DEVELOPMENT OF A COMPACT LOX AND LNG ELECTRIC PUMPS FOR MICRO-LAUNCHERS

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ABSTRACT:

Over the past 10 years, the demand for launching micro and nano satellites has increased considerably. To answer this need, several micro and nano-launchers are now being developed worldwide. Considering the technological improvements of small electric motors and batteries, small launchers using electric pump-fed cycles are now feasible. However, there are currently no market solutions of cryogenic pumps meeting the requirements of a small launcher: compactness, high-performance and high flow.

Therefore, due to the criticality of the pumps, Omnidea has initiated the development for both Liquid Oxygen (LOx) and Liquid Methane (LCH4), with emphasis, in a first approach, on researching the use of innovative additive manufacturing (AM) techniques and building capabilities for technology validation.

The fuel changed from Liquid Natural Gas (LNG) to LCH4 along the development of this work to cope with the rocket engine new requirements.

1. DESIGN ENVELOPE

The pumps were designed based on the requirements of range of the outlet pressure and flow rate envelope. Table 1 summarizes the outlet envelopes of the pumps.

Table 1 - Pumps outlet envelope

Pump	Param.	Low	Nominal	High
LOx	Pres.	77.3 bar	83.7 bar	86.1 bar
	Flow rate	5.34 kg/s	6.13 kg/s	6.45 kg/s
LCH4	Pres.	76.0 bar	90.4 bar	99.4 bar
	Flow rate	1.78 kg/s	2.05 kg/s	2.15 kg/s

Setting the lowest pressure and highest temperature combined as the worst operating condition within the inlet envelope, the starting point for the design is obtained, Table 2.

Table 2 – Inlet envelope

	LOx		LCH4	
Range	Min.	Max	Min.	Max
Temp.	90 K	107 K	112 K	132 K
Pres.	1.7 bar	5.3 bar	1.5 bar	4.5 bar

The mass of the pumps was not a requirement considered at this stage of development, as the initial prototypes are intended for bench testing only. The efficiencies of the pumps are an important factor to design electric-driven rocket pumps [1], as they define the battery requirements. The efficiency of a pump is directly related to its specific speed, while keeping all other parameters constant. However, a high speed and high-power electric motor is very difficult to source, as most high-speed motors are engineered for much lower power applications. As a result of this, the design was driven by the availability of high speed and power electric motors. Lower revolutions per minute (RPM) motors also simplify the procurement of other commercial off-the-shelf (COTS) parts like seals, bearings, shaft couplers, etc…,

After some deliberation with potential suppliers, it was decided that a maximum speed of 40000 RPM was an achievable compromise between motor availability and pump efficiency.

The pump test bench was designed to be able to operate both pumps in the worst condition of their respective envelope, i.e., high outlet pressure of the fuel pump and high mass flow of oxidizer pump, and its parts/materials compatible with both LOx and LCH4. Like the pumps, the test bench will also use COTS whenever available, otherwise it will source specialized designs from local manufacturers.

2. PUMP DESIGN

Pump design was treated as an iterative process, as beyond the initial geometry (obtained through analytical calculations) considerations of manufacturability, forces involved, shaft layout, seals, bearing cooling and lubrication, rotodynamic behaviour and other aspects also had to be taken into account. Higher fidelity modelling was also employed for certain aspects of the pump design (e.g., FEA modelling of the impeller under load). Figure 1 shows a simple flow chart summarizing the process (minus the AM optimization loop).

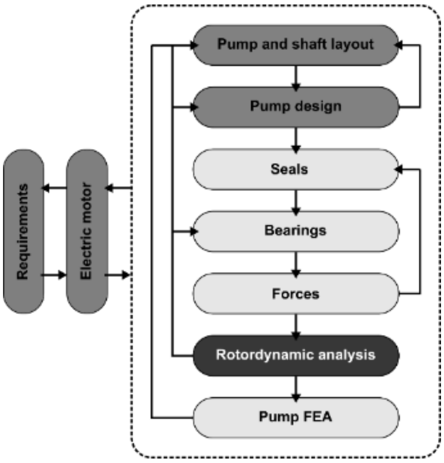

Figure 1 - Pump design flow chart, adapted from [2].

For the pump design, a few calculation tools were developed in-house and implemented using Scilab [3]. The main tool is the PUMPCALC code, which uses the CoolProp Library [4] to gather the thermophysical properties of the propellants and accepts several geometric inputs considering design factors and first guesses of efficiencies to execute several iterative calculation loops to obtain the overall geometry of the pump. The results coming from this tool are then formatted into parametric inputs for CAD modelling which creates (with minor adjustments) the inducer, impeller, and volute 3D models of the pump.

The second tool used for the design process was the ROTORDYN code that uses a finite element method based on [5].

