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CFD Analysis of the Eccentricity Ratio on Journal Bearing due to Differences in Lubrication Type

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***Abstract*—Lubrication on journal bearings is vital in reducing metal-to-metal friction and separating components. Often there is a decrease in lubrication conditions, which will cause failure or a change in roughness to the inner surface. Direct metal-to-metal contact between the bearing's inner surface will gradually deform the bearing. In diesel engine applications, lubrication aims to minimize failure or damage to the journal bearing. This paper investigates a 3D CAD model of a journal bearing using the ANSYS Computational Fluid Dynamics software. The effect of the eccentricity ratio is analyzed for semi-synthetic oil R3 10W-30 and synthetic oil. The result shows that synthetic oil has a considerable pressure compared to R3 10W-30, especially at a higher eccentricity ratio.**

***Index Terms*—ANSYS, CFD, eccentricity ratio, journal bearings, lubrication**

I. INTRODUCTION

THE science of mechanics and dynamics is very important in machining construction. Generally, some problems often occur due to the rotation of the engine in large torque, high loads, and high rotational speeds that can cause damage. It is very important to diagnose the damage carefully, for example, in area applications at high precision, such as gas turbines, electric generators, steam turbines, pumps, automobiles, and diesel engines. Journal bearings are one of the engine elements that are very influential on rotary machines because they can support high loads and speeds. Journal is described as a cylinder bearing that envelops a rotating shaft. Journal bearings, called hydrodynamic rotor bearings, are used to receive or support radial loads or simply as an introduction to the smooth transmission of torque. *Journal bearings* work only under limited conditions (metal-to-metal contact) during engine start-up and shutdown. Because there is often contact between the two, the lubrication must be good to minimize the occurrence of friction between the journal and the bearing, which results in damage to the journal bearing[1]–[4]

Figure 1 shows the basic geometry of the journal bearing in *a steady-state* configuration. The hydrodynamic movement of the journal bearing causes pressure in the lubrication. In the journal bearing lubrication, there is a thin oil film to neutralize the load, which separates the inner surface of the bearing and the surface of the journal. When the steady state condition is reached, the journal will move from the bearing with a central distance *e,* called eccentricity. *Journal bearings* have an eccentricity ratio value ($\boldsymbol{\varepsilon}$), i.e., $\boldsymbol{\varepsilon}. = e / c$, this equation becomes an essential measure of the load capacity of the bearing and measure of the thickness of the lubricating film that authorizes the journal and bearings[5–7].

In recent years, previous research has discussed fluid analysis and geometric variations either by simulation methods, experimental or both in overcoming damage or failure in bearings. Computational Fluid Dynamics (CFD) is a modeling technique widely used to simulate and evaluate based on fluids that work against components, such as hydrodynamic journal bearings, by matching actual conditions without the need to test first [8–11]. Previous research analyzed the influence of the ratio of eccentricity value to pressure distribution with a wide range of software such as ANSYS [12], COMSOL [13], MATLAB [14], and Fortran [6]. Lubrication fluids used are generally types of SAE, with various series, one of which is 10W-30 for the automotive industry, along with the development of SAE products 10W-30 or 40 that can be used in rotating machinery [15]. The results of simulations and experiments explain the failure mechanism comprehensively since the decreased lubrication conditions cause a decrease in bearing performance [16]. Full synthetic type lubrication has the same primary material as other types of lubricants, namely petroleum preparations. But the manufacturing process is carried out more perfectly because the basic material goes through processing again and selects the best through long stages. This oil is better than semi-synthetic mineral oil because the oxidation stability or resistance is better. Besides that, it is better to be stable when the temperature is high and there is protection against damage to synthetic oil. In that case, it will affect the machine's performance [17], [18].

In some studies, looking for an efficient oil to reduce the wear of contact parts is to improve the lubrication performance of the oil by surface and texture modification [19], [20] or by adding additives using colloidal suspension nanofluids from nanoparticles (NP) with an oil base [21–25]. Over the past three years, nanoparticles have been studied extensively as additives to lubrication. They have been used to reduce the coefficient of friction between moving parts and consequently lower wear with different concentration levels of addition [26].

There is often a decrease in lubrication conditions, which will cause failures or changes in roughness to the surface in the bearing, which usually occurs in direct contact between metal-to-metal. Therefore, a study was conducted comparing the type of semi-synthetic oil R3 10W-30 with synthetic oil to find better lubrication for journal bearings in the application of diesel engines, aiming to minimize failure or damage to the journal bearings. In solving this problem, the method used for simulation is the CFD Fluent method on ANSYS software.

