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Design and Fabrication of a Journal Bearing Test Rig for Pressure and Temperature Variation Evaluation

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Abstract

Journal bearings are certainly considered one among the important shafting elements used in the industries. Therefore, a prototype test rig was designed to facilitate experimental studies for journal bearing pressure variation, and temperature variation. The test rig consisted of oil tank/pan, manometer, thermometer, frame, base plate, electric motor, drive shaft, journal bearing assembly, etc. Several test trials were conducted, with the goal of verifying that the theoretical mathematical outcomes derived were in complete accordance with the experimental results of pressure variation. Also, analysis was carried out on the rotational speed, radial bearing load, pressure variation, temperature variation due to heat generated, heat dissipated, and amount of artificial cooling. The results obtain showed that average experimental pressure 1045.29 N/m^2 was found to be harmonious with theoretical pressure 1045.25 N/m^2. Besides, an average 13.165 watts of heat was dissipated from the test rig, thus, a minimal amount of artificial cooling (26.926 watts) is required. It was equally observed that the journal bearing speed is directly proportional to the generated pressure. Similarly, as the speed increased, the generated pressure and operating temperature of the journal bearing increases as well.

1. Introduction

Journal bearings also known as plain bearings are one of the important shafting elements used in the industries especially in the ship shafting propulsion systems [1]. The design of journal bearings test rig is considered important to avoid boundary lubrication problems [2]. Also, a properly designed journal bearings can operate efficiently for a very long time with minimum maintenance. Nevertheless, when a journal bearing suffers damages or failures, the consequences can be disastrous and this can lead to the damage of the machine and frequent downtime during plant operations [3]. More so, after a period of operation, especially under harsh conditions, journal bearings often wear due to damage such as loss of babbitt metal due to the effect of frictional force [4]. Such damages increases significantly especially if system balance is not achieved during operation of the bearing. For instance, it is expected that as heat is generated during operation of the bearing, such generated heat should equally be dissipated to obtain zero or minimum amount of artificial cooling mainly to avoid loss of load capacity, significantly reduction of the oil film thickness because of the decrease in the active surface area. Also, experimental determination of oil film pressure has been a difficult or even infeasible task [6]. It was mainly studied by mathematical means. In addition, under real-world operating conditions, there are usually many practical aspects that make it difficult to experimentally determine the true oil film pressure at a particular point in time. Such as extremely thin or thick oil film which causes defect in geometry or dynamic variability in the bearing.

The journal bearing test rig is an apparatus for investigating the distribution of pressure in the sliding contact bearings, and the load carrying capacity at different bearing loads, operating temperature and speed [5]. Dynamically, loaded journal bearing designs need to be cost effective in terms of time and effort to avoid bearing failure by maintaining bearing performance at optimal levels. Journal bearings should function properly with a sufficient thickness of lubricating film. In addition, the operating temperature rise should be small enough to prevent the bearings from melting or seizing. It is equally important that the quantity of lubricant flow should be sufficient to check the rise in the operating temperature, while avoiding excessive churning losses [6-7]. The operating characteristics of journal bearings can be divided into static state and dynamic states [8]. Static characteristics include bearing load capacity, lining temperature, power loss, and the amount of oil required during operation. Bearing load capacity is often measured either by eccentricity, which is directly related to minimum film thickness, or by maximum pad temperature [9]. The dynamic performance of a bearing is usually characterized by its stiffness and damping characteristics. However, these properties interact with the rotor system determines a machine's overall vibrational behaviour [10].

Furthermore, journal bearing designers usually try to select the design variables within restrictions by a trial-and-error technique using bearing design charts obtained from the bearing characteristic analysis. Although, this approach only guarantees acceptable solutions, but it does not necessarily produce the optimum solutions. Besides, even in the case that the bearing designers manage to get the optimum solutions successfully by such a method, a significant amount of working time and cost will be needed to complete the optimum design of high-speed journal bearings [11-12]. Thus, evaluating useful parameters of journal bearings such pressure, operating temperature, speed, and how they affect journal bearings when in use for optimum performance is necessary and this can best be achieved if a prototype machine is produced. Besides, the development of the journal bearing test rig in the past faces several limitations in its design, such as vibration, lack of enough clearance in shafting fitting, pressure not attaining steady state. Therefore, as a result of the aforementioned limitation [12], it is necessary to develop an improve journal bearing test rig that can establish a better relationship between applied load, speed, operating pressure, coefficient of friction, heat generated, heat dissipated, and amount of artificial cooling require for optimum journal bearing performance.

