EXPERIMENTAL AND THEORETICAL INVESTIGATIONS OF HEAT GENERATION IN RADIAL BALL BEARING

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Abstract: *The usefulness of a radial ball bearing (RBB) is to support radial loads and to reduce the rotational friction. However, during its operating conditions, an unanticipated and vicious heating of balls in radial ball bearing takes place which results in degradation of its performance as well as accuracy. For this reason, it is important to calculate the heat generation in the bearings because if generated heat is not dissipated from the bearing surface, causes a temperature rise in inner race and gives rise to premature failure. In the present work, experimentally, heat generation is calculated by varying the bearing failure parameters such as external load, load position, and rotational speed. Furthermore, a theoretical model for estimation of heat generation in the radial ball bearing is proposed. Later, a correlation between the experimental and theoretical model is carried out. Ultimately, the proposed model reveals that it demonstrates better in estimating heat generation up to 0.45 m of load position without cooling condition. On the other hand, the predicted heat generation above 1.8 N of the external load has a minimal deviation with experimental values at with cooling condition. A better agreement observed between experimental results and the theoretical model.*

Index Terms: *RBB, Heat generation, External Load, Load Position*

1. INTRODUCTION

Machining with high speeds (HSM) is a modern technology in comparison with unconventional machining which enables to decrease production costs whereas increases productivity and quality of workpiece. Even though HSM is known for long time, but there is still lack of understanding in thermal characteristics of machining. Ever since, thermal effects on the machine tools become more significant. Bryan [1] reported that a total thermal error arising from various sources is about 40-70%. The major heat sources during high speed machining process include heat generation at tool-chip interface, spindle motor and the bearings. Among these, heat generated by machining and spindle motor are carried away by coolant. However, heat generation due to rubbing contact at the bearings is becoming dominant one causing thermal expansion. Generally heat generation in bearing where rotating at low speeds is not often studied such as bicycles and clocks. However, in case of bearings rotating rapidly for instance in automotive engines, machine tools, and aircraft engines, etc., heat generation becomes more relevant. So, bearings is one of the most important element commonly used in machinery due to their high load carrying capacity and also constrains the relative motion only to a desired motion.Therefore, adequate knowledge is required on thermal parameters to determine the heat generation in the bearing during its running state.

Several researchers made an attempt to comprehend the thermal behavior of the bearing during its working condition. Bossmanns and Tu [2, 3] developed a model of a high speed spindle to calculate the heat generation analytically in angular contact ball bearing by considering the effect of rotational speed, preload and lubrication. Subsequently, Li and Shin [4] put forward an integrated thermodynamic model of the whole system by using finite element method for high speed spindles and coupled it with spindle dynamic model through thermal expansion as well as heat generation. Jin et al. [5] determined the heat generation rate in bearing by considering the rotational speed and external load of the machine tool. Later, Moorthy and Raja [6] proposed an analytical model to estimate the heat generation in angular contact bearing by considering the change in diametrical clearance after assembly and during rotation. Thus, observed that at speeds (>2000 rpm), heat generation from the proposed model has minimal deviation compared to experimental values. Takabi and Khonsari [7] studied experimentally and analytically about the evolution of temperature in a deep-groove ball bearing with respect to time for different rotational speed and load in an oil-bath lubrication system. As a result, the predicted values of temperature of the oil, outer race and housing are found to be in good agreement with those measured experimentally. Mitrovic et al. [8] presented the influence of induced temperature on thermal expansion and parameters of radial ball bearing. Furthermore, finite element analysis is carried out for radial ball bearing to calculate bearing parameters at several operation temperatures. Later, concluded about the developed procedure by elucidating the problems related to the running state of bearing with respect to optimum operation temperature. Dong et al. [9] experimentally measured the temperature field distribution by using fiber Bragg grating sensors to understand the reasons behind the heat generation in bearing. In addition to that, author analyzed the thermal field distribution by performing simulation in ANSYS. Li [10] carried out experimental investigation on thermal characteristics of a spindle bearing system. Furthermore, used the thermo-mechanical coupling model and heat transfer model to analyze the parameters like applied force, preload, lubricating state, surface morphology and rotational speed numerically. The significant effect on the high rotational speeds, preload oil viscosity, and heat transfer coefficient on thermal failure of the bearing is revealed. William et al. [11] experimentally investigated heat transfer of the bearing rolling element and described that the lubricant distribution within the bearing happens rapidly enough to permit one to tolerate the bulk temperature & spatially constant convection coefficient. Nicola et al. [12] studied the influence of bearing grease composition on friction in rolling/sliding concentrated contacts and found that in high speed region, friction is governed by base oil viscosity as well as base oil type. Therefore, observed that for a given viscosity, friction is lower in the synthetic base oil greases whereas for a given oil type, friction reduces with higher temperature. Wei et al. [13] researched on fatigue life analysis of rolling bearings based on quasi-static modeling by increasing rotational speed. The author noticed that fatigue life of inner raceway and rollers is up while of outer raceway decline. Therefore, concluded that, the rotational speed does not much affect the order in which the damage appear on bearing parts. Hannon [14] studied the heat transfer of a bearing rolling element by proposing a new rolling-element bearing heat transfer model and presented equations to predict the conduction through the bearing housing, shaft, and raceways. Jeng and Huang [15] predicted temperature of ball bearings made up of steel and hybrid ceramic with different lubrication systems by using a ball bearing test rig. As a result, the predicted temperature cannot cover ball bearings under moderate speeds. Siyuan et al. [16] developed a thermal network model for double-row tapered roller bearing lubricated with grease, which is commonly used in high-speed railways. The load distribution and kinematic parameters in bearing are obtained by developing a quasi-static model. Later, the temperature of bearing at different rotational speed, filling grease ratio and roller large end radius are investigated. Hitonobu et al. [17] experimentally investigated wear of hybrid radial bearings (PEEK ring- PTFE retainer and alumina balls) under dry rolling contact. Their rolling contact wear tests were carried out at high rotational speed in dry condition. The peek adhesion film including PTFE and graphite wear practices was observed on the race way of the bearing's inner ring during the test. Due to the PEEK-PTFE adhesion film accumulation, the wear rate decreased to one tenth of all-PEEK bearing. Moreover, observed that, alumina balls were not stuck and the operation temperature is less than the glass transition temperature of PEEK.

From the literature, it was found that only few researchers tried to evaluate the heat generation in bearing system; particularly, in radial ball bearing, experimentally as well as theoretically. In the present study, experimentally, heat generation is calculated at different temperatures on bearing housing under the influence of process parameters such as external load, load position and rotational speed. Later, theoretically, heat generation in the bearing system is predicted by using Buckingham's pi- theorem. Finally, theoretical and experimental results are correlated and discussed at the end.

2. EXPERIMENTATION

In the current research work, test bearing is a radial ball bearing (SKF 6201) with inner diameter of 12 mm, outer diameter of 32 mm and width 10 mm used during experiments.

The bearing is seated on a mild steel shaft with a tight fit and in housing with loose fit. A DC Servo motor whose maximum speed is 10,000 rpm is coupled with the shaft. The estimation of heat generation for the above bearing is carried out in bearing test apparatus (Fig. 1) at with and without cooling the bearing during its operating condition. An external load is applied on the shaft by using hydraulic jack at certain load positions. The load readings are sensed with the help of S-type load sensor and displayed in weighing machine. The hydraulic jack is adjustable within the range of 1-20 N based on the required load to be applied on the shaft. Furthermore, the experiment is performed on bearing without lubrication in the raceways of bearing housing operated at room temperature as well as at low temperatures (-4^0 to 10^0 C) under different bearing parameter such as external load, load position and rotational speed.

