Δp = 210 [bar], Q = 46.17 [L/min]*

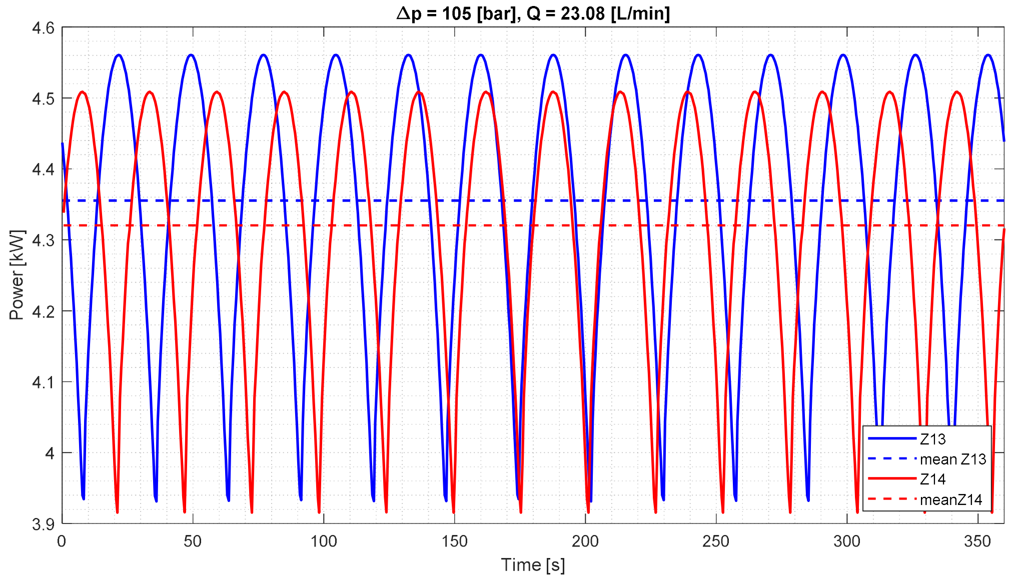



*Figure 71 - Power consumption Δp = 105 [bar], Q = 23.08 [L/min]*

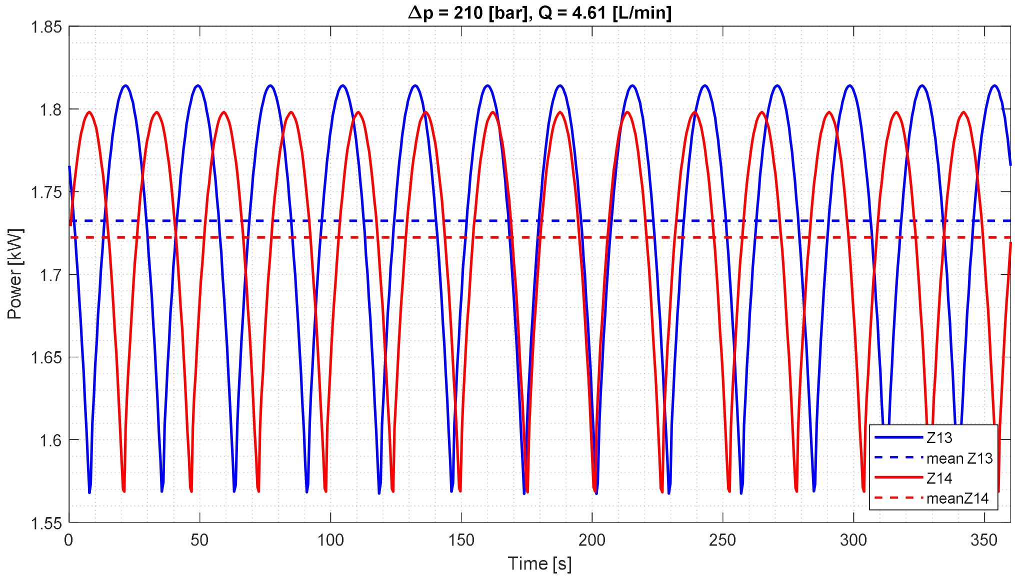



*Figure 72 – Power consumption Δp = 210 [bar], Q = 4.61 [L/min]*

What can be seen in these plots, is that the power consumption doesn't have a smooth curve. In fact, as mentioned before it is the result of the multiplication of the torque and the velocity. The rotation velocity is constant, so the shape shown is the torque. The torque curve depends on the pressurization and depressurization of the Tooth Space Volumes. This can be seen also in the frequency of the curve: because of the higher number of teeth the Z14 gear set has a higher frequency than the Z13.

Moreover, the Z14 gear set has a lower power demand for almost all the range, but not for the maximum flow maximum pressure case. However, the power difference doesn't seem so relevant.

To summarize is shown a chart that summarize the mean power consumption, for both the gear-sets in the three operating condition just shown.

| **Power consumption [kW]** | **Z13** | **Z14** |
|---|---|---|
| **Δp = 210 [bar], Q = 46.17 [L/min]** | 17.43 | 17.46 |
| **Δp = 107 [bar], Q = 23.08 [L/min]** | 4.35 | 4.32 |
| **Δp = 210 [bar], Q = 4.61 [L/min]** | 1.73 | 1.72 |

## Flow ripple

The flow ripple is very important, because it affects the motion of the actuator. A flow ripple equal to zero stands for a perfectly smooth motion of the actuator, on the other hand, a high flow ripple could lead to the cogging of the actuator. The flow ripple is calculated as the difference between maximum and minimum flow value, over the mean flow value. Below are shown the flow ripple and the mean flow value for the three operating conditions. At the end of this section is presented a summarizing chart of the flow ripple value.

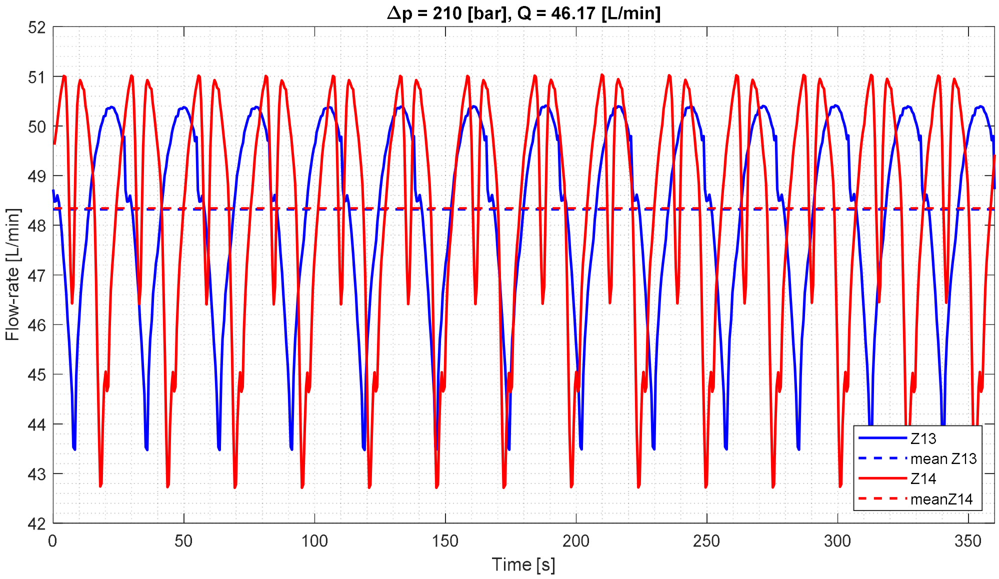



*Figure 73 – Flow-rate Δp = 210 [bar], Q = 46.17 [L/min]*

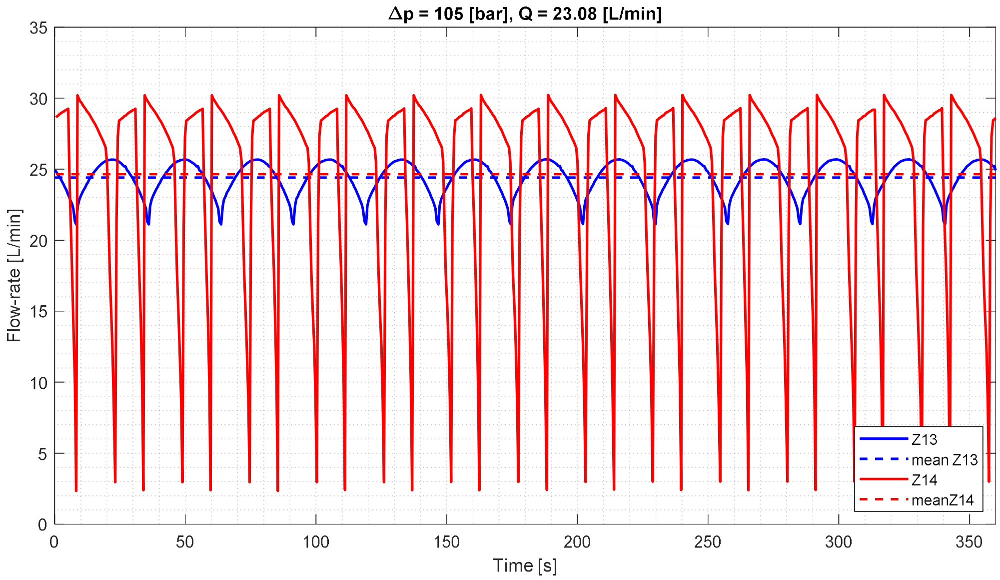



*Figure 74 - Flow-rate Δp = 105 [bar], Q = 23.08 [L/min]*

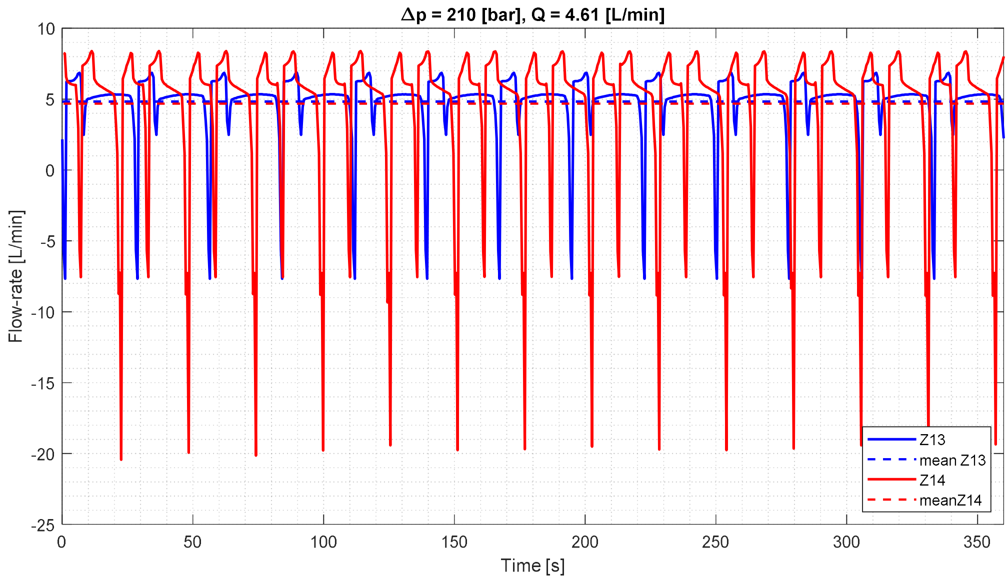



*Figure 75 - Flow-rate Δp = 210 [bar], Q = 4.61 [L/min]*

First, is noticeable the dependency of the flow-rate curve shape from the kinematic flow-rate of the machine. Once again, the curve is periodic, and the frequency of the Z14 gear set curve is higher than the Z13. The mean flowrate is higher than the flowrate expected. This happens because the hypothetical volumetric efficiency used to select the rotation velocity is lower than the actual volumetric efficiency. However, since the mean flow-rate is the same, also the volumetric efficiency is the same.

Below is shown a chart that shows the volumetric efficiency.

| **Volumetric efficiency** | **Z13** | **Z14** |
|---|---|---|
| **Δp = 210 [bar], Q = 46.17 [L/min]** | 0.97 | 0.97 |
| **Δp = 107 [bar], Q = 23.08 [L/min]** | 0.98 | 0.99 |
| **Δp = 210 [bar], Q = 4.61 [L/min]** | 0.97 | 0.94 |

Additionally, the plots highlight a very strong cross-port flow through the relieve groove of the Z14 gear set. More precisely the phenomenon is highlighted by the flow that periodically became negative. A negative value of flow physically means that that fluid goes from the high pressure port to the low pressure port, so invert its direction. This factor is not neglectable, because it can bring to the cogging of the actuator that for obvious reasons is not acceptable.

The problem just mentioned are pointed out also by the flow ripple value. See the chart below for the flow ripple values.

| Flow ripple | Z13 | Z14 |
|---|---|---|
| **Δp = 210 [bar], Q = 46.17 [L/min]** | 0.1436 | 0.1721 |
| **Δp = 107 [bar], Q = 23.08 [L/min]** | 0.1879 | 1.1318 |
| **Δp = 210 [bar], Q = 4.61 [L/min]** | 3.0138 | 6.1572 |

The flow ripple a very low velocity is high also using the Z13 gear set. This is un avoidable, and common in many commercial pumps. The solution to this problem can be limiting the minimum rotation velocity to 500 rpm, and eventually bleed the excess flow sending it to the low pressure zone of the circuit. As a confirmation of what just mentioned, below are shown plots that show how for both the gear sets, independently from the outlet pressure, the low rotation velocity affects the flow rate, promoting the cross port flow. This phenomenon is evident in the Z14 gear-set plot.

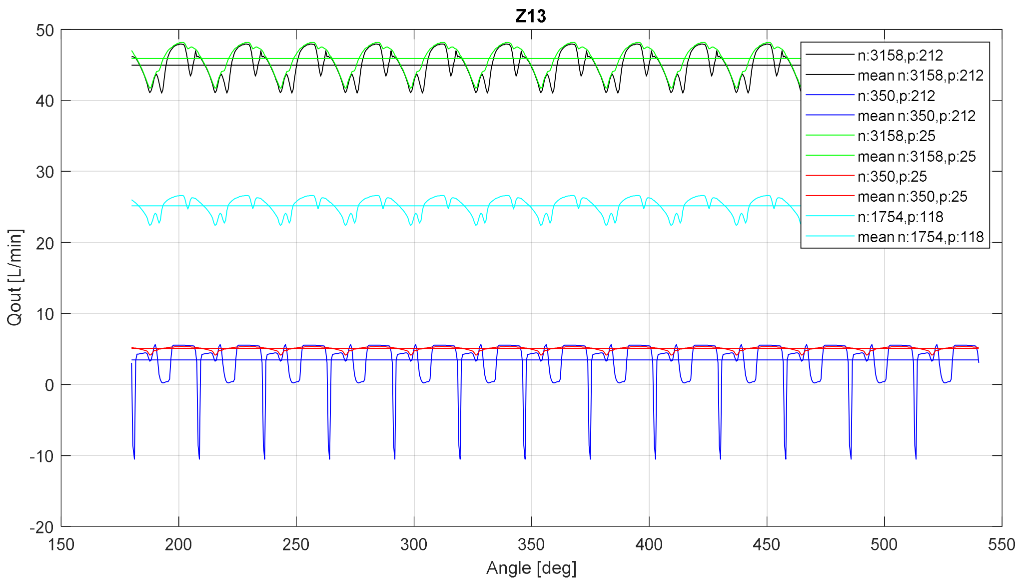



*Figure 76 - Z13 gear set flow-rate*

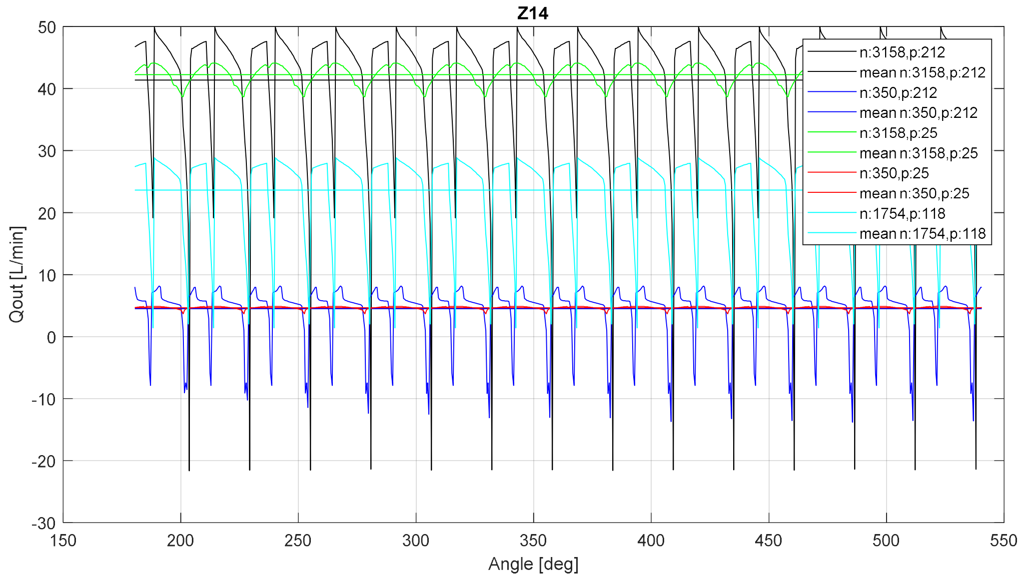



*Figure 77 - Z14 gear set flow-rate*

## Torque ripple

The torque ripple affects the electric motor efficiency and dynamic. A torque ripple equal to zero would reduce the losses of the electric motor and would lead to a reduction of vibration and noise emission. The torque ripple is calculated as the difference between maximum and minimum torque value, over the mean torque value. Below are shown the torque ripple and the mean torque value for the three operating conditions. At the end of this section is presented a summarizing chart of the torque ripple value.

