PAPER • OPEN ACCESS

LUMEN Turbopump: Preliminary Thermal Model

To cite this article: Anirudh Mukund Saraf *et al* 2021 *J. Phys.: Conf. Ser.* **1909** 012052

View the article online for updates and enhancements.

You may also like

- Heat generation of ball bearing for rocket-engine turbopump in low-temperature gas hydrogen
  Hiromitsu Kakudo, Takashi Yokoyama and Satoshi Takada
- 1st International Seminar on Non-Ideal Compressible-Fluid Dynamics for Propulsion & Power
- On the hydrodynamics of rocket propellant engine inducers and turbopumps
  L d'Agostino

This content was downloaded from IP address 129.247.247.240 on 26/11/2021 at 07:37

LUMEN Turbopump: Preliminary Thermal Model

Anirudh Mukund Saraf[1*], Tobias Traudt[1], Michael Oschwald[1]
[1]German Aerospace Center - Institute of Space Propulsion
Langer Grund
74239 Hardthausen, Germany

E-mail: *Anirudh.Saraf@dlr.de

Abstract.
The DLR LUMEN (Liquid Upper stage deMonstrator ENgine) rocket engine comprises of two separate turbopumps, one each for Liquid Oxygen (LOX) and Liquified Natural Gas (LNG) supply. In both turbopumps, an identical bearing block separates the pump section from the turbine section. In the current design, the bearing block features an oil-jet lubrication system to cool and lubricate the bearings during operation. Cryogenic conditions at the pump-interface and high temperature conditions at turbine-interface impose thermal constraints on the selection of a suitable lubricant for the bearing assembly and the design of the lubrication system.

As a starting point to address this challenge, this paper presents a preliminary uncoupled finite element thermal model of the LUMEN LOX turbopump. The thermal model investigates the thermal behavior of the housing and rotor components in the bearing housing from start-up to shutdown of the turbopump. The model employs thermal worst case conditions to establish the operational thermal envelope of the turbopump system. Spatial and temporal evolution of the temperature at critical points within the bearing housing are reported.

The model, albeit the uncertainties rooted from its empirical nature, provides beneficial insight into the thermal characteristics of the turbopump assembly. This serves as a first estimate of the thermal constraints for the selection of lubricant oil.

1. Introduction and background

Turbopumps are high speed rotary fluid machines used in rocket engines. Their main purpose is to intake propellants from the tanks and deliver them at a higher pressure to the combustion chamber using suitable plumbing. In it's simplest form, a turbopump consists of a single shaft supported by bearings, with a pump impeller mounted at one end and a turbine rotor on the other. Due to high rotational speeds and loads, these bearings generate heat and must be cooled. Most turbopumps use the pumped propellant to cool the bearings by bleeding a small quantity of propellant from the downstream high pressure region of the pump. A few turbopumps use the conventional oil lubrication such as in the Titan, Atlas and Thor turbopumps [1]. The bearings in the DLR LUMEN (Liquid Upper stage deMonstrator ENgine) turbopumps are planned to be oil cooled and lubricated. This unconventional choice of the conventional lubrication system is attributed to the modular design approach followed in the LUMEN project [2]. While bearings cooled with cryogenic propellants are difficult to design, oil lubricated turbopump bearings come with their own challenges. One of them is the thermal conditions that the lubricating oil is exposed to in the bearing cavity of the turbopump. The cryogenic conditions at the pump and the high temperature conditions at the turbine introduce thermal constraints on the lubricant oil. If the oil reaches a temperature lower than it's pour

Content from this work may be used under the terms of the Creative Commons Attribution 3.0 licence. Any further distribution of this work must maintain attribution to the author(s) and the title of the work, journal citation and DOI.
Published under licence by IOP Publishing Ltd

point, the viscosity increases to an extent that it ceases to flow. On the contrary, if the oil reaches it's flash point temperature, the oil vapors becomes ignitable. For choosing a suitable oil, it is critical to understand the worst case thermal conditions it would experience in the bearing cavity. Thus, a preliminary thermal study of the turbopump is proposed which evaluates the component temperatures of the turbopump. A thermal worst case approach is followed, in which conservative boundary conditions are selected such that they lead to the highest (hot case) and lowest (cold case) possible temperatures in the turbopump bearing cavity during it's operation. The hot and cold cases are simulated because the temperatures obtained in the bearing cavity in these two cases serve as operational temperature extrema for the selection of the lubricant oil.