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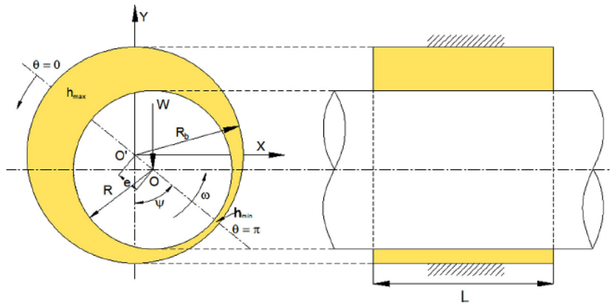
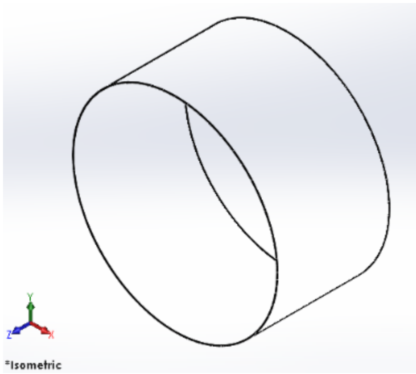


Fig. 1. Basic geometry of journal bearings [27].

II. METHODS

The sliding bearing simulation model, referred to as the journal bearing studied in this study, models the bearing oil film by varying the eccentricity ratio, which will affect the deformation of the bearing. In addition to varying the eccentricity ratio, it compares shell R3 type lubrication R3 10W-30 with synthetic oil. The research method used is to use CFD Fluent in ANSYS.



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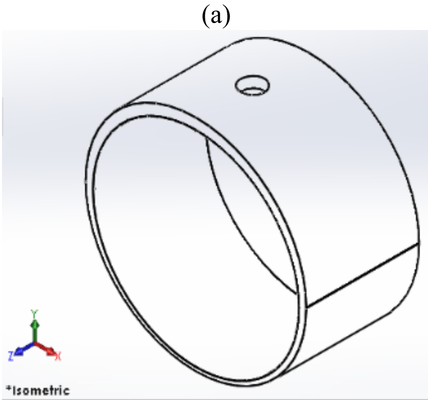
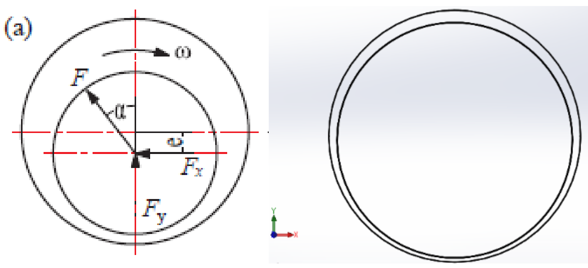


(b)

Fig. 2. Modeling journal bearings (a) film oil (b) bearings.



(a) (b)

Fig. 3. Scheme load carrying capacity of oil film (a) model attitude angle equal to zero (b) modeling journal bearings.

The geometry parameters refer to the previous study in table 1, using secondary data from the previous study. The material Babbit alloy uses in bearings [16].

TABLE I
GEOMETRY PARAMETERS [16]

Feature	Value	Parameters	Value
Journal diameter [mm]	52.692	The density of bearing alloy [kg/m3]	7310
Bearing bore diameter [mm]	52.764	Young's modulus	55
The total wall thickness of bearing [mm]	1.825	Poisson's ratio of bearing alloy	0.3
Bearing outside diameter [mm]	56.414	Tensile yield strength [MPa]	66.4
Width of bearing [mm]	28.86	Tensile ultimate strength [MPa]	85.5
		Compressive ultimate strength [MPa]	96.3
Bearing alloy thickness [mm]	0.25		
Oil inlet diameter [mm]	6		

