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NUMERICAL INVESTIGATION ON THE VIBRATION REDUCTION OF ROTATING SHAFT USING DIFFERENT GROOVE SHAPES OF TILT BEARING

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Abstract

Vibration control is very important for high-speed rotors. Oil film damping is considered an effective vibration-damping method, especially for long shafts in gas turbines, ships, and other high-speed rotating equipment. The existing groove in the internal surface of the tilt bearing increases the amount of oil that flows through the bearing; this is more effective in suppressing the vibration of the rotor system carried by the plain bearing. In order to suppress the vibration of the rotor system, which is supported by sliding bearings, a different groove-shaped oil flow (GSOF) is studied and analysed in this paper. A different shape of grooves in bearings was set up and measured to study the vibration-damping effect of the flow oil shape with GSOF. ANSYS software presents significant benefits to engage Fluent for oil flow with Transient structural for vibration measurements. This paper uses these terms to perform the simulation numerically to explore the groove-shaped damper's damping effect under the rotor system. The study identified three enhancements of vibration and settling time. First, the circular groove showed a 35.71% reduction in amplitude and 10% increase in stilling time; the next one is the circular groove which reduced the amplitude by 42.85% and the settling time by 0%. The third modification was the inclined groove which reduced the amplitude by 42.85% and the settling time by 12%. The last one was the triple-inclined groove, which reduced the amplitude and settling time by 57.14% and 20%, respectively.

Keywords: tilt bearings, bearing groove, rotor system, ANSYS transient, fluent, vibration reduction, finite elements.

List of Symbols/Acronyms

FEM – Finite element method;
GSOF– Groove-Shaped oil flow;
HSFD – Hydrolically-squeezed film damper;
ISFD – Integral-Squeeze film damper;
SFDs – Squeeze Film Dampers;
TP – Tilting pad;
l – distance [mm];
Δ – measurement tolerance [mm].

1. INTRODUCTION

The effect of vibration on high-speed rotor systems is essential, especially suitable for important modern industrial equipment such as aero-engines, industrial gas turbines, and centrifugal compressors [1]. The equipment is more likely to vibrate significantly due to high speed and rotor eccentricity [2]. Excessive vibration can cause malfunction or severe damage accidents if the equipment is not controlled. Using an oil flow shape damper (OFSD) becomes another choice for some researchers to reduce thevibration of rotating machinery rotors carried by rocker arm bearings. It has been extensively studied by many researchers. A 45° inclined side of the groove of the internal surface of a tilting pad (TP) bearing with an internally pressurized oil film damper [3].

Moreover, B. Ertas et al. used integral squeeze film damper ISFD to solve the problem of sub-synchronous vibration at full load and full speed operation of a 46 MW, 6230 kg, multi-stage, multi-purpose turbine [4]. Control the vibration of the bearing-rotor system by adjusting the oil supply pressure to change the stiffness and damping characteristics of the squeeze film damper [5]. Through theoretical and experimental studies, this paper discusses the vibration-damping effect of plain bearings with HSFD. Cai Wan Et al. Systematically analyzed the dynamic behavior of the PSFD transmission mount system and found that PSFD can improve the dynamic stability of the transmission mount system [6]. A. Benzidine and M. Thomas study the nonlinear dynamic behavior of a rigid rotor supported by a hydrostatic squeeze film damper [7]. The majority of conventional squeeze film dampers (SFDs) and structure-improved dampers use passive control strategies, which are currently more useful for damping vibration. These researchers have contributed to the field of active control. An electro-

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hydraulic actuator was introduced by Cai-Wan et al. to a static SFD compensation. They discovered that a typical squeeze film damper might have its unsynchronized vibration reduced. [8]. Based upon a standard squeeze film damper, a system of actively controlled lubrication using magnetorheological fluids was studied by Ajay et al. They found that it is possible to reduce the amplitude by as much as 70% near the critical speed [9]. When successfully passed the initial critical speed, R.P. Spada and R. Nicoletti utilized the Udwadia-Kalaba approach to trajectory control of nonlinear systems, lowering the rotor's lateral vibration at the control point. [10]. The field of rotor system damping has used passive control techniques widely. There are limited research on the use of sliding bearing rotor systems, which are more commonly used for rolling bearing rotor systems' vibration damping. This study investigates a sliding-bearing rotor system's vibration reduction using a hydraulically controlled squeeze film damper (HSFD). To reduce the vibration of the bearing rotor system, the oil supply pressure is altered to alter the stiffness and damping properties of the squeezing film damper. [11]. In this paper, a groove-shaped internal surface of the tilt-bearing damper is considered to be used for the vibration reduction of a double rotor system. The internal shape of the bearing and the oil supply pressure change the mass flow rate of oil and the damping characteristics of the oil film damper to control the vibration of the bearing rotor system.

