Effect of the Central Radial Groove on the Bearing Pressure Distribution Based on the Artificial Neural Network

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Abstract- **An experimental study has been implemented to study the effect of the central radial groove on the bearing pressure distribution. This study is based on the artificial neural network in the prediction of the complex and uncertain positions. Both width and depth of the groove have been varied at some magnitudes in order to investigate their effects on the pressure distribution and the stability of the bearing. Also, the effect of the groove parameters on the noise at the bearing situation in the systems have been analyzed and discussed. The results show that the use of neural network in the prediction of some points with range is very powerful in the minimization of the overall cost of groove design.**

KEY WORDS: Bearing Design, Pressure Distribution Analysis, Groove Design, Noise, Neural Network.

I. Introduction:

Journal bearing is a very important subject in many studies because it relates with the efficiency of any system or mechanism that it is being a part of it. The design of the components of the journal bearing has a great effect on the robustness of the bearing. Many problems and undesirable phenomena can be removed or reduced by best optimization of the journal bearing assembly. These problems may be ;vibration, noise, oil whirl, etc.

The geometry of the rotating masses of the journal bearing has a great effect on the reduction of the response of these masses to the vibration effects. Also, in a high speed machines, the isolation of the system from the surroundings can play an important role for removing the vibration and noise. The position of the journal center is eccentric with the bearing center. A lubricant fills the small annular gap or clearance between the journal and the bearing. The amount of eccentricity of the journal is related to the pressure that will be generated in the bearing to balance the radial load. Therefore, the design of the inner rotating journal surface is a very important issue. Many researches had investigated the optimum bearing parameters, which maximize the load carrying capacity in the radial direction. A study for indicating of the stability for whirl, for the design of regular and reversible rotation type herringbone grooved journal bearing was proposed by Yasume [1] in 1992.

In 2010, Sami et. al.[2] presented a study on steady state analysis of a hydrodynamic short bearing supplied with a circumferential groove. In this study , a theoretical prediction of the steady state characteristics of a hydrodynamic short bearing with a circumferential central feeding groove was discussed. In 2013, Krishan et. al. [3] examined the effect of surface texture on the hydrodynamic journal bearing performance. A single axial groove for plain journal bearings is common in industrial applications [4]. Ahmed et. al. [4] in 2013 investigated a thermo-hydrodynamics performance of single groove journal bearing for various oil supply pressure values at different groove positions. Flow rates from semi-circular and irregular groove cross-sections were presented by Hoge et. al. [5] in 1997. Comparison of calculated oil flow for a production-level grooved bearing showed good agreement with experimental measurements from a bench test rig. The equation for the pressure buildup in these bearing was subjected to a first-order perturbation with respect to parallel as well as skewed displacements of the center lines of journal and bearing (Hirs [6] in 1968). Promising applications in machines were also discussed in this study.

1.1 Objective of the present work

The main objective of this work can be summarized as follows:

Propose a new model of grooved journal bearing.

Implement a predictable strategy for investigating the properties of the complex and uncertain positions in the bearing based on the Artificial Neural Network.

Analyze the effect of the groove parameters on both noise and pressure distribution in either radial or axial direction.

2. Theoretical Background

In the section of bearing geometry shown in Fig.(1), the point (o) represents the centroid of the bearing ,while (c) represents the centroid of the journal.

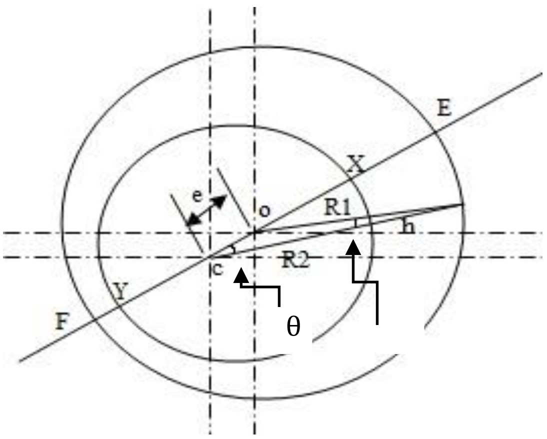

Fig.1 Section of bearing geometry

When the connection between the points (o) and (c) has been done, then, the great thickness of oil film will be at the line FE which can be represented by the segment XE. However, the smallest thickness of oil film is represented by YF. From Fig.(1), the following equation can be formulated:

$$CA = R_2 + h = CD + DA \qquad (1)$$

$$R_2 + h = e\cos(\theta) + R_1\cos(\alpha) \qquad (2)$$

$$h_{max} = C + e \qquad (3)$$

$$h_{min} = C - e \qquad (4)$$

Where h_{max} is the maximum thickness of oil film and h_{min} is the minimum thickness of the oil film. From the triangle (OCA),

$$\frac{e}{\sin(\alpha)} = \frac{R_1}{\sin(\theta)} \qquad (5)$$

Then,

$$\cos(\alpha) = \sqrt{1 - \left(\frac{e\sin(\theta)}{R_1}\right)^2} \qquad (6)$$