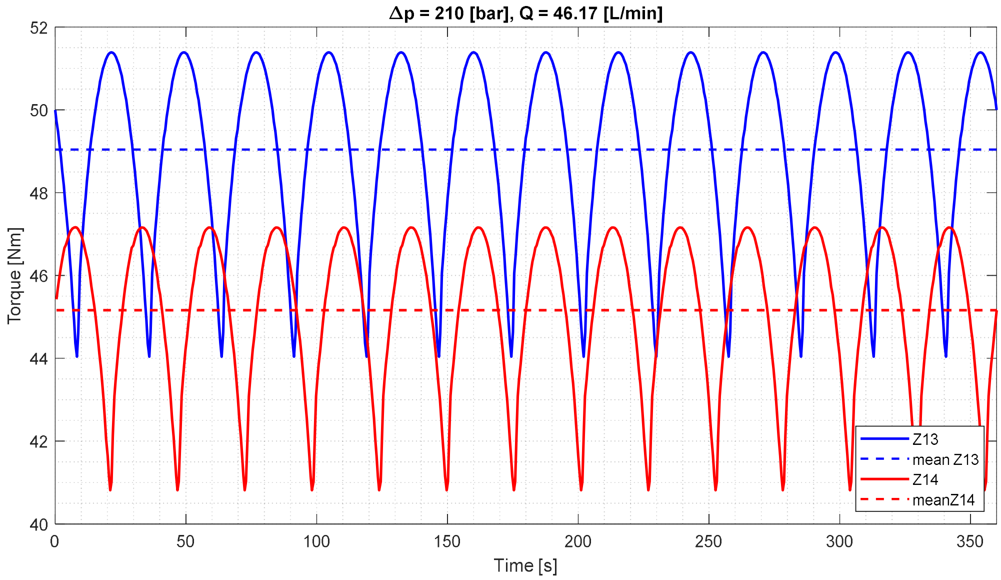



*Figure 78 – Torque Δp = 210 [bar], Q = 46.17 [L/min]*

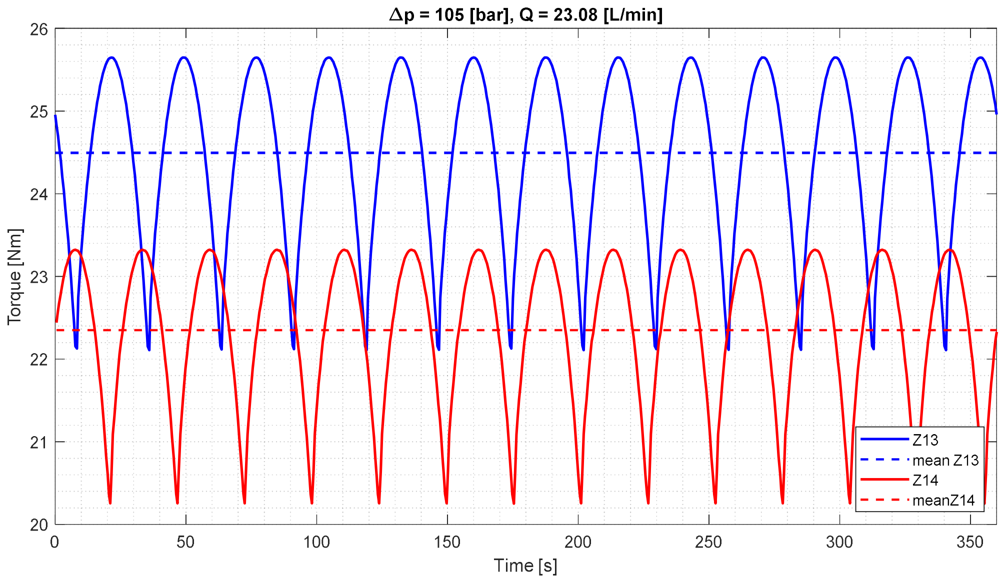



*Figure 79 – Torque Δp = 105 [bar], Q = 23.08 [L/min]*

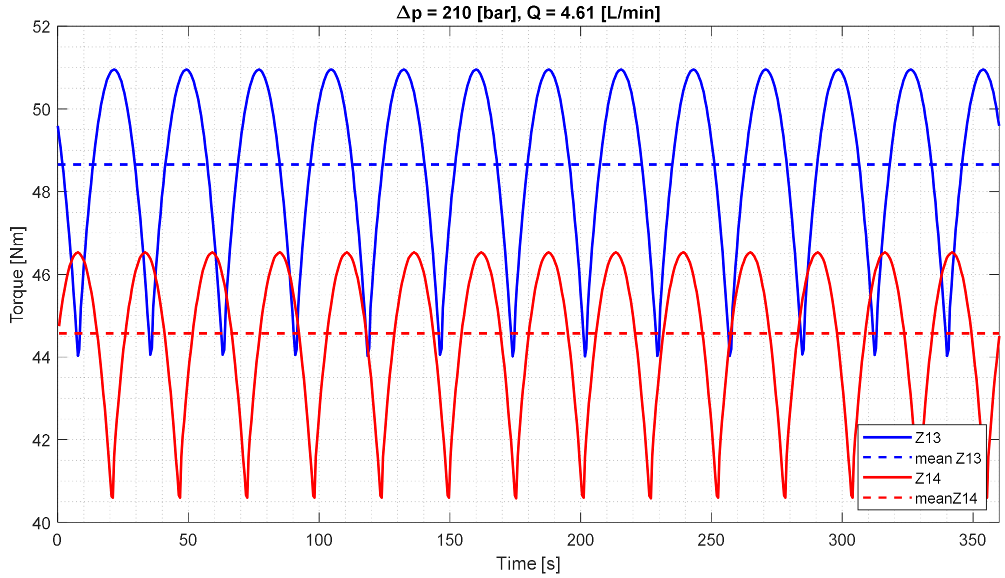



*Figure 80 - Torque Δp = 210 [bar], Q = 4.61 [L/min]*

Looking at the plots it's important to notice that a smaller theorical displacement lead to a lower torque.

$$T \cdot \omega = Q \cdot \Delta p$$
$$T = \frac{n \cdot thDisp \cdot \Delta p}{k \cdot n}$$
$$T = \frac{thDisp \cdot \Delta p}{k}$$

However, the mean torque is not a critical parameter, it's not important as the torque ripple or the power consumption. The power consumption has already been analyzed and the different rotation velocity reduce the difference between the two gear set, the torque ripple, as shown below is different, but not as different as the mean torque value.

| **Torque ripple** | **Z13** | **Z14** |
|---|---|---|
| **Δp = 210 [bar], Q = 46.17 [L/min]** | 0.1498 | 0.1426 |
| **Δp = 107 [bar], Q = 23.08 [L/min]** | 0.1445 | 0.1373 |
| **Δp = 210 [bar], Q = 4.61 [L/min]** | 0.1426 | 0.1334 |

## Volume

The last objective function of the optimization is the minimization of the volume. The volume occupied by the pump is indirectly valuated by the longitudinal length of the pump defined as the sum of the nominal interaxis of the gear and the outer radius of both the gears.

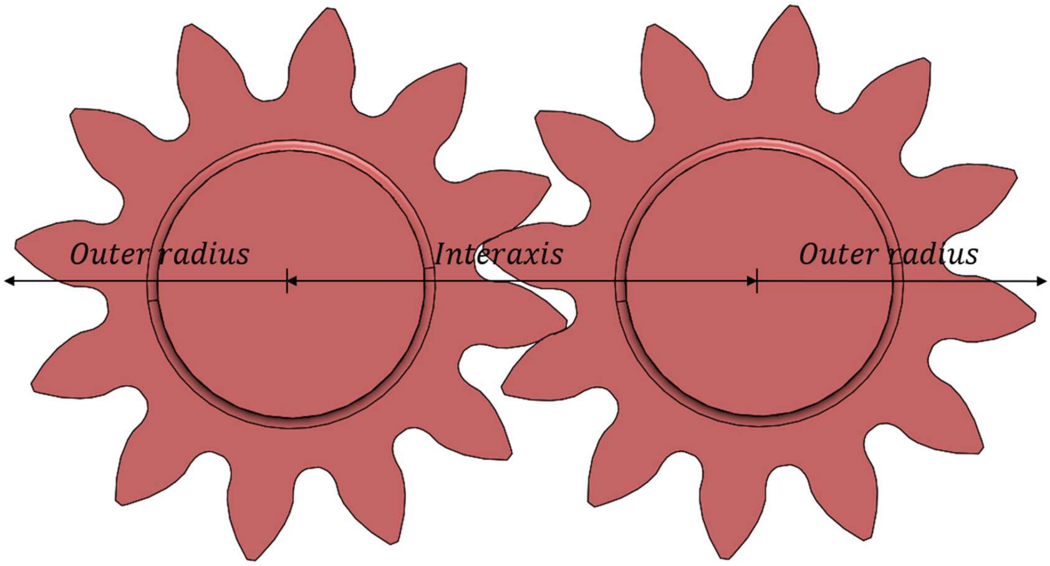


*Figure 81 - Longitudinal length of the pump*

| **Footprint [mm]** | **Z13** | **Z14** |
|---|---|---|
| **Interaxis** | 37.51 | 37.05 |
| **Outer radius** | 21.75 | 21.30 |
| **Length** | 81.01 | 79.65 |

The difference of volume footprint is 1.36 millimeters, so neglectable.

### Selection

After a careful examination it’s possible to see that the Z14 gear set is slightly better in power consumption, torque ripple and volume footprint. However, the flow ripple is very high, and unacceptable. On the other hand, the Z13 gear set is more balanced, and even though it has not the best result in terms of power consumption, torque ripple and volume footprint, its performances are more balanced and moderate flow ripple. For this reason, it was decided of proceeding with the Z13 gear set. Additionally, the poor performances demonstrated by the flow rate of the 14 teeth gear set added another reason to verify the gear set procedure design step by step to validate the C++ model of HYGESim.

## Z13 gear-set examination

Below are shown the steps and the results obtained using HYGESim in all its modules. Also, to a better understanding of the plots, is reminded to the reader that the x axis refers to the reference system described in the previous section. Noticed that the device hereafter analyzed can handle the fluid that goes in both directions. For this reason, it’s not exact speaking about high pressure zone, and low pressure zone, but for the sake of clarity in the following paragraphs, the explanations will refer to a well-defined operating condition where the high pressure zone and low pressure zone location is known.

## Geometry module

Below are shown some geometric features of the gear sets.

### Volumes

In the plot below are shown two corresponding Tooth Space Volumes. By corresponding tooth space volumes is meant the two volumes that belong to the two gears and came together in the meshing zone.

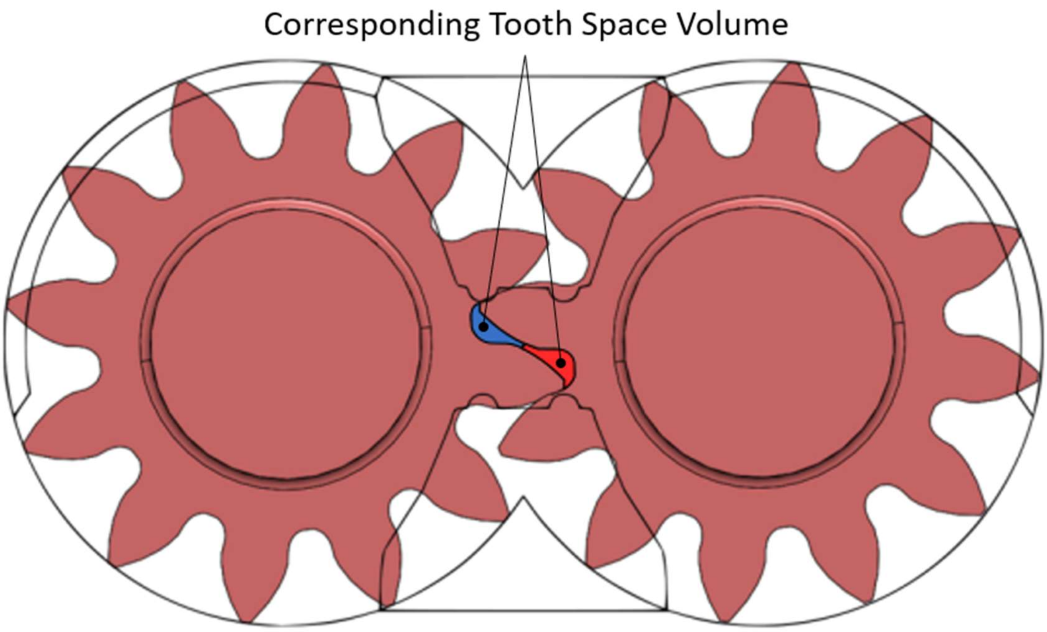



*Figure 82 - Corresponding tooth space volumes*

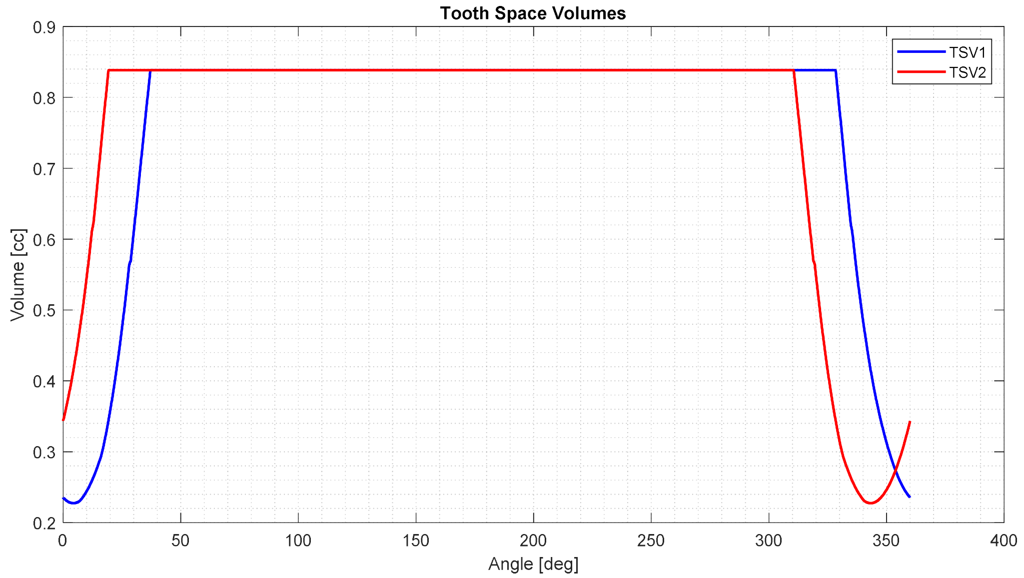



*Figure 83 - Corresponding tooth space volumes*

Summing the two Tooth Space volumes, calculating the differential value of the combined volume, and multiplying it for the number of teeth it's possible to calculate the kinematic displacement.

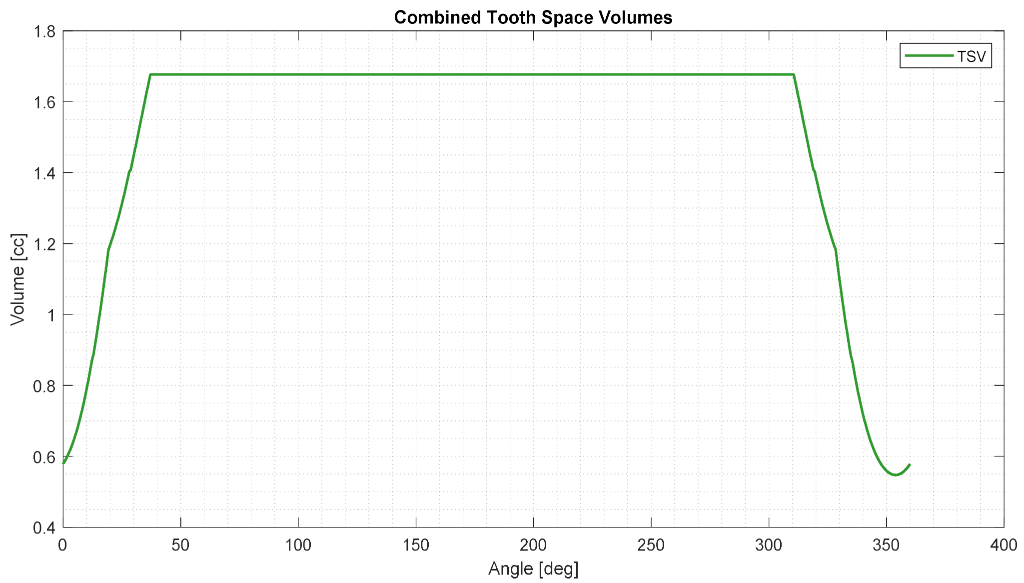



*Figure 84 - Combined TSV*

### Depth-wise connections

Below are shown the opening area of the depth wise connections. Usually, by connection is meant the connection of the TSV with another area part of the pump. Again, the opening area values are represented as a function of the rotation angle of the drive gear. Before the plots is shown a cross section of the pump. The angle $\alpha$ is named casing angle, and it's the angle that define when the casing starts to close the depth-wise connection of the Tooth Space Volume. Usually it's possible to find a low-pressure casing angle, and a high-pressure casing angle, but in this case, since the ports are symmetric they are equal. For this reason, they will be called only casing angle. In this pump the casing angle is 56.30 degrees. A deeper description of the design will be made in the latest section, for now, the reader should focus only on the angle in which the TSV start to be closed by the casing, and the angle in which it is completely closed. The same behavior can be noticed in the opening toward the high pressure area. The connection of the TSV and the high pressure zone is called HP connection, the connection with the low pressure zone instead is called LP connection.

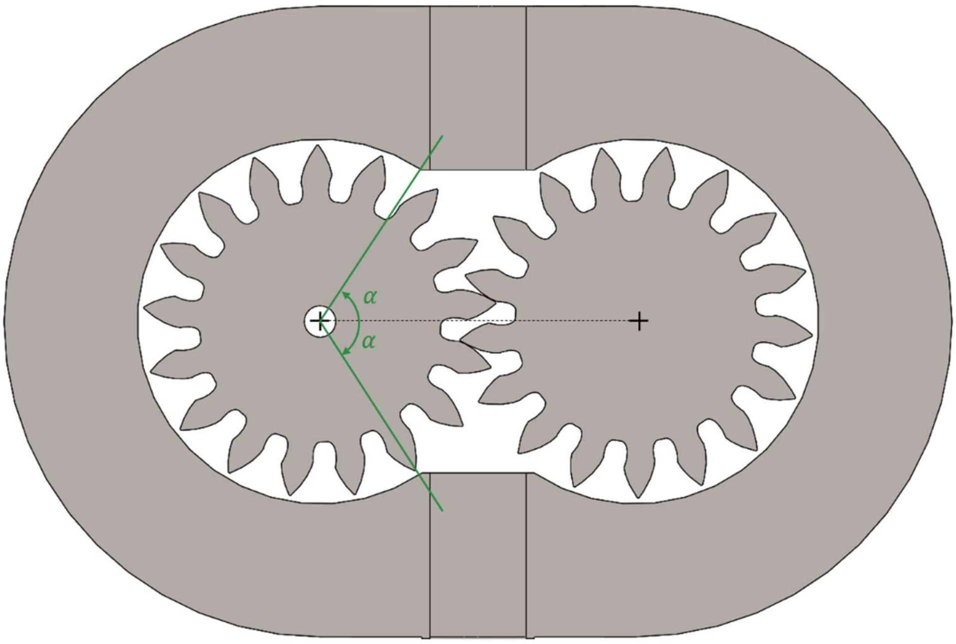



*Figure 85 - Casing angles*

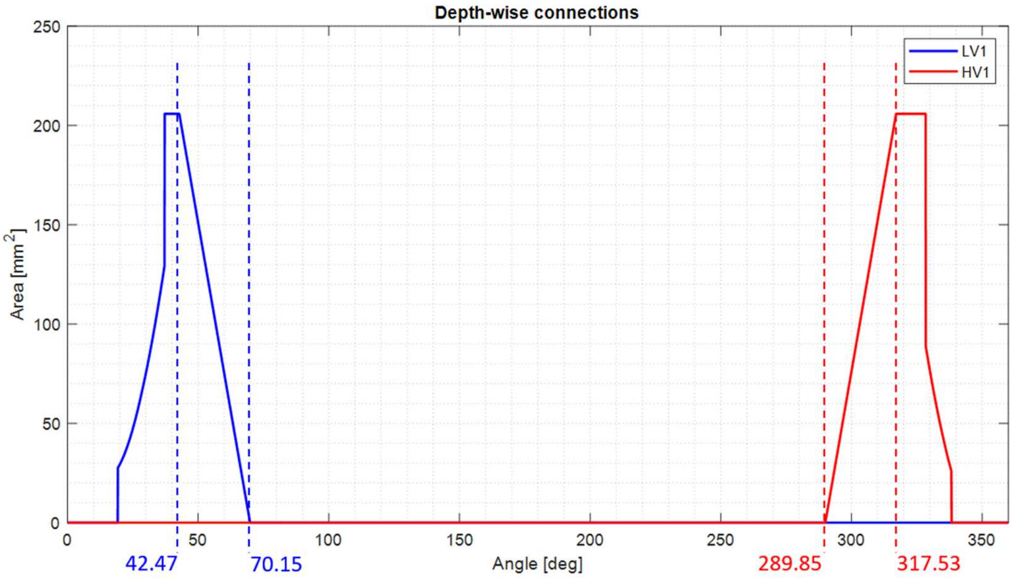



*Figure 86 - Tooth Space Volume depth-wise connection*

In the plot above is shown the HV and LV areas. What can be noticed is that LV is non zero only right after the meshing zone, and on the other hand HV is non zero only before the meshing zone. Also, notice that because of the meshing, the two curves are not symmetrical. By meshing is meant that the drive gear pushes the driven gear by the contact of their tooth, and the contact break the symmetry. However, the slope that describe the closing of LV and the opening of HV are symmetrical. Considering that the casing angle is 56.3 degrees, LV starts to close at 56.3 – (360/13)/2 = 42.47 degrees, and it's completely closed at 56.3 + (360/13)/2 = 70.15 degrees. The same concept can be applied to the HV connection. It starts to open at 360 – 56.3 – (360/13)/2 = 289.85 degrees and it closes at 360 - 56 + (360/13)/2 = 317.53 degrees.