2. 3D MODELL DESIGN

Figure 1 shows a 3D model of the unbalanced rotating shaft. The geometry was accomplished by using SolidWorks 2020 software. An electric motor drives the model. This equipment's operating range is 0 to 250 RPM. A flexible clutch connects the shaft to the motor. The other end of the shaft is supported by a thrust bearing to prevent displacement in the axial direction. The shaft is included three tilt bearings where the oil flows inside them to reduce vibration. The distance between the bearings is equal, which is 450 mm. The diameter of the rotating shaft is 50 mm. The two unbalanced rotating discs

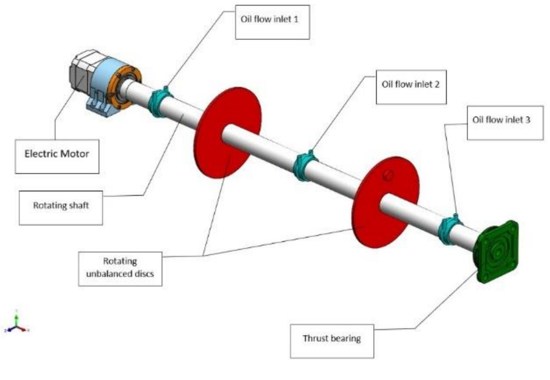


Fig. 1. 3D model designed by Solid Works software

are located in the midspan of the bearings, as shown in figure 1. Each disc has 200 g mass to produce the vibration for the shaft. In this work, five different bearing designs will be considered, as shown in figure 2.

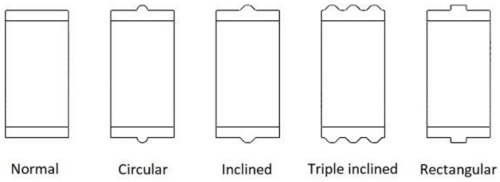


Fig. 2. Cross section of testing tilt bearings to figure out which approach is the optimum

3. MESH SETTINGS

After modelling the geometry, meshing is required for analyzing the body. Tetrahedral elements have been used to mesh the model because it adapts to the sharp curves and edges of the oil flow and rotating parts. A grid-independent test is used to validate the mesh, which is done by reducing the element size to get the accepted level of tolerance, of which the Grid Independence test contains. Changing the mesh size from coarse to fine and comparing the output results for each mesh is how this is accomplished. When changing the mesh, the results are alittle affected, and there is no need to create a minor element size and select that minimum mesh size for our final solution output. For the standard bearing, the nodes and elements are 77176 and 301176, respectively, and for the triple inclined design, they are 88323 and 359904, as shown in Figure 3.

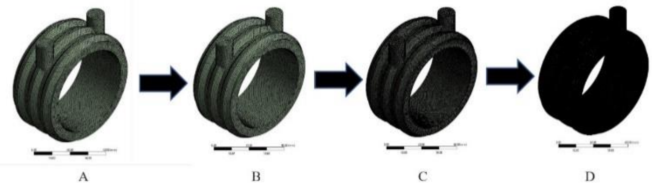


Fig. 3. Shows mesh different element size for mesh validation (a) 1mm, (b) 0.75mm, (c) 0.5mm, (d) 0.25mm

The same approach is used in the transient structural model meshing as in FLUENT. Figure 4 illustrates the changing element size from 5mm to 1mm, giving stable results for the amplitude at 2mm and 1mm sizing. The nodes and elements statistics of 5mm size are 173307 and 40227, whereas the nodes and elements number for 1mm size are 4610547 and 587716, respectively. The optimum size is 2mm because the amplitude does not change much at 1mm element size.