Other simpler tools were developed to assist the mechanical design of the pump:

- cryoMAT: Calculates several material properties, such as the thermal/contraction expansion ratio using data from [6].
- FLANGE: Calculates the flange design based on a method for the design of flanged joints from [7].
- SHAFT_HUB: Calculates the torque transfer of a cone type shaft hub connection based on [8].

Only the PUMPCALC and ROTORDYN tools and the mechanical systems design will be further discussed in this paper.

2.1. Pump layout

The basic concept of each pump is simple. It consists of a centrifugal pump with an inducer to achieve better anti-cavitation characteristics. The pump is axially fed and has a single-turn volute for the outlet. The main seals are of the floating ring type. The pump has at least two bearings and an electric motor integrated into the same shaft. The propellant will cool down the electric motor and bearings and then, if needed, it will be sent back to the pump's inlet by a hydrodynamic seal. Lubrication will be provided by the bearing cage material. This layout is presented at Figure 2-A and represents the targeted layout for a flight-rated design.

This layout provides the simplest and lightest electric pump system for a rocket engine, without couplings, the smallest possible number of seals, no external lubrication or cooling system for the electric motor, and even the Electronic Control Unit (ECU) can be cooled from this arrangement. However, at the time being, a suitable COTS electric motor compatible with cryogenic temperatures was not found. In addition, one needs the bearing cooling parameters to design the minimal possible leak for this purpose, in order to not compromise the pump efficiency.

For the test bench, a layout with an external electric motor (with its own bearing and cooling system), and the pump shaft supported by COTS bearings with oil lubrication is proposed, Figure 2-B.

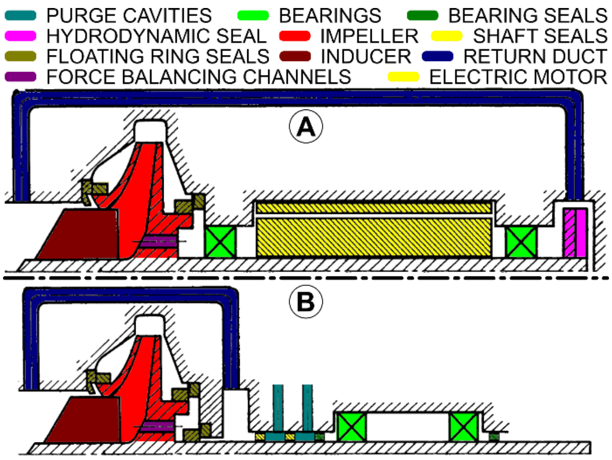

Figure 2 - Pump layouts.

This layout provides great flexibility for the electric motor and the dedicated bearing and lubrication system can provide the needed data for future flight-ready hardware. Although this layout requires a more complex sealing solution to isolate the propellant from the lubrication system, it is feasible with COTS. The flexibility of the power source enables the use of electric motors with less speed and power so one can map the pump characteristics through scaling factors and using safer fluids, like liquid nitrogen (LN2). A later test using the target propellants and adding a faster and more powerful electric motor, or even a turbine, can be implemented to achieve the design operating conditions of the pump.

2.2. PUMPCALC

The PUMPCALC tool starts by setting the nominal outlet condition, defined by the input properties of the propellant via CoolProp. It then calculates the basic parameters of the pump: head, net positive suction head (NPSH), volumetric flow rate, etc., and computes the basic inlet geometry: axial inducer inlet speed triangles, and blade configuration. Then, it calculates the outlet speed triangles and blade geometry for the centrifugal impeller and iterates the flow rate factor to define the impeller external diameter. Most of the calculation is based on [9], [10] and [11], which refer to high speed centrifugal pumps with axial inducers.

Later, the tool calculates the cavitation characteristics of the inducer's outlet and centrifugal impeller's inlet and matches the cavitation behaviour to define the complete geometry of the blades.

At this point it is important to emphasize that rocket pumps (where high speed, high efficiency, small size, and low weight are key objectives for optimization) follow an important design distinction from industrial pumps: cavitation is only avoided at levels that may affect performance. As they are not intended for life cycles of hundreds of thousands of hours, they are designed to work under controlled levels of cavitation. Even a reusable rocket motor will have the pump designed with a few hours of total life [9].

For rocket pumps, the inducer and centrifugal impeller are designed as a single unit and the matching algorithm will combine the inducer outlet and impeller inlet blades geometry to make sure that both will have the same cavitation behaviour. If the inlet conditions are ideal, the design shows that an axial inducer may not be needed, as in [12], and it will consider an inducer-less alternative. An example result of this inducer and impeller cavitation matching algorithm is shown in Figure 3.

