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Mechanical design and manufacture of a boundary layer pump

Lapo Bufalari[2], *Eduardo* Anselmi[1*], *Mirko* Rinchi[2], *Kevin* Howard[1], *Lorenzo* Talluri[2], *Daniele* Fiaschi[2]

[1]Centre for Propulsion and Thermal Power Engineering, Cranfield University, Bedford MK43 0AL, UK
[2]Department of Industrial Engineering, University of Florence, 50121 Firenze, Italy

Abstract. This paper describes the current efforts to develop and manufacture a first prototype for a boundary layer pump as a mean to assess future and more complex designs. Following an approach of "learning by doing", a previous design was re-assessed from a mechanical/workshop point of view. Budget constraints and in-house manufacturing capabilities were taken into consideration to deliver a new design, suitable for quick production. Challenges such as disc holding, gap spacing, pump intake, discharge nozzles, and tolerances were addressed. Structural analysis has been conducted; where every single component has been modelled and sized accordingly to standard practices. As a support of structural analysis, FEM analysis was also performed with the aim of identifying, discussing, and fixing any potentially critical issues, particularly regarding the bolts holding together the discs into the power shaft. Finally, modal analysis was performed in order to test the dynamic response of the rotor: its critical frequencies would be far from the working range of the machine. This paper gives an overview of the critical issues to be taken into account during the mechanical design of boundary layer pump prototypes for different working fluids in the field of power generation and thermal management.

1 Introduction

In many fields of application such as power generation and aeronautical engines, rotor blades are the most commonly used component due to their high performance. However, despite their high efficiency, they introduce numerous defects and limitations like shocks, vibrations, and complexity during modelling, manufacturing and maintaining. In 1913, Nikola Tesla patented a new type of pump [1], as known as boundary layer pump (or friction pump or Tesla pump), with the purpose to overcome all these problems. Tesla pump is made of a number of thin, smooth, flat, parallel discs arranged normal to a shaft and fastened rigidly to it with small spacers between discs. Fluid comes into the pump through holes or slots near the shaft, and then enters in the gap between discs. The shaft is connected to an external drive, which allows the spinning of the discs and causes the

* Corresponding author: e.a.anselmipalma@cranfield.ac.uk

centrifugal forces. The main working principle of function is the viscosity of the fluid of work: friction and centrifugal forces generate a differential pressure between the inner and outer part of the discs. Despite robustness and ease to manufacture, the boundary layer pumps have efficiencies lower than the traditional rotor blade ones.

2 Literature review

Many research studies have been carried out to gain a better understanding of its functions and to determine the factors, which may influence its efficiency and performance, in particular pump but also turbine prototypes are taken into account because of their similar design. In 1950, Leaman [2] wrote a M.Sc. thesis about the design, construction and experiment testing of air Tesla turbine. He focused on the study and measurement of losses caused by bearings; using new deep grove bearings in place of the old ones, the efficiency significantly increased. In 1952 Armstrong [3] wrote a thesis about investigation of the performance of a steam Tesla turbine, aimed at determining the ways to improve the original Tesla's design. In 1960, Hasinger and Kehrt [4] investigated analytically and experimentally a Tesla pump using water as working fluid. Combining the results with the mathematical model, the predictions indicated that a rotor efficiency up to 60% would be possible. In 1963 Rice [5] did an experimental investigations of multiple discs rotor. Three different prototypes were designed and tested: a water pump, an air compressor and an air blower. The results were compared with non-dimensional model, in order to provide general rules for modelling a Tesla rotor. In 1973, Morris [6] investigated the influence of the gap spacing using water as working fluid. The investigation started from a modified commercial pump equipped with an overhung shaft. In 1987, Darby [7] presented an experimental evaluation of the performance and characteristic of a water Tesla pump. Multiple configurations were used in this project, in order to obtain and compare different results. In 2007, Bloudiček [8] worked on the design, manufacturing and testing of a water Tesla turbine, with the objective of achieving a significant effectiveness. In 2014, Wang [9] designed and built an experimental test rig for a shear force pump. The objective of the paper was investigating the internal flow dynamics and clarifies the loss mechanism. In 2015, Dodsworth and Groulx [10] presented a document about the study of a Tesla pump aimed at finding a relationship between discs' gap and efficiency. In 2020, Talluri et al. [11] investigated the optimal design procedure of Tesla turbines for ORC applications. A through optimization method was performed by evaluating losses of each component and by introducing an innovative rotor model. In 2019, Martinez-Diaz [12] published an article on the effects of turbulization on the pump discs' performance. The objective was increasing the performance of this pump developing a new experimental study based on the turbulization of flow by the placement of turbulizers into the inter disc channel output. In 2019, Nicotra [13] wrote his M.Sc. thesis on the numerical model and design procedure to realize Tesla pump prototype using water as working fluid. In 2020, Heng [14] published a paper about the preliminary experimental work on a centrifugal pump model, specifically designed to transport slurry and multiphase flows.