Substituting equation (6) in equation (2) assuming $R_1 << 1$ yields:

$$R_2 + h = e\cos(\theta) + R_1\left(1 - \frac{e^2\sin^2(\theta)}{2R_1^2}\right)$$

Therefore ,

$$h = (R_1 - R_2) + e\cos(\theta)$$

$$= c + e\cos(\theta)$$

$$= c\left(1 + \frac{e}{c}\cos(\theta)\right)$$

$$\therefore \quad h = c(1 + \epsilon\cos(\theta)) \qquad (7)$$

Where $\epsilon = e/c$ and $c = R_1 - R_2$.

Oil pressure is given by :

$$P = 3U\eta\,\frac{dh/dx}{h^3}\left(y^2 - \frac{l^2}{4}\right) \qquad (8)$$

Where,

P= oil pressure

U= Surface speed of shaft

dh/dx= Film thickness gradient in the direction of motion

η=dynamic viscosity

h= film thickness

l=axial bearing length

y=c-ordinates in x-direction

from the geometry, x=Rθ ,then, dx=Rdθ , therefore

$$\frac{dh}{dx} = \frac{dh}{d\theta}\left(1/R\right)$$

$$\therefore \quad \frac{dh}{dx} = \frac{-c\epsilon\sin(\theta)\,d\theta}{Rd\theta}$$

$$= -c\epsilon\sin(\theta)/R \qquad (9)$$

Substituting eq.(9) in eq.(8) :

$$P = \frac{3U\eta c\epsilon\sin(\theta)}{Rc^3(1+\epsilon\cos(\theta))^3}\left(\frac{l^2}{4} - y^2\right) \qquad (10)$$

3. Neural Network Analysis

3. 1 Artificial Neural Networks

Artificial neural networks (ANNs) attempt to emulate their biological counterparts. McCulloch and Pitts (1943) [7] proposed a simple model of a neuron, and Hebb (1949) [7] described a technique which became known as 'Hebbian' learning. Rosenblatt (1961) [7], devised a single layer of neurons, called a perceptron , that was used for optical pattern recognition [7].

3.2 Operation of a Single Artificial Neuron

The basic model of a single artificial neuron consists of a weighted summer and an activation (or transfer) function as shown in Fig.(2). This figure shows a neuron in the jth layer, where

x_1 ….. x_i are inputs

w_{j1} …. w_{ji} are weights

b_j is a bias

f_j is the activation function

y_j is the output

The weighted sum s_j is therefore,

$$s_j(t) = \sum_{i=1}^{n} W_{ji}\,x_i(t) + b_j \qquad (11)$$

Equation (11) can be written in matrix form,

$$s_j(t) = W_j X + b_j \qquad (12)$$

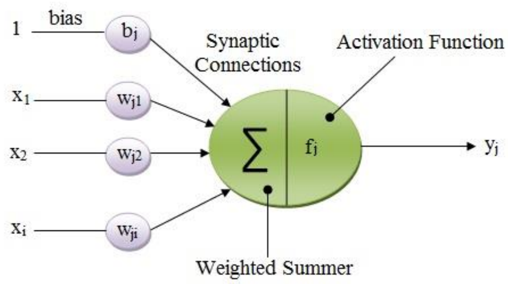

Fig.2 Basic model of a single artificial neuron.

3.3Network Architecture

An ANN is a network of single neuron jointed together by synaptic connections. Fig.(3) shows a three – layer feedforward neural network.

The feedforward network shown in Fig.(3) consists of a two neuron input layer, a one neuron output layer and a three neuron intermediate layer, called a hidden layer. Note that all neurons in a particular layer are fully connected to all neurons in the subsequent layer. This is generally called a fully connected multilayer network, and there is no restriction on the number of neurons in each layer, and no restriction on the number of hidden layers.

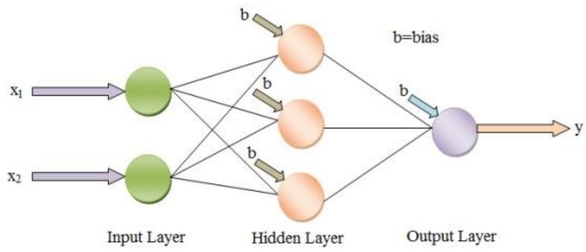

Fig.3 Basic structure of a three – layer feedforward neural network.