Since oil lubricated turbopumps are uncommon, thermal research of such turbopumps is sparse. Among propellant cooled turbopumps, a full turbopump transient thermal model of the MC1 Engine turbopump is presented by Roman [3], in which, the rotor and housing components are analysed as two separate 2D FE models with same boundary conditions between them. Thermal boundary conditions are obtained from different work groups and are not detailed in the report. The results of the thermal model were changes to the base design by adding a heat shield between the disk cavity and the fuel pump to minimize the impact of the unknown turbine environment. Van Hooser, Bailey and Majumdar [4] accomplished the numerical prediction of the Fasttrac turbopump's axial thrust and internal flows using the Generalized Fluid System Simulation Program (GFSSP). The program employs a finite volume approach to predict the flow characteristics. One of the reported results using this method was the pump boundary temperatures during the start transient, which could potentially be used as a thermal boundary in a non coupled thermal model of the turbopump. Thermal models of machine spindles, electric motors, automotive turbochargers and jet engines were also reviewed since they closely represent the bearing and shaft system with a separate lubrication system as planned in the LUMEN turbopumps. The work presented in this paper is a starting point for an extensive thermal investigation of the LUMEN Turbopump design. Within this framework, this paper presents a holistic uncoupled 3D FE transient thermal model for the thermal study of the LUMEN LOX turbopump.

2. Thermal model

2.1. Model and mesh

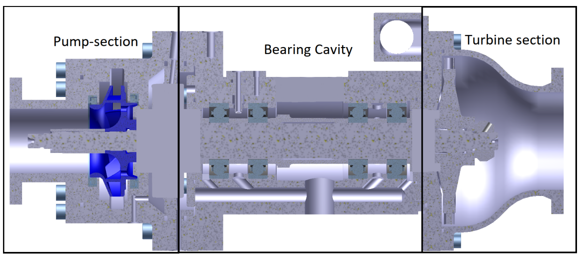


Figure 1: CAD mode of the LUMEN LOX turbopump.

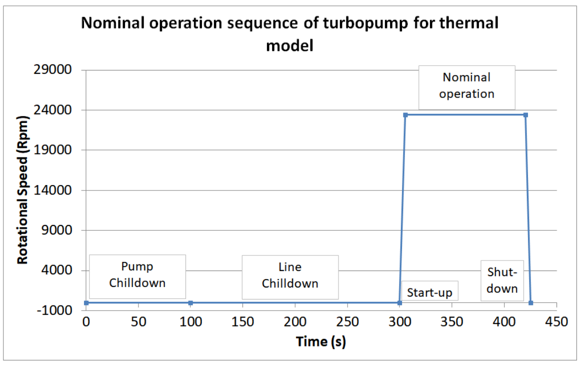


Figure 2: Operation profile of the turbopump used in the thermal model

The thermal analysis is performed in ANSYS® *Academic Research Mechanical, Release 19.1*. A CAD model of the turbopump is illustrated in Figure 1. The shaft of the LUMEN turbopump is supported by two pairs of bearings in O-configuration. The left pair of bearings form the locating bearings and the right pair forms the floating bearings. The bearing housing is equipped with oil jet lubrication nozzles with suitably sized oil-drainage holes [2] and seals separate it from the pump and turbine sections. The nominal operation speed of the turbopump is $23400\ rpm$ and the nominal turbopump operation sequence used for the thermal analysis is shown in Figure 2. LOX pump chill-down is the first step in the sequence, followed by a dwell time which corresponds to the downstream pipeline chill-down. The turbopump then starts-up to attain the operational speed, and it continues to run at that speed until it is shut to halt at the end of operation time. The Thermal Geometric Model (TGM) is a simplification of the CAD model. Geometric details such as small holes, chamfers and fillets are eliminated to reduce meshing and computational effort. The blades on the impeller and the turbine are simplified in form for similar purposes. The rolling elements of the bearing are not included in the thermal model. Since the turbopump could be closely considered as an axially symmetric system, only 1/8th of the model is simulated with cyclic symmetry boundary conditions, owing to the impeller blade number of 8. The model is discretized with hexahedral and tetrahedral elements and a conformal mesh is obtained using ANSYS® Mechanical mesh tool. The mesh element size is reduced step wise until mesh convergence is obtained at an element size of 0.001 m. Meshing revealed a total of 468561 elements and 1885023 nodes. The simplified model used as the TGM along with the model mesh is displayed in Figure 3. Materials under cryogenic conditions exhibit a significant change in material properties with change in temperature [5]. Within the turbopump assembly, the material which experiences cryogenic conditions is Inconel in the pump section. Hence, known temperature-dependent material properties are used for Inconel in the cryogenic temperature range [6].