In order to research the thermal characteristics of radial ball bearing during experiment, resistance temperature detector (RTD) sensors are employed at inner and outer race of radial ball bearing as shown schematically in Fig. 2. These sensors ((1, 3) and (2, 4)) measure the temperature at inner race and outer race of bearing.

Therefore, the sensed temperature values are displayed in digital VR06 paperless recorder. Since, the genesis of internal generation of

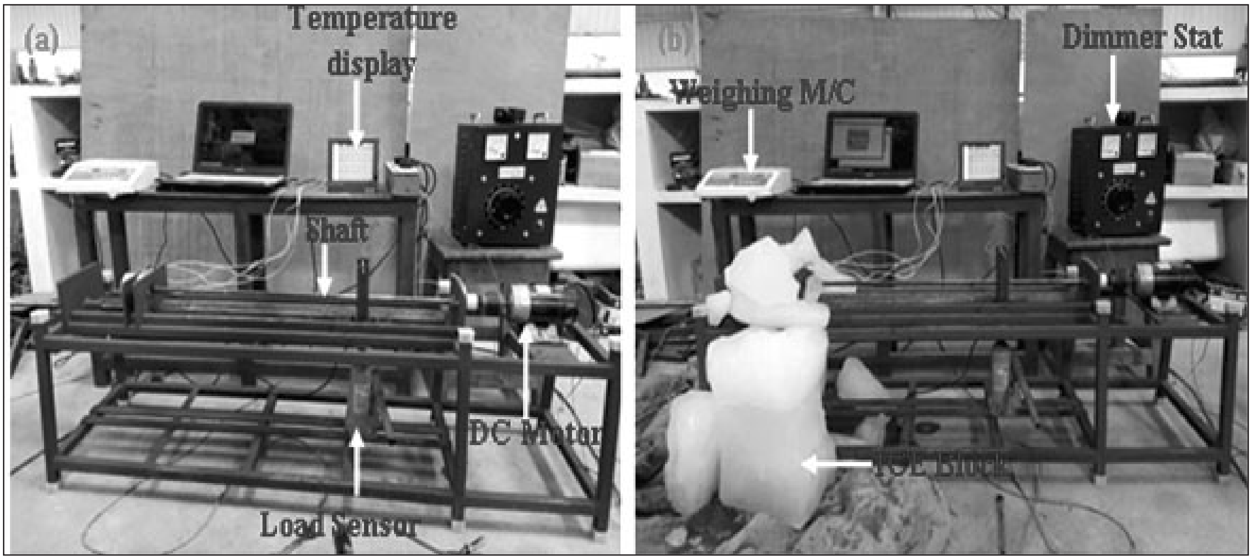

Fig 1. Overview of Experimental Setup (a) Without Cooling System, (b) With Cooling System

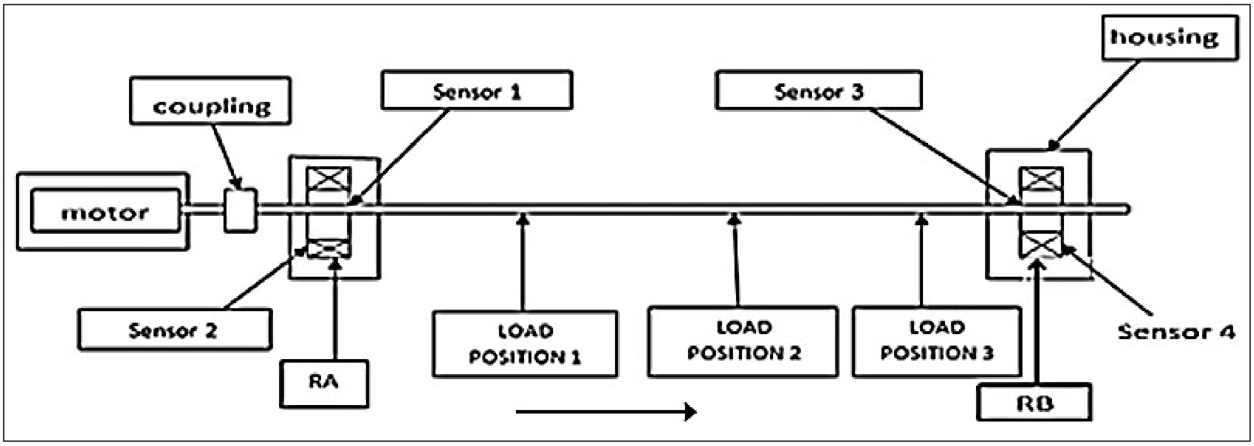

Fig 2. Layout of the Experimental Setup

heat in a radial ball bearing is the local contact between rolling element and races during operating condition produces heat. Schematically the transfer of generated heat to the environment through the bearing is shown in Fig. 3. A part of the generated heat goes to races while remaining part goes to rolling elements. However, important consideration is to avoid the failure of a bearing, in order to do so the total heat generated in bearing should be effectively transferred from surface of outer race to environment via convection mode. Thereby, equilibrium conditions are reached in short time. In the present study, the total internal heat generation is dissipated via conduction through the balls from inner to outer race and then via convection to environment from bearing housing.

In the experiment, bearing is considered as heat source, and the heat flow (Q) is calculated by assuming it as one dimensional steady state heat transfer with conduction and convection (Figure 4). The entire heat transfer processes in the radial ball bearing is well-ordered in block diagram which makes easy to investigate the thermal behavior of the bearing system. The heat generation in radial ball bearing is calculated experimentally by considering the process parameters such as external load, load position and rotational speed respectively. Fig. 5 depicts the heat transfer process including conduction and free convection within as well as in between the bearing elements. In the current study, the heat transfer due to radiation is neglected owing to low surface temperatures.

After conducting preliminary experiments, the operating rages of input parameters are found. These ranges are used to design the experiments by the Taguchi method. Total of 9 experiments are designed using the Taguchi's L_9 orthogonal array shown in Table 1 with the help of MINITAB 14 package.

3. ANALYSIS OF VARIANCE (ANOVA)

In the present research work, ANOVA is used for investigating the most significant influencing process parameter on performance characteristic of bearing system. It is measured by determining the percent (%) of contribution which is the ratio between sum of squared deviations

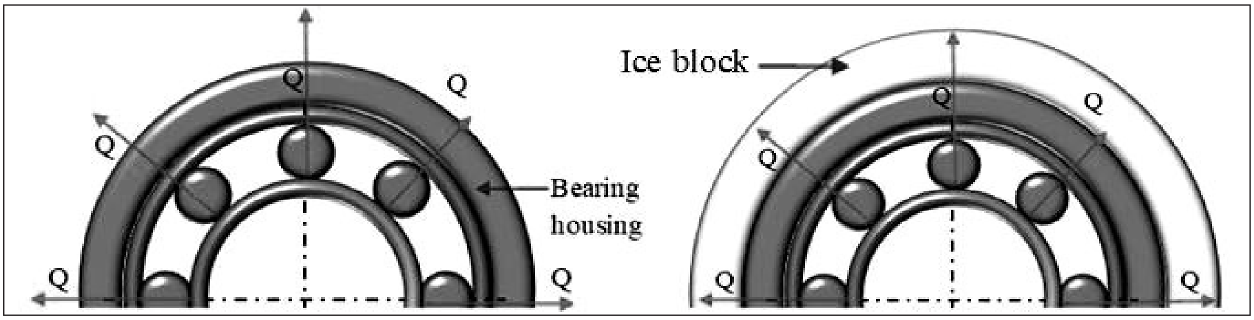

Fig 3. Schematic Representation of Dissipation of Heat from Bearing to (a) Environment (b) and Ice Block