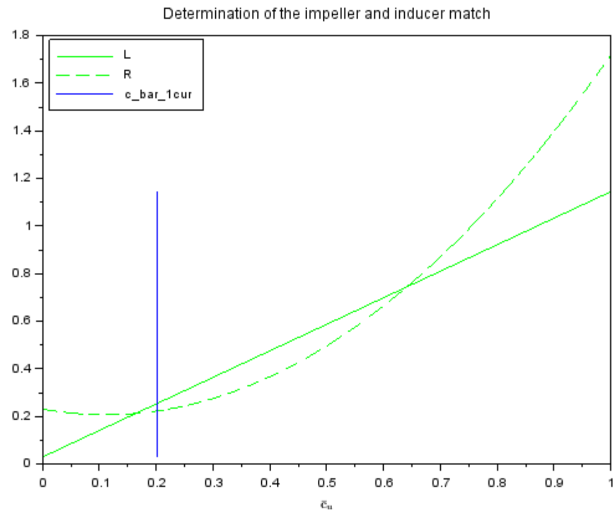

Figure 3 – Left (continuous) axial inducer and right (dashed) centrifugal impeller cavitation characteristics for blades geometry matching.

The "L" curve (continuous) of Figure 3 is the cavitation characteristic of the inducer, and the "R" curve (dashed) is the cavitation characteristic of the impeller, both as a function of the flow swirl coefficient, $\bar{c}_u$, [10] and [11].

$$\bar{c}_u = \frac{c}{u} \qquad \text{Eq. 1}$$

Where c is the absolute velocity and u is the peripheral velocity, [13]. A geometry is considered valid within a certain range of the swirl coefficient and when $L \geq R$, as shown by the selected coefficient shown by the vertical line at the figure.

The tool then iterates the blade constraining factor from the first guess, by repeating the loop until an acceptable iteration error (per [10], [11] and [9]), is achieved. It finishes the calculation of the geometry of the blades, volute, and outlet diffuser. Each of these calculations has their own iteration loops which seek for minimal pressure losses within recommended design factor constraints.

The volumetric efficiency algorithm, that calculates the leakage recirculation of the main free floating seal rings, is a combination of an iterative design from [9], [10] and [13]. It can also be manually iterated and even combined with inner channels connecting the inlet of the centrifugal impeller with the back face of the pump in order to manage the axial forces acting over the pump.

The disk friction losses are estimated using the methodology from [13]. The volute losses are based on the algorithm from [14].

The tool then iterates the first guesses of the volumetric and hydraulic efficiency until convergence. Afterwards, it checks if all of the calculated design parameters are within the recommended values from the references and print out the geometric parameters for the CAD design.

A plot with the cavitation characteristics, the matching algorithm curves, the head characteristics, from [10] and [11], and the efficiency as a function of the specific speed using data from [15] and [9] as an example, is displayed in Figure 4.

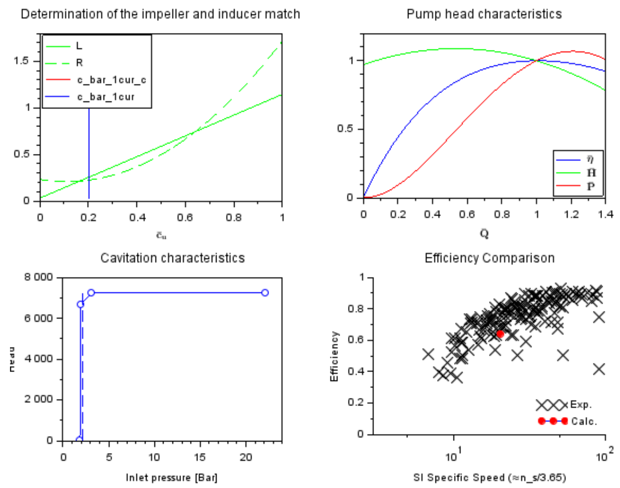

Figure 4 - Plot with some of the outputs from the PUMPCALC code

The speed and power required for the lower and higher outlet conditions of the envelope are calculated by combining the nominal head characteristic of the pump with the rocket motor pressure versus the flow rate requirements, [10] and [11].

The design tool was verified using Simcenter STAR-CCM+ CFD software, Figure 5, by comparing results from key areas as the average speeds, pressure distribution, leakage rate, etc.

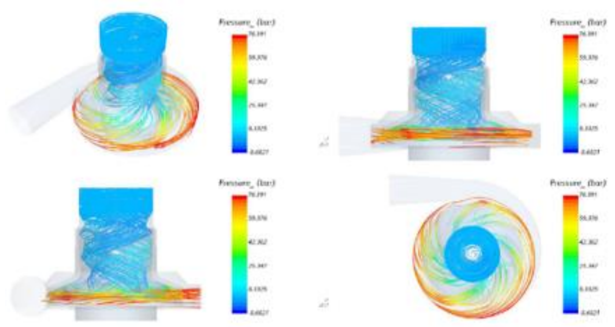

Figure 5 - Example of one of the CFD validation studies.