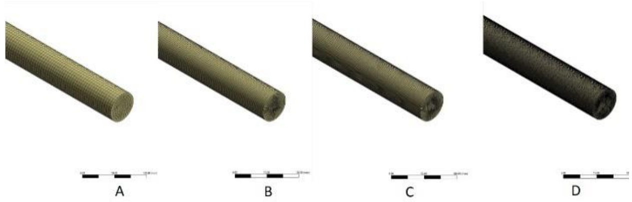


Fig. 4. Shows mesh different element size for the rotating shaft. (a) 4mm, (b) 3mm, (c) 2mm, (d) 1mm

4. SIMULATION STUDY USING ANSYS FLUENT AND TRANSIENT STATIC STRUCTURAL

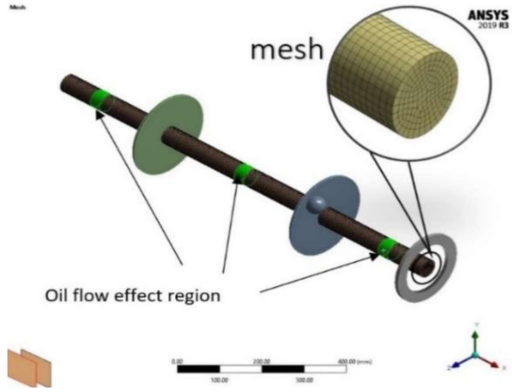
The finite element method (FEM) is considered the most proper simulation technique for denoting the physical behaviour of structures and systems [12]. The majority of problems in daily life do not have appropriate methods in the engineering sciences, hence numerical solutions were created to address the specific instances. In this work, ANSYS software has been used to simulate the dynamic model of the rotating shaft. Because ANSYS can analyze the oil flow and its pressure effect on the walls of the bearings and shaft. The properties of the oil are illustrated in Table 1 [3]. The fluent results are transferred to transient structural software to analyze the eccentricity of the shaft due to the unbalanced rotor. The 3D model is applied in the analysis. The dimension is taken into account in the three-dimensional model of the double rotors, and the mechanical properties of the shaft are given in table 2. The rotor model has been entered into ANSYS 19.3 FEM software, in which different steps of FEM are used. First, for analysing the body, we need to use tetrahedral elements to mesh the model; Tetrahedral mesh was used to support the intricate curves and edges of the oil flow and rotating parts. For the normal bearing, the nodes and elements are 77176 and 301176, respectively as shown in Figure 5(b), and for the triple inclined-side design, they are 88323 and 359904, as shown in Figure 3.

Table 1. The properties of lubrication oil

(Properties)	(Density at 20°c g/cm³)	(Specific gravity at 20°c)	(Viscosity at 40°c (cst)	(Viscosity at 100°c (cst)
value	0.8503	0.8514	31.5	5.1

Table 2. The Mechanical properties of the rotating shaft

Properties	Density kg/m^3	Tensile Yield Strength MPa	Young's modulus GPa	Poisson s Ratio
value	7850	250	205	0.3



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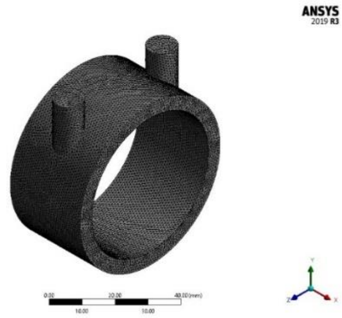

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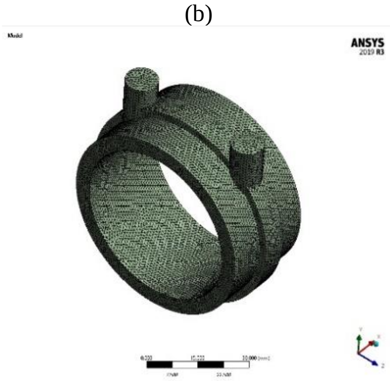

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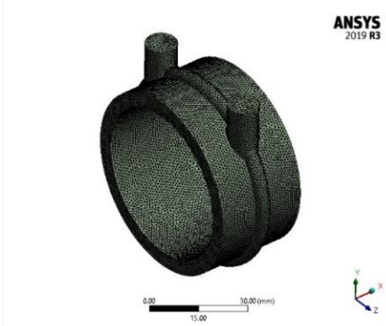

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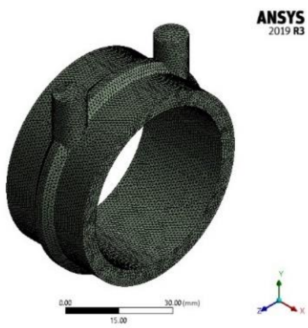