Below is shown a picture that ease the understanding the reading of the just shown plots.

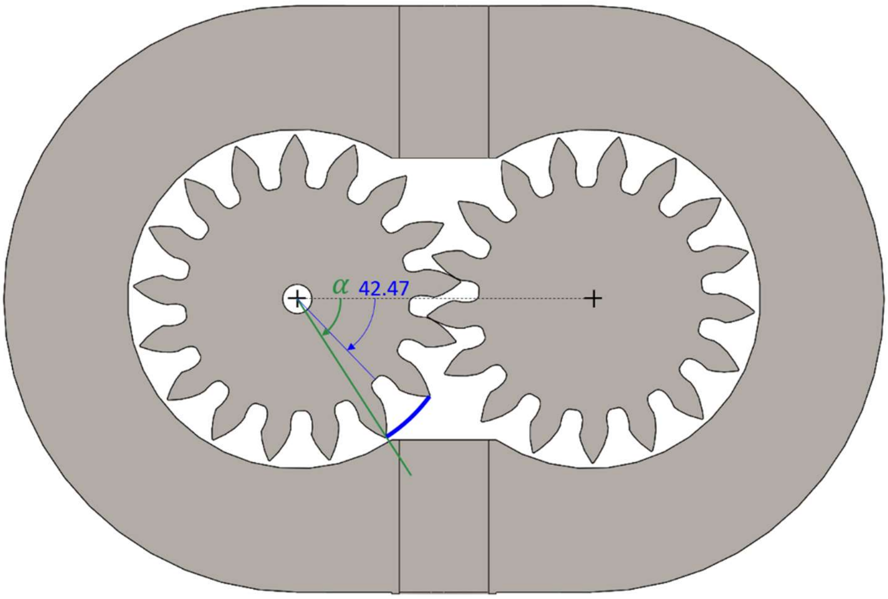



*Figure 87 - Beginning of the closing phase of LV1*

In the plot just presented the connection areas refers to the drive gear. The areas of the driven gear are very similar to the one just presented but they are translated of a small angle. More precisely the angle necessary for the meshing of the two gears.

For the sake of completeness will be added also the so called FG and CL connection. These connections are connection generated in meshing zone, and for this reason they are equal to zero outside it. The FG connection is the connection between two corresponding Tooth Space Volumes. The CL connection is the connection between a Tooth Space Volume and the previous Tooth Space Volume of its corresponding one. Even if they are difficult of interpret they doesn't show anomalous trends.

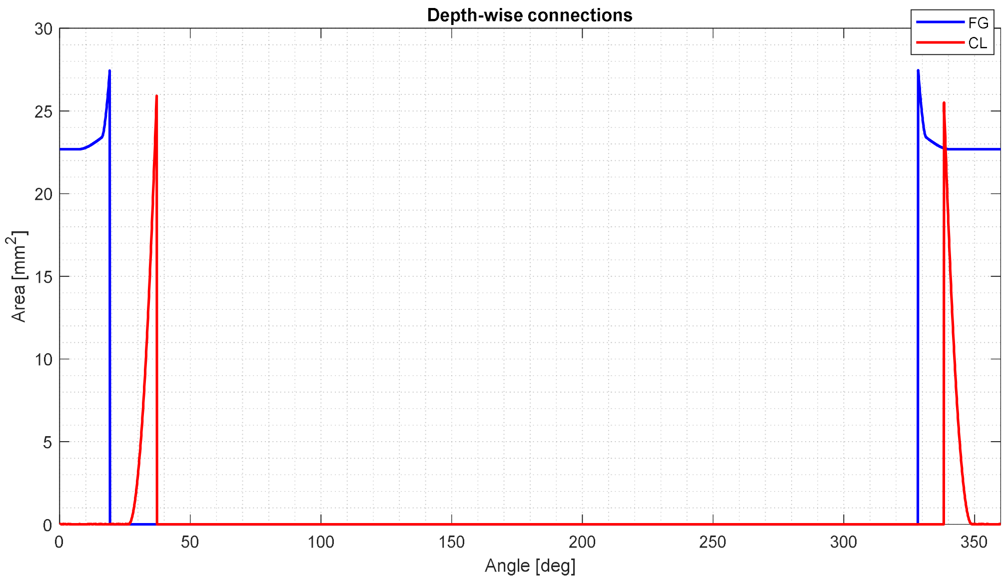



*Figure 88 - FG and CL connections*

## Face-wise connections

Another very important connection that can be used to verify the correct calculation of the geometry module is the connection between the Tooth Space Volume and the back-flow groove. For the sake of simplicity this connection is named HS connection.

Before moving further is reminded to the reader that the back-flow grooves are very important to ensure a smoother pressurization of the Tooth Space Volume and to allow it to reach the high pressure before it links to the high pressure zone. To allow this the back-flow grooves are connected to the high pressure zone. But the device analyzed in this thesis is a four quadrant hydraulic machine, so, the high pressure zone position is not defined and can be in both ports. For this reason, the grooves are not connected to any port, but they just connect the adjacent teeth in the pressurization area. Below is shown an example of plate with this peculiarity.

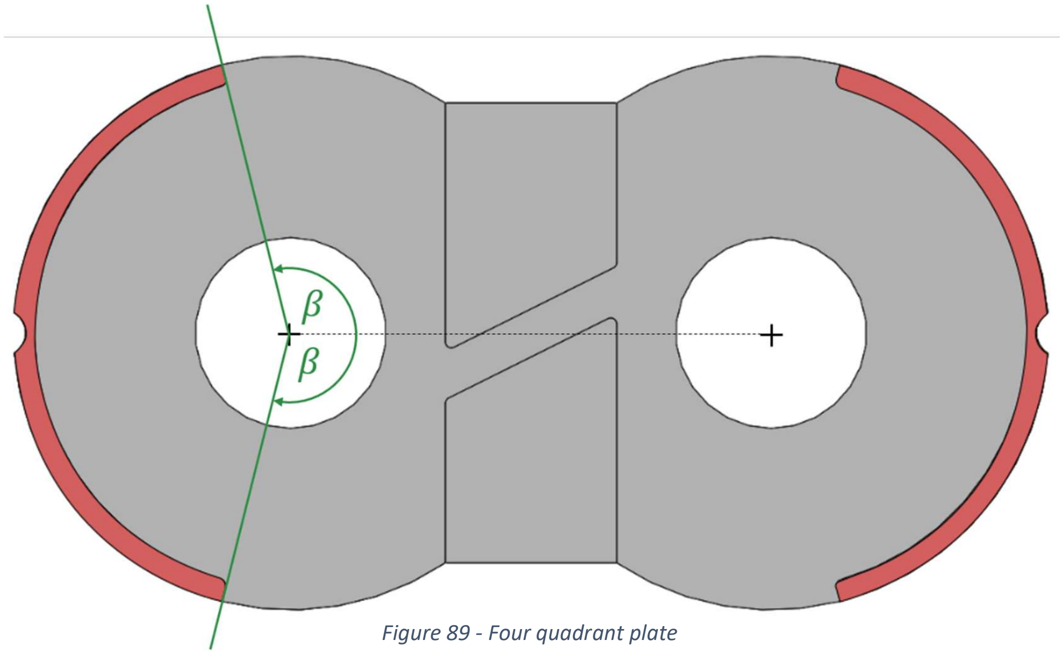



*Figure 89 - Four quadrant plate*

More precisely the angle $\beta$ is 104.5 degrees. Again, since the device has to be able of pumping in both direction the geometry of the back-flow grooves is symmetrical.

Such a plates geometry gives the following plot as a result.

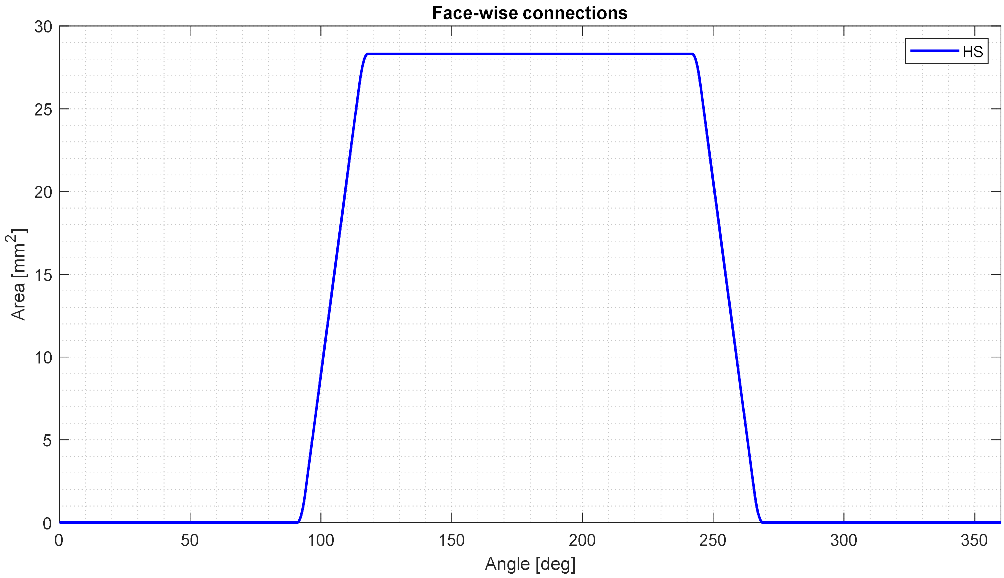



*Figure 90 - Back-flow grooves connection*

Again, can be noticed that the HS connection doesn't start exactly at 104.5 degrees, and the maximum area is reached only after 104.5 degrees. The reason for such a result is the selection of the reference system likewise the previous section.

Is reminded to the reader that in the fluid dynamic module this connection is modelled as an orifice. To properly use the orifice equation the smallest area is needed. For this reason, in the fluid-dynamic module this area is limited to a smaller value. More precisely the smaller value is the cross section area of the back-flow groove. The following images show the area used as soon as the Tooth Space Volume starts to overlap the back-flow groove and the area of the back-flow groove cross section used after a small rotation.

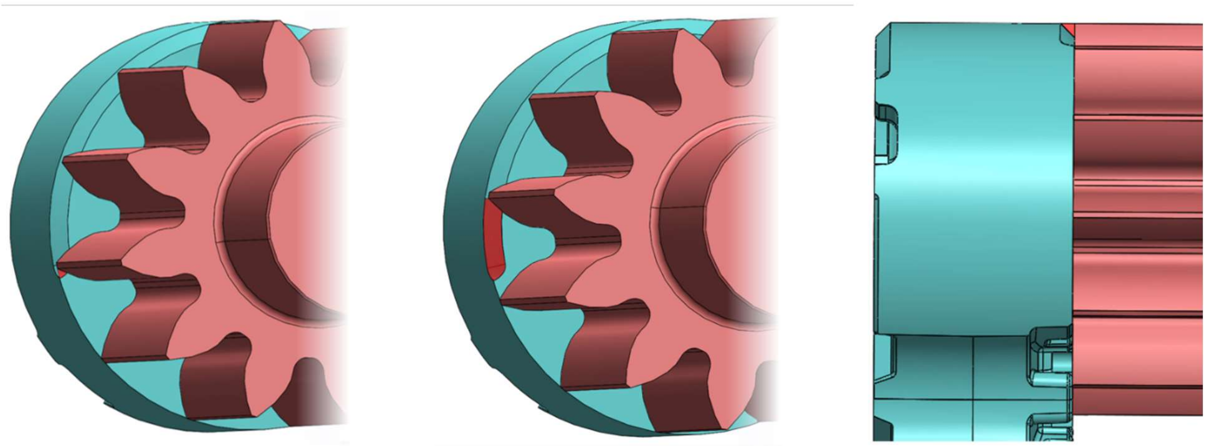


*Figure 91 - Limitation of the HS connection in the fluid-dynamic module*

The results given by the geometry module seems feasible.

For completeness below are shown the other face-wise connection: the connections of the Tooth Space Volume with the high pressure and low pressure zone through the relieve grooves. Likely the FG and CL connection, the physical interpretation is slightly more difficult, but still they can be used to verify that the results are feasible.

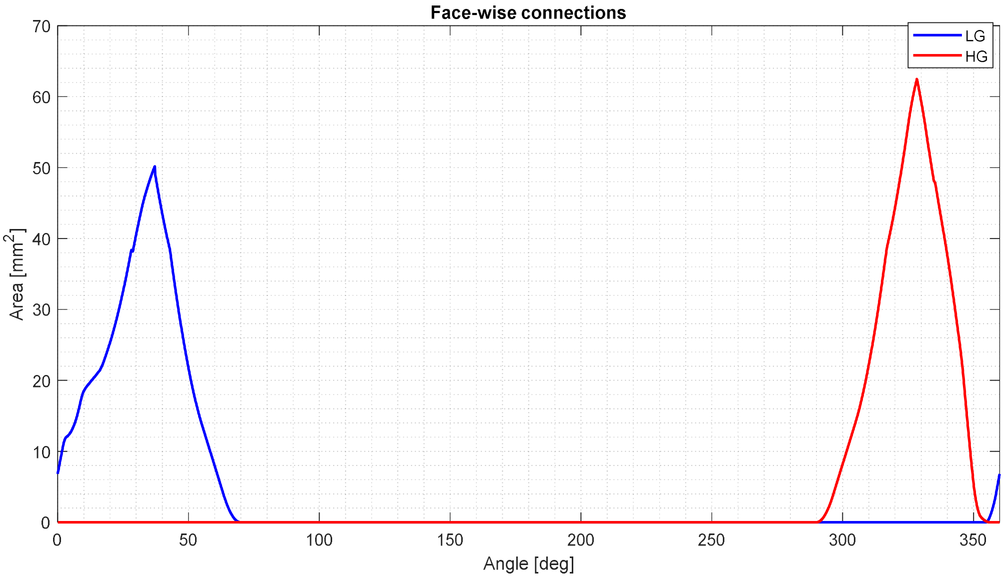



*Figure 92 - LG and HG connections*

Obviously, the connection with the low-pressure zone is right after the meshing zone, and on the other hand the connection with the high pressure zone is right before. Also, zooming in the meshing zone is noticeable that there is a small rotation range in which both the areas are non-null. This opening allows the cross-port flow. Below is shown a zoom of this point.

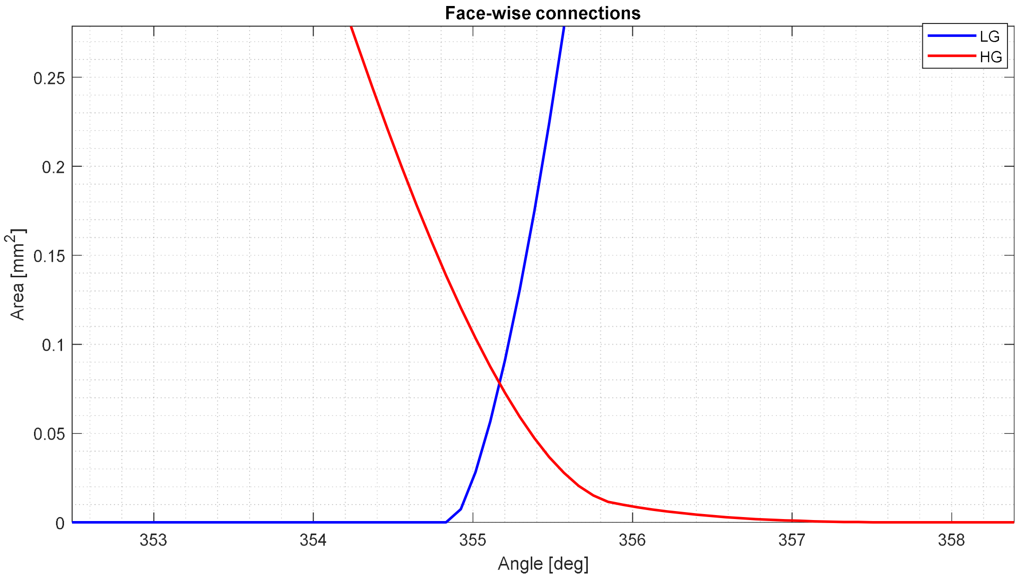



*Figure 93 - LG and HG connection, meshing zone zoom*

## Fluid-dynamic module

Below are explored the results of the fluid-dynamic module. First, will be analyzed the results of the model run in F mode, and then in FFI mode. Both the models calculate the pressure in the volumes of the pump, and the flows through its connections. However, the F mode, is characterized by the gear fixed position. The user can place the gear everywhere, but they are fixed, and not affected by the typical micromotion due to the pressurization of the volumes that surround them. The following step is run the simulation in FFI mode. The FFI mode calculates the forces generated by the pressure in the volumes that surround the gears, and then moves the gears consequently. Therefore, the FFI mode includes the Loading Module and the Micro-motion Module.

### F mode

As said before the F mode only calculate the pressure and the flows in the external gear machine.
To better understand the results presented it's important to keep in mind the reference system presented in the section before. Also, when it comes to the analysis of the fluid-dynamic module can be very useful to see to the Tooth Space Volume pressurization profile. The curves presented below show how the pressure of the Tooth Space Volume changes with the rotation of the gear.

All the results presented show what happen if the machine rotate at 2000 rpm, and if its outlet is connected to an infinite volume. More precisely, the results are obtained using the ideal Volume Termination circuit.

First, are presented the results obtained keeping the gears in the center of the casing.