Several manual iterations of the input data are required to achieve the final result that is within the recommended factors, realistic geometric and mechanical dimensions for the CAD design and additive manufacturing, Figure 6.

Figure 6 - CAD render of the pumps. Left: LOx pump; Right: LCH4 pump.

2.3. Additive Manufacturing optimization

Throughout the design process, many strategies to deal with overhanging surfaces during the AM production of the pump were discussed. A test part with two blades and one flow path of the impeller was designed and manufactured to verify this issue, Figure 7.

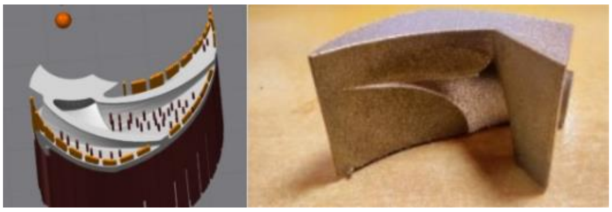

Figure 7 - One of the support strategies printed.

The input factors and parameters of the PUMPCALC tool have also been tweaked to achieve a design of the pump with minimal overhanging surfaces, Figure 8.

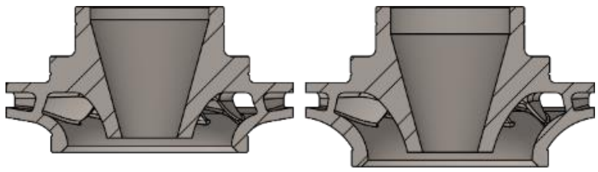

Figure 8 – A cut view of a LOx pump as originally designed (left) and optimized for AM (right).

2.4. Seals design

It was determined during design that the pumps would require four sealing systems:

- Main seals, that affect the volumetric efficiency of the pump and balance the hydraulic forces of the system.
- Shaft seals, which prevent leakage from the main seals to the environment or towards the bearing lubrication cavities.
- Lubrication system seals, that prevent oil from leaking into the environment or into cavities contaminated by the fluid being pumped.
- Cryogenic static seals, used at the main component interfaces, pump's inlet and outlet flanges.

The main seals are of the floating ring type. Two seals are required for the front and back faces of the impeller. The final design will use three seals, thus adding a higher pressure cavity between the back main seal and the shaft, and connected by channels to the area between the inducer and impeller to minimize the axial forces.

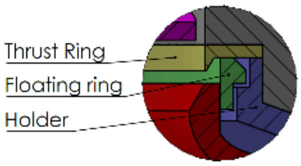

Figure 9 – Floating ring main seal schematic.

The shaft seals will have two cavities where there may be residual fluid and oil leaks to be purged by nitrogen gas (GN2). The first cavity isolates the pump's fluid and the second, the lubrication oil.

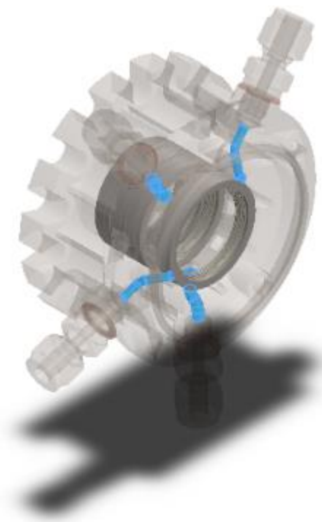

Figure 10 – Shaft seals with the inlet and outlets of the two cavities purged by GN2.

The lubrication system will use a pair of frictionless labyrinth seals, both to isolate the shaft's seals cavity and the ambient. The last group of seals are the static seals of the pump assembly, fittings, and interface flanges, Figure 11. The shaft, oil and static seals are all COTS.

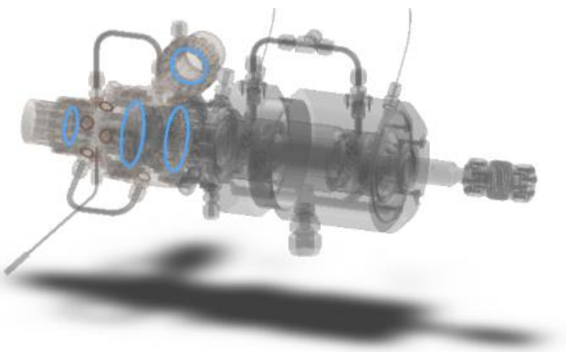

Figure 11 - Cryogenic static seals of the pump assembly and flanges.