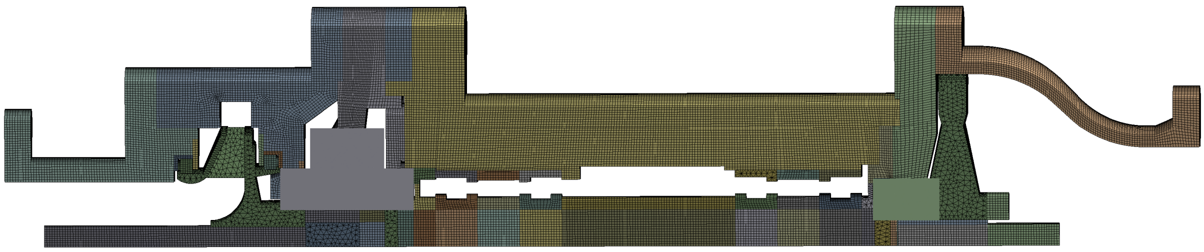


Figure 3: Simplified CAD used as the TGM along with the model mesh

2.2. Loads and boundary conditions

The thermal loads and boundary conditions considered in this study are shown in Figure 4. The friction power due to bearings is applied as a source of heat generation at the inner and outer races of the bearings. Temperature boundary conditions are applied at the pump and turbine interfaces and analytically calculated convective coefficients are utilized to simulate heat transfer to external air and internal heat transfer in the bearing cavity. The temperature rise in the bearing cavity is mainly dictated by the heat generation in the bearings and the heat flux into the bearing cavity from the turbine section. One the other hand, the temperature drop in the bearing cavity is mainly due to the chilldown heat transfer in the pump section. The boundary conditions in the hot and cold cases are obtained by selecting the turbopump design points from the operational envelop that result in the hottest and coldest temperatures in the bearing cavity. For example, at the hot case design point, the combination of heat generation in bearings and heat flux from turbine at that design point result in highest temperatures in the bearing cavity. A nominal case is also simulated, the boundary conditions of which are obtained from the nominal design point of the turbopump. Thermal contact resistance is not included in this preliminary model. A description of the boundary conditions is presented in the following sections.