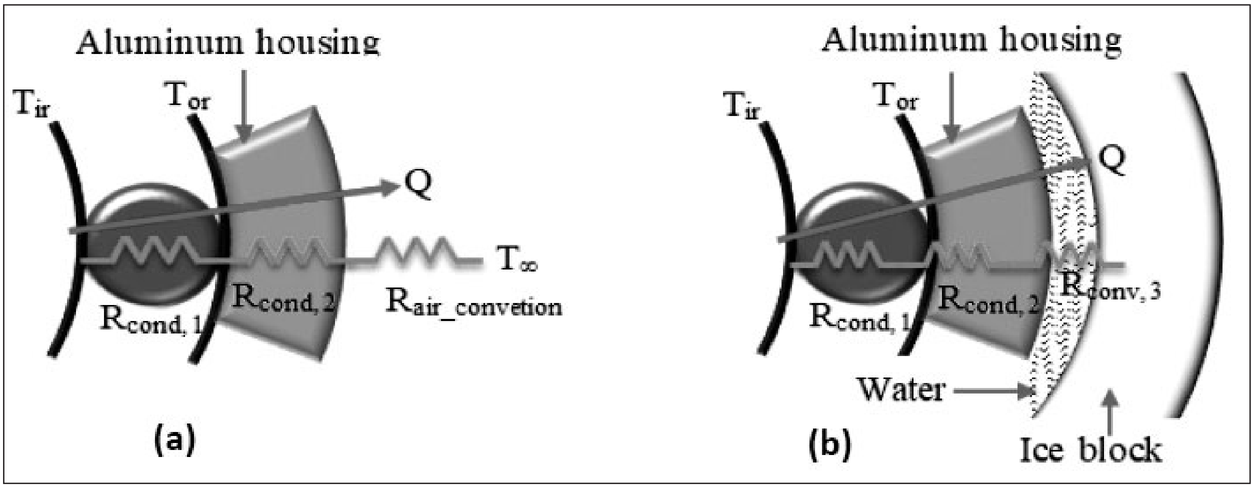

Fig 4. Schematic Diagrams of Thermal Resistances at (a) Without Cooling (b) With Cooling Condition

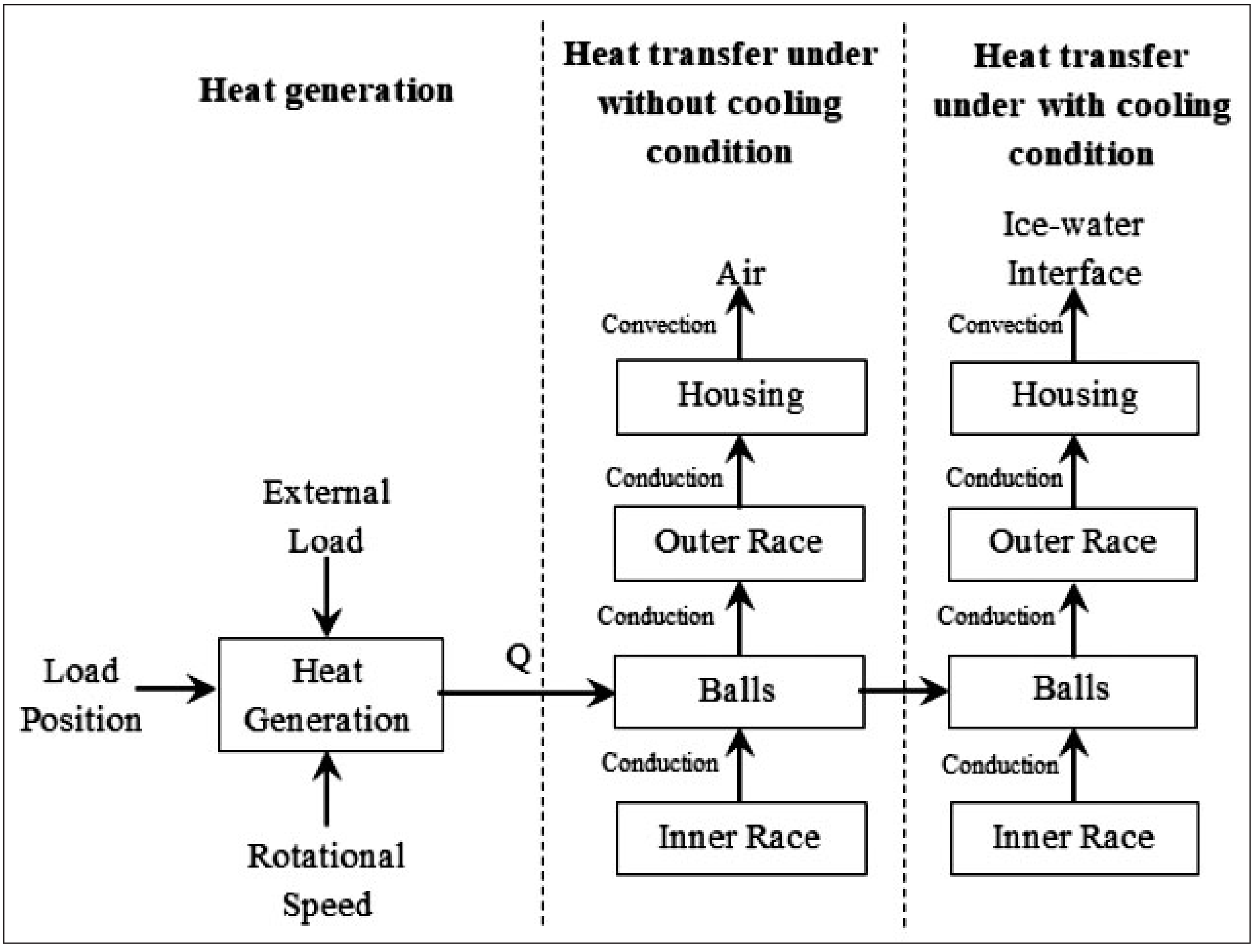

Fig 5. Block Diagram of Heat Transfer Process at With and Without Cooling Conditions

Table 1: Orthogonal Array (L_9) Experiments and Measurements

S. No.	External load (kg)	Load position(m)	Rotational speed(rpm)	T2(^{0}C)	T3(^{0}C)
1	1	0.25	800	35.8	35.1
2	1	0.5	1600	43.4	42.8
3	1	0.75	2400	49.8	49.4
4	1.5	0.25	1600	36.4	35.7
5	1.5	0.5	2400	49.6	49.2
6	1.5	0.75	800	45.6	45.1
7	2.5	0.25	2400	37.6	37
8	2.5	0.5	800	39.2	38.7
9	2.5	0.75	1600	46.8	46.4

of each factor and the total sum of squares. The three factors namely external load, load position and rotational speed are considered in the present analysis for analyzing as well as to determine their influence on heat generation in the bearing at with and without cooling condition.

The influence of these factors at various levels is examined by ANOVA (Table 2 and 3). Optimal value for each bearing parameters is obtained by analyzing the heat generation. The optimal conditions for bearing parameters are identified with increasing heat generation using Taguchi method. S/N ratios for each performance characteristic at all levels of process parameters are evaluated by larger-the-better criteria. The main influencing plots of S/N ratios for each response are presented in Fig. 6 and Fig. 7.