2. Materials and Method

Design considerations were selected and this was mainly to identify some problems which could hindered effective performance. As such, effort was geared to identify these factors and constraints as put together; functionality, reliability, manufacture, durability, cost, and safety. Two conceptual designs were drawn. First design concept consists of an electric motor, journal shaft, bearing sleeve, oil tank, weight hanger, plumber block, frame etc. The power required to drive the shaft inserted in the bearing sleeve is given by the electric motor. To aid a smooth operation, a lubricant would be supplied through oil tube connected to the bearing from the oil tank (reservoir), which also creates a pressure head at required height. Due to rotation of shaft and load on the sleeve, a load bearing pressure is created. Also, in this region of load bearing pressure, a certain amount lubricant is pumped out through the holes provided through the sleeve. The difference inlet height and outlet pressure heads help in calculating the pressure at the specific point on the sleeve. Figure 1 shows the isometric view of design concept one.

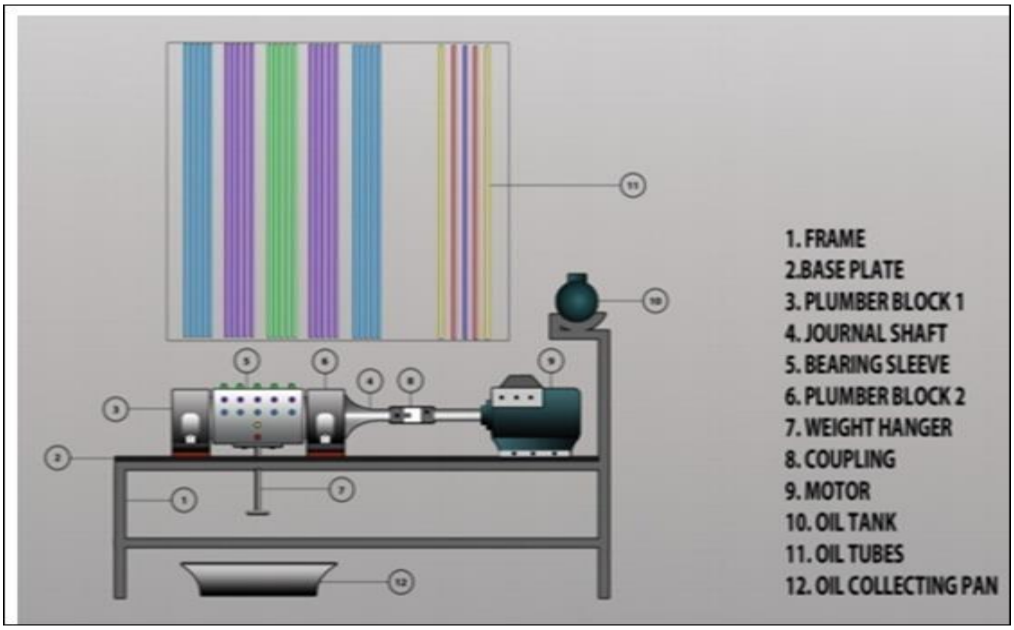
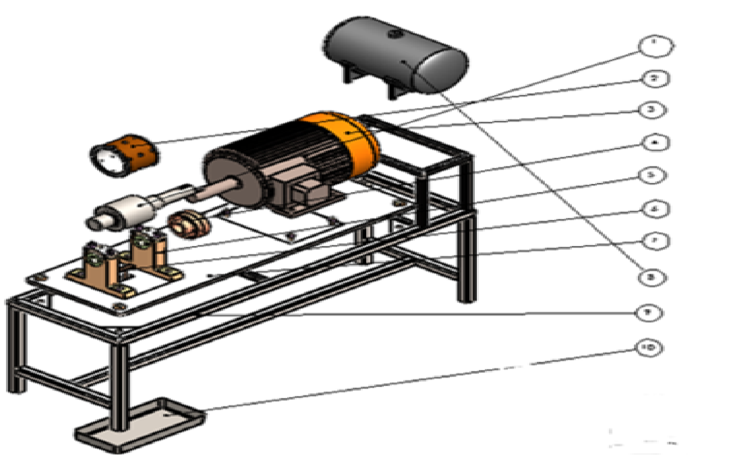