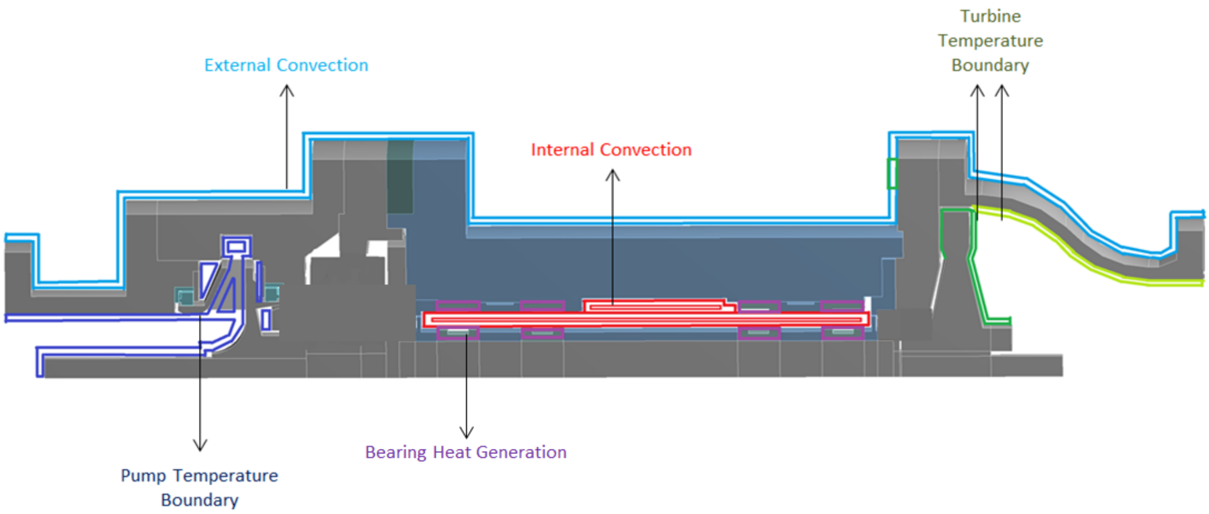


Figure 4: Loads and boundary conditions used in the turbopump thermal model

Pump During chill-down, LOX boils in the pump, cooling the pump casing and the impeller. The thermal behaviour during line chill-down in a rocket engine is studied in good detail by several authors, and models have been developed to predict chill-down characteristics. However, models to predict the flow and heat transfer of the impeller during chill-down are scarcely available. Goode B [7] performed experiments to investigate boiling heat transfer coefficients when an impeller is dipped in a pool of liquid nitrogen. Blade surface temperatures were measured at several locations. The temperature-time plot revealed a near linear temperature drop. Some thermocouples recorded a near linear temperature drop upto a knee temperature value after which the temperature dropped to the liquid nitrogen temperature immediately. Lin T Y [8] developed a FE model to predict chill-down time of a hydrogen pump. The author reports a comparison of experimental and model predicted temperatures of the pump interior points showing a near linear like temperature drop from initial temperature to saturation temperature of the medium. Therefore, in this paper chill-down of the pump is approximated as a temperature boundary condition with a linear temperature drop from initial temperature to the inlet temperature of LOX as shown in Figure 5. The time for LUMEN LOX Turbopump line chill-down is estimated by system analysis to be 250s and that of the pump to be 15s nominally. For the hot case, it is assumed that the pump chill-down takes longer by 15s and the overall chill-down is shorter by 50s and vice-versa in the cold case.

Turbine The thermal boundary layer on the turbine blade under supersonic flow determines the heat transfer between the blade and the hot gas. The interaction of the shock wave with the boundary layer on blade surfaces complicates the evaluation of heat flux to the blade as it makes the heat flux non uniform [9]. Evaluation of the heat transfer coefficients on the blade surface using CFD (Computational Fluid Dynamics) are left for a detailed thermal model in future. In this study, the heat transfer to the turbine is approximated using a temperature boundary, where, the inlet manifold and blade temperature is set to the inlet stagnation temperature. Whereas at the turbine exhaust and the disk surface at exit, temperature evaluated from adiabatic flow in the turbine is applied. This is obtained using the turbine efficiency, pressure ratio and the

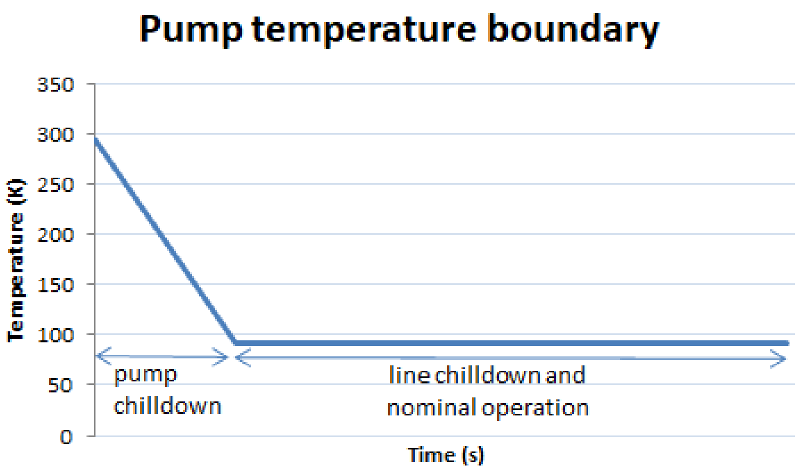


Figure 5: Linear temperature boundary approximation for chill-down of pump

ratio of specific heats of the gas. This approach leads to over-estimation of the temperatures in the cold case. However, it does not affect the lowest attained temperature in the bearing cavity, since the lowest temperature is expected at the end of chill-down, before the turbine start-up.

External housing and internal gap convection The external surface of the turbopump exchanges heat with the environment. This is modelled empirically by calculating the heat transfer coefficients for natural convection from a horizontal cylinder obtained from Incropera and de Witt [10]. The ambient temperature used for external heat convection is the same as that considered as the initial temperature of the turbopump and is based on expected annual average temperature at the test bench. Within the bearing cavity, the heat transfer between the shaft and the housing interior is addressed as a problem of heat transfer between concentric rotating cylinders with a fluid medium between them. Such a flow is called the Taylor-Couette flow. Fénot M *et al* [11] have reviewed this topic in detail and have reported empirical heat transfer coefficients for the heat transfer between the inner and the outer cylinder. The flow conditions are characterized by the Taylor number (Ta), which can be interpreted as the ratio of centrifugal forces to viscous forces on the fluid medium. Taylor number is mathematically described in equation 1.