From Fig. 6, it is inferred that the optimal process parameters for increasing temperature in bearing

Table 2: ANOVA for Bearing System at With Cooling Condition

Source	DF	SS	MS	F	% Contribution
load	2	9.167	4.5834	5.91	52.66
Load position	2	2.736	1.368	1.77	15.72
Speed	2	3.955	1.9774	2.55	22.72
Residual Error	2	1.55	0.775		8.9
Total	8	17.408			100

Table 3: ANOVA for Bearing System at Without Cooling Condition

Source	DF	SS	MS	F	%contribution
Load	2	0.06054	0.03027	0.33	5.77
Load position	2	0.58825	0.29412	3.17	56.08
Speed	2	0.21439	0.10719	1.15	20.44
Residual Error	2	0.18583	0.09291		17.71
Total	8	1.049			100

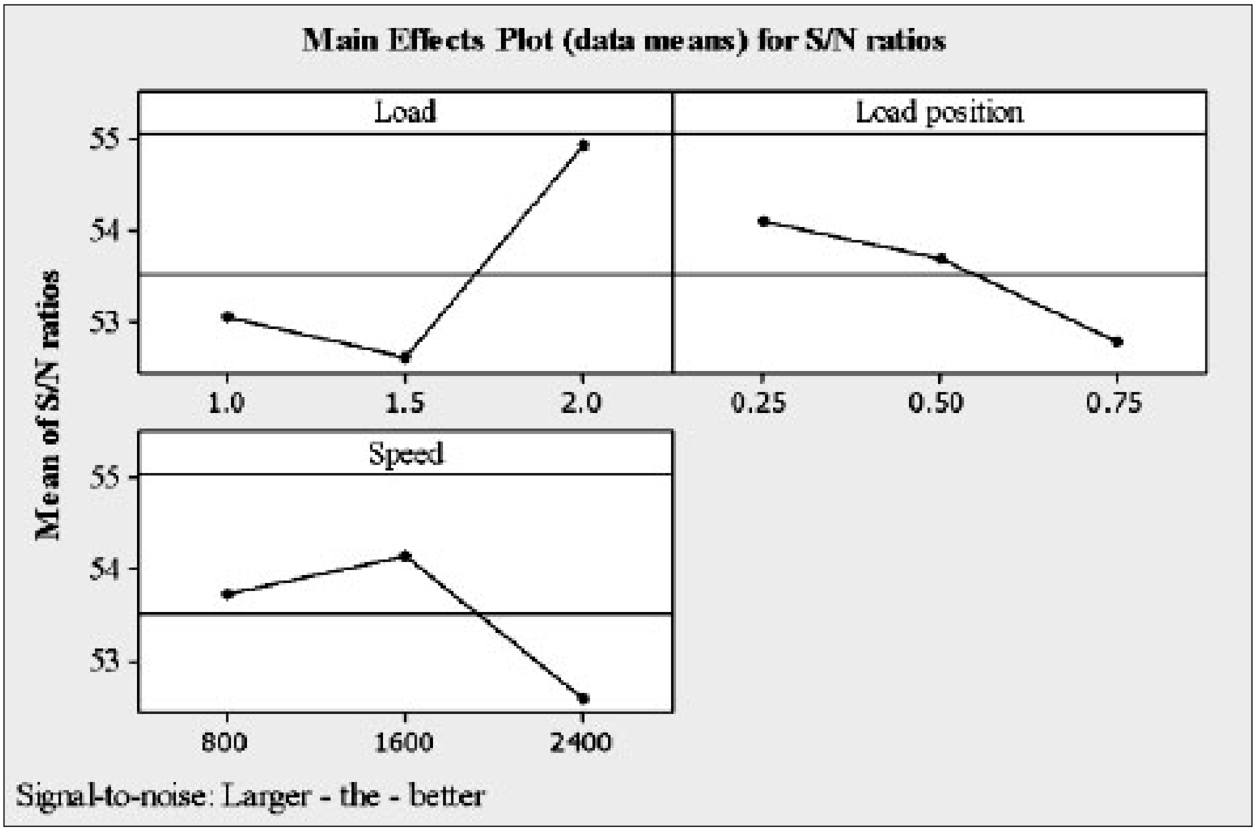

Fig 6. Effect of Process Parameters on Bearing System at With Cooling Condition

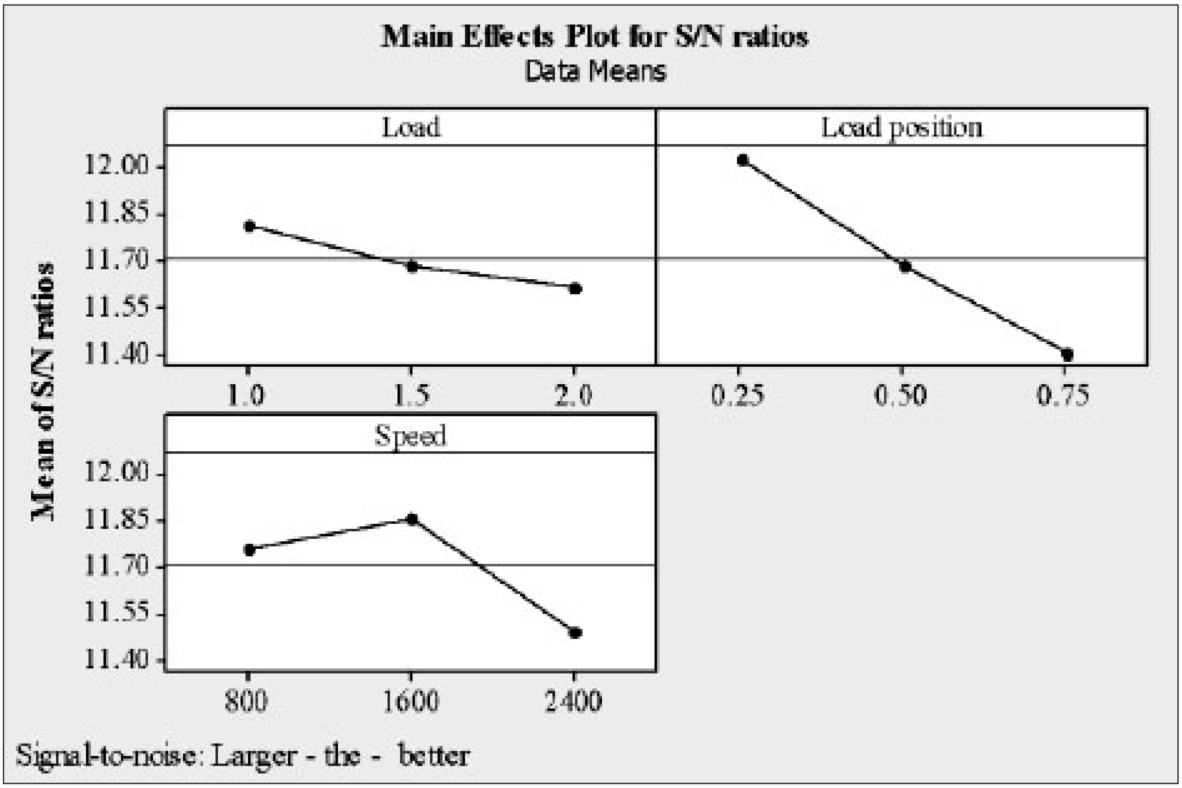

Fig 7. Effect of Process Parameters on Bearing System at Without Cooling Condition

are external load (1 N), load position (0.25 m) and rotational speed (1600 rpm). Whereas, from the main effects plots for heat generation with cooling (Fig. 7), it is noticed that external load (2 N), load position (0.25 m) and rotational speed (1600 rpm) has better impact by way of increase in heat generation when compared to other process parameter levels.