4. Experimental Results

Experimental results are obtained at different speeds, 17, 25 ,30 33, 40, 46 and 48 Hz. These results have been done on a journal bearing having groove of ratio w/h<1, w/h=1 and w/h>1 as shown in Figs.(4, 5 and 6) ,where w is the width of groove and h is the depth of groove . The results show that the geometry of the groove have a great effect on the pressure difference generated in the bearing.

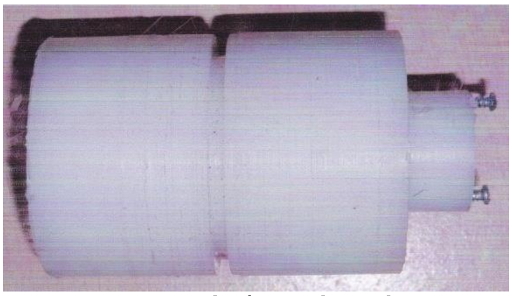

Fig.4 Sample of grooved journal.

4.1 circumference results

For small values of speeds (17Hz), the pressure difference can be reduced by optimum design of groove geometry as shown in Fig.(7). However , in the range of (θ=50-100) and (θ=250-330) ,the bearing without groove gives lower pressure difference (where θ represents the angular position on the bearing). For any range of rotational displacement, there is an optimum value of the ratio w/h. As it is clearly appeared in Fig.(7),in the range of (θ) that lies between 100 and 200 degrees ,the acceptable value of w/h is greater than one, while from θ=0-30 deg. ,the corresponding w/h is that equal to one. These ranges will be varied when the speed will be increased (25Hz) as shown in Fig.(8).In this case ,at the range of θ=75-100 and θ=26-330 the case of without groove bearing is better than that of grooved bearing. Also, in this speed , the behavior of the pressure difference is the same as that of the (17Hz)one. When the speed is increased (30Hz), the grooved bearing gives better behavior except at θ=75-100 with high similarity at all the w/h ranges cases. At the high speeds (30 and 40Hz) , the ratio of w/h>1 gives lower pressure difference except at θ=75-100 where the without groove case has produced better behavior. However, at 40Hz speed case and at the small values of θ , the behavior of w/h=1 case is the best of the other cases. As shown in Fig.(9) , the pressure will fluctuated randomly over all the circumference position when the speed has been set at high magnitudes(46Hz).

4.2 Axial Results

In the axial and at small range of speed (17Hz), the ratio of w/h=1 gives lower pressure except at the positions (15-16mm) and (50-55mm) as shown in Fig.(10). However when the ratio increases to w/h>1, the bearing without groove will be better especially at (15-25mm) positions. When the speed has increased to the values, 25, 30, 33, 40 and 48Hz, the geometry of the groove designed as w/h=1 can give acceptable pressure difference as shown in the Fig.(11).However, when w/h has been increased ,then, the pressure will increase especially at some critical positions as shown in Fig.(12).

4.3 Sound Level Analysis

The sound level is investigated in this paper experimentally. The grooved journal bearing has given a lower dB as shown in Fig.(13) especially at w/h=1. However, when w/h has been increased to greater than one ,then, the sound level will be greater than that of without groove bearing especially at 25-40Hz range of rotational speeds.

4.4 ANN results

Neural network is considered in this work to predict the pressure difference in some ranges of speeds and angular displacements and this is tested at w/h>1 ratio. The results show that the correct optimization of ANN parameters such as training algorithm and the number of hidden nodes and layers will produce a very efficient net that can give a very fit outputs as compared with the experimental targets. From Fig.(14), it can be concluded easily that ANN can give an acceptable prediction curve with correlation coefficient factor (R) of(1) and training Root Mean Square Error (RMSE) of (0.009×10^{-17}) ,Fig.(15).This result is obtained after a very hard optimization which led to choose the Levenberg–Marquardt (trainlm) and one hidden layer of(33) nodes which agree with [8]. For making such comparison ,when the Batch Gradient Descent (traingd) training algorithm is considered, then, the correlation coefficient factor (R) of (0.899) and training Root Mean Square Error (RMSE) of (0.0989) have explained why the prediction curve is deviated hardly from the actual one as shown in Fig.(16). However , this result is obtained after the choice of more than (11) types of training algorithm which are available in [9] and more than (100) values of ANN hidden nodes numbers.