$$T_a = \frac{\omega^2 R_1 ({R_2}^2 - {R_1}^2)}{\nu^2} \tag{1}$$

Where, ω is the rotational speed, R_1 and R_2 are the inner and outer cylinder radii and ν is the kinematic viscosity of the medium. For the speed and radii in the bearing cavity of the LUMEN LOX turbopump, Nusselt number correlation provided by Tachibana as presented by Fénot M *et al* [11] is suitable and it is described in equation 2. This equation is used to evaluate the heat transfer within the bearing cavity at different rotational speeds in each case - hot, nominal and cold.

$$Nu = 0.092(TaPr)^{0.33} \tag{2}$$

Where Nu and Pr are the Nusselt and Prandtl number of the flow respectively. It must be noted however that the medium considered in the annular gap is air, where as in reality, a two phase flow of air and oil is expected as suggested by Hannon W [12]. Such a medium with averaged air oil flow properties is not implemented in this preliminary study, since using air favours the worst case approach.

Table 1: Summary of loads and boundary conditions for nominal, hot and cold cases

Quantity/Case	Nominal Case	Hot Case	Cold Case
Pump chill-down time (s)	15	30	0
Total chill-down time (s)	250	200	300
TP operation time (s)	120	180	60
Turbine exhaust temp (K)	456.6	401.4	553.6
Turbine blade temp (K)	532.1	488.2	607.1
Pump left bearing power(W)(1/8th)	38.66	63.61	23.30
Pump right bearing power(W) (1/8th)	19.33	32.25	23.30
Turbine left bearing power(W)(1/8th)	17.67	44.72	13.90
Turbine right bearing power(W)(1/8th)	17.67	44.72	13.90
Ext. convection coefficient (W/m^2K) (Section 2.2)	Diameter based	Diameter based	Diameter based
Int. convection coefficient (W/m^2K) (Section 2.2)	Gap based	Gap based	Gap based
Turbine inlet manifold temp (K)	532.1	488.2	607.1
Initial temperature (K)	290.15	305.15	275.15

Bearing heat generation LUMEN Turbopumps use two pairs of hybrid roller bearings in which the races are made of steel and the rolling elements of ceramic [2]. The heat generated in these rolling bearings can be attributed to the friction between rolling elements, races and the cages. Several heat transfer models for roller bearings were reviewed [12, 13, 14, 15]. It was noted that every model has some empiricism associated with it and that the heat generated is dependent on the specific bearing and the bearing assembly. Heat generation data for the bearings used in the LUMEN turbopump was available at a design point from the bearing manufacturers. This data was extrapolated to hot, nominal and cold cases by using the suitable rotational speeds, axial load and radial loads for each case. This made the model specific to the LUMEN bearings and configuration which is desired in a bearing thermal model. However, in this preliminary study, conduction in rolling elements is not evaluated and the heat generated is divided equally between the inner and outer races. This is an approximation and heat-partition methods are planned to be implemented in a future model.

Summary of loads, boundary conditions and cases The thermal model is simulated for three cases: nominal, hot and cold case. A summary of the loads and boundary conditions for each case is shown in Table 1. It was observed that the heat generated in the bearings is a larger contributor to the highest temperature in the bearing cavity compared to the turbine heat flux. Furthermore, it was derived from the system analysis that for design points with higher rotation speeds, the stagnation temperature at the turbine inlet is lower and the bearing performance and load is higher. Hence, turbine blade and exhaust temperatures are lower in hot case compared to cold case while the bearing powers in hot case are more compared to the cold case. Additionally, it must be noted that the pump side bearings dissipate more power because they are locating bearings.