Percentage contribution of external load, load position and rotational speed on heat generation at without cooling condition is 5.77%, 56.08% and 20.44% and with cooling is 52.66%, 15.72% and 22.72% respectively. Therefore, ANOVA results indicate that load position has significant influence on heat generation at without cooling condition whereas with cooling, external load has significant influence on heat generation. The temperature at inner and outer race is measured under various external loads and load positions are shown in Fig. 8 and 9.

4. THEORETICAL INVESTIGATION

Dimensional analysis is a mathematical method of obtaining the equations governing certain natural or physical unknown phenomena by balancing the fundamental dimensions as mass, length, time and temperature of the problem considered. It is known as the principle of dimensional similarity. This method can be applied to all type of fluid resistances, heat flow problems and many other problems in fluid mechanics and thermodynamics. The equation produced by this method also gives non-dimensional constants which governs the problem under consideration. This non-dimensional constants are essential in engineering practice, as it enables the behaviour of the problems of same type to be predicted provided the linear dimensions are geometrically similar, particularly, the heat lost by convection (natural or forced) is extremely complex process and controlling variables are numerous. It is difficult to study the effect of each variable on the process, but it is possible to investigate experimentally and express scientifically the result for any geometrical configuration has its own different indices values.

4.1. Finding Non Dimensional Parameters

The Non Dimensional parameters which influence the bearing characteristics are found conveniently by using Buckingham π- Method. In the present study, 24 parameters are considered during the investigation. While in other methods, Non dimensionless numbers are difficult to compute. Therefore, firstly, the bearing parameters are studied for analyzing the bearing characteristics are summarized including with their dimensions

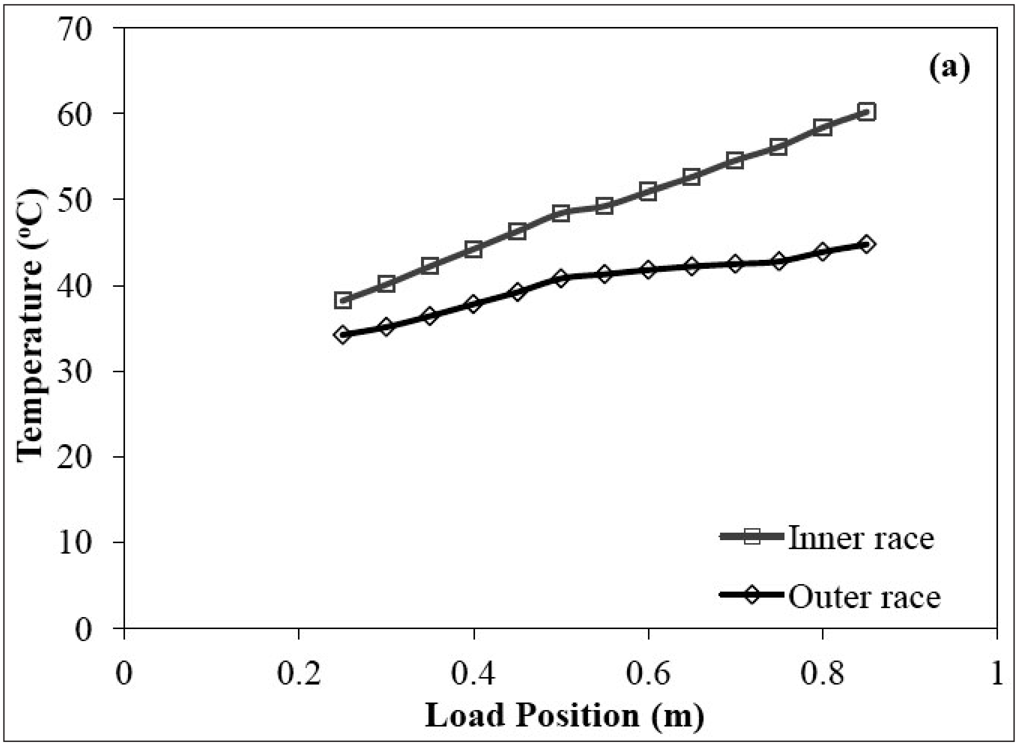

Fig 8. Variation of Temperature on Inner and Outer Race with Respect to Load Position Keeping Other Parameters Constant (External load 1 N and Rotational speed 1600 rpm) at Without Cooling Condition

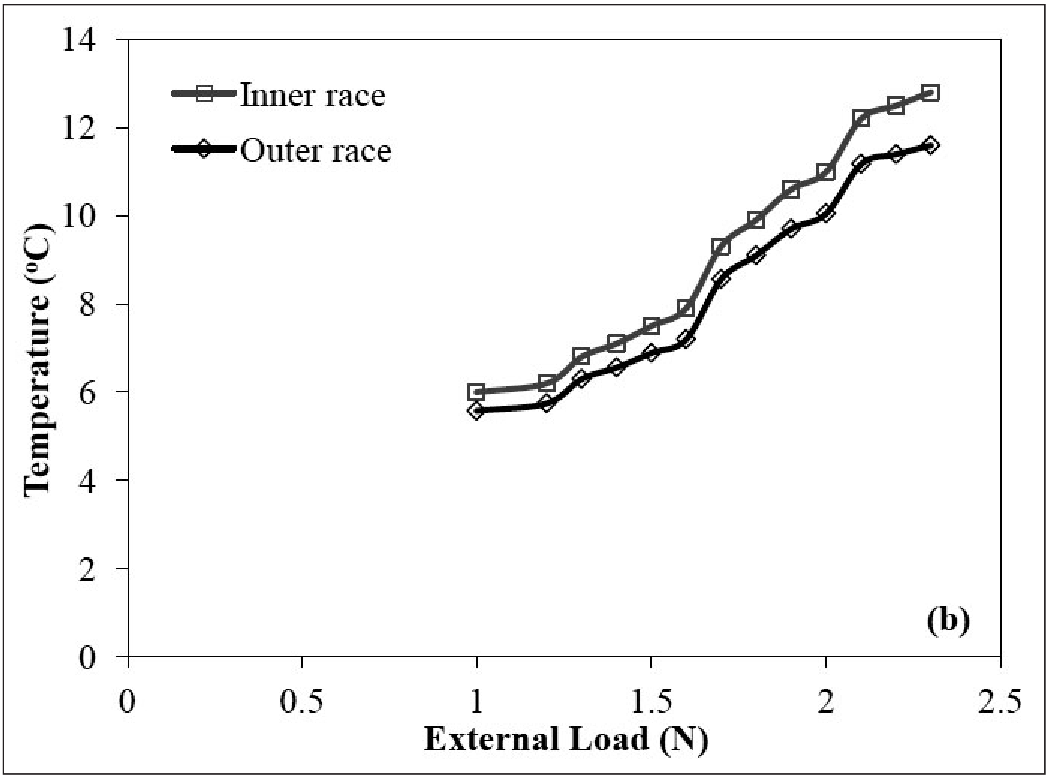

Fig 9. Variation of Temperature on Inner and Outer Race with Respect to External Load Keeping Other Parameters Constant (Load position 0.25 m and Rotational speed 1600 rpm) at Without Cooling Condition

and finally π-terms are evaluated in section (ii).