2.5. Pump FEA

The LOx pump impeller and inducer were simulated up to 150% of the maximum speed, (neglecting the pumped fluid pressure distribution) to help the selection of the pump's material and to achieve a blade mechanical design compatible with the available AM materials, Figure 12.

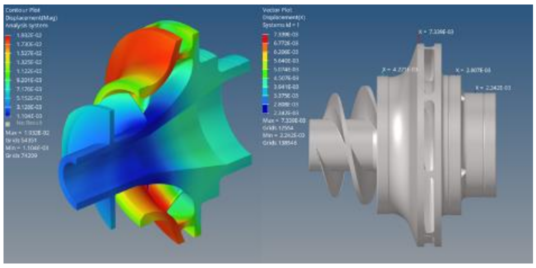

Figure 12 - LOx pump FEA analysis for the SS316L AM material.

2.6. ROTORDYN

The ROTORDYN tool implements a finite element model of a flexible shaft with rigid disks [16]. The tool creates a mesh for the shaft automatically (a Euler-Bernoulli beam element) from the segments data, as in Figure 13. Then it combines the shaft, disks, and bearings elementary matrices to calculate the Campbell's diagram, modes of vibration and unbalance response.

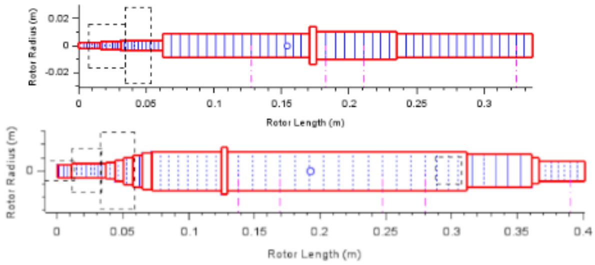

Figure 13 - Example of meshes for the first and one of the latest iterations for the shaft design using the ROTORDYN mesh.

The software is validated by reproducing literature examples with less than 5% error from [16],

The first design iteration of the complete pump, shaft, seals, and bearings assembly, shown in Figure 14, had an operation speed range above the first two modes of vibration as shown in Figure 15.

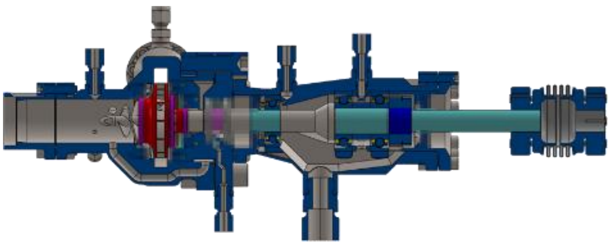

Figure 14 - First mechanical design iteration.

Even though it is common for rocket pumps to operate between natural frequencies, the vibration amplitudes and forces while going through the first two modes of vibration cause interference at several key locations.

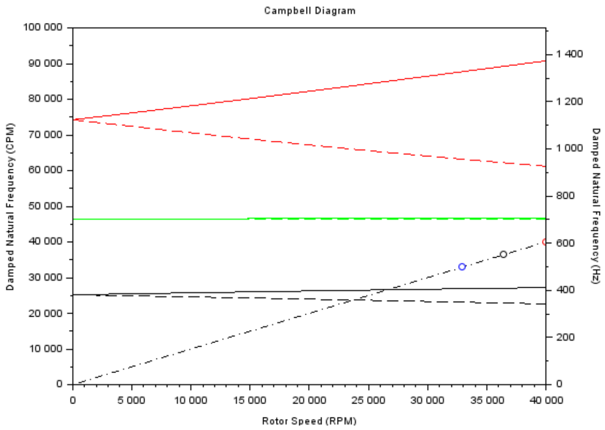

Figure 15 - Campbell's diagram of the first design iteration.

From this first iteration to the final design shown in Figure 14 and Figure 17, there were fourteen iteration designs for the shaft/seals/bearings. The final design can withstand, with small modifications, the LCH4 pump. The range of operation for the final shaft is comfortably below the first mode of vibration as shown in Figure 16.

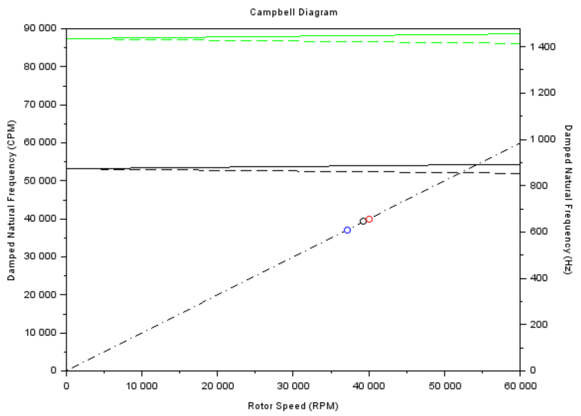

Figure 16 - Campbell's Diagram for the final design of the shaft and bearings.