Figure 1. Isometric view of first design concept

Second design concept uses an electric motor, journal shaft, oil tank, manometer bored holes, journal bearing, coupling, etc. The entire system will be operated by the power input from the motor and torque will be transmitted into the journal shaft through the mounting couplings. Oil from the oil tank is drawn into the journal sleeve through the input holes which are 24 in numbers for the measurement of pressures and temperature variation through the help of sensors incorporated. The load carrying capacity of the journal sleeve can also be calculated as there is room for the load variation on the journal. Figure 2 shows the isometric view of design concept one. To ensure the best concept was selected for detail design, a decision matrix was drawn based on the following design considerations; cost, safety, functionality, and maintenance. Table 1 shows the decision matrixes used for selecting the best conceptual design. Based on the ranking, the second design concept with the highest ranking of 76.1 was selected for detail design.



S/N	Part Description	Quantity
1	Journal Shaft	1
2	Journal Sleeve	1
3	Electric Motor	1
4	Coupler	2
5	Plumber Block	2
6	Weight Holder	1
7	Base Plate	1
8	Oil Tank	1
9	Frame	1
10	Oil Pan	1

Figure 2. Isometric view of second design concept

Table 1: Decision matrix for selection of the best conceptual design

	First Design Concept		Second Design Concept	
Criteria	**Rate**	**Score (1)**	**Rate**	**Score (2)**

	Weight				
Functionality	3.5	7	24.5	8	28
Safety	2.8	6	16.8	9	25.2
Cost	2.2	7	15.4	7	15.4
Manufacturability	1.5	8	12.0	7	10.5
Total			68.7		76.1

*Weight Factors from 1 – 4 and Rating 1 – 10 *Score=Rating x Weight

The exploded view of the machine is shown in Figure 3.

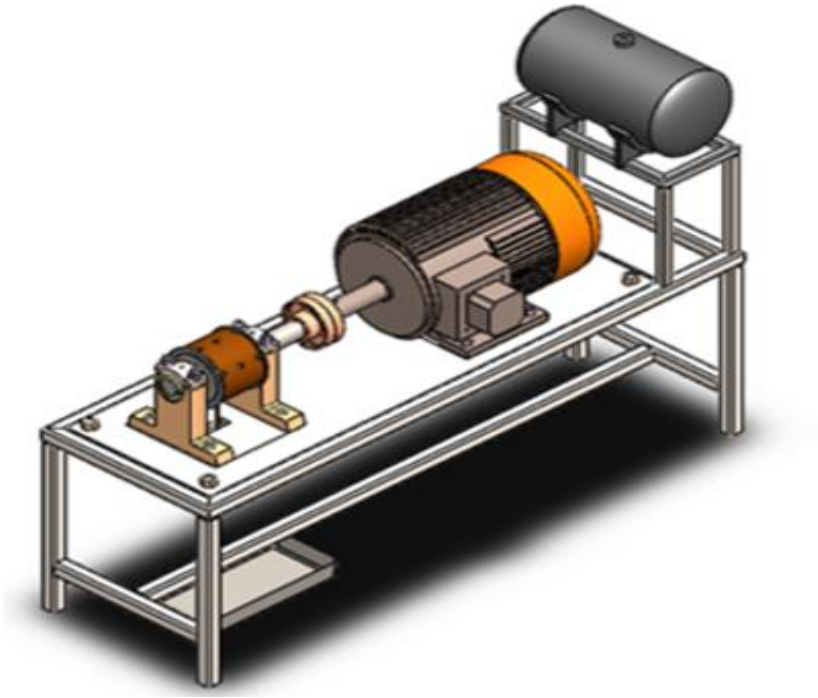


Figure 3. Isometric view of the journal bearing test rig

Based on the aforementioned design considerations that led to the selection of the best conceptual design; the following design specifications were drawn:

i. 1hp will be required for the operation of the journal bearing test rig

ii. A minimum force not greater 15 N will be for the operation of the journal bearing test rig

iii. A minimum torque not greater than 3 Nm will required for the operation of the journal bearing test rig

iv. A portable machine frame of area of not more than 0.6 m^2.