3 Mechanical design of the Tesla pump prototype

A new mechanical evaluation was carried out based on an existing model [13]. Standard components were sized and modelled according to respective standards rules, while other parts were designed taking into account budget constraints and in-house manufacturing capabilities, with the goal of speedup the prototype manufacturing as quick as possible. The internal volute follows a logarithmic spiral using the same parameters as found in Nicotra's model. The two most critical parts of the prototype were the inlet and outlet. Water enters into the pump through a pipe screwed onto the casing. While this inlet solution proved to be the most optimal one, it could entail reverse flow phenomena inside the pump not reaching the rotor pack. The outlet section proved to be very complex to model, since its shape and size were fixed by the geometry and the shape of the volute. It was necessary to introduce a new part, connected with screws to the casing, allowing modification of the section from rectangular to circular through a diffuser. A pipe with the same size as the inlet was screwed onto this part.

4 FEM simulations

FEM simulations were used as support to the mechanical design, in order to identify critical issues and fix them. Two different software (Solidworks and Ansys) were used and the related results were compared. Two types of static analyses were conducted, using a simplified rotor model: the first one considering only the gravitational force, while the second one added the centrifugal force contribution to simulate real working conditions at 5000 rpm. The in-house components were made in Aluminium. This material is cheap, easy to manufacture and prevents rust. In this project, it was assumed that standard components were realized with the same material, such as a structural steel, in order to ease the modelling process. Both of the software had a large materials database making the selection of the proper one easy. The challenge was finding, within the database, materials closely matching the ones actually planned to be adopted, possibly being similar in the two software. This choice was done because different inputs, such as different materials, can significantly affect the final results of the mechanical analysis. In Solidworks Simulations, 1060 Aluminium Alloy was chosen for the in-house parts and AISI 304 for the standard components. In Ansys, Aluminium Alloy and Stuctural Steel were selected for the two parts. In the simplified model, the bearings were not taken into account; constraints were required to simulate the bearings. In Solidworks Simulations, specific constraint called "Bearing Support" was used. Self-alignment was chosen to prevent unrestricted off-axis shaft rotation. In Ansys environment, the bearings were modelled as a cylindrical constraint with tangential displacement free to move. In this model, it was assumed that constraints are completely stiff. This assumption could be a problem and gives results far from the real conditions because actually, bearings are flexible, allowing radial and axial deformations. In FEM analysis, meshing is a critical process, which influences the accuracy of the results and the computational efficiency of the analysis. It requires a careful trade-off between accuracy and computational effort, since a finer mesh will generally provide more accurate results but will also increase computation time. Same parameters were used for meshing process in both of the software in order to obtain similar meshes. The quality of the mesh is critical to the accuracy and reliability of simulations. In this project, quality check of the mesh was performed taking into account two parameters: Aspect Ratio and Jacobian. Both of the meshes respected the high-quality mesh standard for the software.

5 Results of gravity analysis

The main focus of the analysis was determining the stress and strain distribution of the most affected component. The results from both software showed very similar stress values distribution and the maximum value was almost the same. It was important to note that the maximum stress values remained within the elastic range, and there were neither plastic deformation nor component failure predicted. The most stressed components were the bolts. These parts had the main role to keep discs and washers in position, supporting the weight of the whole rotor pack. The maximum strain was extremely limited, such as can be approximated to zero. In this case, the maximum deformation values of specific components may differ in the two software. In Solidworks, the first disc close to the shaft resulted having the highest deformation, while in Ansys, the shaft resulted having the highest value. These results sound consistency with the real working condition of the rotor.

6 Results of rotational analysis

As from the previous simulation, in both of the software, the maximum stress values resulted very similar, largely below the yield stress. From Solidworks results, the most stressed component was one of the bolts keeping discs and washers in the fixed position, with maximum value at 2.785 MPa. Ansys achieved similar results with the bolt resulting the most stressed component at the very similar value of 2.50 MPa as shown in figure 1 below. The results highlighted an interesting observation; e.g. both of the software addressed an unique maximum value of stress for one component only. With a slight deepening the analysis, it was easy to verify that the three bolts had similar stress values due to the rotational speed and centrifugal force. In both of the software, the flange resulted in the maximum strain that it could be assumed negligible because it is close to zero. These results look consistent with the real rotor working condition.