3. Results

In Figure 6, the turbopump bearing cavity is shown with the nomenclature used to describe the results. Figures 7, 8 and 9 are shown to compare the temporal evolution of average temperature

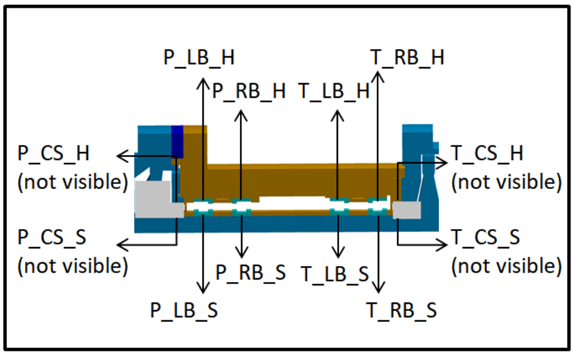


Figure 6: Component nomenclature: P and T refer to Pump side or Turbine side, R and L refers to Right or Left, B refers to bearing, CS refers to seals and S and H refers to whether the bearing or seal is in contact with shaft or housing

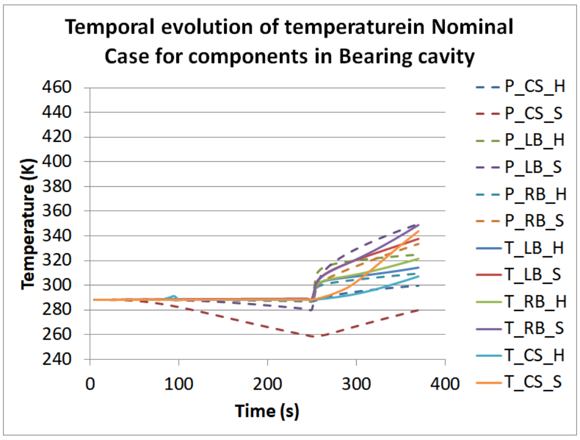


Figure 7: Temporal evolution of temperature in Nominal case

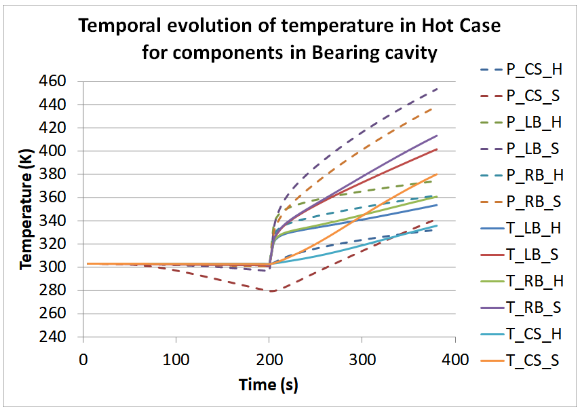


Figure 8: Temporal evolution of temperature in Hot case

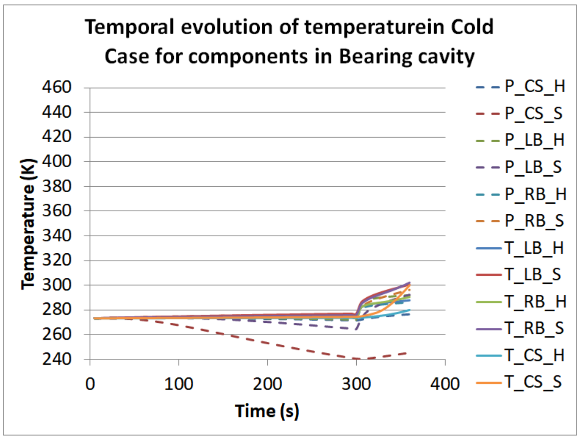
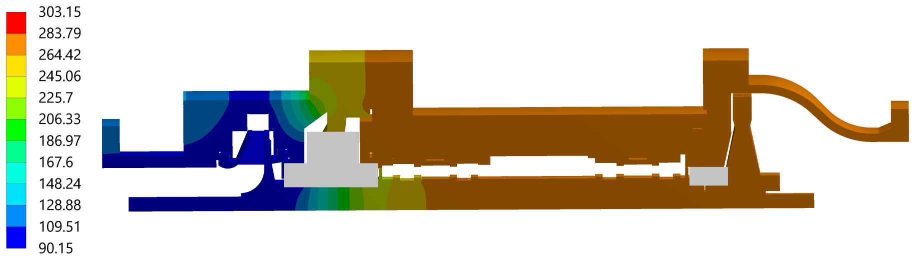


Figure 9: Temporal evolution of temperature in Cold case

of the component parts in the bearing cavity in nominal, hot and cold cases. These are also the components with which the lubricant oil comes in direct contact with. The dashed lines represent the pump side components and the solid lines represent the turbine side components. In the hot case, it can be seen that the components with highest temperatures are the pump side bearings despite being closest to the cryogenic conditions at the pump. This is attributed to the fact that the pump side bearings are locating bearings and are thus both radially and axially loaded leading to a greater friction heat generation. Comparing Figures 7, 8 and 9, after 200s, the temperature drop of the pump side seals is close to 20K. This shows that the pump chill-down time is less significant compared to the overall chill-down time, as in the cold case a total chill-down time of 300s leads to an additional 15K drop in the seal temperature. In all three cases, it is observed that the minimum temperatures for all components occur as expected at the end of chill-down and the maximum temperatures are observed at the end of operation time. To visualize the spatial temperature distributions in the turbopump at these two points in time, Figure 10 compares the turbopump temperatures in the hot and the cold cases.