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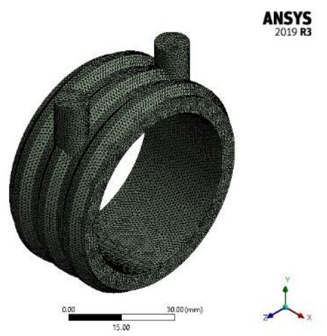


(f)

Fig. 5. Shows mesh distribution for transient analysis of normal bearing (a). b), c), d), e), f) Show the mesh elements for the fluent analysis of all bearings

5. RESULTS AND DISCUSSION

5.1. Analysis of oil flow

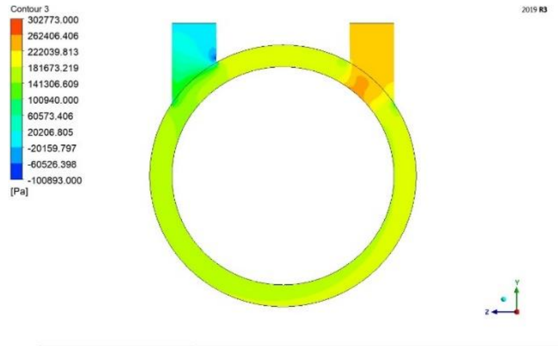
The oil flow was made as geometry in SolidWorks and transferred to the ANSYS geometry feature. The next step in FLUENT is Mesh, the selection of optimum mesh needs to try more than one mesh and compare the results. A suitable setup parameter was used to perform the fluid flow. Simulating the flow standard k-ε model has been used because it has solving two independent transport equations in two steps. Since it was first introduced by Launder and Spalding, the standard model in ANSYS Fluent, which belongs to this class of models, has taken the lead in actual engineering flow computations [13]. It is a semi-empirical model, and the phenomenological and empirical approaches are used to derive the model equations. The turbulence-kinetic-energy (k) and its dissipation rate (ε) model transport equations serve as the foundation for the conventional k- model. The precise equation is used to obtain the model transport equation for k.

In contrast, Physical reasoning was used to develop the model transport equation for, which differs significantly from its mathematical counterpart. In the derivation of the k-ε model, the assumption is that the flow is fully turbulent, and the effects of molecular viscosity are negligible. Therefore, the standard k-ε model is valid only for fully turbulent flows. As the strengths and weaknesses of the standard k-ε model have become known, modifications have been introduced to improve its performance. The turbulence kinetic energy, (k), and its rate of dissipation, (ε), is obtained from the following transport equations (1) and (2):

$$\frac{\partial}{\partial t}(\rho k)+\frac{\partial}{\partial x_i}(\rho k u_i)=\frac{\partial}{\partial x_i}\left[\left(\mu+\frac{\mu_t}{\sigma_k}\right)\frac{\partial k}{\partial x_j}\right]+G_k+G_b-\rho\varepsilon-Y_M+S_k \quad (1)$$

$$\frac{\partial}{\partial t}(\rho \varepsilon)+\frac{\partial}{\partial x_i}(\rho \varepsilon u_i)=\frac{\partial}{\partial x_i}\left[\left(\mu+\frac{\mu_t}{\sigma_\varepsilon}\right)\frac{\partial \varepsilon}{\partial x_j}\right]+C_{1\varepsilon}\frac{\varepsilon}{k}(C_b+C_{3\varepsilon}C_b)-C_{2\varepsilon}\rho\frac{\varepsilon^2}{K}+S_\varepsilon \quad (2)$$

These parameters obtain models, which is the most important parameter for accurate results. The simulation shows the profile of oil inside the casing, and no need for the casing geometry in the simulation. Figure 6 represents the results obtained from ANSYS FLUENT. In the flow analysis carried out on the geometry of the tilt bearing, the maximum pressure is about 303KPa for the standard bearing (without enhancement) and 2150 KPa for the circular-shaped groove bearing. Rectangular,