The simulation method on CFDs uses Fluent in Ansys Workbench 19 R3, with the *hexahedral meshing* of defined boundary conditions. The properties of shell oil R3 10W-30 are shown in table 2 and synthetic oil in table 3, which during fluid simulation, lubricating oil was considered *incompressible* in the study. Viscosity models are established as laminar flows. The oil input inlet hole is assumed to be the front of the oil film model, and its outlet is at the back of the oil film model. The inlet pressure was 0.6 MPa, and the exit pressure was assumed to be atmospheric. The inner surface of the oil film is set as a "moving wall" with a rotational speed equal to the journal speed of 3000 rpm, and the outer surface of the oil film is designated as a "stationary wall" A no-slip condition is imposed; on both walls. *The initial vapor* fraction of the volume in the stream at the inlet and outlet boundaries is set as zero. A mixed model is used to continue processing to a two-phase flow. The mass transfer mechanism is regulated as cavitation from the liquid phase to the vapor phase. The cavitation model is set as Zwart-Gerber-Belamri, and the vaporization pressure is set as 0 Pa. In the study, the Shell R3 10W-30 lubricating oil temperature was 70°C, and the corresponding dynamic axle viscosity was 0. 022314 Pa.s. In this study, there are several variations of the variables used shown in Table 2.

TABLE II
PROPERTIES OF SEMISYNTHETIC OIL R3 10W-30[16]

Feature	Oil	Oil Vapor
Density [kg/m^3]	881	1.225
Kinematic viscosity [cSt]	75.1(40)°C 11.5(100)°C	-
Dynamic viscosity [Pa.s]	6.7(-25)°C	1.79.10-5
Cavitation pressure [Pa]	0	-
Pour point []°C	-36	
Flash point []°C	220	

TABLE III
PROPERTIES OF SYNTHETIC OILS [28]

Feature	Value
Density [kg/m^3]	823.2
Viscosity [Pa.s]	0.025631(70)℃

TABLE IV
VARIABLE SIMULATION

Types of Lubricants	Eccentricity Ratio, [mm]ε	Eccentricity, [mm]e	*Radial Clearance* (C)
Shell R3 10W-30	0.1	0.0036	0.036
	0.3	0.0108	
	0.5	0.018	
	0.7	0.0252	
	0.9	0.0324	
Oil synthetic	0.1	0.0036	
	0.3	0.0108	
	0.5	0.0180	
	0.7	0.0252	
	0.9	0.0324	

III. RESULT AND DISCUSSION

This research uses CFD simulations. In the process of several simulations, results have been obtained at the value of the eccentricity ratio of 0.9 with a rotation speed of 3000 rpm. It can be seen that it is very influential on the pressure of *the oil film*; the greater the eccentricity value of the ratio, the more the pressure increases because the oil film is also thinner so that the oil film is sandwiched between two fields, namely the journal and the bearing, from the pinching of the oil film, the opportunity for the journal to move will be large from the distance of the center, while if the ratio of eccentricity is getting smaller the value, then the contour of the oil *film* pressure is not very significant the change is seen in figure 4. the pressure on the oil film. When the ratio value is small, it is seen at 0.1, and it tends to spread the pressure because the chances of the journal moving from its center distance are smaller. For example, the maximum pressure value of the oil film for R3 10W-30 journal bearings at an eccentricity ratio of 0.1 is 0.600671 MPa. In comparison, at an eccentricity ratio of 0.9, it is a sharp increase of 33,8125 MPa. The maximum pressure for synthetic oil with a ratio of 0.1 is 0.605637 MPa, while at an eccentricity ratio of 0.9, it increased sharply by 38.8075 MPa.

From the explanation above, the effect of pressure on bearing deformation is shown in Fig. 5 and Fig. 7. The stress distribution on the surface of the bearing bush produced because there is a load from the oil film layer is basically in harmony with the pressure of the oil film layer. Surface tension increases gradually as the constellations or eccentricity in the area increase. The visible visualization contours for oil film pressure and bearing deformation are centered on the inner surface of the bearing when the eccentricity ratio is 0.9 so that it decreases sharply after reaching the maximum value. It can be said that the oil film pressure and total deformation work linearly. The greater the value of the eccentricity ratio, the more the oil film pressure value and the bearing statement's deformation value. According to previous studies [16], [29] can cause failure or catastrophe and reduce engine efficiency.