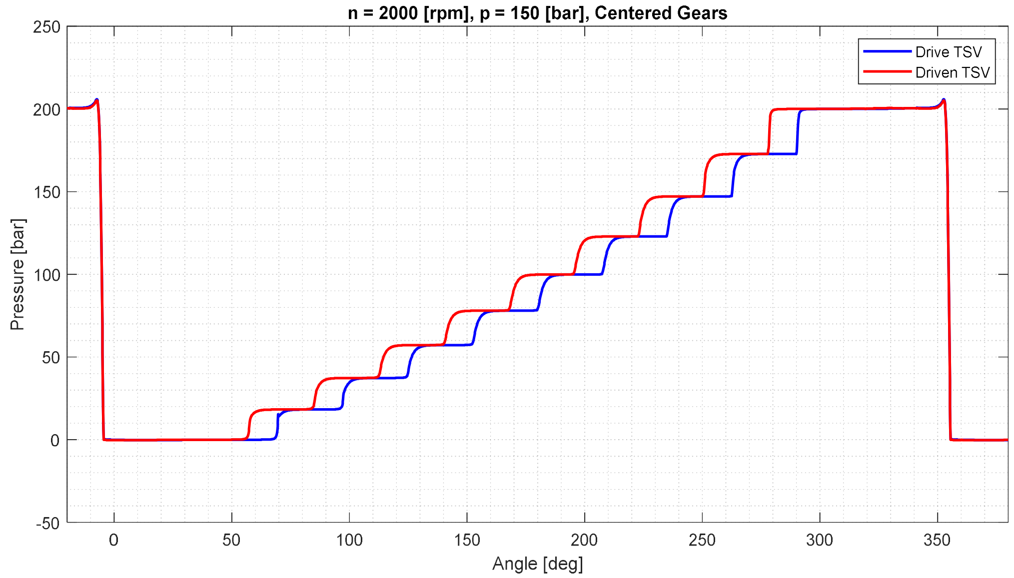



*Figure 94 - Tooth Space Volume pressure profile, centered gears*

Here can be notice 4 phases. The first is the one right after the meshing one.
In the first phase the Tooth Space Volume fill sucking fluid from the suction port.

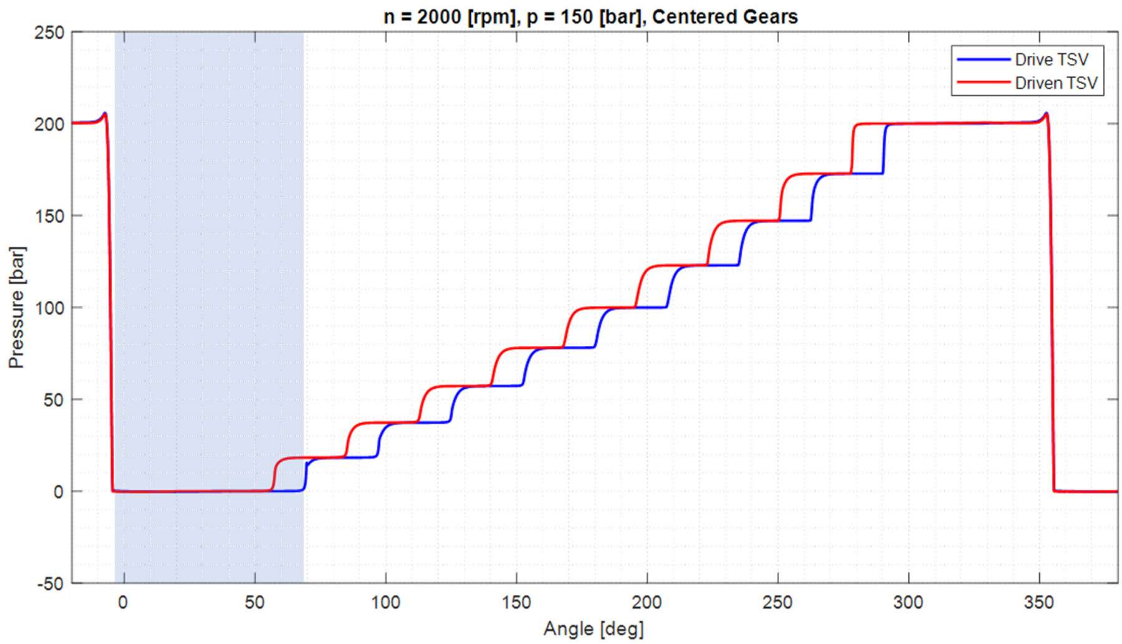



*Figure 95 - Filling phase*

In this phase the cavitation phenomenon can happen. In the figures below is shown the Tooth Space Volume position and a zoom of the curve right after the meshing zone. From the curve is possible to see that the cavitation phenomenon happens.

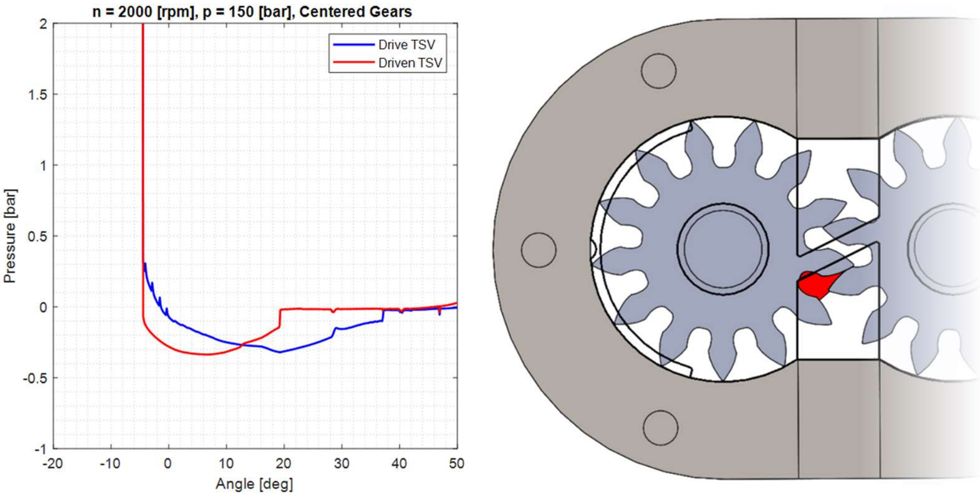



*Figure 96 - Zoom in the cavitation zone, and corresponding TSV position*

The second phase is the pressurization phase. In this phase the Tooth Space Volume gradually increase its pressure until it reaches the outlet pressure value.

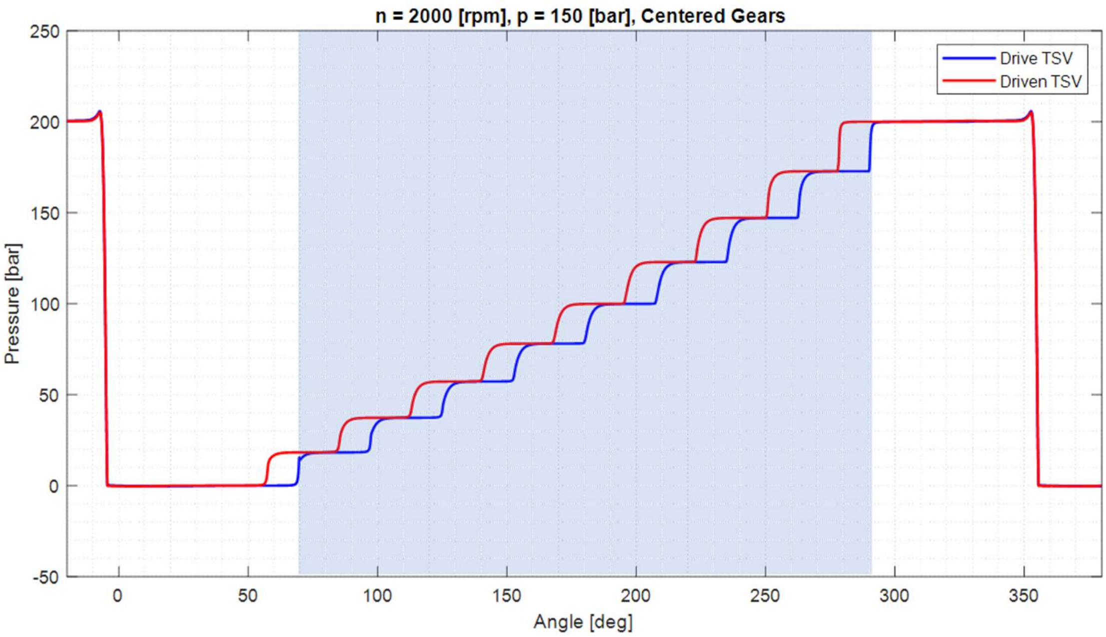
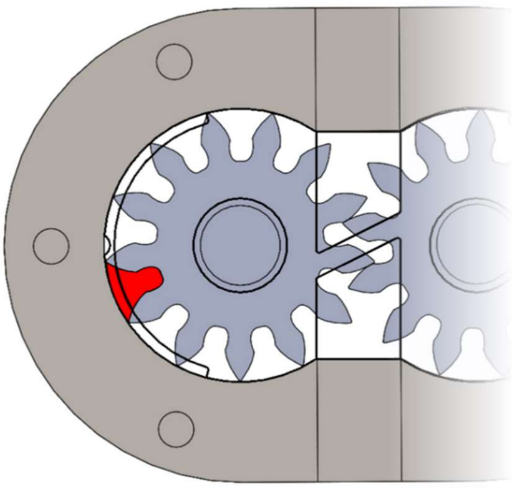



*Figure 97 - Pressurization phase*

The peculiar trend of the pressure profile in this zone can be explained thinking at the Tooth Space Volume as a chamber insert in a long chain of chambers and flow constrictions. At one side of this chain there is a high pressure source, and at the other side there is a low pressure source. Depending on the position of the Tooth Space Volume and the number of flow constrictions between itself and the high pressure source and the low pressure source the Tooth Space Volume build-up a different pressure. More precisely, looking at the plot above, every step of the curve represents a balance point in which the pressure is stable. Than, the rotation of the gear moves the Tooth Space Volume closer to the outlet reducing the number of flow constrictions, and increasing the pressure in the Tooth Space Volume itself.



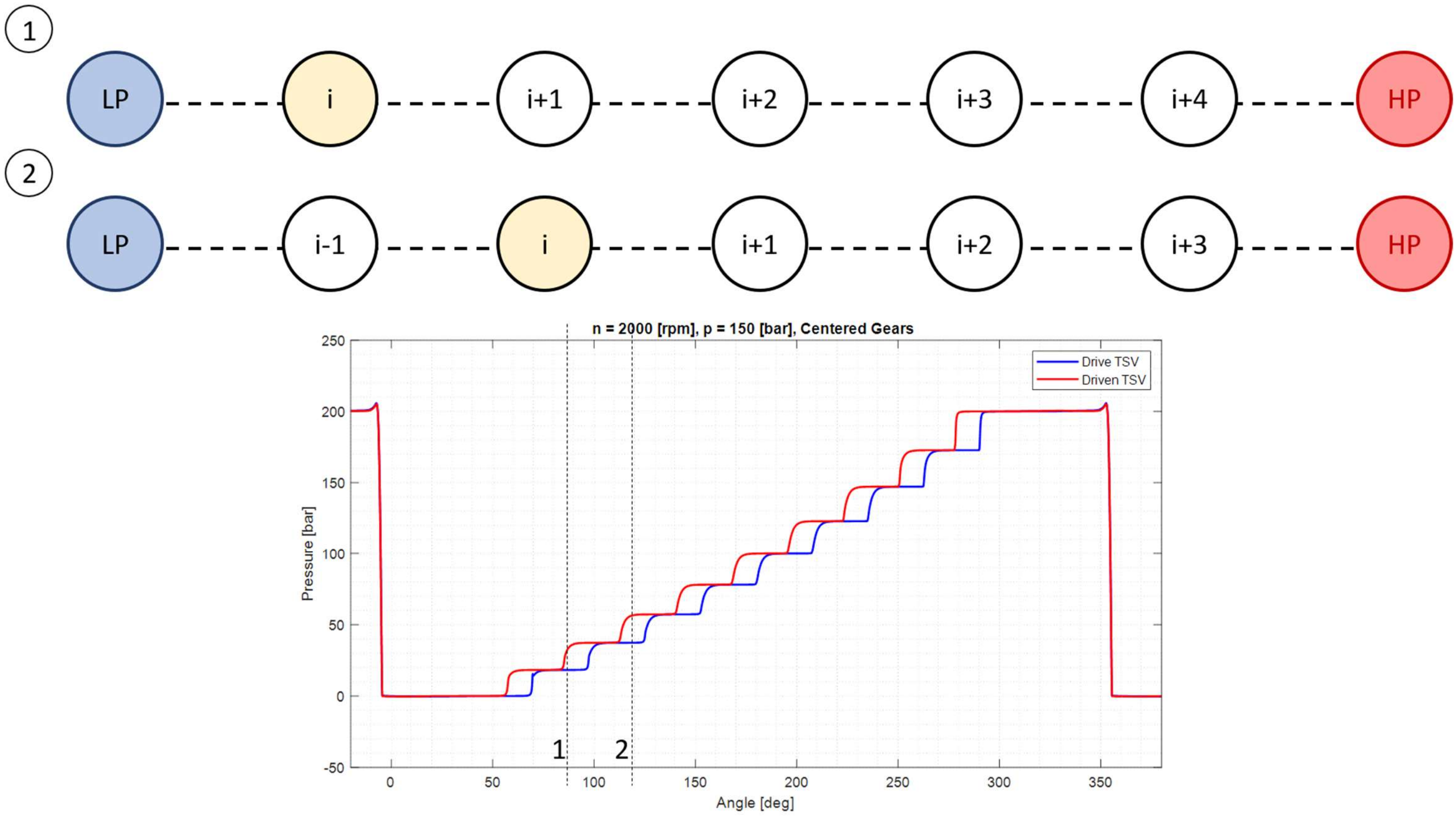



*Figure 98 - Chain of chambers*

The third phase is characterized by the connection of the Tooth Space Volume to the high pressure volume.

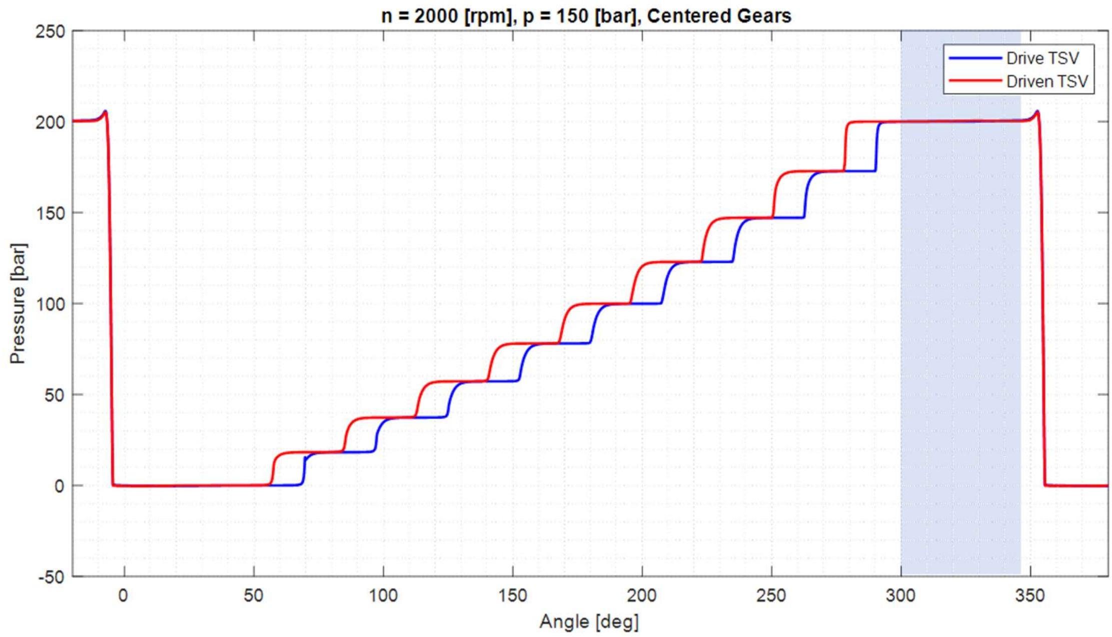



*Figure 99 - High pressure connection phase*

The fourth phase is the meshing phase. Here the Tooth Space Volume is reduced, and the fluid is squeezed toward the outlet port. In the plot can be seen that there are small peaks right before the depressurization. Those peaks happen because the fluid is almost trapped, and the connection with the outlet and inlet volume through the relieve grooves are not large enough.

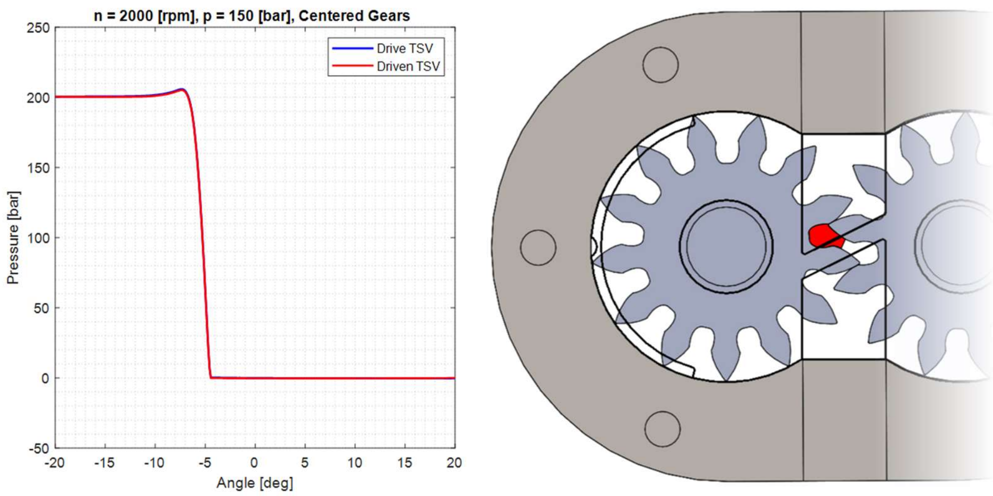



*Figure 100 - Meshing zone pressure peaks*

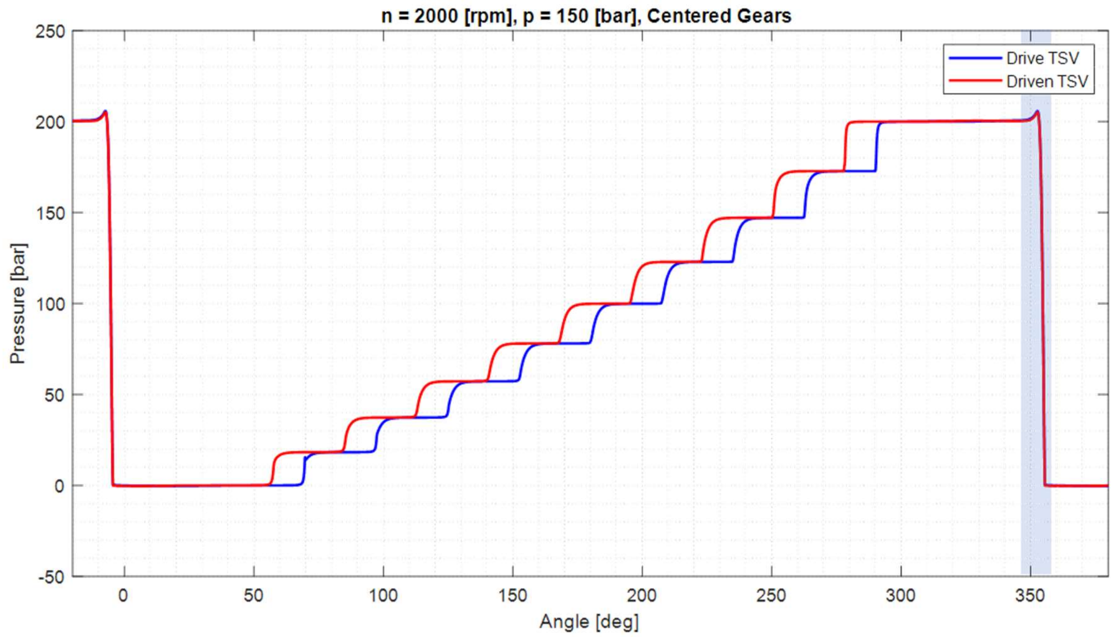



*Figure 101 - Meshing zone*

The results just presented are strongly unnatural. They are unnatural because in normal operating conditions the gear moves, because of the non-homogeneous pressure distribution. More precisely they move toward the inlet port. For this reason, the next step consists in moving the gears toward the inlet port. Since it's just an assumption the gears are arbitrarily moved without any variation of interaxis. Also, they are moved of 60 microns, that is the nominal radial clearance of the needle bearing.