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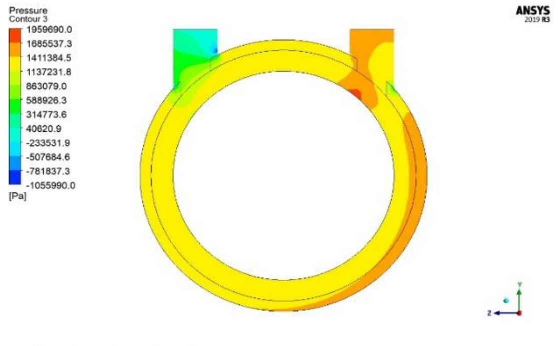

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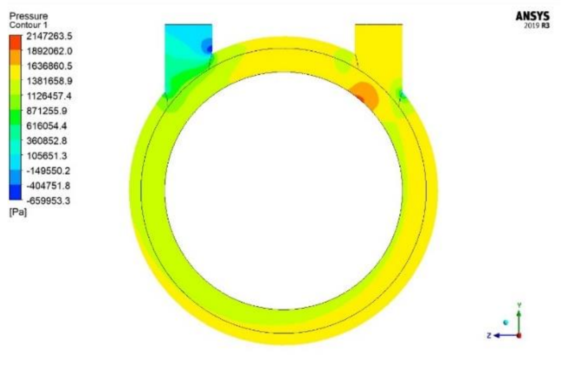

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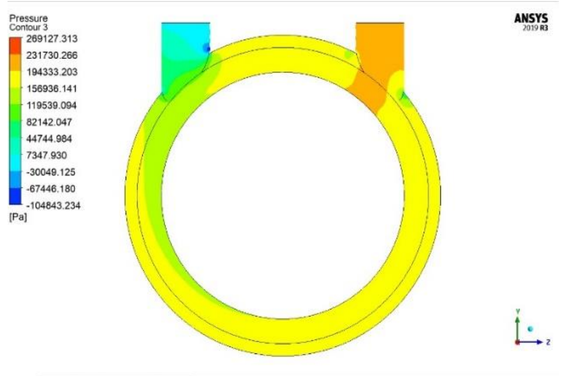

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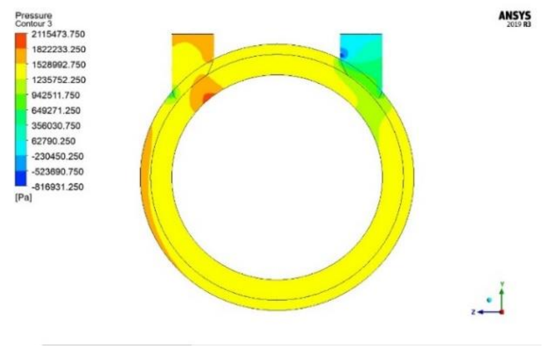


e)

Fig. 6. Pressure distribution of oil inside the bearing, a) Standard bearing shape, b) Rectangular groove, C) Circular groove, d) Inclined-side groove, e) Triple inclined-side grooves

inclined, and triple inclined have a maximum pressure of 1960 KPa, 269 KPa, and 2115 KPa, respectively; figure 7 concludes these results. The triple-inclined shape has the best pressure distribution on the shaft and bearing walls. These results transferred to the Transient structural to show the effect of these results' pressure on the vibration of the shaft.

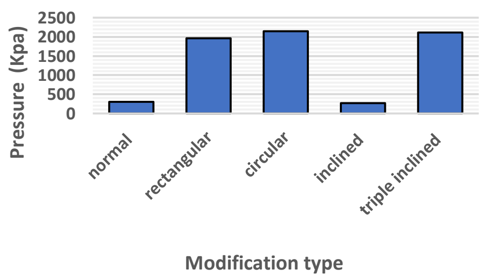


Fig. 7. The maximum pressure for each bearing shape

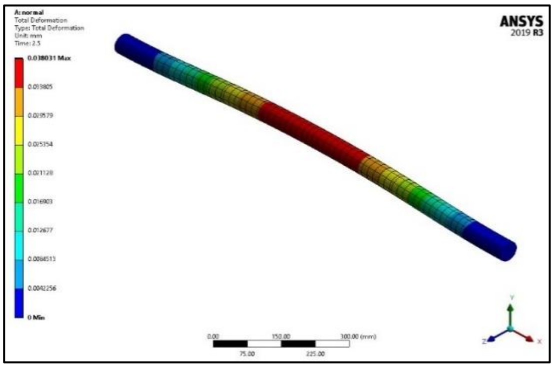
5.2. Transient Analysis of the Rotor

When the FLUENT simulation finished, the results transferred to transient structural in ANSYS, transient structural. A technique used to study a structure's dynamic reaction to any kind of time-dependent stress is transient dynamic analysis, often known as time-history analysis. When a structure is subjected to any combination of static, transient, and harmonic loads, this kind of analysis can be used to identify the time-varying displacements, strains, stresses, and forces that the structure experiences. A transient dynamic analysis's solution to the fundamental equation of motion is