The dimensioned view of the machine is shown in Figure 3.7. The machine frame has a composite shape of a rectangle. Therefore, the area of the machine frame was taken as area of a composite rectangle and it is given by Equation (1).

$$A_{mf} = A_1 + A_2 \tag{1}$$

The volume of journal sleeve is calculated as follow;

$$V_i = \pi(R^2 - r^2)h \tag{2}$$

The weight of the journal bearing is given by Equation (3),

$$W = m \times g \tag{3}$$

Also,

$$m = \rho V \tag{4}$$

The torque (τ) was calculated using Equation (5)

$$P = \tau\omega \tag{5}$$

Figure 4 shows the dimensioned view of the shaft.

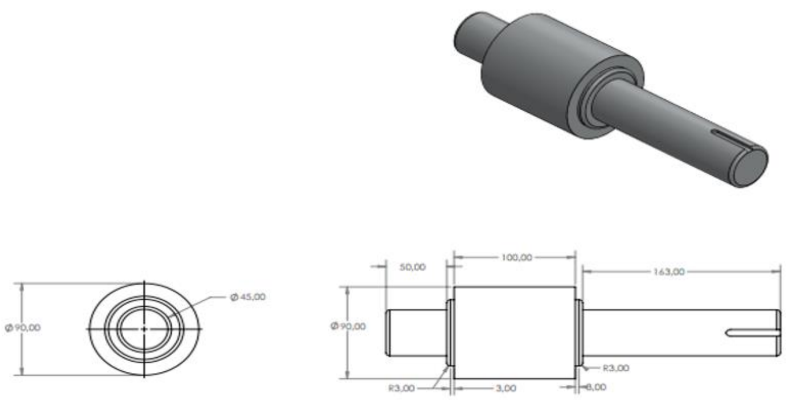


Figure 4. Dimensioned view of the bearing test rig shaft

Let;

τ = Shear stress induced due to twisting moment, and

σ_b = Bending stress (tensile or compressive) induced due to bending moment.

According to maximum shear stress theory, the maximum shear stress in the shaft,

$$\tau_{max} = \frac{1}{2}\sqrt{(\sigma_b)^2 + 4\tau^2} \tag{6}$$

But,

$$\sigma_b = \frac{32M}{\pi d^3} \tag{7}$$

$$\tau = \frac{16T}{\pi d^3} \quad (8)$$

Substituting Equation (8) and (7) into Equation (6)

$$\tau_{max} = \frac{1}{2}\sqrt{(\frac{32M}{\pi d^3})^2 + 4(\frac{16T}{\pi d^3})} \quad (9)$$

Thus,

$$\tau_{max} = \frac{\pi d}{16}[\sqrt{M^2 + T^2}] \quad (10)$$

Also,

$$\frac{\pi}{16} \times \tau_{max} \times d^3 = [\sqrt{M^2 + T^2}] \quad (11)$$

$$T_e = [\sqrt{M^2 + T^2}] \quad (12)$$

The expression $\sqrt{M^2 + T^2}$ is known as equivalent twisting moment and is denoted by Te. The equivalent twisting moment may be defined as that twisting moment, which when acting alone, produces the same shear stress (τ) as the actual twisting moment. Figure 5 shows the picture of the fabricated journal bearing test rig machine

Figure 5. Picture of the fabricated journal bearing test rig machine

3. Results and Discussion

The pressure distribution, at different rotating speeds, around the journal bearing was measured by the aid of manometer, installed all around the circumference of the bearing. The bearing pressure, coefficient of friction, heat generated, heat dissipated and amount of artificial cooling was determined using Equation (13) to equation (16). On carrying out an experimental study on the journal bearing using the test rig under consideration, it is important to check the consistency and validation of its derived results. Thus, the pressure distribution, at different rotating speeds, around the journal bearing, was measured by the aid of manometer and compared to the theoretical result of bearing pressure determined.

$$P = \frac{F}{A} \tag{13}$$

$$H_g = fW_r \frac{\pi DN}{60} \tag{14}$$

$$f = 0.326\left(\frac{\mu N}{P}\right)\left(\frac{D}{C}\right) + k \tag{15}$$

$$H_d = \frac{(\Delta T+18)^2}{K} LD \tag{16}$$

$\Delta T = T_B - T_A = \frac{1}{2}(T_0 - T_A)$ = The difference between the bearing surface temperature T_B and the temperature of the surrounding air T_A