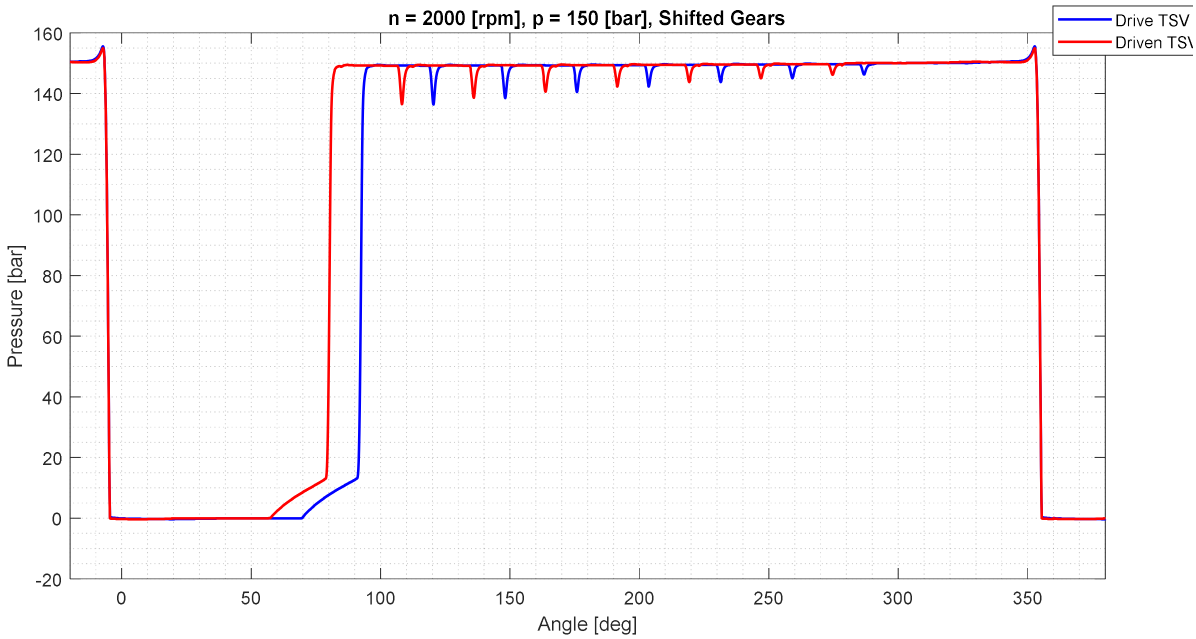



*Figure 102 - Tooth Space Volume pressure profile, shifted gears*

The shifting of the gears toward the inlet port, modify the pressure profile plot as shown above. This trend can be explained thinking to how the fluid constrictions are modified by the motion of the gears. More precisely, instead of a discrete variation of number of fluid constriction, there is also a variation of every fluid constriction itself: the fluid constrictions areas become bigger as they move closer to the outlet port. The image below should ease the reader understanding.

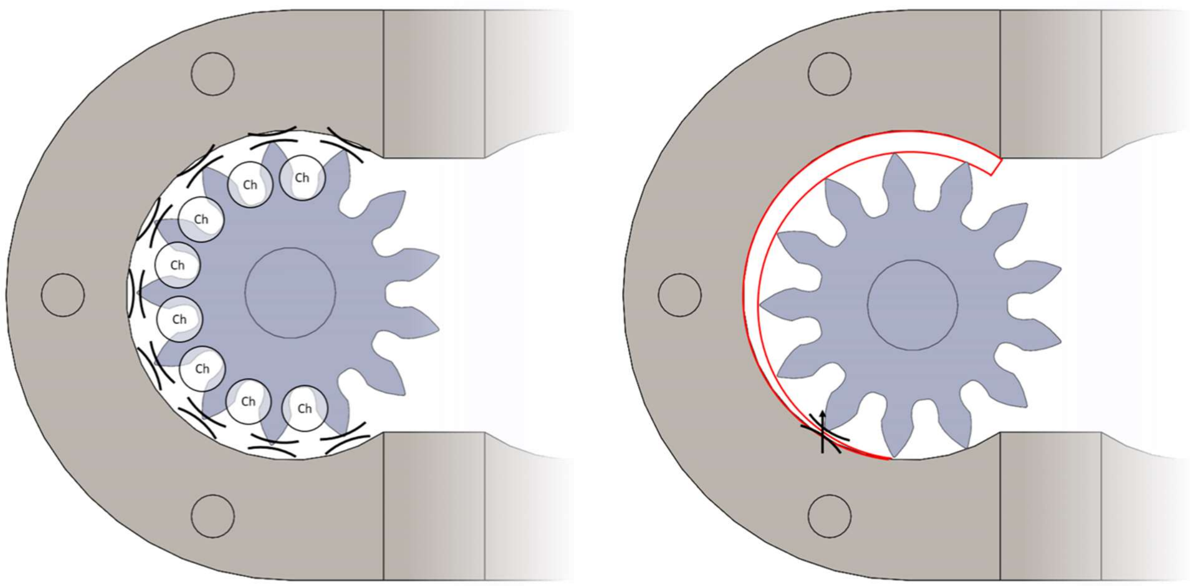



*Figure 103 - Centered gear and shifted gear fluid constrictions*

The bigger fluid constriction areas toward the outlet port, and the very small fluid constriction areas close to the inlet port allow the Tooth Space Volumes to reach the outlet pressure earlier and keep it for almost all the pressurization phase. Another thing that can be seen is the peaks that point out the sudden decrease of pressure during this phase.

The origin of these peaks is the approaching of the Tooth Space Volumes to the back-flow grooves.

In fact, the connection of the Tooth Space Volume pressure with the back-flow groove represent a sudden decrement of the fluid constriction and increasing of flow toward that Tooth Space Volume. This cause a perturbation in the following Tooth Space Volumes pressure. More precisely, the perturbation will be weaker as the Tooth Space Volume is closer to the outlet port and further from the back-flow grooves beginning.

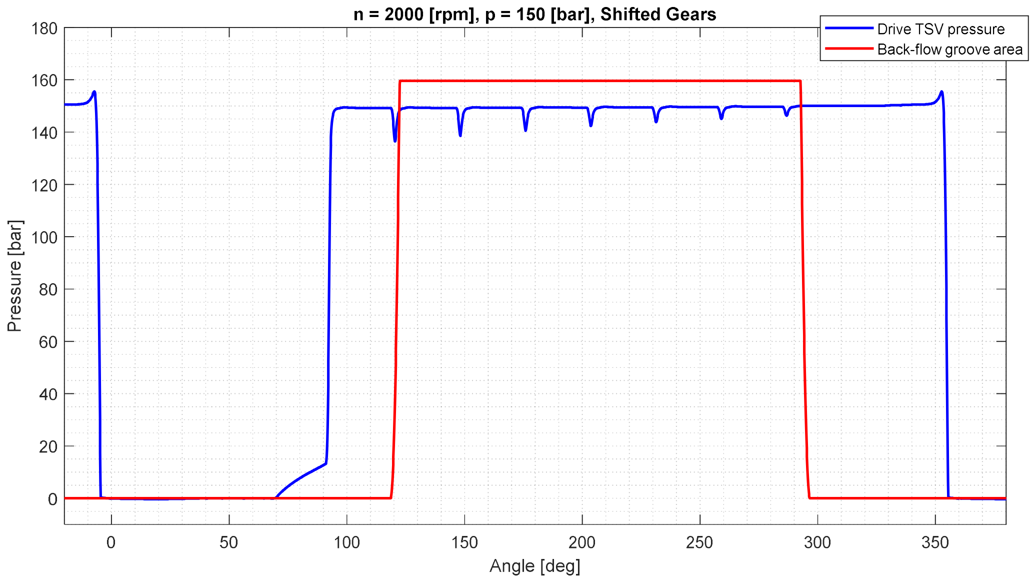



*Figure 104 - Pressure perturbation in the pressurization area*

After having explored the Fluid-Dynamic model in F mode, it's necessary to move toward the FFI mode. Is reminded to the reader that the FFI mode still calculate the flows and the pressure but taking in account the motion of the gears due to the forces generated by the pressure.

### FFI mode

In the plot below is possible to see how the Tooth Space Volume pressurization profile is similar to the one obtained in the last section. This demonstrate that the assumption of the gear kept toward the inlet port is completely feasible. Also, is reminded to the reader that the simulations were run imposing a velocity of 2000 rpm, and an outlet pressure of 150 bar. Additionally, is brought to the attention of the reader that since the model is moving to the actual case, the inlet pressure is set to 2 bar to simulate the presence of the accumulator instead of the tank. First, will be presented the results obtained using the ideal Volume Termination circuit, and then the more realistic Restriction Termination circuit.

Looking closer to the curves is possible to see differences, for example the reduction of the depressurization peak in the pressurization phase, the modification of the pressure peaks in the meshing zone, and the different trend in the filling phase.

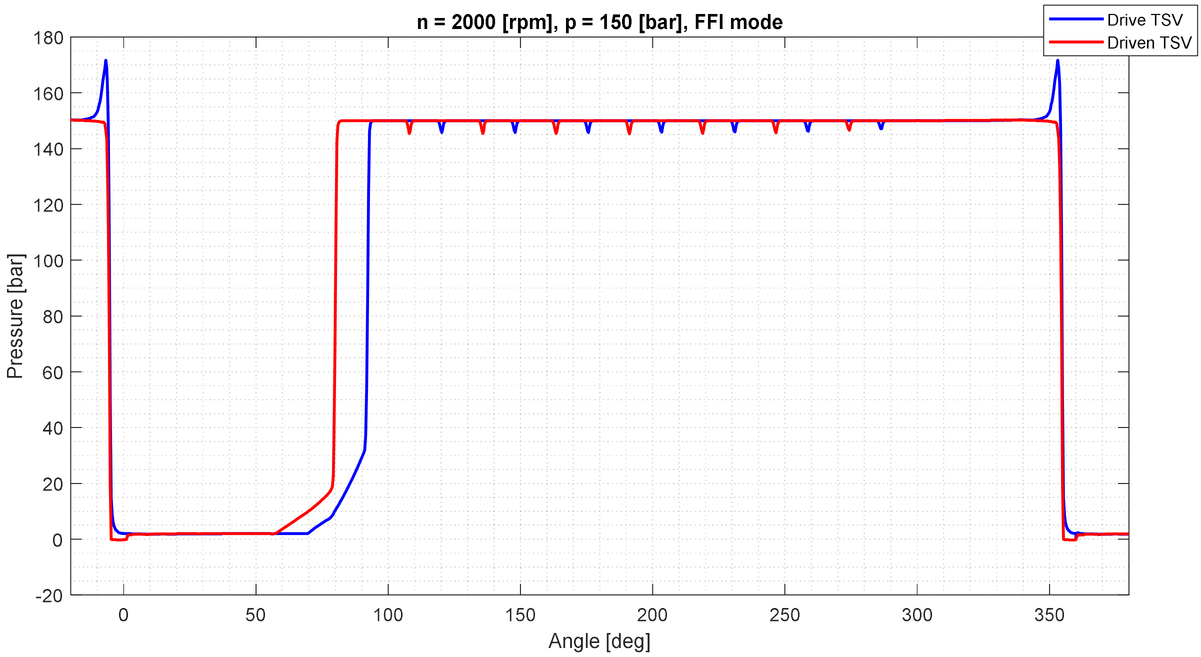



*Figure 105 - Tooth Space Volume pressure profile, FFI mode*

Below are shown some details that distinguish this case from the previous one.

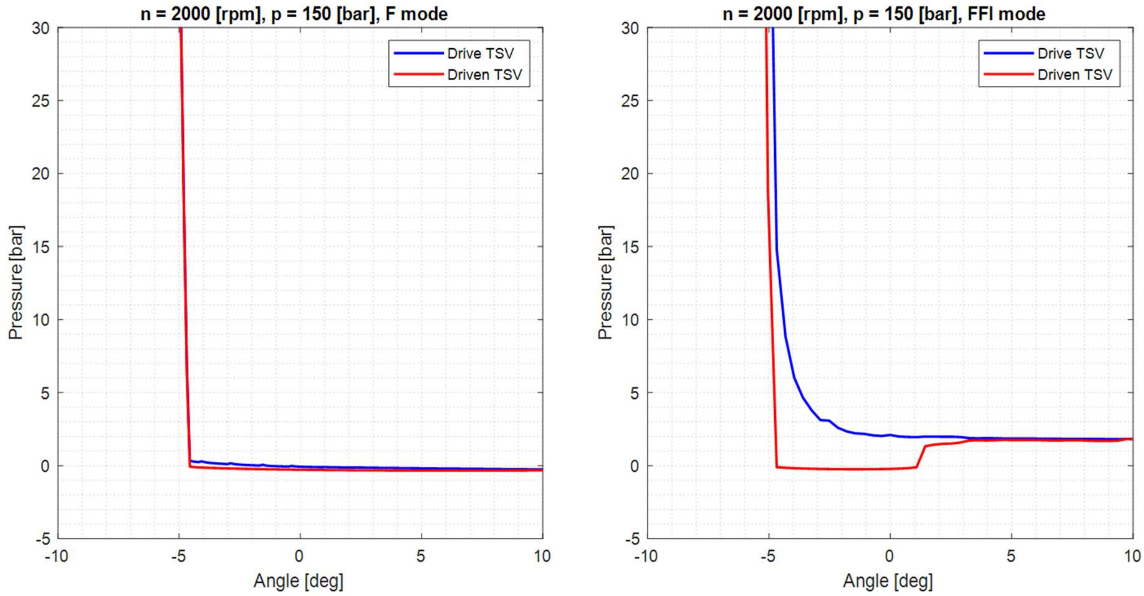



*Figure 106 - Filling phase comparison*

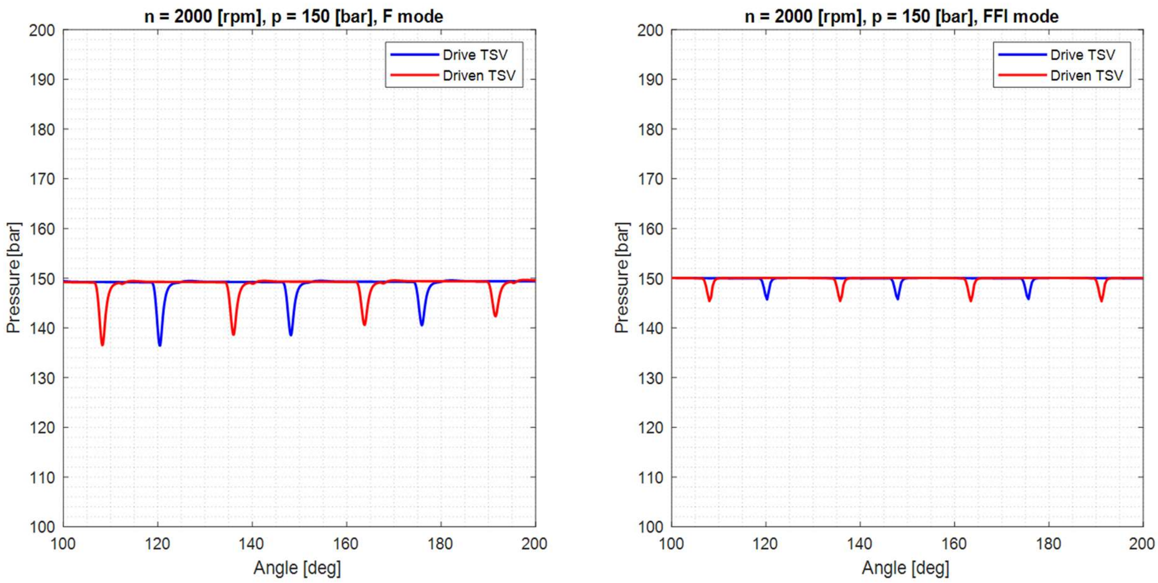



*Figure 107 - Pressurization phase peaks comparison*

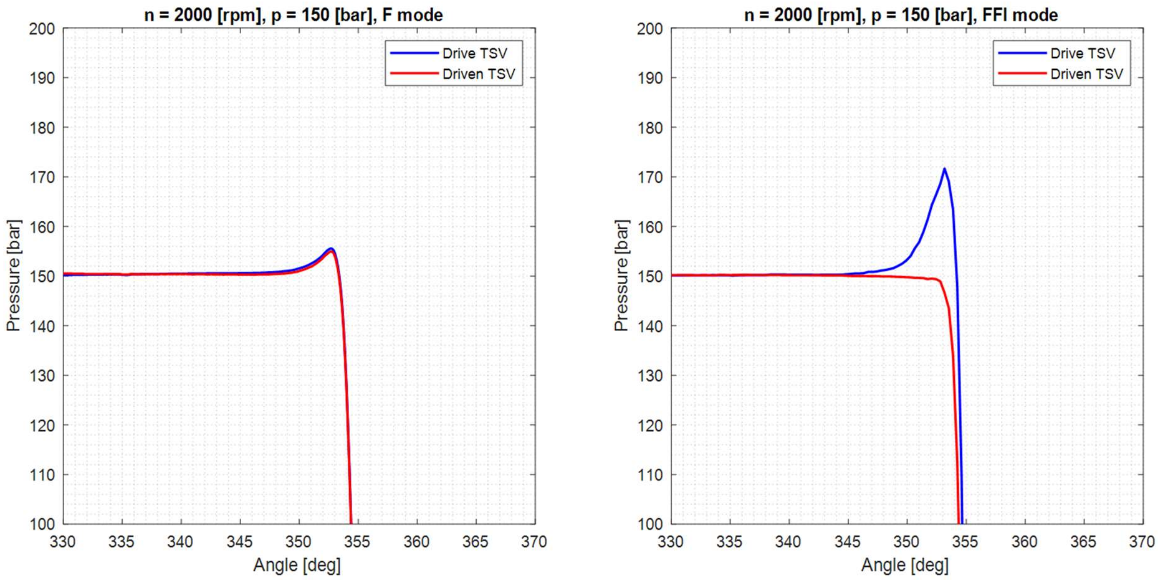



*Figure 108 - Meshing zone peaks comparison*

One of the peculiarities of the Volume Termination circuit is that is noticeable the outlet flowrate, but the pressure ripple is almost absent because the infinite volume capacity absorbs it. Below are shown these two features of the simulation.