$$\{F(t)\} = [M]\{\ddot{u}\} + [C]\{\dot{u}\} + [K]\{u\} \tag{3}$$

where:
[M] = mass matrix
[C] = damping matrix
$[K]$ = stiffness matrix
$\{\ddot{u}\}$ = nodal acceleration vector
$\{\dot{u}\}$= nodal velocity vector
$\{u\}$ = nodal displacement vector

At any given time, t, these equations can be seen as a set of equilibrium equations that are "static" and also account for inertia forces ($[M]\{\ddot{u}\}$) and damping forces ($[C]\{\dot{u}\}$). To resolve these equations at discrete time points, the software use either the Newmark time-integration approach or an enhanced technique known as HHT. The integration time step is the amount of time that separates subsequent time points. The five different bearings' geometry used and the best design selection depended on the Transient Structural results. Figure 8 shows the total deformation of the normal bearing (0.038mm) and triple inclined bearing (0.015mm), which was the lowest value. Furthermore, Figure 7 illustrates the amplitude of deformation in millimetres concerning change in time. Figure 9(a) shows the vibration of the shaft using normal bearings, the highest amplitude is 0.035mm, and the settling time is 1 second. Figure 9(b) shows the highest amplitude of 0.0225mm and settling time of 1.1s for the rectangular bearing shape. Figure 9(c) shows the highest deformation value at 0.02mm and the system stability in approximately 1 sec for circular-shaped bearings. Figure 9(d) is used to evaluate the shaft's dynamic reaction attached to the inclined shape bearing, the maximum deformation is 0.02mm, and the stability time is less than 0.88s. The settling time for the inclined bearing is 0.8s, and the maximum deformation is 0.015mm, as shown in figure9(e). The previous analysis shows the damping effect of lubrication's shape and how it is important to reduce the vibration of the rotating system. Unbalanced rotating discs can indeed produce continuous vibration. The rotor system in many practical applications, such as turbines, compressors, electric motors, and pumps, a heavy rotor is mounted on a light-weight, flexible shaft that is supported in bearings. There will be unsteady rotors due to manufacturing errors. These unbalance and other influences, such as the stiffness and damping of the shaft, gyroscopic effects, and fluid friction in bearings, will cause a shaft to bend in a complicated manner at certain rotational speeds, known as the whirling, whipping, or critical speeds. Whirling is described as the whirl of the plane made by the line of centres of the bearings and the bent shaft. This is because an unbalanced disc has an uneven distribution of mass, which creates a centrifugal force that acts on the disc as it rotates. This force causes the disc to vibrate, resulting in noise, instability, and potential damage to the system. Continuous vibration can occur when the unbalanced disc rotates at a constant speed. The uneven distribution of mass causes the centrifugal force to be unbalanced, which in turn causes the disc to vibrate continuously. The frequency and amplitude of the vibration will depend on the speed of rotation, the degree of imbalance, and other factors, such as the stiffness of the supporting structure. To lessen the effects of continuous vibration, it may be necessary to balance the rotating disc by adding or removing mass in specific locations. Balancing can be achieved through various methods, including trial and error, computer simulations, or specialized equipment such as a balancing machine. Once the disc is balanced correctly, the continuous vibration should be significantly reduced or eliminated [21].



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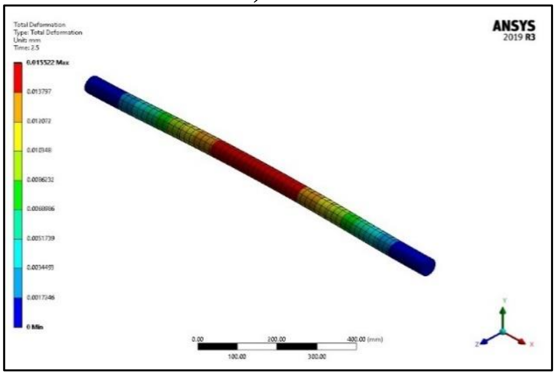
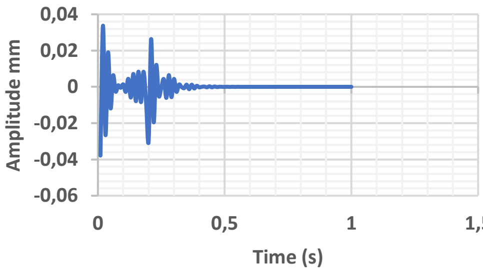


b)