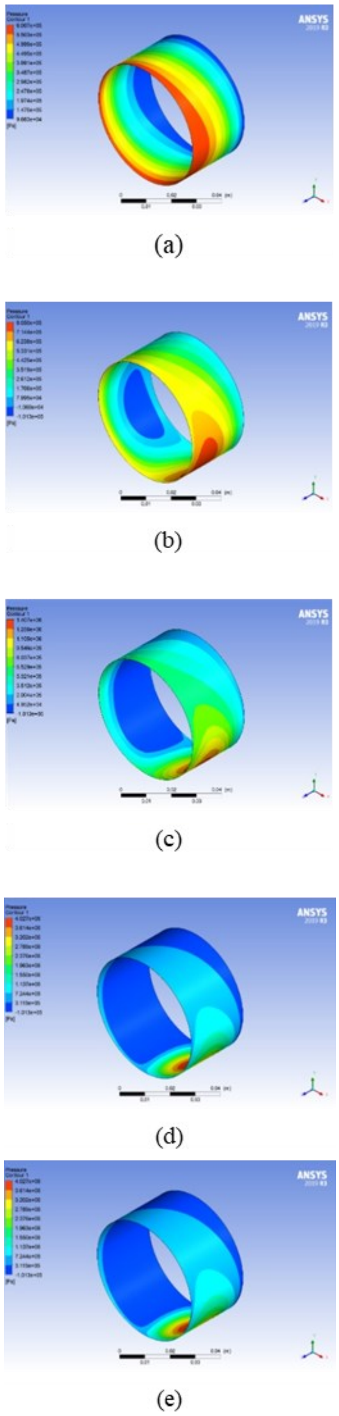
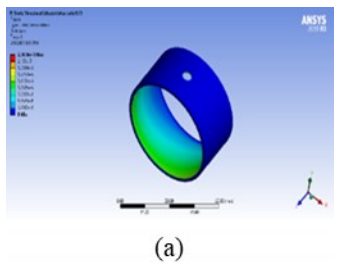
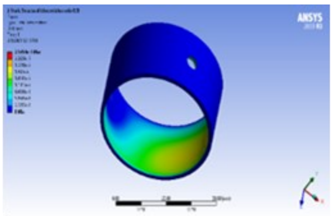


Fig. 4. Simulation results on Shell R310W-30 oil film pressure ε, (a) 0.1 (b) 0.3 (c) 0.5 (d) 0.7 (e) 0.9.





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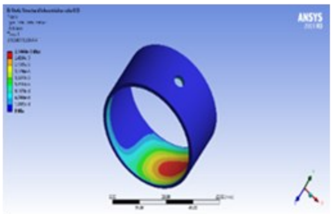

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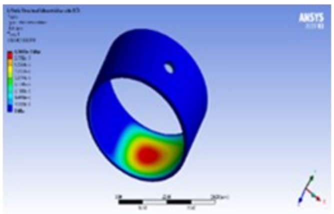

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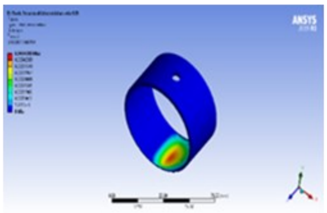
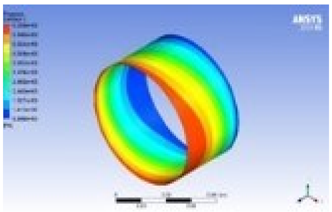


(e)

Fig. 5. Total deformation on bearings on Shell R3 oil 10W-30 ε (a) 0.1 (b) 0.3 (c) 0.5 (d) 0.7 (e) 0.9.



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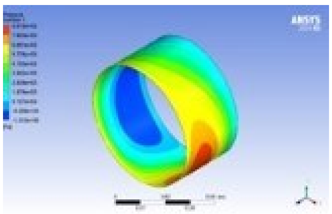

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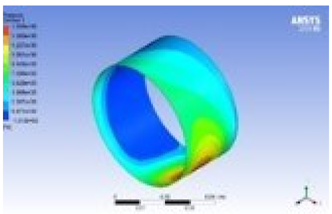

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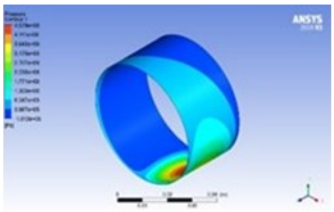

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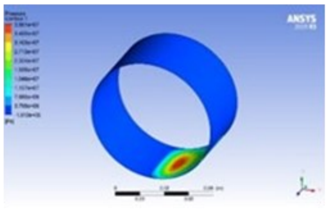

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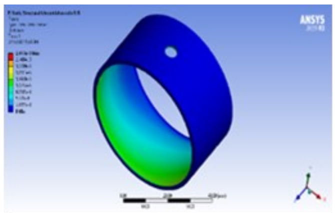
Fig. 6. Simulation results on synthetic oil film pressure ***ε***, (a) 0.1 (b) 0.3 (c) 0.5 (d) 0.7 (e) 0.9.