4.2. Evaluating π-terms

In compliance with the mentioned list in Table 4, π-terms are evaluated for the considered bearing parameters by taking four repeating variables namely Power (P), Heat Generated in the bearing (Q), Heat transfer coefficient(H), Thermal conductivity (K). Therefore, 20 π-terms are obtained which is represented in equations (Eq. 1-20) or in matrix form (Eq. 21). The constants (a1, b1, c1 ……..d20) present in equation (1) to (20) are obtained by using Buckingham pi-theorem.

$$\Pi_1 = H^{a1}Q^{b1}K^{c1}P^{d1}.F_r \quad (1)$$

$$\Pi_2 = H^{a2}Q^{b2}K^{c2}P^{d2}.g \quad (2)$$

$$\Pi_3 = H^{a3}Q^{b3}K^{c3}P^{d3}.n \quad (3)$$

$$\Pi_4 = H^{a4}Q^{b4}K^{c4}P^{d4}.Ta \quad (4)$$

$$\Pi_5 = H^{a5}Q^{b5}K^{c5}P^{d5}.W \quad (5)$$

$$\Pi_6 = H^{a6}Q^{b6}K^{c6}P^{d6}.F_b \quad (6)$$

$$\Pi_7 = H^{a7}Q^{b7}K^{c7}P^{d7}.\mu \quad (7)$$

Table 4: List of Bearing Parameters

S. No.	Parameters	L	M	T	Θ
1	Reaction force (F_r)	1	1	-2	0
2	Acceleration due to gravity (g)	1	0	-2	0
3	Shaft speed (n)	0	0	-1	0
4	Power (P)	2	1	-3	0
5	Heat generated in the bearing (Q)	2	1	-2	0
6	Surface temperature of the bearing (T_a)	0	0	0	1
7	Load carrying capacity (N)	1	1	-2	0
8	Frequency of the bearing (F_b)	0	0	-1	0
9	Dynamic viscosity (μ)	-1	1	-1	0
10	Bearing Size (D_{ball})	1	0	0	0
11	Clearance of ball bearing (C_{ball})	1	0	0	0
12	Thermal conductivity (K)	1	1	-3	-1
13	Convective heat transfer coefficient (H)	0	1	-3	-1
14	Shaft density (ρ)	-3	1	0	0
15	Flexural rigidity (EJ)	1	3	-2	0
16	Moment of inertia (I)	4	0	0	0
17	Stiffness (s)	0	1	-2	0
18	Thermal conductivity (K_{ref})	1	1	-3	-1
19	Shaft speed (n_{ref})	0	0	-1	0
20	Load carrying capacity (n_{ref})	1	1	-2	0
21	Frequency of the bearing ($F_{b\ ref}$)	0	0	-1	0
22	Bearing size ($D_{ball\ ref}$)	1	0	0	0
23	Dynamic viscosity (μ_{ref})	-1	1	-1	0
24	Temperature initial ($T_{initial}$)	0	0	0	1

Table 5 Simplified Dimensionless Variables

$\pi_{simplified}$				
$\pi_a = \frac{\pi_3}{\pi_{14}}$	$\pi_b = \frac{\pi_5}{\pi_{15}}$	$\pi_c = \frac{\pi_6}{\pi_{10}}$	$\pi_d = \frac{\pi_8}{\pi_{12}}$	$\pi_e = \frac{\pi_{16}}{\pi_{17}}$
$\pi_f = \frac{\pi_4}{\pi_9}$	$\pi_g = \frac{\pi_7}{\pi_{11}}$	$\pi_h = \frac{\pi_{18}}{\pi_{19}}$	$\pi_i = \frac{\pi_2}{\pi_{20}}$	$\pi_j = \frac{\pi_{18}}{\pi_{19}}$
$\pi_k = \frac{\pi_{16}}{\pi_{17}}$				

$\Pi_8 = H^{a8}Q^{b8}K^{c8}P^{d8}.D_{ball}$ (8)

$\Pi_9 = H^{a9}Q^{b9}K^{c9}P^{d9}.T_{initial}$ (9)

$\Pi_{10} = H^{a10}Q^{b10}K^{c10}P^{d10}.F_{b\ ref}$ (10)

$\Pi_{11} = H^{a11}Q^{b11}K^{c11}P^{d11}.\mu_{\ ref}$ (11)

$\Pi_{12} = H^{a12}Q^{b12}K^{c12}P^{d12}.D_{ball\ ref}$ (12)

$\Pi_{13} = H^{a13}Q^{b13}K^{c13}P^{d13}.K_{\ ref}$ (13)

$\Pi_{14} = H^{a14}Q^{b14}K^{c14}P^{d14}.n_{\ ref}$ (14)

$\Pi_{15} = H^{a15}Q^{b15}K^{c15}P^{d15}.N_{\ ref}$ (15)

$\Pi_{16} = H^{a16}Q^{b16}K^{c16}P^{d16}.\rho$ (16)

$\Pi_{17} = H^{a17}Q^{b17}K^{c17}P^{d17}.EJ$ (17)

$\Pi_{18} = H^{a18}Q^{b18}K^{c18}P^{d18}.I$ (18)

$\Pi_{19} = H^{a19}Q^{b19}K^{c19}P^{d19}.S$ (19)

$\Pi_{20} = H^{a20}Q^{b20}K^{c20}P^{d20}.C_{\ ball}$ (20)

$B = -(A_S)^T((A_Z)^T)^{-1}$ (21)

Where B is solution matrix, A_S is a sub matrix of decisive quantities and A_Z is the residual sub matrix.

Therefore, the evaluated π-terms is summarized below

$\Pi_1 = H^{-1}Q^{-1}K^1P^0.F_r$ (22)

$\Pi_2 = H^1Q^2K^{-1}P^{-2}.g$ (23)

$\Pi_3 = H^0Q^1K^0P^{-1}.n$ (24)

$\Pi_4 = H^{-1}Q^0K^2P^{-1}.T_a$ (25)

$\Pi_5 = H^{-1}Q^{-1}K^1P^0.W$ (26)

$\Pi_6 = H^0Q^1K^0P^{-1}.f_b$ (27)

$\Pi_7 = H^{-3}Q^{-2}K^3P^1.\mu$ (28)

$\Pi_8 = H^1Q^0K^{-1}P^0.D_{ball}$ (29)

$\Pi_9 = H^{-1}Q^0K^2P^{-1}.T_{initial}$ (30)

$\Pi_{10} = H^0Q^1K^0P^{-1}.fb_{\ ref}$ (31)

$\Pi_{11} = H^{-3}Q^{-2}K^3P^1.\mu_{\ ref}$ (32)

$\Pi_{12} = H^1Q^0K^{-1}P^0.D_{ball\ ref}$ (33)

$\Pi_{13} = H^0Q^0K^{-1}P^0.K_{\ ref}$ (34)

$\Pi_{14} = H^0Q^1K^0P^{-1}.n_{\ ref}$ (35)

$\Pi_{15} = H^{-1}Q^{-1}K^1P^0.N_{\ ref}$ (36)

$\Pi_{16} = H^{-5}Q^{-3}K^5P^2.\rho$ (37)

$\Pi_{17} = H^{-5}Q^{-7}K^5P^4.EJ$ (38)

$\Pi_{18} = H^4Q^0K^{-4}P^0.I$ (39)

$\Pi_{19} = H^{-2}Q^{-1}K^2P^0.S$ (40)

$\Pi_{20} = H^1Q^0K^{-1}P^0.C_{\ ball}$ (41)

Therefore, the phenomenon of heat generation in the bearing system can be expressed as,

$$\pi_1 = f(\pi_2, \dots \dots \dots \dots \dots \dots \dots \dots \dots \pi_{20}) \quad (42)$$

Since, the number of dimensionless parameters is still more. Therefore, they are further simplified by the following operations given in Table 5.