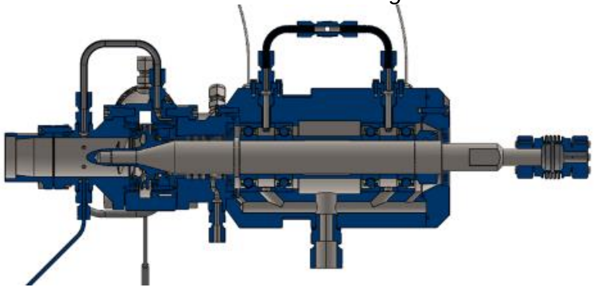

Figure 17 - The final design of the LOx pump.

The bearings are COTS and follow the manufacturer design guidelines for the spacers, preloads, tolerances, etc. The final configuration displayed in Figure 18 uses two pairs of hybrid bearings in “O-configuration” for better stiffness [17].

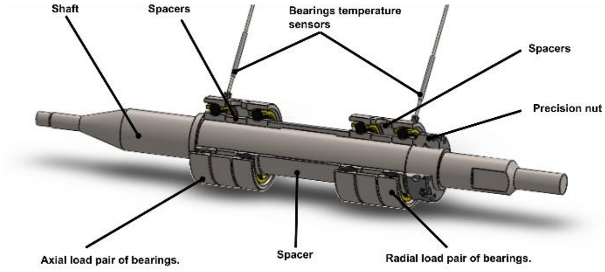

Figure 18 - Bearings architecture for the final design.

2.7. Lubrication System

The lubrication system will use direct oil injection to the balls bearing race, Figure 19. Each pair of bearings will have an injector with dedicated orifices to impinge the oil directly at the ball-race interface.

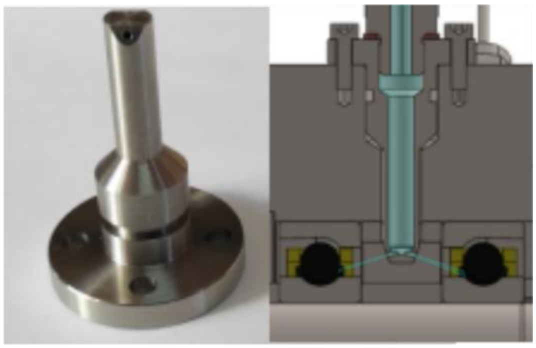

Figure 19 - An oil injector besides a cut-away vision of the assembly.

The oil pressure and temperature will be monitored right before injection. A second temperature sensor will be placed at the oil drain. The channels for draining the oil back to the circuit are shown in Figure 20.

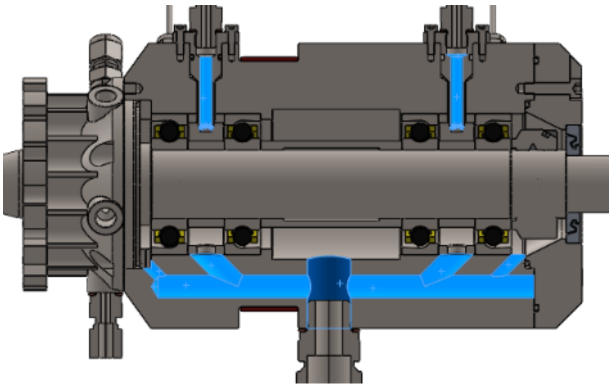

Figure 20 - Oil injection and collection.

The oil will be drained to a dry-sump oil tank that uses baffles to keep the oil free of gas bubbles. From the tank, the oil will be pumped by a gear driven electric pump, filtered, and sent back to the oil injectors. The dry-sump tank volume will be filled with GN2 and its pressure controlled to avoid issues with the oil labyrinth seals and the purging cavities of the shaft seals. A schematic of the system is seen at Figure 21.

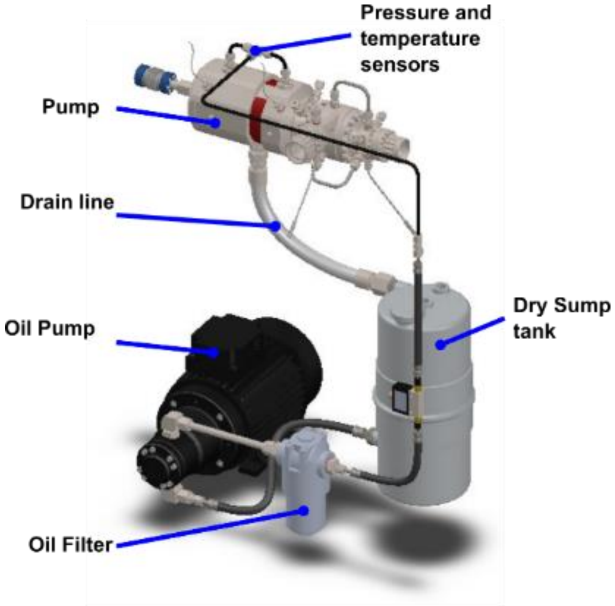

Figure 21 - Lubrication system schematic.