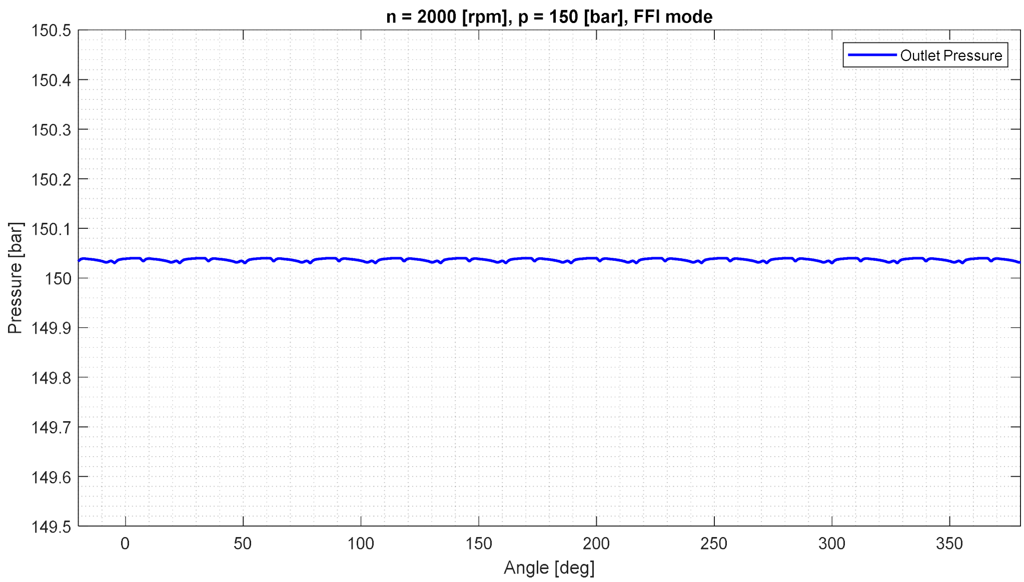



*Figure 109 - Outlet pressure*

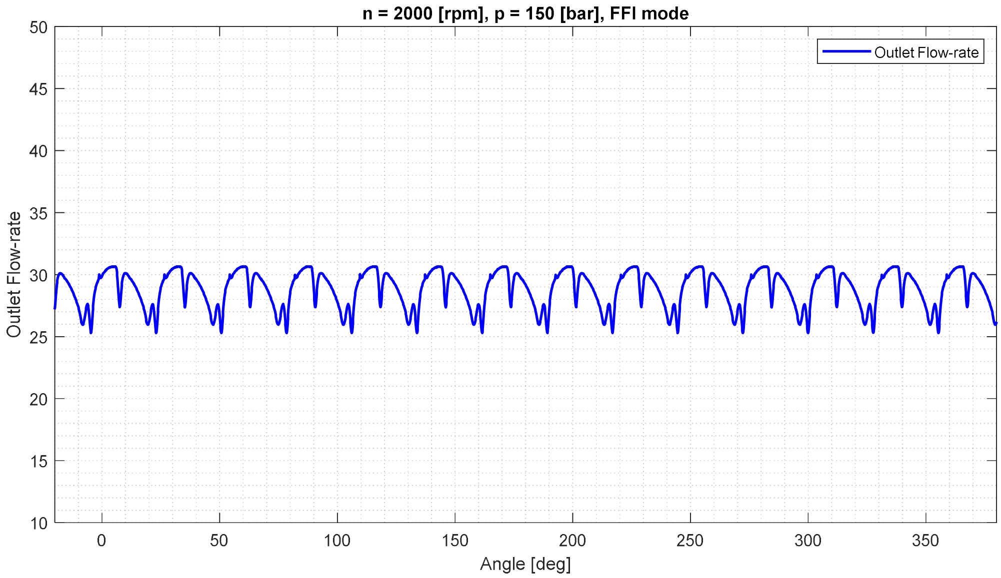



*Figure 110 - Outlet flow-rate*

Looking at the curve is possible to recognize the trend of the kinematic flow-rate, perturbated by peaks.

The peaks are due to the connection of the Tooth Space Volumes to the back-flow grooves. The concept is similar to the one presented before, about the peaks in the pressurization phase. More precisely, the Tooth Space Volume that connects to the back-flow groove represent a decrement of the flow constriction that lead to a redirection of the fluid toward the Tooth Space Volume itself instead of the outlet port. Notice that the outlet flow-rate is a periodical function with period 360/13 degrees. The peaks happen with a doubled period, because the just mentioned connection happen both in the drive gear and in the driven gear.

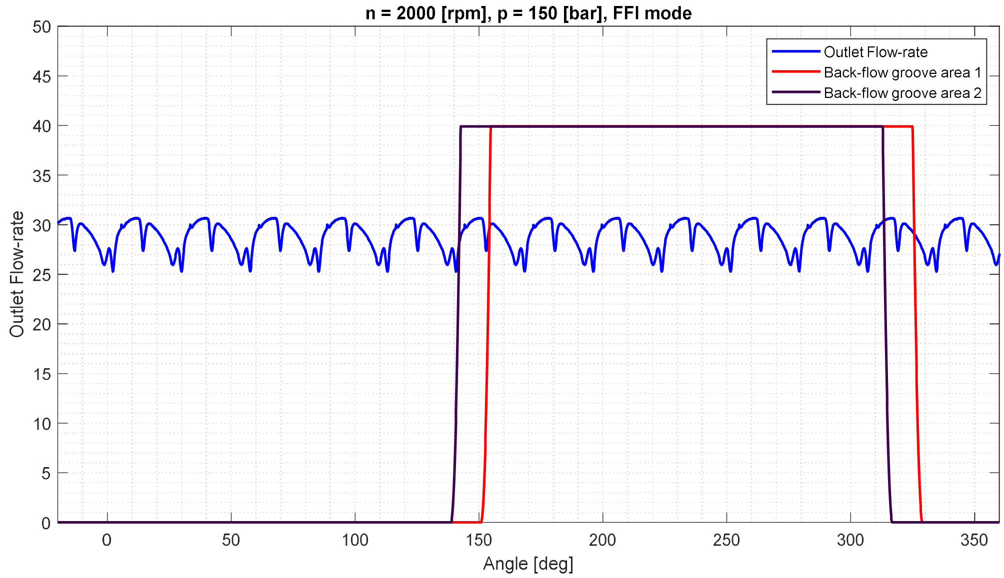



*Figure 111 - Outlet flow-rate profile and back-flow groove areas*

Below are presented the results obtained with the usage of the Restriction Termination circuit.
Is reminded to the reader that the Restriction Termination circuit simulate the presence of a load connected to the pump. To simulate the presence of the load is added an orifice in front of the outlet port. The image beside represents the used circuit.

The orifice diameter is set to 2.2 mm. This diameter value lead to an higher mean pressure than the previous case.

Again, is firstly presented the Tooth Space Volume pressure profile.

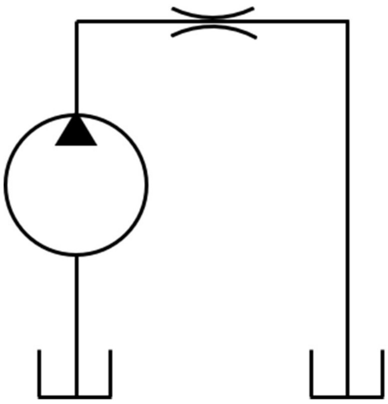


*Figure 112 - Restriction Termination circuit*

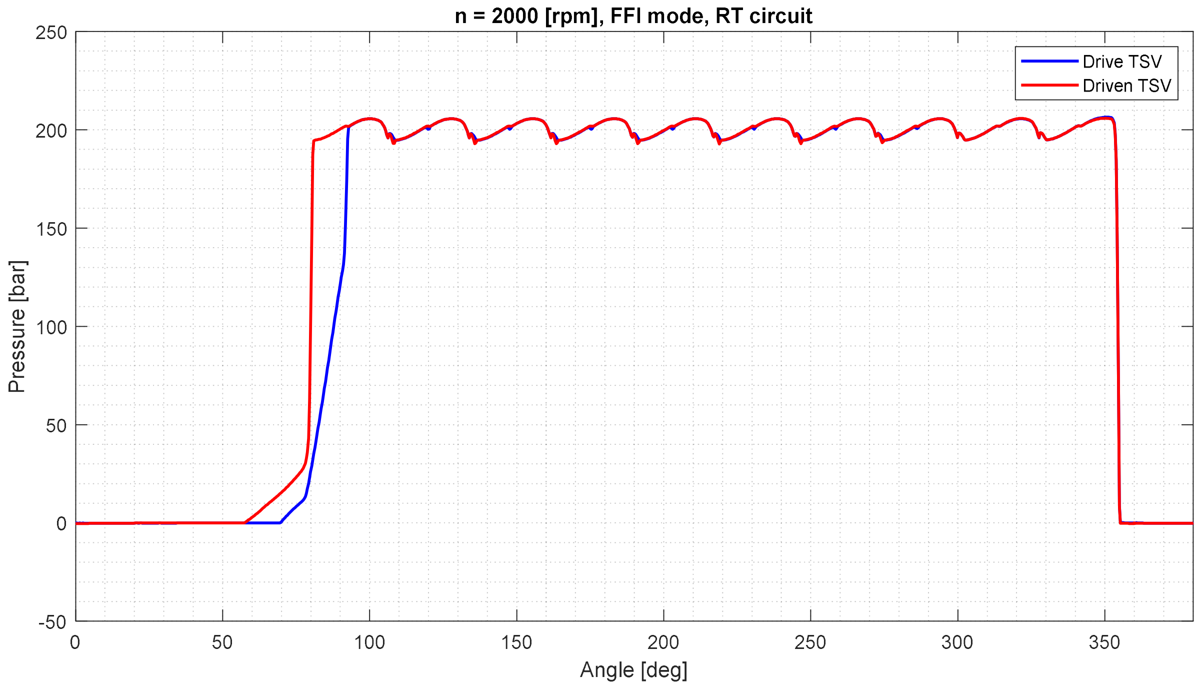



*Figure 113 - Tooth Space Volume pressure profile, RT circuit*

The biggest difference in this case is the profile in the pressurization phase. As in the case before the Tooth Space Volume reaches the same pressure value of the high pressure volume. Also, in this case, because of the Restriction Termination circuit is possible to appreciate the outlet pressure ripple aside the outlet flow ripple. More precisely, the oscillation of the pressure in the Tooth Space Volume pressure profile is the outlet pressure ripple. Additionally, it’s still possible to see the connection of the Tooth Space Volumes to the back-flow grooves.

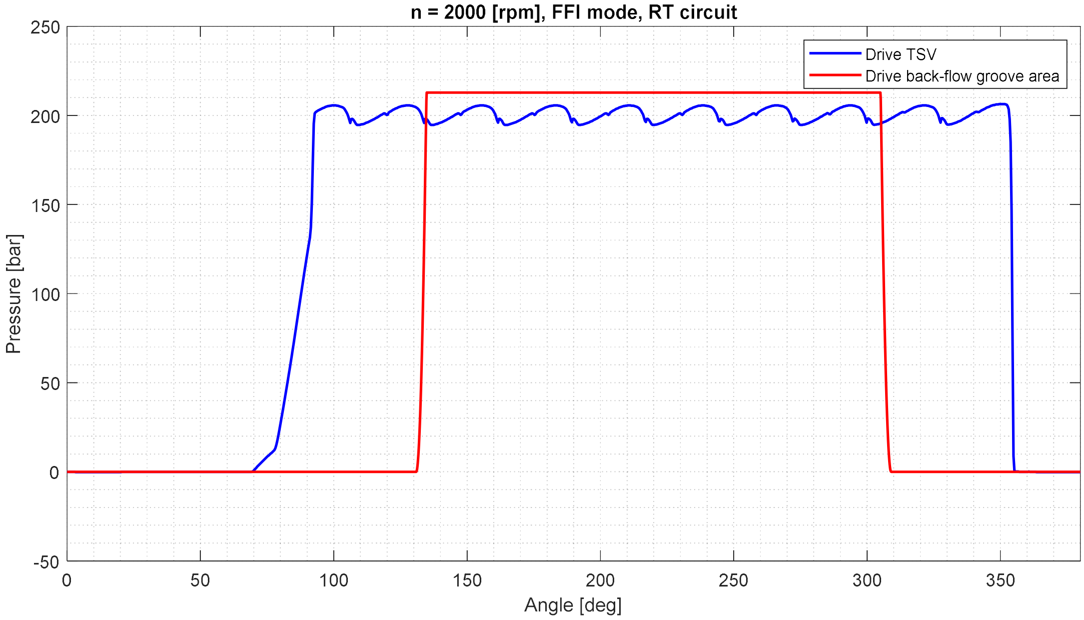


*Figure 114 - Tooth Space Volume profile and back-flow grove area connection*

Below is shown the outlet pressure profile, and the outlet flow-rate.

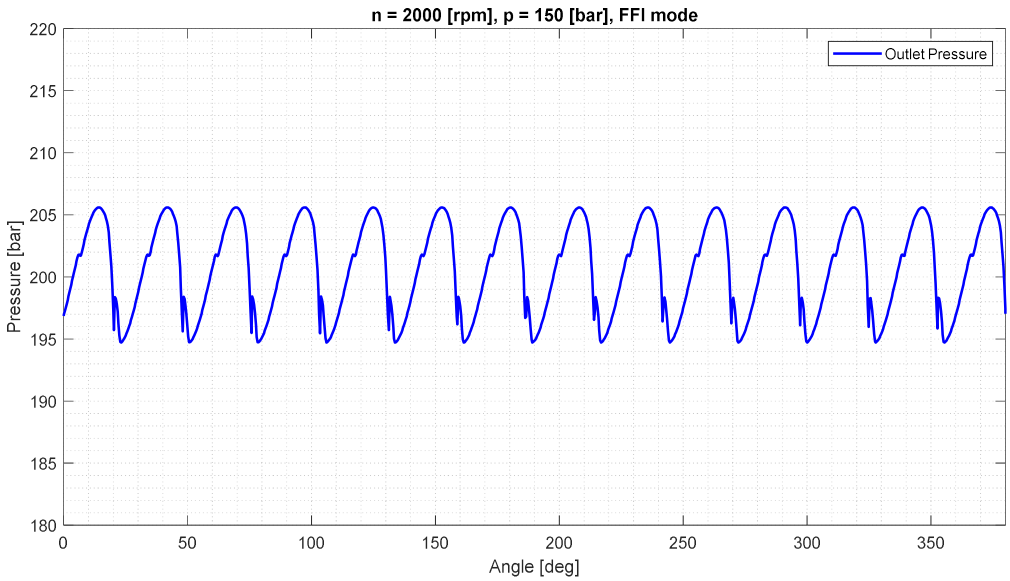



*Figure 115 - Outlet pressure profile*

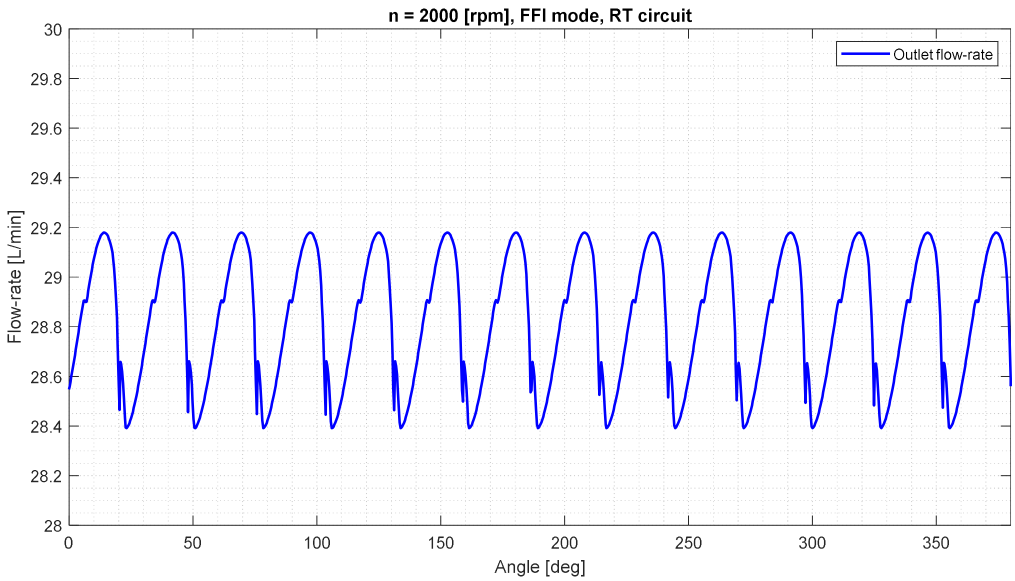



*Figure 116 - Outlet flow-rate*

## Loading Module

The FFI mode allow the user to calculate the forces applied to the bearing elements by the gears.

Is reminded to the reader that the forces are generated by the pressure in the volumes of the external gear machine. To a better understanding of the results presented below, the reader must know that the forces are decomposed in the X direction and the Y direction, and the reference system is shown below.

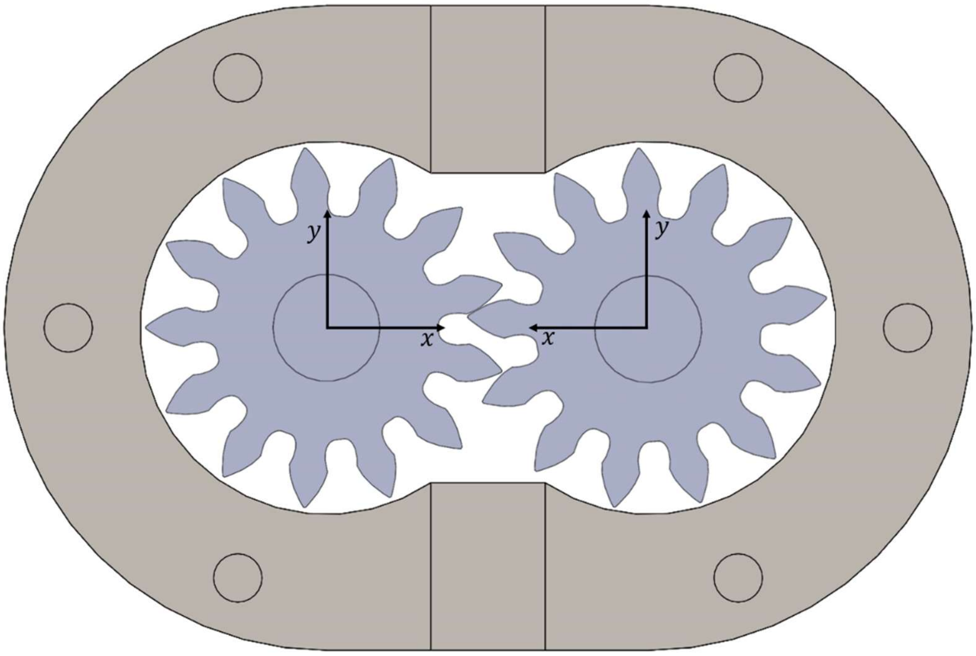



*Figure 117 - Loading Module reference system*

Below are shown the forces applied on both the needle bearing.