Assuming constant, $k = f(\pi_a, \pi_b, \pi_c, \pi_d, \pi_e, \pi_f, \pi_g, \pi_h, \pi_{13})$

Where '*k*' represent the constant variables which don't vary during the experimentation. Therefore, the equation (42) is rewritten as,

$$\pi_1 = f(k, \pi_i, \pi_j, \pi_k) \quad (43)$$

The derived terms for heat generation (Q) can be formulated using power law as,

$$\pi_1 = k\, \pi_i^{c1}\, \pi_j^{c2}\, \pi_k^{c3} \quad (44)$$

4.3. Solution of Governing Equation by Regression Analysis

The solution of governing equation (44) is solved by multiple regression analysis technique and the constants c_0, c_1, c_2, c_3 are obtained by following mathematical operations. Among 'n' experiments to be performed, the response of i^{th} experiment is calculated as,

$$y_i = c_0 + c_1 x_{i1} + c_2\, x_{i2} + c_3\, x_{i3} \quad (45)$$

Therefore, sum of all the 'n' experiments is written as,

$$\sum_{i=1}^{n} y_i = n\, c_0 + c_1 \sum_{i=1}^{n} x_{i1} + c_2 \sum_{i=1}^{n} x_{i2} + c_3 \sum_{i=1}^{n} x_{i3} \quad (46)$$

To determine the four unknowns we require four simultaneous equations, so the other equations are obtained by multiplying one by one separately to equation (46) as follows,

$$\sum_{i=1}^{n} y_i x_{i1} = n\, c_0 \sum_{i=1}^{n} x_{i1} + c_1 \sum_{i=1}^{n} x_{i1}\, x_{i1} + c_2 \sum_{i=1}^{n} x_{i2}\, x_{i1} + c_3 \sum_{i=1}^{n} x_{i3}\, x_{i1} \quad \text{.........(47)}$$

$$\sum_{i=1}^{n} y_i x_{i2} = n\, c_0 \sum_{i=1}^{n} x_{i2} + c_1 \sum_{i=1}^{n} x_{i1}\, x_{i2} + c_2 \sum_{i=1}^{n} x_{i2}\, x_{i2} + c_3 \sum_{i=1}^{n} x_{i3}\, x_{i2} \quad \text{..........(48)}$$

$$\sum_{i=1}^{n} y_i x_{i3} = n\, c_0 \sum_{i=1}^{n} x_{i3} + c_1 \sum_{i=1}^{n} x_{i1}\, x_{i3} + c_2 \sum_{i=1}^{n} x_{i2}\, x_{i3} + c_3 \sum_{i=1}^{n} x_{i3}\, x_{i3} \quad \text{............(49)}$$

Equations (47) to (49) are solved simultaneously for the unknowns. Therefore, after determining the unknowns, equation (44) is used as model equation to obtain the heat generation in a bearing system.

5. RESULTS AND DISCUSSION

5.1. Variation of Heat Generation with Load Position under Without Cooling Condition

From the experiment, we noticed that the rise of heat generation in the radial ball bearing is in proportion to the load position but the relationship is non-linear. While, the hear generation is varying linearly with the load position. Heat generation is calculated for various values of load position by keeping all other parameters as constant without cooling the bearing. It can be seen in Fig. 10 that during the low-load position range, the deviation between the two curves is small because at the initial operating condition wear of balls is less. However, with the increase in load position, heat generation increases. This is due to the fact that as the load position increases the interaction between the balls and surfaces (inner and outer race) increases drastically causes a conversion of some of the power into heat. This may leads to decrease in diameter of balls due to high wear at the contact point between balls and surfaces. Due to this, air entraps in clearance between the surfaces and contact point thereby reduces the thermal conductance. Whereas, theoretical modeling doesn't take this into account thereby more deviation is observed at higher load position.

However, as the load position continues to increase, the difference becomes larger and larger because in theoretical modeling it is assumed that only balls act as a representative carrieer of generated heat from inner to outer race. Moreover, it can be seen that the curves depict the same trend and the proposed theoretical model has better predictive ability at lower load positions (< 0.4 m).

5.2. Variation of Heat Generation with External Load under Cooling Condition

Theoretically, as the applied load increases, the magnitude of reaction forces on the bearing increases which results in an increase in heat generation. Moreover, from the experiment it is evident that rise of heat generation in the radial ball bearing is in proportion to the load but the relationship is non-linear. According to the trend observed in Fig. 11, it can be deduced that at low-load range, the difference between the two curves is more because the forces acting between the rotating and stationary surfaces in radial ball bearing acts at higher contact areas resulting less amount of heat generation. However, with the increase in load, the heat generation increases drastically because the force acts at smaller contact area resulting less amout of stress at the contact in turn alter the amount of heat generation. Since, the effect of forces at the contact surfaces is not accounted in theoretical

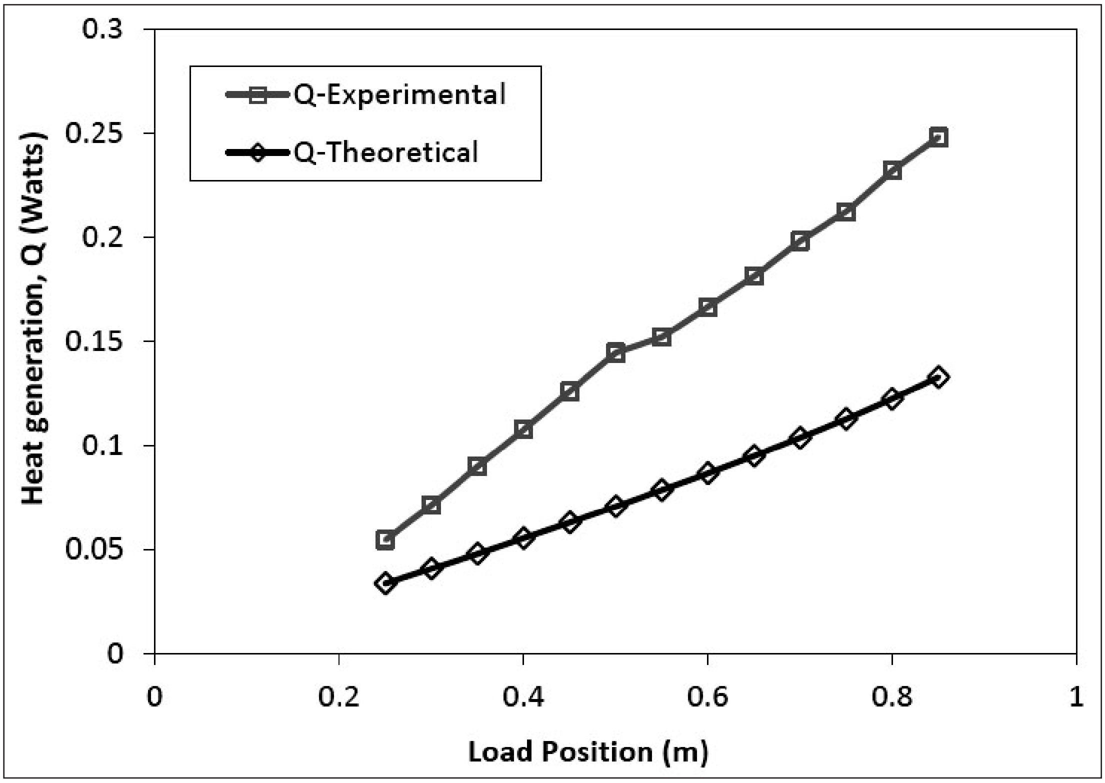

Fig 10. Comparision of Heat Generation with Load Position Keeping Other Parameters Constant (External load 1 N and Rotational speed 1600 rpm) Under Without Cooling Condition