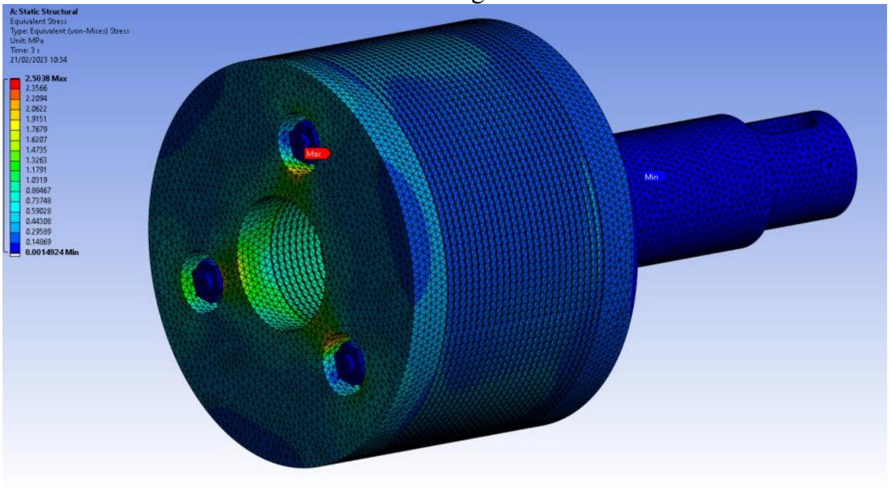

Figure 1 - Stress distribution in rotational analysis in Ansys

7 Results of modal analysis

This analysis aimed at analysing the dynamic behaviour of the rotor, with a special focus on the identification of its critical frequencies and vibration modes. To achieve this, a modal analysis was conducted using the two software, e.g. Solidworks simulation and Ansys mechanical. The required input was the number of frequencies. After running multiple simulations, twelve frequencies were selected as the optimal amount to catch the rotor's behaviour, enough to cover the real range of frequencies. A comparison of the results obtained from both of the software indicated a discrepancy less than 1% between the corresponding values. The first critical frequency was approximately 670 Hz, this value was much higher than the maximum operating conditions of the rotor (83 Hz at 5000 rpm). This suggested that the rotor is operating in a safe zone and there is no evidence of resonance phenomena occurring. Given the complexity of these properties and factors, the analysis of each individual mode was very hard. However, a criterion to divide the modes into three categories was identified. To explore this phenomenon, several simulations were performed with different number of discs (zero, one, five, complete assembly) and the observed results appeared consistent across all the scenarios. Specifically, three categories were observed in each simulation, providing strong evidence for the criterion's validity. The first identified category was that of inflectional modes, the first and second modes of vibration were bending modes. Rotor and, specifically, flange, screws and discs pack moved on a plane perpendicular to the rotational axis. The second category consisted in a torsional vibration mode. The third mode was purely torsional, because there was a movement only around rotor rotational axis. Discs vibration represented the third and final category. Starting from the fourth mode onward, vibration modes of the rotor were a mix of modes of the single discs. While all other components remained motionless, the discs only had motions perpendicular to their plane. In figure 2.a, 2.b and 2.c, the three categories are shown. Because each disc had its own vibration mode and each rotor mode was a combination of 25 disc modes, it was impractical to conduct an in-depth analysis of every single mode of this category, as it would require a significant amount of time and computational power without giving further relevant information. It was important to investigate the underlying reasons of these behaviours. For each category, the components and their influence on the rotor vibration were thoroughly analysed. It revealed that, for the first and second category, the screws holding discs and washers in position were the main components affecting the vibration modes. A simulation of the shaft and the three screws was performed by examining the combination of the screws free movements.