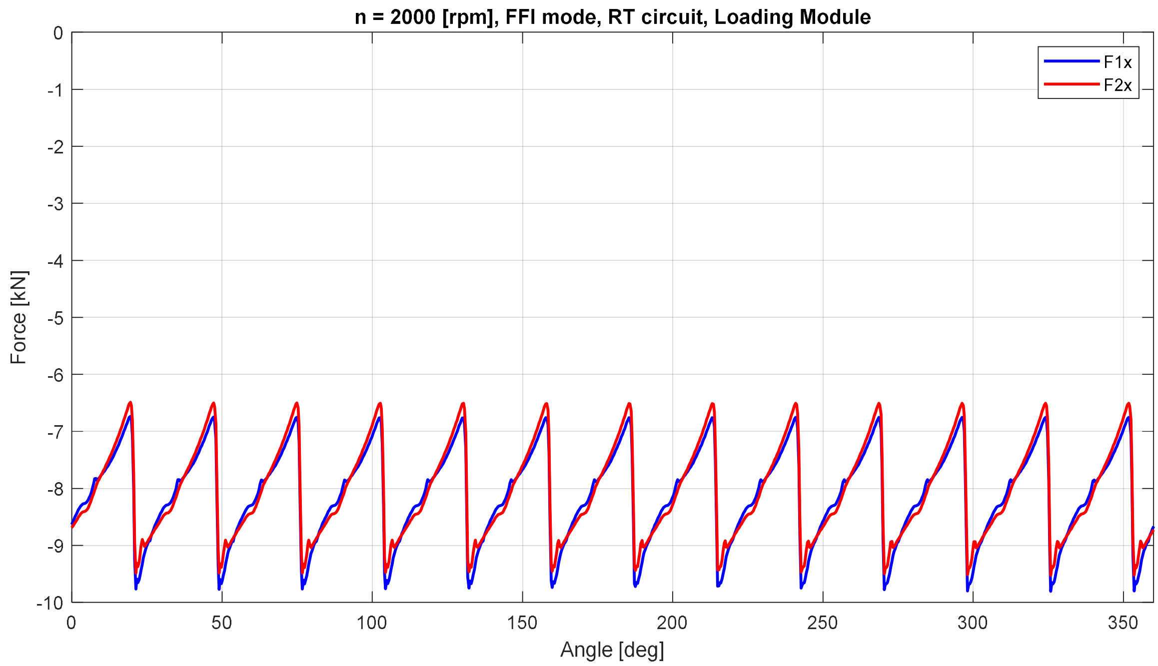



*Figure 118 - Forces in the X direction*

As can be seen from the plot the forces in the X direction push the gears toward the outside. Also, the trend is similar.

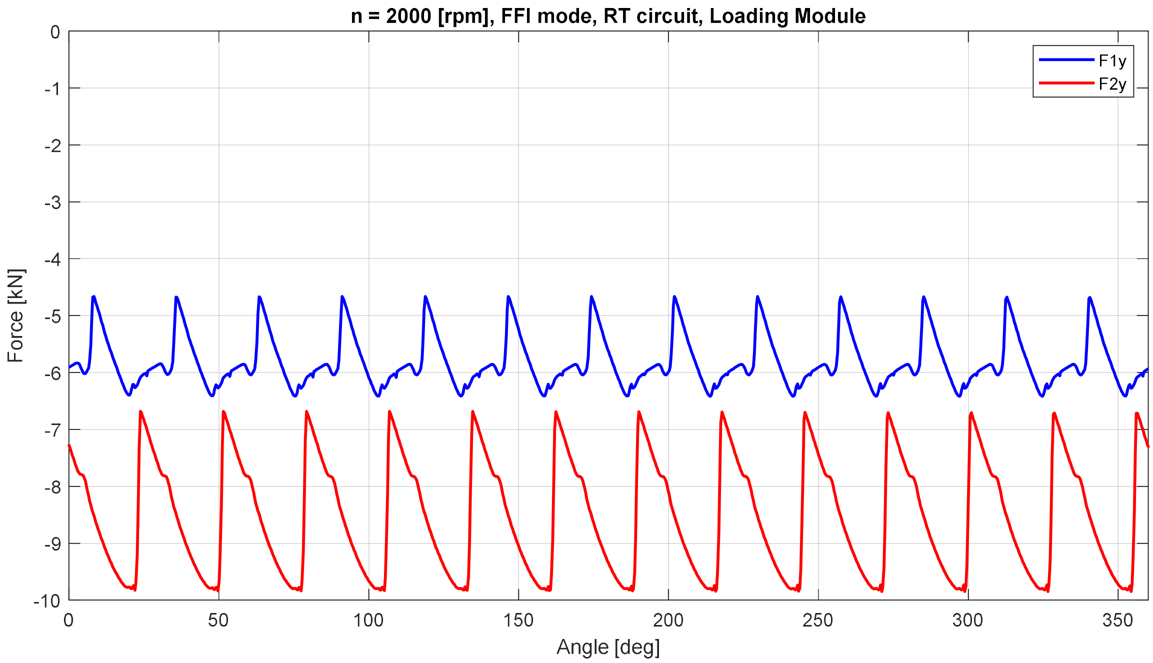



*Figure 119 - Forces in the Y direction*

Is noticeable that the force acting on the drive gear is noticeably smaller than the force acting on the driven gear. This phenomenon happens because on the drive gear, the pressure generated force and the contact force generated by the contact of the two gears act in the opposite direction. Focusing on the driven gear, instead, they act in the same direction. In the following image is shown the pressure generated forces, and the contact generated forces.

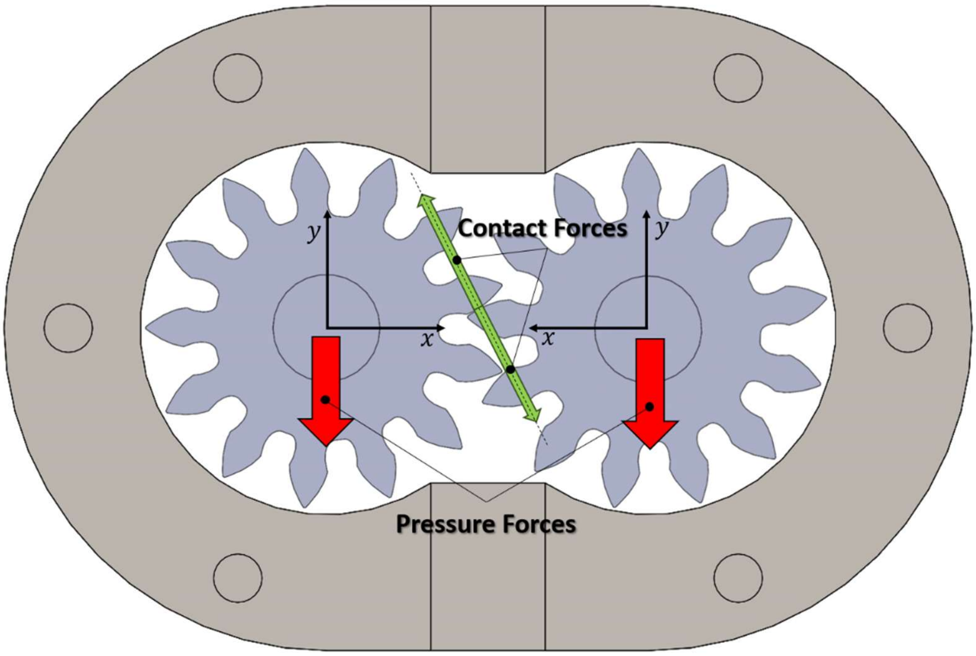



*Figure 120 - Forces schematic acting on the gears*

Obviously, the overall forces in the Y direction push the gears toward the inlet port.

Knowing the forces acting the gears, it's also possible to calculate the angle that define the acting direction of the forces.

| | **Action Force Angle** |
|---|---|
| **Drive Gear** | - 144 [deg] |
| **Driven Gear** | - 133 [deg] |

Below is shown a schematic that ease the figuring of the forces direction.

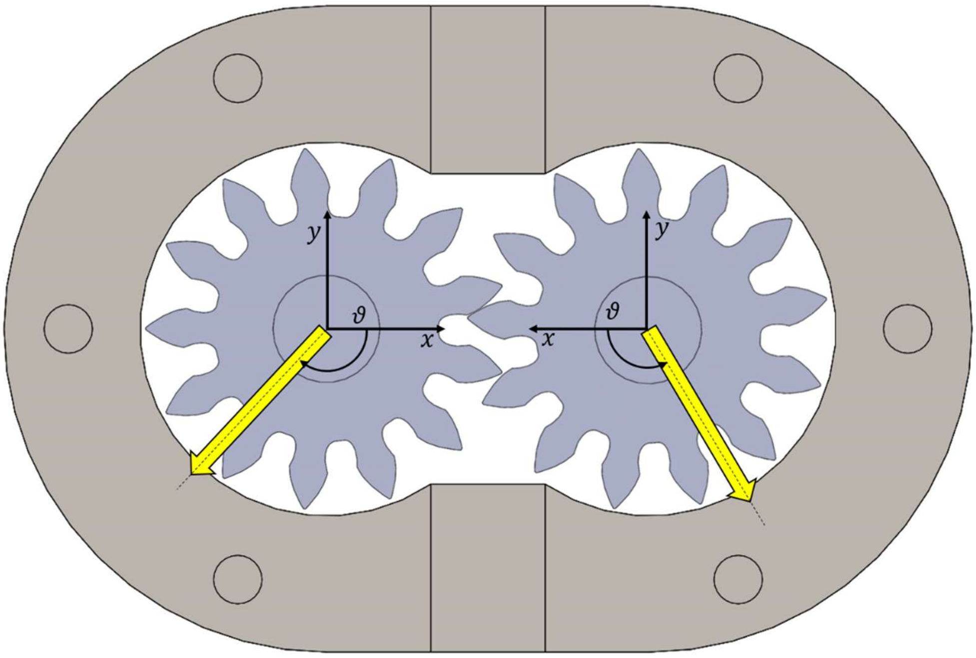



*Figure 121 - Forces direction*

## Micro-motion module

For the micro motion module, the reference system is the same. Since needle bearings were adopted the model presented at the beginning of the thesis is not useful anymore. So, it was developed a simple model that consider the micromotion of the gear in the same direction of the forces, and with a magnitude equal to the nominal radial clearance of the bearings. Actually, the team is still facing the details of the designing phase, so this module is representative and not definitive.

Below are shown the motion in the X and Y direction of the drive and driven gears.

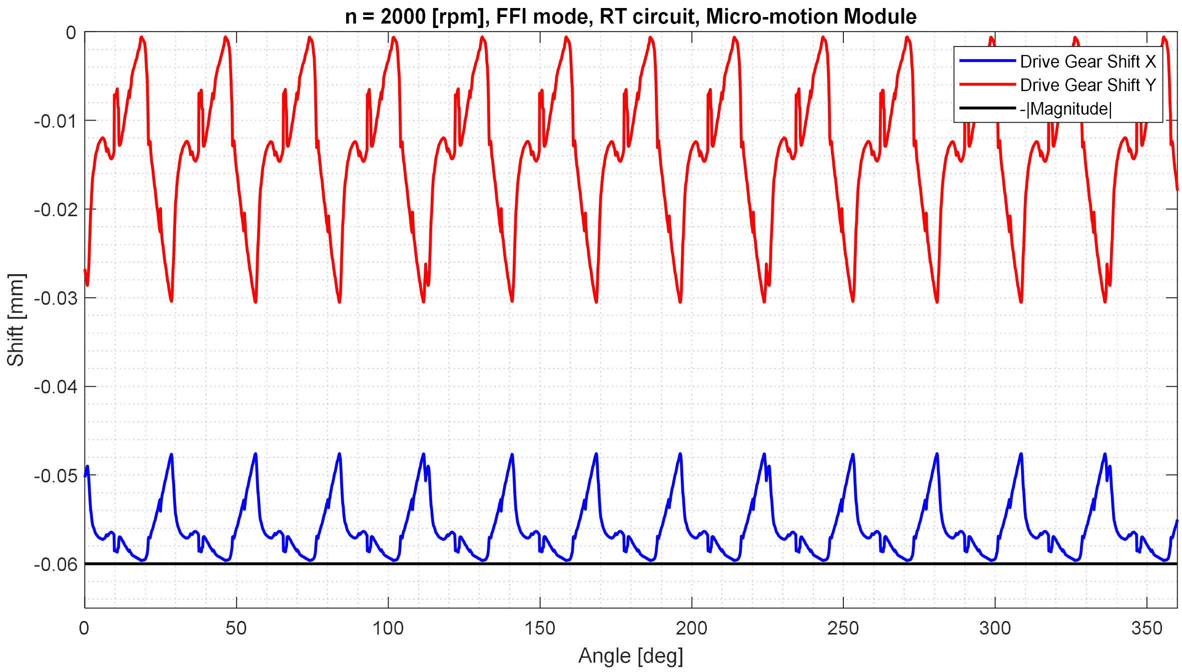



*Figure 122 - Drive gear micro-motion*

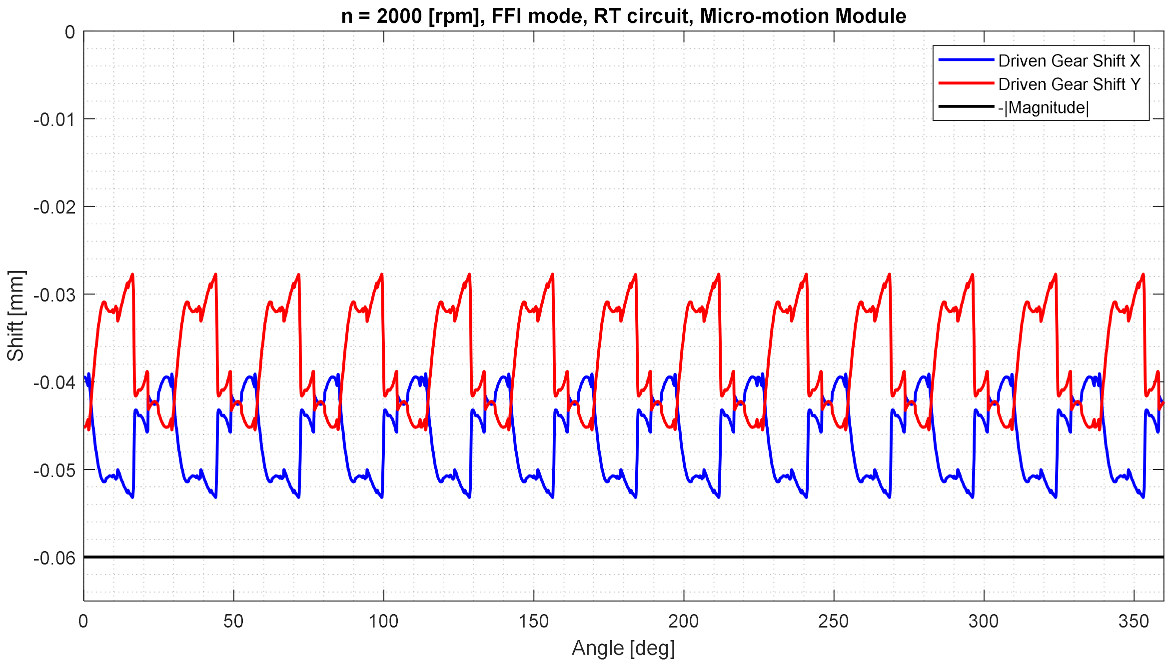



*Figure 123 - Driven gear micro-motion*

As mentioned above, the direction of the motion is the same of the loading module. For the sake of completeness is represented below the mean direction of the gears shift.

| | Action Force Angle |
|---|---|
| **Drive Gear** | • 144 [deg] |
| **Driven Gear** | • 133 [deg] |

And below a graphical representation of the gear shifting.

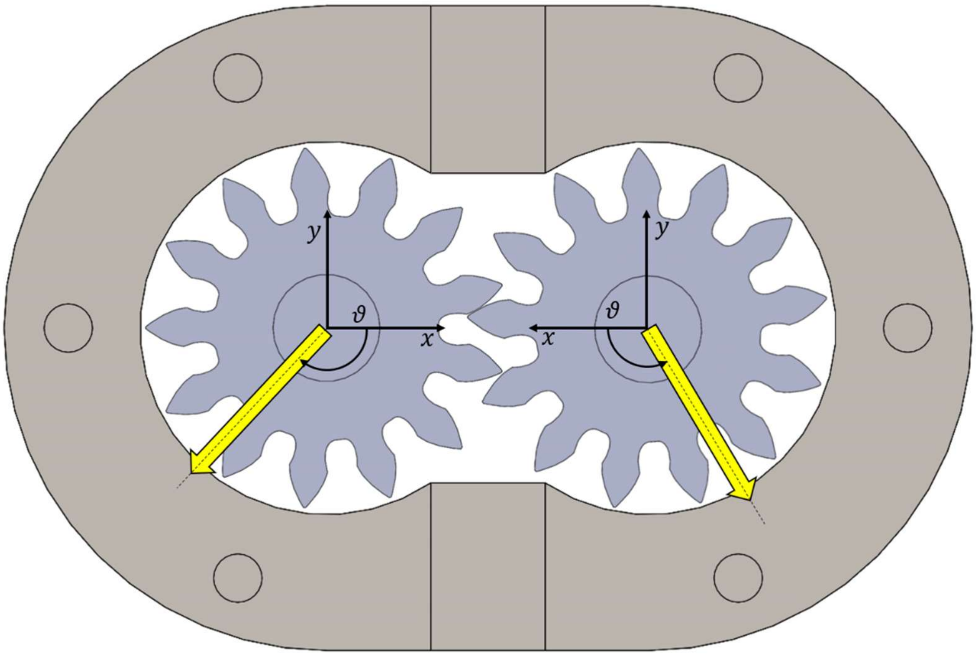



*Figure 124 - Forces direction*

## Design

The last part of the thesis is focused on the design of the device.
One of the objective functions of the project is minimizing the size of the device. To achieve this goal the hydraulic machine was integrated in the electric machine. The novelty of this design was worthy of a patent which progress is ongoing.

To achieve a good integration, the electric machine designed by an E.C.E. (Electronic and Computer Engineering) team, is thin and presents a big radius. The active part of the electric machine is concentrated toward the outer radius. The size ratio of the two machines proof the higher power density of the hydraulic compared to the electric machines.

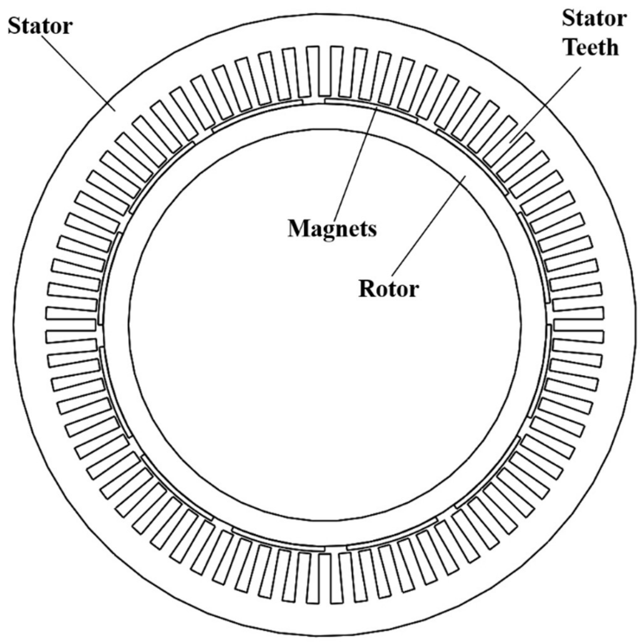



*Figure 125 - Electric machine front view*

In the image beside is possible to see the stator and the rotor that constitute the electric machine. Notice that in the stator are shown the teeth used to install the windings. In the rotor instead are shown the magnets. Both the stator and the rotor are a compound of layers placed together one on top of the other to avoid losses.