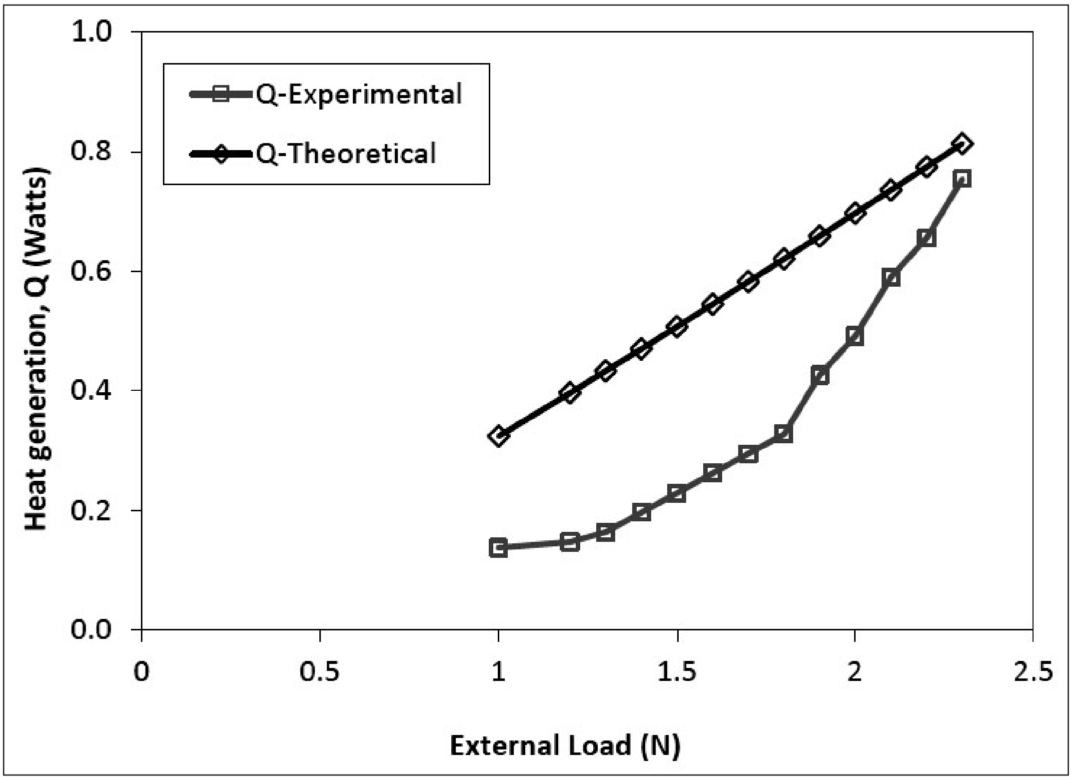

Fig 11. Comparision of Heat Generation with External Load Keeping Other Parameters Constant (Load position 0.25 m and Rotational speed 1600 rpm) Under Cooling Condition

modeling which leads to higher deviation at lower values of load (< 1.8 N). However, the proposed model at higer loads in the range (1.8-2.3 N) predicts better when compared with the experimental values.

CONCLUSIONS

The present experimental study focuses on investigating the influence of heat generation on the bearing system using Taguchi method. In the present study, load, load position and rotational speed are considered as process parameters. The level of influence of process parameters on performance characteristic is identified using main effects plots of signal noise ratios and also the most influence of process parameters are identified by employing ANOVA. Thus, the presented results reveal that load position is considered as an important parameter in bearing system at without cooling condition. On the other, with cooling condition, load is more significant parameter in bearing analysis. Furthermore, a comprehensive theoretical model is presented for estimating the heat generation in radial ball bearing. The proposed model demonstrates better in estimating the heat generation up to 0.45 m of load position at without cooling condition and with cooling condition, above 1.8 N of load. Therefore, the proposed theoretical model could be used effectively for predicting the heat generation at the above specified conditions.

Acknowledgement

The authors are thankful for the financial support provided by N.B.K.R. Institute of Science and Technology, Department of Mechanical Engineering.

Declaration of conflicting interests

The author(s) declared no potential conflicts of interest with respect to the research, authorship, and/or publication of this article.

Nomenclature

F_r	Reaction forces
g	Acceleration due to gravity
n	Shaft speed
P	Power supply
Q	Heat Generated in the bearing
T_a	Surface Temperature of the bearing
H	Convective heat transfer coefficient
EJ	Flexural rigidity
S	Stiffness
$D_{ball\ ref}$	Bearing size
n_{ref}	Shaft speed
N	Load carrying capacity
F_b	Frequency of the bearing
μ	Dynamic viscosity
D_{ball}	Bearing size
C_{ball}	Clearance of ball bearing
K	Thermal conductivity
ρ	Shaft density
I	Moment of inertia
K_{ref}	Thermal conductivity
μ_{ref}	Dynamic viscosity
$T_{initial}$	Temperature initial
$F_{b\ ref}$	Frequency of the bearing
$R_{cond,\ 1}$, $R_{cond,\ 2}$	Thermal conduction resistance of ball and aluminium housing
$R_{air\_convection}$	Thermal resistance of air convection
$R_{conv,\ 3}$	Thermal resistance of ice-water interface

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Appendix A

Heat generation calculation during experiment and by theoretical model

1 D steady-state heat conduction with convection through bearing housing is calculated at with (Equation A.1) and without cooling (Equation A.2) condition by the following formulas

$$Q = \frac{(T_{ir} - T_{\infty})}{\left\{\frac{\ln\left(\frac{R_i}{R_o}\right)}{2\pi K_b L_b} + \frac{\ln\left(\frac{R_o}{R_h}\right)}{2\pi K_h L_h} + \frac{1}{h_{air} A_s}\right\}} \qquad (A.1)$$

Where R_i, R_o, R_h is radius of inner race, outer race and housing, , is width of the bearing and housing, T_{ir} is temperature at inner race, is ambient temperature, is thermal conductivity of bearing, is thermal conductivity of bearing housing.

$$Q = \frac{(T_{ir} - T_{\infty})}{\left\{\frac{\ln\left(\frac{R_i}{R_o}\right)}{2\pi K_b L_b} + \frac{\ln\left(\frac{R_o}{R_h}\right)}{2\pi K_h L_h} + \frac{1}{h_{ice-water} A_s}\right\}} \qquad (A.2)$$

Where is surface area of bearing housing, is heat transfer coefficient at ice-water interface which can be calculated by the formula given by Chen et al. [18].

Appendix B

Values of various constants used during experiment and theoretical modeling

Radius of inner race (m)	= 0.006
Radius of outer race (m)	= 0.016
Radius of housing (m)	= 0.036
Thermal conductivity of bearing (w/m k)	= 45
Thermal conductivity of bearing housing (w/m k)	= 205
Width of the bearing (m)	= 0.01
Width of the housing (m)	= 0.025
Average room temperature (^{o}C)	= 30

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