$\frac{\mu N}{P} = 30 \times 10^{-6}$

g = 9.81 m/s^2

$\frac{D}{C} = 1000$

Figure 6 shows the results of comparative analysis of theoretical pressure with experimental pressure at different journal bearing speed. The outcome revealed that both pressures obtained are approximately equal. Thus, there is a good agreement between experimental and theoretical pressure results and this is an indication that the designed and fabricated journal bearing test performance was satisfactory and can be used in College of Engineering and other higher institution laboratory as a training apparatus. More so, it can also be used for commercial purpose. It was equally observed that the journal bearing speed is directly proportional to the generated pressure. Similarly, as the speed increased, the generated pressure increases as well. According to [13], an increase in speed generally increases the temperature of bearing thereby increases the pressure. Furthermore, the findings from this research work also agreed with the research work of [14], that reported that temperature, coefficient of friction increases with increasing load and rotational speed.

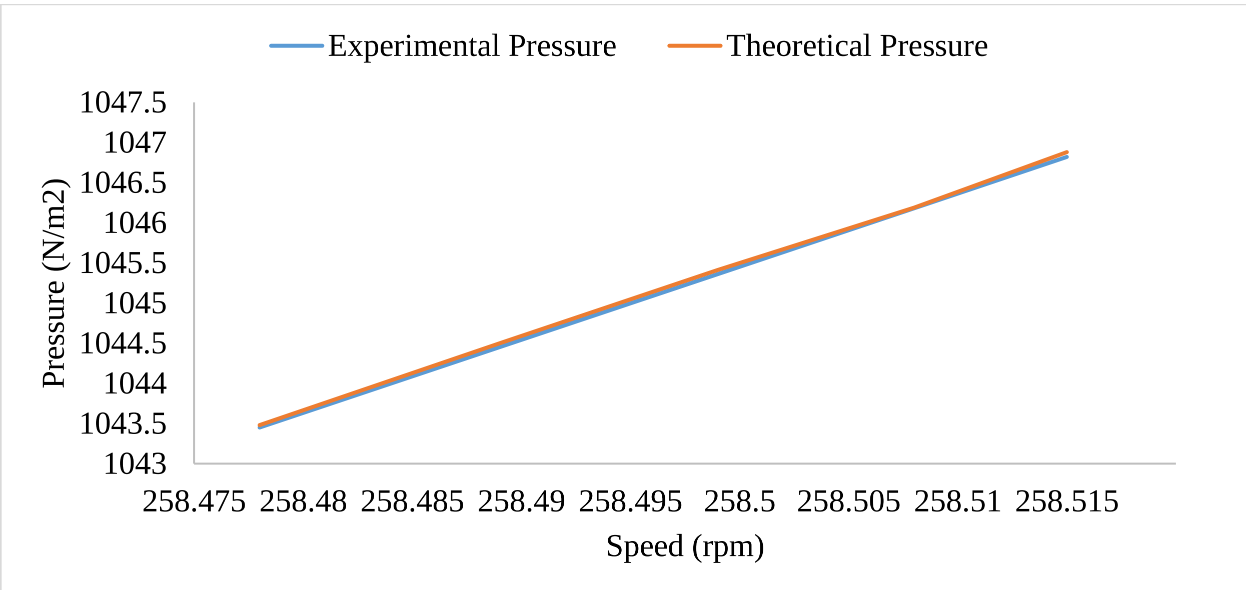


Figure 6. Graph of journal bearing speed against pressure generated

Figure 7 shows the performance evaluation of heat produced as a result of variation in the bearing radial load, operating and ambient temperature, and the applied speed. It was observed that as speed increases, the heat generated, heat dissipated, and amount of artificial cooling increase as well. The heat dissipated, which is the amount of heat that is removed from the system is a critical parameter that requires continuous monitoring. More so, zero heat dissipation or a drop in rate of heat dissipation as against the rate of the heat generation will leads to rise in the bearing operating temperatures, thereby increasing the hardness of bearing ring. As such, the sliding element decrease, thus, resulting in a plastic deformation, lubricant deterioration, heat imbalance failure, and eventual breakage of the bearing components. Also, according to [15], thermal changes significantly affect the bearing performance as lubricant viscosity is a strong function of temperature. Moreover, excessive rise in temperature can cause oxidation of the lubricant and, consequently, leads to failure of the bearing. However, as depicted in Figure 7, sufficient heat is dissipated from the bearing test rig, thus minimal amount of artificial cooling required. This again is an indication of a better design and thus, confirming the functionality and reliability of the designed and fabricated test rig system.