The stator is press fitted in the device casing, and in the rotor is press fitted a drive flange used to transmit the motion to the hydraulic machine and vice versa. The drive flange is coupled with the drive gear through a spline.

In the image below on the left are also shown the fans used to cool the electric machine. In the image on the right instead is shown how the stator is press fitted in the external casing of the device.

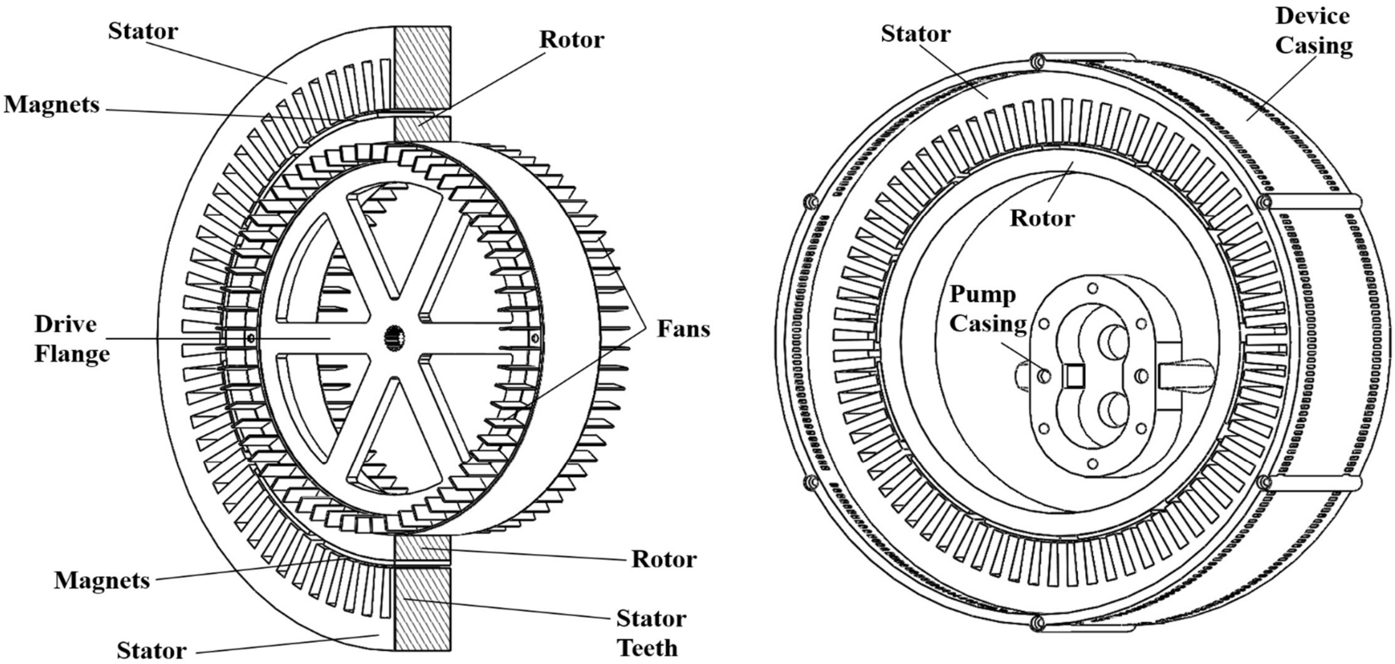



*Figure 126 - Electric machine and electric machine casing*

As can be seen in the image above, the hydraulic machine casing is contained inside the electric machine. In the image below is shown the hydraulic machine, focusing on its components. More precisely, what is shown in the image below is what is missing in the assembly of the machine shown in the right image above. Notice that to carry the load of the pump, but also the gyroscopic moment generated by the translation of the rotating electric machine, were selected needle bearing instead of journal-bearing. Additionally, notice that between the nut and the shaft sealing there should be the drive flange, but for the sake of clarity it was omitted in the image.

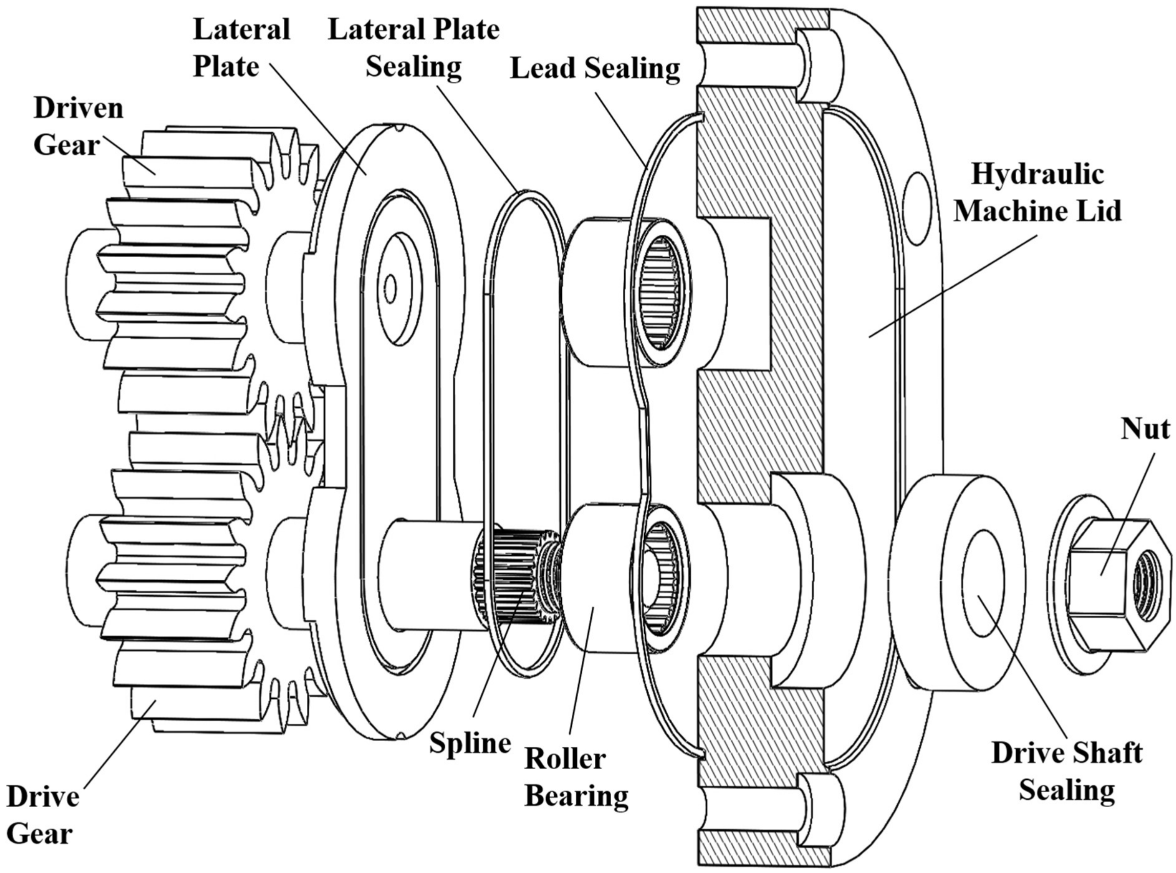



*Figure 127 - Hydraulic machine exploded view*

In the following image is shown a cross sectional view of the entire device.

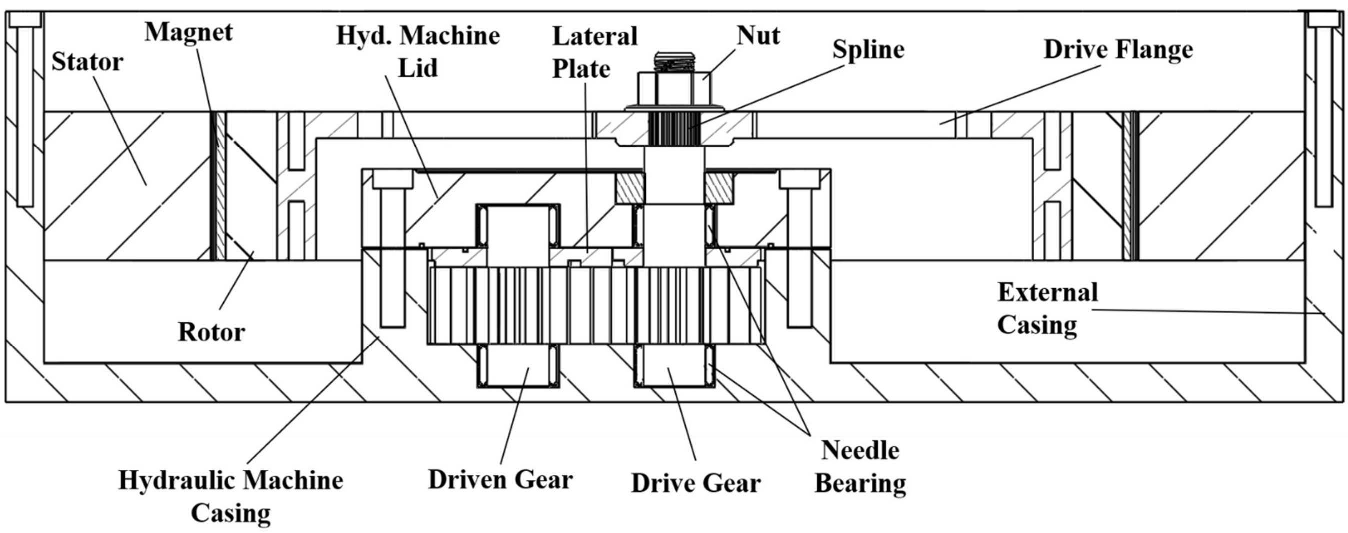



*Figure 128 - Cross sectional view of the device*

Here is shown an image to give an idea of how the components are disposed in the casing.

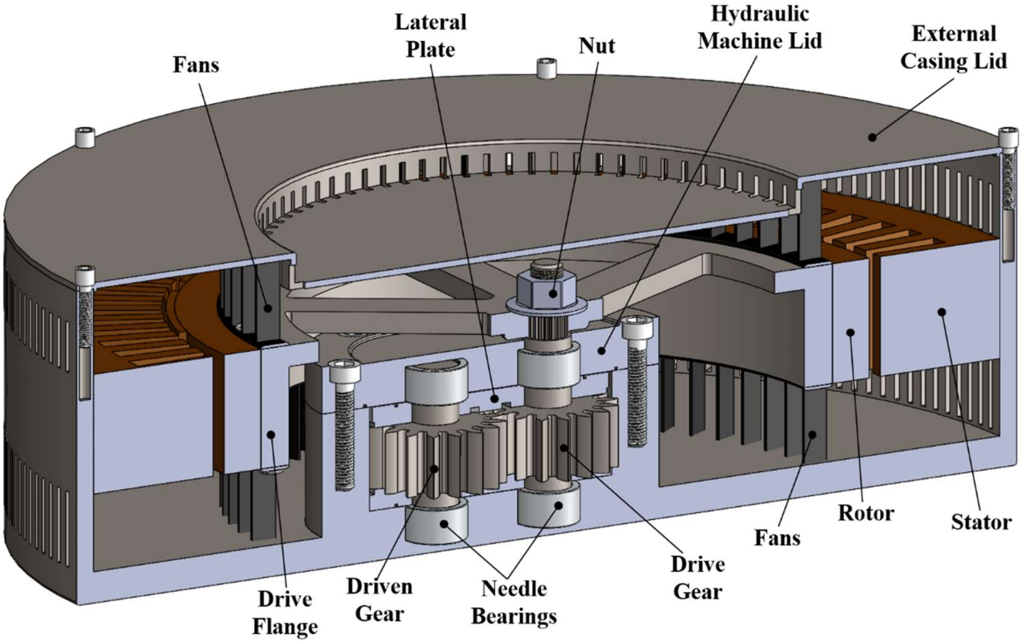



*Figure 129 - Section of the concept design*

In the end, the device closed as should appear mounted on the off road machine. The diameter of the device is 330mm, the depth 100 mm. To connect the hoses to the hydraulic machine was chosen the boss o-ring connections.

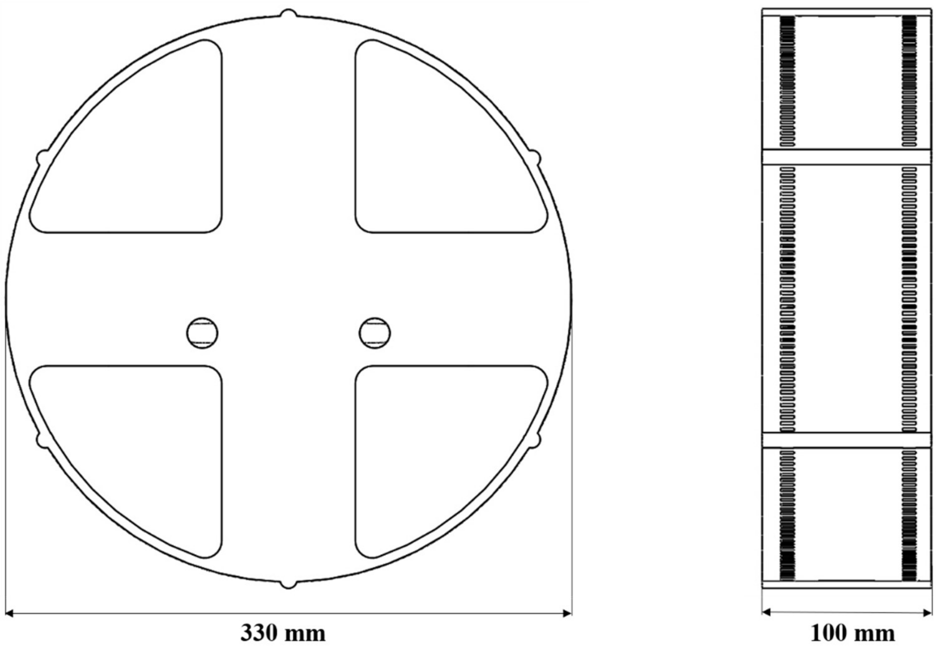



*Figure 130 - Closed device dimensions*

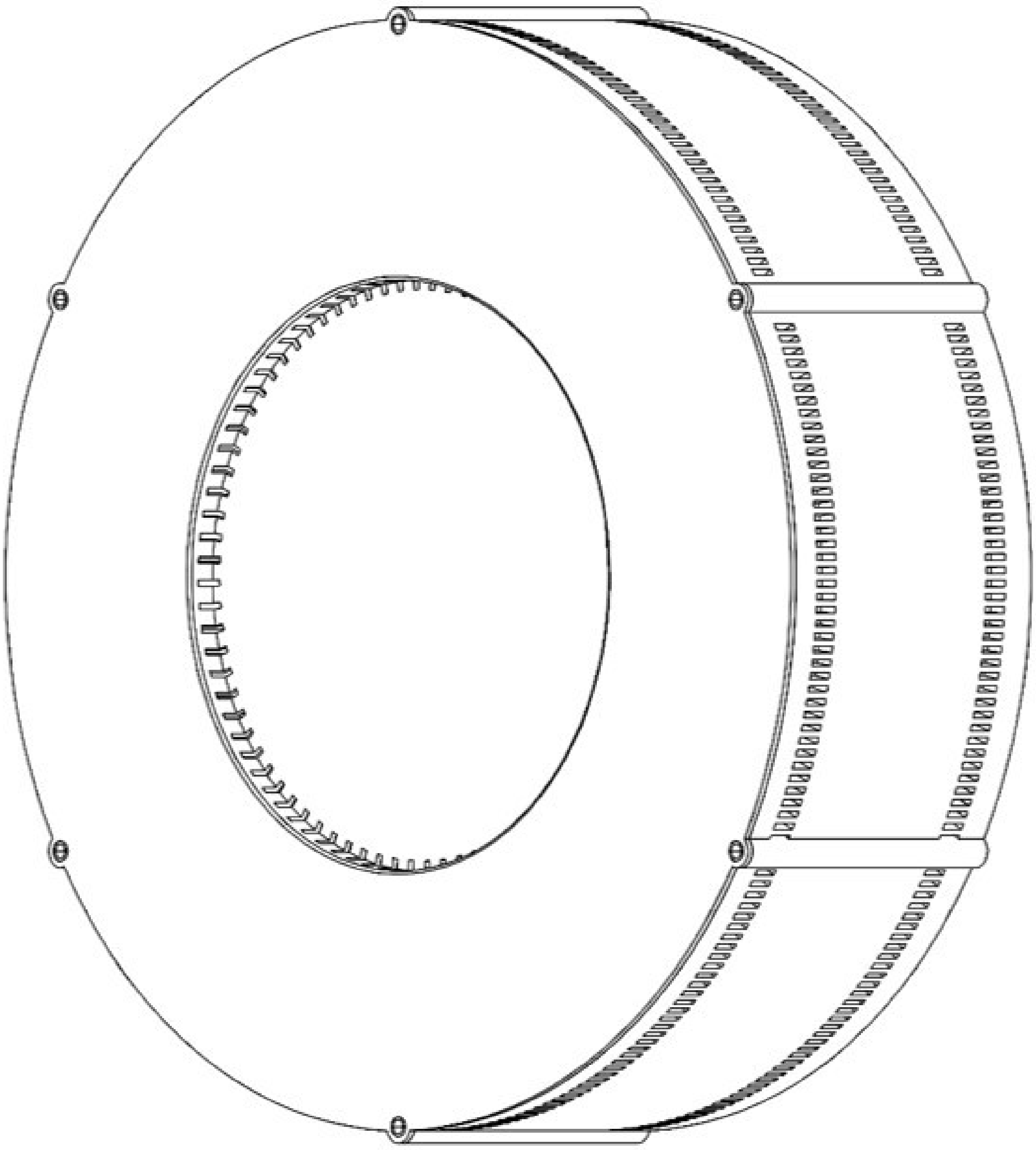


*Figure 131 - Assembled design*

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To Nicola, that I've always named Guidi. I've always been skeptical, and I've never believed in forever lasting friendship. However, I've to admit that every year that pass it seems more feasible. After more or less 12 years, even though the thousands of miles between us, still at my side.

To Viola, that, after I destroyed everything (she would know what I mean), took my hand and brought me where the sun shine. With infinite patience, she took care of me, and helped me to start again. She taught me how to be a better person, a normal person or even just a person. She taught me most of what I know. I couldn't ask for anyone better than you. I swear.

To Carla, that was with me most of the time spent in Turin, helped me in my personal growth, and taught me what does it mean being by someone side, taking care each other. I will always thank her for making me understand what the sense of beauty is.

To Michael and Sergio, that together with Carla, were always with me, and helped me to overcome all the difficulties and hard moments that life offered me.

To Annalisa and Sangbeom, that were my reference system in my adventure in the United States. They have been friends, guides, and examples. Thank to them my concept of friendship become deeper and closer to the concept of family. Sometimes the choices that we make in our life tear us apart, I want to hope that this will never happen to us.

As last, but not least, I would like to thank Tom and Rituraj, colleagues and friends, that followed me and helped me in achieving every goals and taught me an almost infinite amount of things.