2.8. Pump Manufacturing

The final assembly design of the LOx pump intended for ground testing is shown in Figure 22. For the LCH4 pump, small modifications should be made to the shaft and a new design of the volute, inlet and main seals are needed, but the shaft seal interface and bearings casing share the same design.

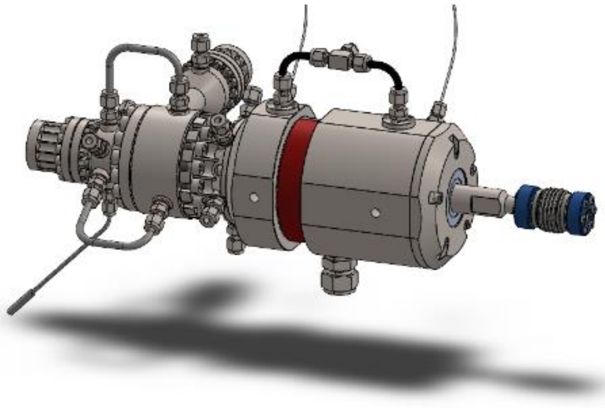

Figure 22 - View of the final design of the LOx pump assembly.

Two pump units, from different suppliers and each one using different strategies for supports, Direct Metal Laser Melting (DMLS) parameters and post treatments were manufactured. The results for the two pieces are shown in Figure 23.

Figure 23 - Left, $50\mu m$ layer with open impeller and bead blasted. Right, $20\mu m$ layer with closed impeller, blead basted and tumbling.

Key components with complex geometries were also obtained with AM techniques: inlet, volute, outlet, the interface part between the volute and the bearings casing, also holding the shaft seals, Figure 24. All components were machined according to their final specifications, as shown in Figure 25, and all other parts were fully machined or COTS.

Figure 24 – Manufactured components of the LOx pump.

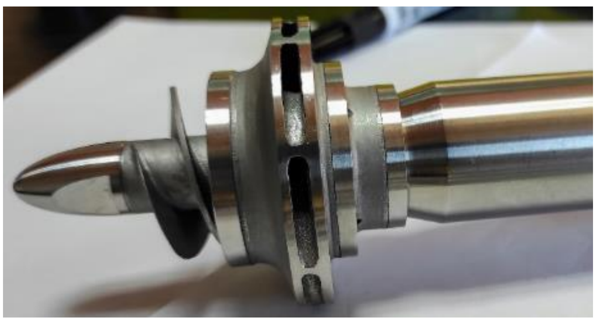

Figure 25 - Pump after machining at the shaft with the pump nut installed.

3. TEST BENCH

In order to test the pumps and characterize their performance it is necessary to measure the flow rate, the pressure and temperature at the inlet and outlet, and the driving motor speed of rotation. The torque is obtained from the electric current required to drive the motor whose torque /current relation is previously known.

The cryogenic test bench was designed to operate in closed loop and to supply the maximum flow rate for both pumps ($< 9\,\mathrm{L/s}$, for LN2). The selected parts and materials of the test bench are compatible with low temperatures and with both LCH4 and LOx, see Figure 26 for further details.

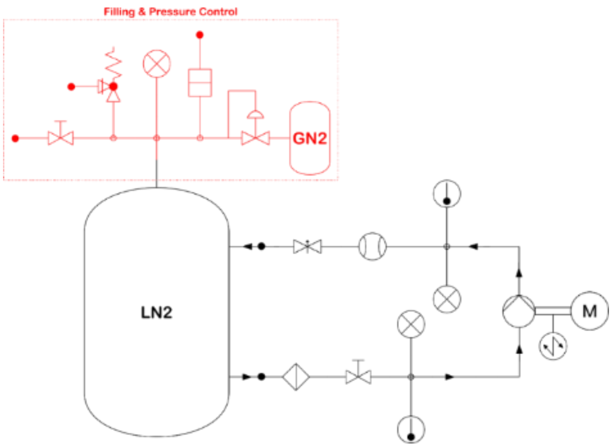

Figure 26 - Cryogenic test bench piping and instrumentation diagram.