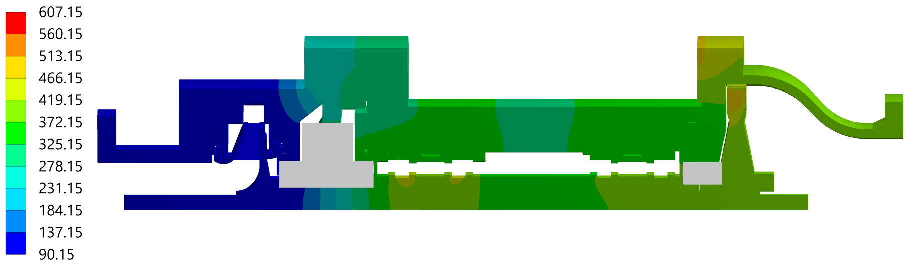
Figures 10a and 10b show the temperature distributions for the cold and the hot case at the end of chill-down, and Figures 10c and 10d show the temperature distribution at the end of



(a) Cold case at end of chill-down

(b) Hot case at end of chill-down

(c) Cold case at end of operation



(d) Hot case at end of operation

Figure 10: Hot case and cold case temperature plot of the turbopump at end of chill-down and end of operation representing the worst cold case and worst hot case

operation. Figures 10a and 10d show a significant difference in temperature of the pump side bearing at the inner and outer races. This temperature difference dictates the amount of thermal contraction or expansion in the bearing material and thus the choice of preload mechanism. A comparison of Figures 10c and 10d shows that even though the temperatures at the turbine are high in the cold case, within the bearing housing, the heat generated from the bearings contributes more to the rise of temperature. Figures 11 summarize the results from Figure 10 by showing the predicted temperature range which the components in the bearing housing experience in the hot and cold cases. It can also be observed that the components in contact with the shaft have more extreme temperatures than those in contact with the housing and this is expected since the thermal mass of the shaft is smaller compared to the housing.