5. Conclusions

Experimental design of grooved journal bearing has been presented. Three types of groove geometry is proposed and investigated in the circumference and axial directions. The results show that:

1. the ratio of groove width to depth has a great effect on the pressure difference generated in the bearing.
2. When the speed is increased, the optimum ratio of groove parameters will be different for any range of angular or axial positions in order to reach the desirable range of oil pressure.
3. The medium ranges of groove parameters ratio can give acceptable sound level (dB).
4. the optimal estimation of the training algorithm and the number of hidden nodes and layers for ANN will led to higher fitting prediction curves and lower error curves. The proposed model has considered a net of two inputs and one hidden layer of (33) hidden node number arranged at one hidden layer.

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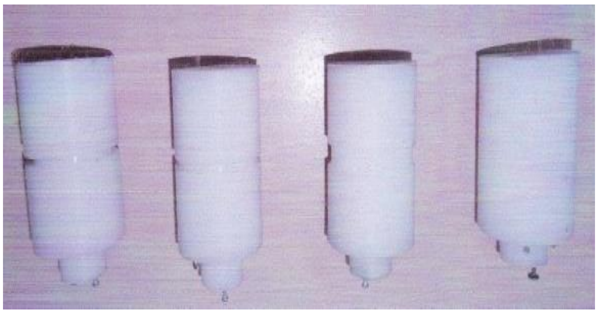