Fig. 8. a) The total deformation of rotating shaft with normal bearings. b) The total deformation of rotating shaft with triple inclined bearings



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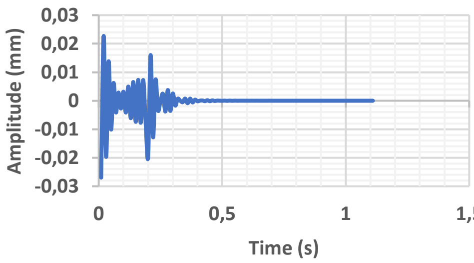

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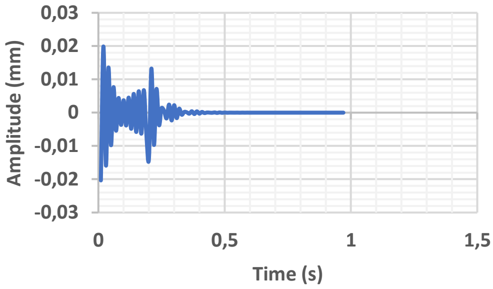

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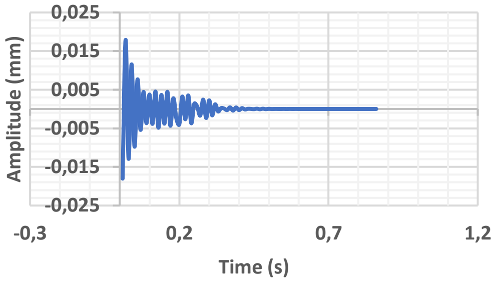

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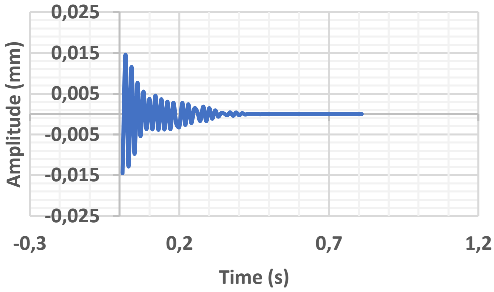

e)

Fig. 9. The directional deformation(mm) vs time(s) a) Standard bearing, b) Rectangular bearings, c) Circular bearings, d) Inclined-side bearings, e) Triple inclined-side bearings

5. CONCLUSION

In this study, dynamic analysis has been performed for the rotor before and after the development of the bearings by using FEM software (ANSYS 19.3) to compare the results in the five cases. In the FLUENT analysis, the pressure distribution of oil on the shaft affects the value of the deformation of the shaft. The increasing flow for any shaped bearing is better than the flat internal surface of the bearing (standard bearing). The presence of sharp edges, as in circular and rectangular channels, in the flow path negatively affects the damping value due to the formation of turbulent oil flow around these edges. The vibration's amplitude in a rectangular groove is 0.022mm, and in a circular shape is (about 0.022mm); this enhancement appears because the circular has fewer edges than the rectangular one. The open channel groove has two 45° inclined sides, which reduces the turbulent flow and makes it more linear. The vibration amplitude improved in the open channel groove to record 0.018mm at the highest value. Increasing the number of channels gives better results when reducing vibration amplitude to 0.015mm for the three open-channel shapes. The different bearing shapes did not improve the amplitude of vibration only but the stability time too. In the first shape (standard shape), the settling time is 1s, while the settling time for the circular one is 0.88s. The dynamic shaft stability for the inclined groove and triple inclined is 0.825s and 0.8, respectively. Due to the turbulence effect, the odd result can be seen only in the rectangular shape (1.15s). From the previous results, the system excitation value decreased by 40.11% at the triple-grooves improvement. The settling time was enhanced by 20%. Therefore, increasing the number of inclined grooves decreases the vibration amplitude and settling time.

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Declaration of competing interest: *The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.*

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