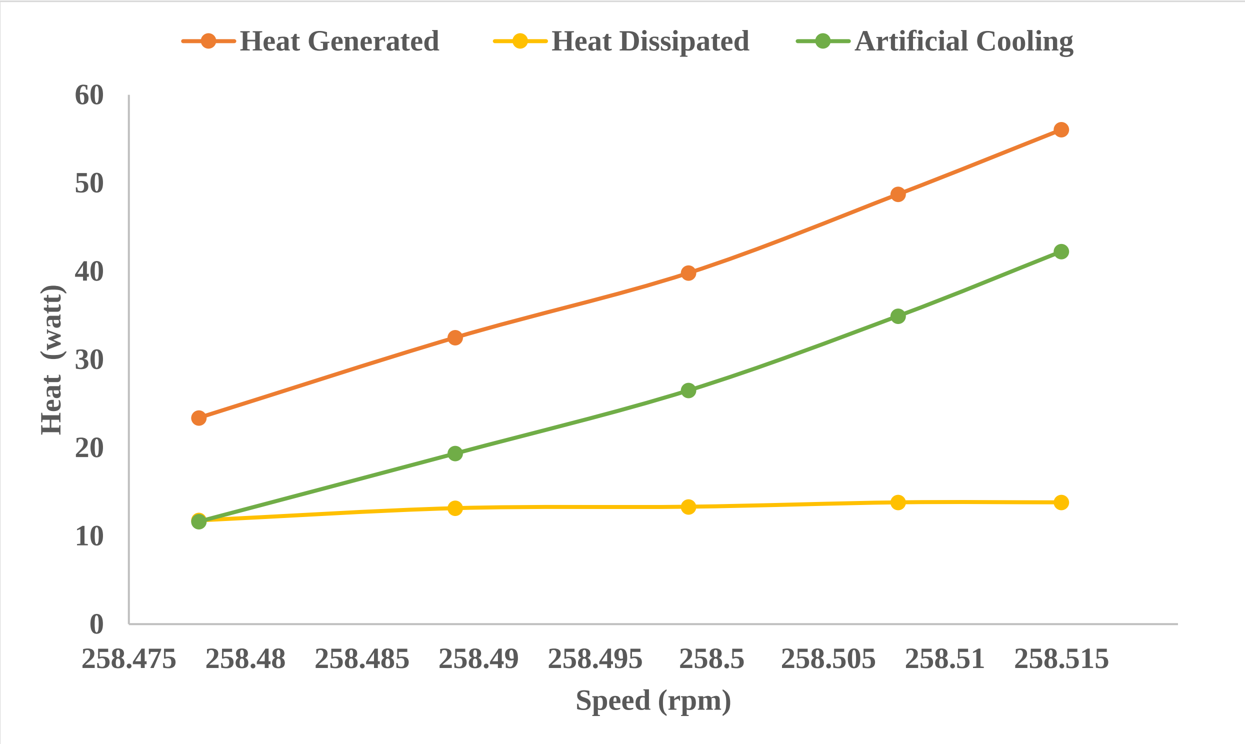


Figure 7. Graph of speed against heat values

The theoretical pressure was calculated using Equation (13) and the experimental pressure distribution measured with the aid of manometer, installed all around the circumference of the fabricated journal bearing test rig. Five different tests were carried out using the measured journal bearing parameters such as diameter, length and the weight of the journal. The calculated theoretical values and experimental pressure distribution readings at each test were measured and the average values calculated. Figure 8 shows the comparative analysis of the average values of theoretical and experimental pressure. The average value of results of experimental pressure were found to be approximately the same with theoretical pressure as depicted in Figure 8.