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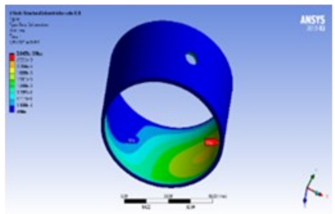

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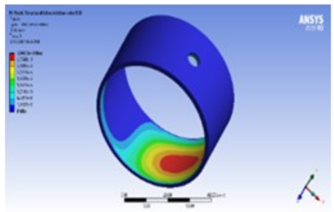

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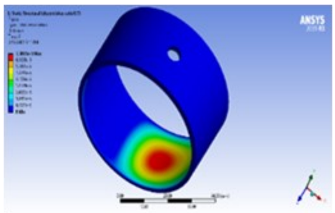

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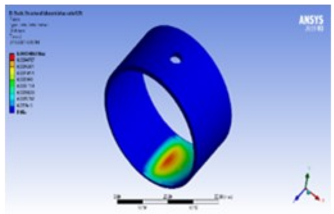

(e)

Fig. 7. Total deformation in bearings on synthetic oil *ε* (a) 0.1 (b) 0.3 (c) 0.5 (d) 0.7 (e) 0.9.

A comparison graph was obtained from the simulation results that represents the ratio of eccentricity to maximum pressure in the journal bearing. It is noticed that synthetic oils are higher than R3 10W-30 because their viscosity value is greater than R3 10W-30 oil. The viscosity value is closely related to pressure. The greater the viscosity, the higher the maximum pressure value produced, and it will affect the lubrication system. The higher the viscosity, the more minor the wear and friction due to its viscosity.

This comparison chart shows that the values ε of 0.1 and 0.3 are not very significant differences, but 0.5 towards 0.7 and 0.9 looks like the line is increasing. The value ε of 0.1 pressure obtained by lubricant type R3 10W-30 is 0. 600671 MPa, while synthetic oil has a maximum pressure value of 0.605637 MPa with an attitude angle equal to zero for all variations. When $\varepsilon\varepsilon$ 0.5 goes towards 0.7 and 0.9, it looks very significant. The Pressure difference is 0.9 at R3 10W-30. Its maximum pressure value is 33.8125 MPa, while synthetic oil its maximum pressure value is 38.8075 MPa at a rotational speed of 3000 rpm. So, this shows the viscosity value is very important in the lubrication system if you want to reduce friction resulting in wear on two or more surfaces.

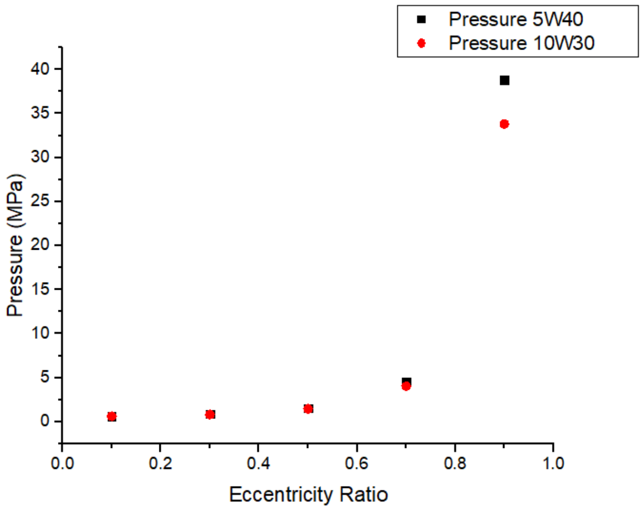

Fig. 8. Comparison chart of the ratio of eccentricity with maximum pressure.

III. Conclusion

The conclusion obtained from this study is that the eccentricity ratio value greatly affects the pressure of the oil film. The greater the value of the eccentricity ratio, the thinner the oil layer due to the smaller the clearance, which will cause the bearing to deform. It was also seen that the difference in the type of lubricant also greatly influenced the results obtained on pressure and deformation. The maximum pressure value increases, causing the deformation value of the bearing to also increase due to the working load and different viscosity values, the viscosity value is closely related to the pressure in the lubrication system, so it is concluded that the greater the viscosity value, the more the pressure value increases. In other words, the greater the viscosity value, the less friction occurs. In this case, synthetic oil is superior to R3 10W-30 semi-synthetic applications where the load is heavy, but the drawback is that the price of synthetic oil is higher than that of R3 10W-30 semi-synthetic oil.

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