Fig.5 Four types of journal

Fig.6 Testing Rig

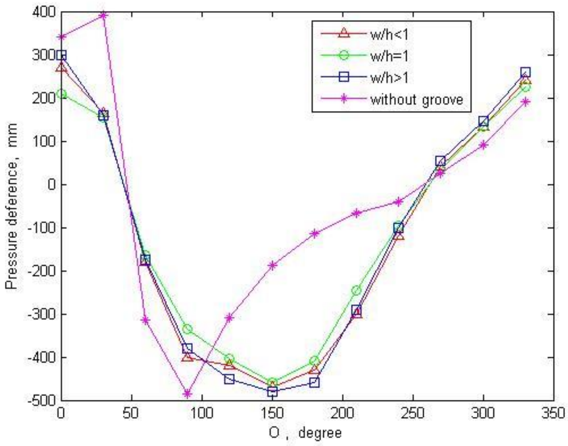

Fig.7 Circumference test at speed=17Hz

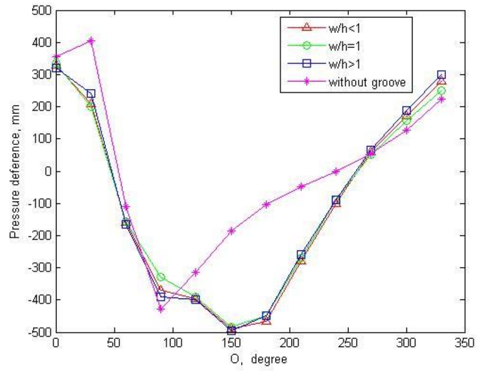

Fig.8 Circumference test at speed=25Hz

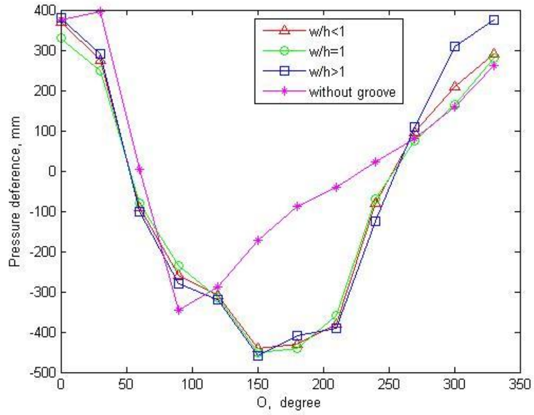

Fig.9 Circumference test at speed=46Hz

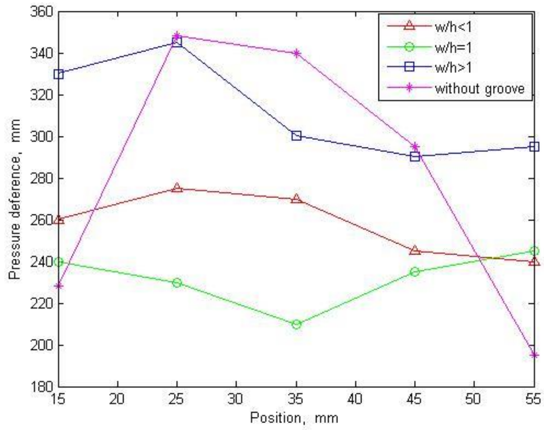

Fig.10 Axial test at speed=17Hz

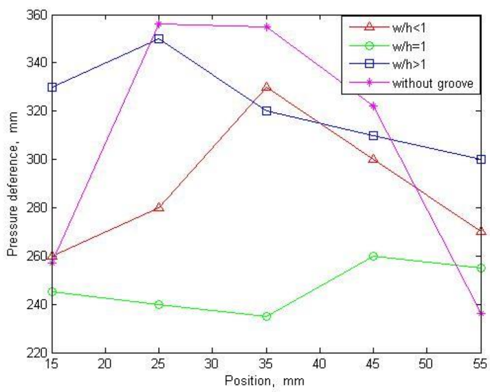

Fig.11 Axial test at speed=25Hz

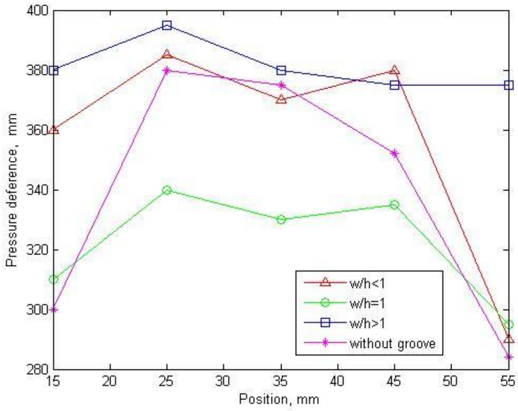

Fig.12 Axial test at speed=48Hz

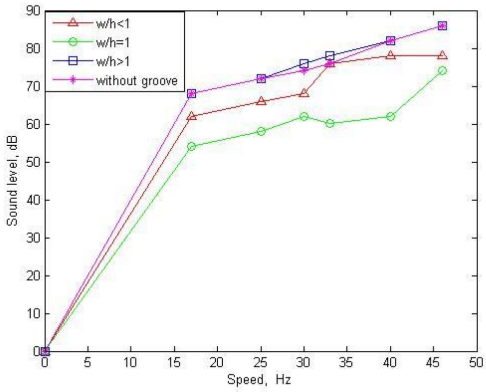

Fig.13 Sound level test at w/h >1

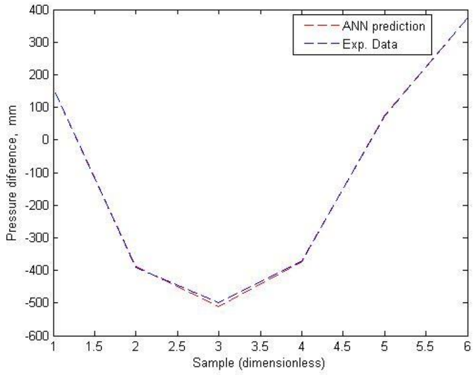

Fig.14 ANN prediction using Levenberg – Marquardt (trainlm)

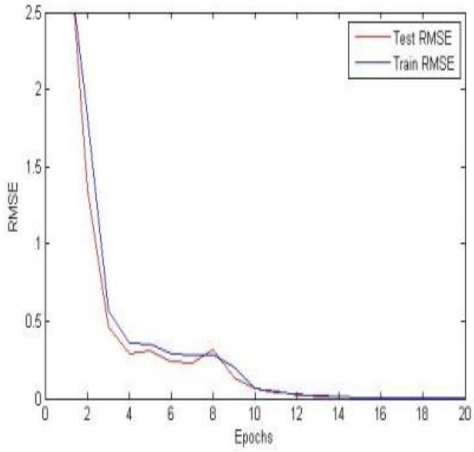

Fig.15 ANN Error Curves using Levenberg – Marquardt (trainlm)

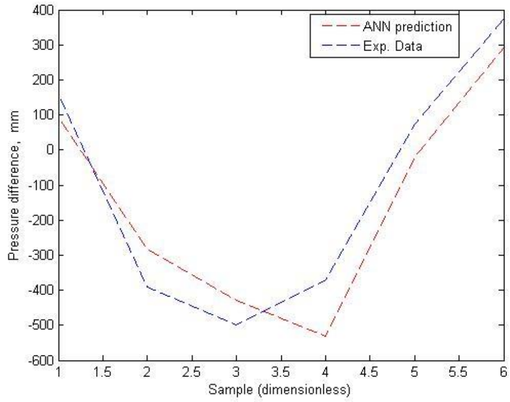

Fig.16 ANN prediction using Batch Gradient Descent (traingd)