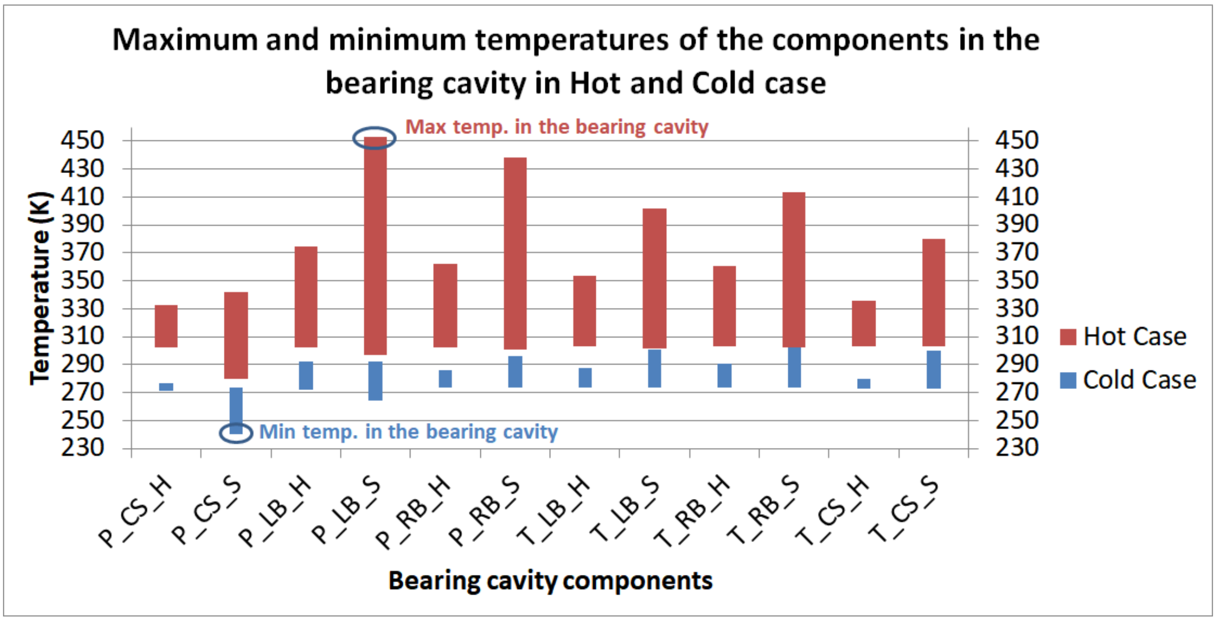


Figure 11: Component temperature ranges in hot and cold case

4. Discussion

Based on the analysis, the following requirements are drawn for the selection of the lubricant oil. Firstly, to reduce the risk of an explosive reaction with LOX, the oil is preferred to be compatible with LOX. This adds redundancy over the inter-propellant seals which ensure that the oil doesn't come in contact with LOX. The coldest temperature predicted in the turbopump bearing cavity is 240.7 K. The pour point of the lubricant oil must be lower than this temperature to ensure continuous flow of the lubricant oil. The hottest temperature predicted in the turbopump bearing cavity is 453.4 K. This sets the lower limit for the flash point of the oil. The lubricant oil should not have a flash point lower than this and in the best case should be non-flammable. Since the viscosity of the oil changes with the change in temperature, the lubricant oil must have a high viscosity index leading to lower changes in viscosity with temperature. From the results of the thermal model, it can be assessed that oil temperature control and flow control is desired for the lubrication system. Since it is seen that there exits a difference in temperature at the inner and outer races of the bearings, oil temperatures and flow rate control to regulate bearing temperatures could prove advantageous for each bearing set.

5. Conclusion

In this paper a preliminary thermal model of a turbopump is presented to asses the thermal conditions within the bearing housing. Simplified CAD and boundary conditions are applied to approximate the complex physical phenomena in the turbopump. A worst case approach is adopted to identify the extreme conditions the lubricant oil is expected to encounter in the bearing cavity. First estimates of temperatures within the bearing housing are obtained and requirements are drawn for the lubricant oil. The thermal model is planned to be compared with forthcoming tests of the LUMEN LOX turbopump. Some of the modelling improvements which are identified for a future model include: prediction of turbine blade heat transfer coefficients, inclusion of a heat partition method for splitting the heat generated between inner and outer races of the bearings, inclusion of thermal contact resistance between components and employing a two phase air-oil mixture model for heat transfer in the annular gap in the bearing cavity.

References

[1] Butner M and Keller R 1971 *Liquid rocket engine turbopump bearings* (United States: NASA Technical Report Server)
[2] Traudt T, Mason T, Deeken J, Oschwald M, Schlechtriem S, dos Santos Hahn R H and Mader C 2019 *Proc. Int. Symp. on Space Technology and Science* (Fukui, Japan)
[3] Roman J 2002 *Proc. Twelfth Thermal and Fluids Analysis Workshop* (Huntsville, Alabama)
[4] Hooser K V, Bailey J and Majumdar A 1999 *35th Joint Propulsion Conference and Exhibit* (Los Angeles: AIAA: AIAA)
[5] Barron R 1999 *Cryogenic Heat Transfer* (Taylor & Francis)
[6] Sweet J, Roth E and Moss M 1987 *Int. J Thermophy* **8** 593–606
[7] Goode B K 2002 *Proc. Twelfth Thermal and Fluids Analysis Workshop* (Huntsville, Alabama)
[8] Lin T Y 1991 *Cryogenics* **31** 411 – 416
[9] Larsson J 1999 *Numerical Simulation of Turbulent Flows for Turbine Blade Heat Trasnfer Applications* Ph.D. thesis Chalmers University of Technology
[10] Incropera F P and DeWitt D P 2002 *Introduction to Heat Transfer* (New York: Wiley)
[11] Fénot M, Bertin Y, Dorignac E and Lalizel G 2011 *International Journal of Thermal Sciences* **50** 1138 – 1155
[12] Hannon W M 2015 *Journal of Tribology* **137**
[13] Sibilli T and Igie U 2017 *Journal of Engineering for Gas Turbines and Power* **140**
[14] Harris T A and Kotzalas M N 2007 *Advanced concepts of bearing technology.* 5th ed (Taylor & Francis)
[15] Wu L and Tan Q 2016 *Entropy* **18**