The low-pressure line, located between the bottom of the tank and the inlet of the pump, is straight and as smooth as possible to minimize pressure losses. However, it contains a flexible metallic hose to compensate the thermal contraction of the piping and also a 150 µm filter to avoid clogging. Between the shutoff valve and the pump there is also a safety relief valve and a long ($\geq 800\,\mathrm{mm}$) and straight seamless pipe to stabilize the flow before entering the pump, Figure 27.

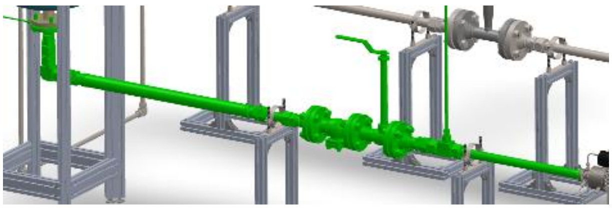

Figure 27 – Low-pressure line highlighted in green.

The high-pressure line, between the outlet of the pump and the top tank interface, also has a flexible hose to withstand thermal contraction and a long seamless tube to stabilize the flow before traversing the volumetric flow rate meter. Right before re-entering the tank, a cryogenic needle valve provides an additional variable load on the pump, thus closing the loop. A safety relief valve is installed between the flow meter and the needle cryogenic flow control valve to safely release any trapped cryogen, Figure 28.

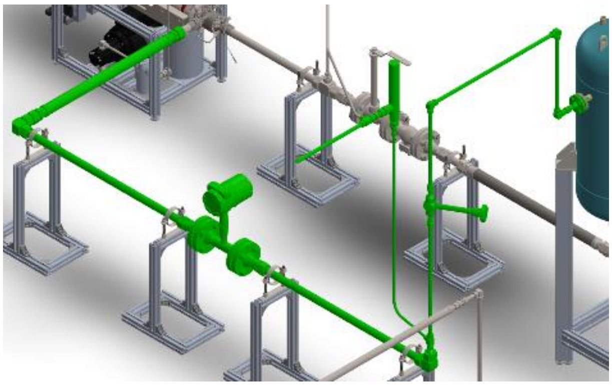

Figure 28 - High-pressure line highlighted in green.

All components of the test bench are COTS, with the exception of the tank, and most of them are cryogenic, i.e., can withstand low temperatures, and also LOx compatible. A custom double-walled tank (vacuum insulation), displayed in Figure 29, was designed by a local manufacturer. It has a volume of $V_{tank} = 100\,\mathrm{L}$ and was designed for a minimal operational temperature $T_{min} = 77\,\mathrm{K}$ at $P_{max} = 15\,\mathrm{bar}$. Such a tank is capable to operate the within the inlet envelopes of either pump.

Figure 29 - Manufactured tank.

The tank has a passive pressurization control system that maintains its pressure at a desired level, either by self-pressurization (by boiling the cryogen) or by an external gas source. A differential pressure gauge is also employed to manage the volume of liquid, particularly during the filling operations, Figure 30.

Figure 30 - Tank pressure and level control system highlighted in green.

For each test, this tank is loaded from a commercial cryogenic Dewar through a flexible transfer line, as depicted in Figure 31.

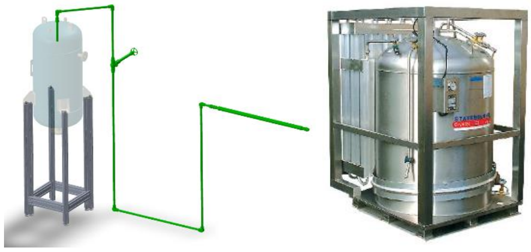

Figure 31 - Tank filling line, highlighted in green.

On this test bench, all feed lines are not vacuum insulated since long-term runs are not expected. In any case, the low and high pressure lines will be covered with insulating foam for cryogenic applications and LOx compatible.

The test bench will be completely installed in a modified 20-feet cargo container and the ergonomics and disposition of the valves, tubes and indicators were placed using virtual reality to ensure that all critical components are easily reachable and free of movement restrictions during its assembly, maintenance, and operation. A complete view of a render model of the test bench is shown on Figure 32.

Figure 32 - Cryogenic pump test bench

4. CONCLUSION

The LOx was successfully designed and is currently being assembled. The LCH4 pump I still in development but can share the same bearing, lubrication shaft seals and shaft design as the LOx pump. Components like seals, bearings, lubrication system, etc. are commercial off-the-shelf and other hardware like the volute, bearing casing, shaft, were designed to be additive manufactured or machined based on the needed complexity. The cryogenic test bench, also under assembly, was also designed to have most of its components commercial off-the-shelf, with the exception of the tank which needed a custom design for its high flow throughput and harsh operating conditions (low temperatures, hazardous and pressurized fluids).

The next phase will include completion of pump assembly, integration in the test bench followed by a water test and preliminary characterization of the pumps with LN2 and scaling factors.

5. FUNDING

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