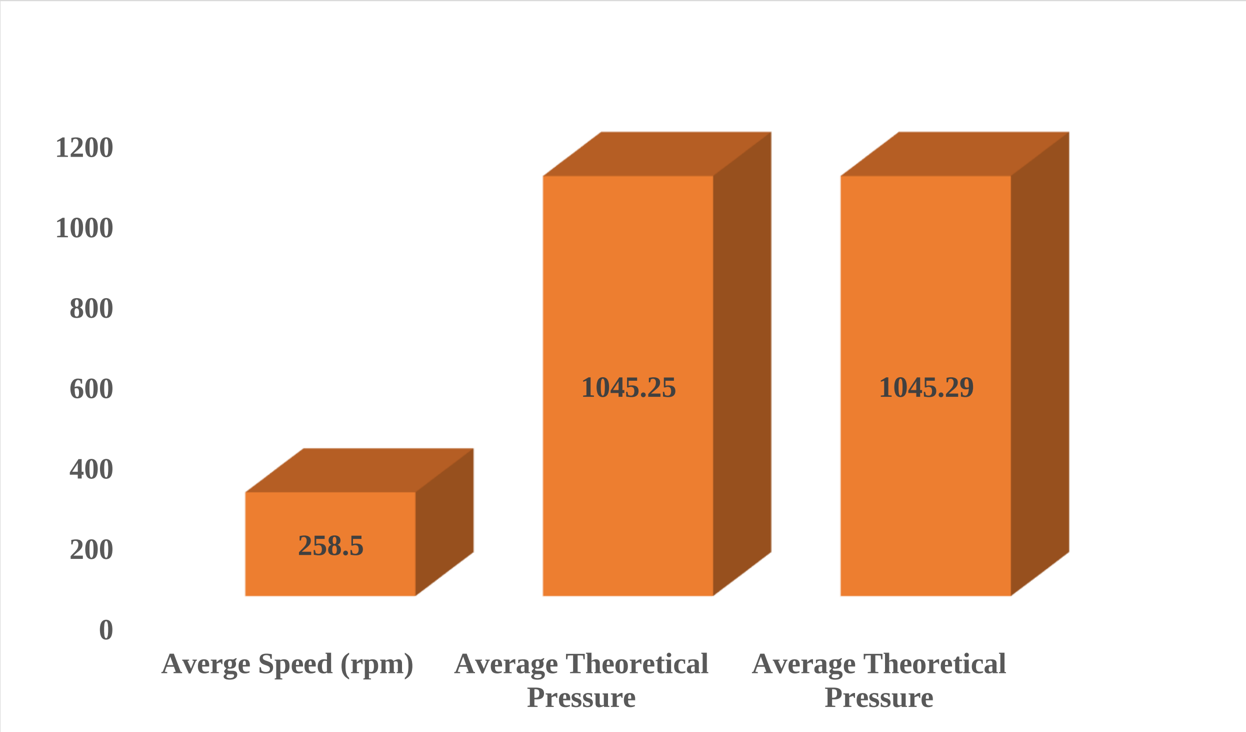


Figure 8. Comparative analysis of average experimental and theoretical pressure

4. Conclusion

A journal bearing test rig for pressure and temperature variation evaluation was successfully design and fabricated. Test performance was carried out on the fabricated machine. Analysis was carried out on the rotational speed, pressure variation, temperature variation due to heat generated, heat dissipated, and amount of artificial cooling. The journal bearing test rig was tested for different speeds and the results of experimental pressure were found to be harmonious with theoretical pressure. Besides, sufficient amount of heat is dissipated from the test rig, thus, minimal amount of artificial cooling is required. More so, the machine performance was satisfactory and the machine can be used for demonstration purpose in higher institution, locally and industrially in small scale.

Nomenclature

A_1	Area of small rectangle (m^2)
A_2	Area of big rectangle (m^2)
A_{mf}	Area of machine frame (m^2)
D	Journal diameter
Exp.	Experimental Pressure
f	Journal coefficient of friction
g	Acceleration due to gravity
h	Required height
H_d	Heat dissipated
H_g	Heat generated
L	Length of journal
m	Mass of journal bearing
N	Journal speed
P	Power
r	Inner radius (m)

R	Outer radius (m)
T	Torque
T_0	Operating temperature of the oil
Theo.	Theoretical Pressure
V	Volume of journal sleeve
W	Weight of journal bearing
Wr	Total radial bearing load

Greek letters

ρ	Density of mild steel
ω	Angular velocity

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