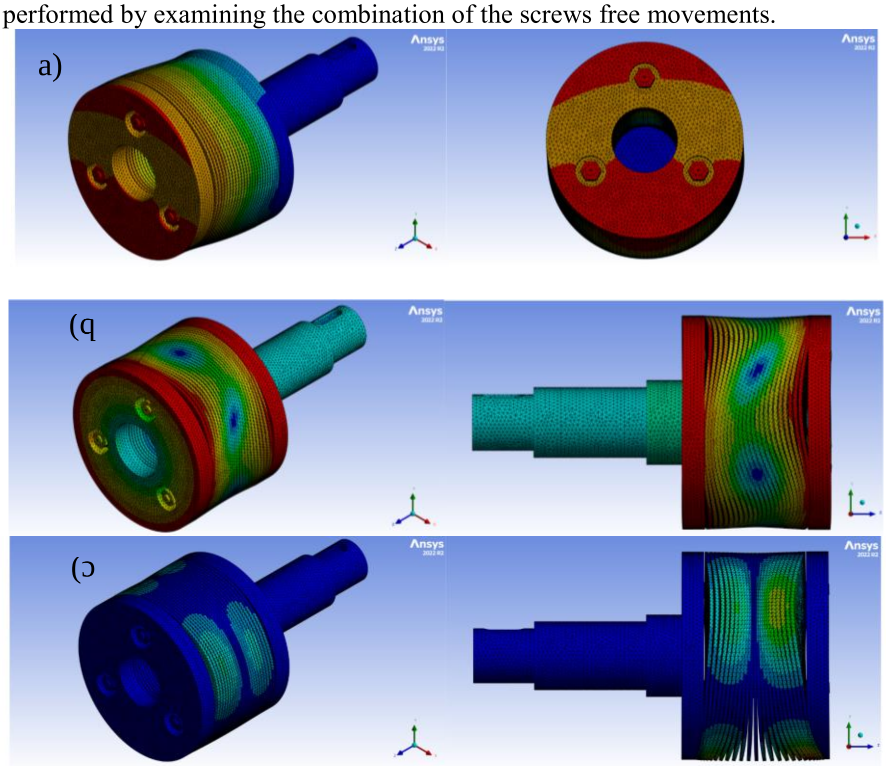

Figure 2 - Vibration modes in Ansys : a) First mode (bending category) at 678 Hz. b) Third mode (torsional category) at 1264 Hz. c) Fourth mode (disc vibration) at 1571 Hz.

In this way, it was possible to characterize and catch the rotor vibration modes. Specifically, in the first and second mode, all three bolts moved synchronously in the same direction. The movements in the first and second mode defined the first bending category of the full rotor. Moving on the third vibration mode, the screws moved in different directions, resulting in a torsional deformation of the rotor (the second purely torsional category). Both software addressed this behaviour. From the fourth mode onward, the discs were the main components affecting the third category of vibration of the rotor. The simulation of the rotor with one disc only revealed that the single disc has only three significant vibration modes. The disc, due to the three bolts, could be divided ideally into three parts, or lobes. Each one can be moved in a specific way. Depending on how the lobes move in combination with each other, the disc will vibrate in specific modes. Three type of vibration were identified: the first one occurred when two of the three lobes moved asynchronously, while the third one still remain; the second one occurred when two lobes moved synchronously in one direction while the third one moved in the opposite direction; finally, the third one occurred when all three lobes moved together synchronously. By combining these movements for every disc, the overall set of vibration modes of the rotor can be determined.

The bearings played a crucial role in the model because, despite in most typical mechanical analyses they are assumed to be perfectly still, actually they are not so. The first step to analyse this occurrence consisted in a free-free modal analysis of the rotor without bearing constraints, allowing all degrees of freedom to be free. Another important modal analysis was performed by using a mathematical model to calculate a real value of the bearing stiffness. The comparison of the critical frequencies values obtained in the three types of simulations was particular interesting. Its significant impact on the first three critical frequencies became evident (bending and torsional modes), whereas the values starting above the fourth frequency were similar because the discs were the main relevant component. It was also noteworthy that the first three frequencies of the stiff bearings had higher values than those of the deformable ones. This condition was in agreement with the expected dynamic behaviour of the system. In all the three analysed cases, there was a noticeable sharing into the same categories: bending, torsional and discs vibration modes. The presence of the same pattern in all the three simulations suggests that the rigid modelling of the bearings is adequately representative of the real dynamics of the bearings, or, at least, sufficient for the purpose of this project.

8 Conclusions

The simplicity and the robustness of the Tesla pump design appear suitable for applications where traditional bladed turbomachinery may fail. The purpose of this manuscript was establishing guidelines for the design of a Tesla pump, by analysing the most critical and stressed components. The presence of two critical components was confirmed: the bearings and the bolts. The formers did not comply with the real modelling conditions, which may lead to an inaccurate solution. On the other hand, the bolts resulted being the most stressed components, having a relevant influence on the values of natural vibrational frequencies of the rotor.

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