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CFD Analysis of Thin Film Lubricated Journal Bearing

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Abstract

The three dimensional CFD analysis were investigated regarding the performance characteristics of a thin film lubricated journal bearing. In the existing literature, several numerical analyses had been reported. Most of these analyses used two dimensional Reynolds equations to find the pressure distribution in the lubricant flow by neglecting the pressure variation across the film thickness. Besides that, most researchers only consider laminar flow. In this paper, three turbulent models which are the Standard k-ε model, Realizable k-ε model and Reynolds Stress Model (RSM) had been used to simulate the characteristics of a plain journal bearing. Three dimensional models had been simulated using ANSYS Fluent software package to accurately predict the performance of the three turbulent models on the journal bearing analysis. Design parameter like static pressure, wall shear stress and dimensionless load carrying capacity were considered and transient analysis was carried out for the analysis with different L/D ratio of 0.25, 0.5, 1.0, 1.5 and 2.0. The results showed that for the cases of thin film lubricated journal bearing, the turbulent models did not give any significant to the simulation results. However for the case of complex geometry, another simulation needs to be conduct to determine an effect of the different turbulent models in the simulation of the lubricant in bearing system.

1. Introduction

Nowadays, the trend in current industries is to use a high speed rotating machinery and carrying heavy rotor load. In each application, one of the significant systems that function to support an applied load by reducing friction between the relatively moving surfaces is a bearing system. In the bearing system, it was found that the lubricant is affected by the gap of bearing [1]. The amount of interaction between the lubricant and solid surface influences the lubricant properties of the bearing clearly. From the literature, a thin film lubricated journal bearing was used to develop positive pressure because of relative motion of two surfaces separated by a fluid. Thin film bearings are those in which although lubricant is introduced, the working surfaces not completely contact each other at least part of the time. These bearing also called Boundary Lubricated Bearing [2].

Recently, following the increasing of technology in computer system, many researchers began to use commercial package of Computational Fluid Dynamics (CFD) in their investigations of performance of the bearing system especially the problem that involved with lubricant [3]. The main advantage of CFD is the researchers can get many

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outputs as part of post-processing stage. The outputs which may get from the CFD simulation are like pressure distribution, velocity distribution and profile, shear stress along the journal surface, plotting the graphs between the different quantities etc. Moreover, the CFD packages are applicable in very complex geometries. There are a number of CFD software package available in the market. Fluent is the most popular and widely used amongst them.

   In this paper, the turbulent flow through journal bearing is analyzed through three turbulence models in order to determine the most suitable models to simulate such cases problems. Moreover, the effect of pressure distribution and wall shear stress occurred on the flow dynamics is studied numerically. A detailed flow insight is presented to explain the characteristics of the thin film lubricant between two surfaces separated by the fluid. One purpose of this study is to determine the static pressure variation on the walls of the bearing. Besides that, the wall shear stress developed on the journal and bearing walls will be carried out numerically. Both analyses for the entire L/D ratio will be present as results of the simulation.

1.1. Journal bearing modeling

   The geometry of a journal bearing used in this study is shown in Fig. 1. Here, the journal rotates with an angular velocity, ω and is in equilibrium position under the external vertical load. The details parameters of the model used in this study will explain later in the rest of this paper.
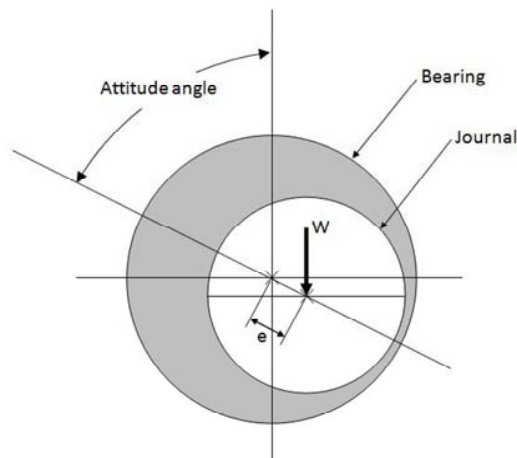

Fig. 1. Schematic diagram of a smooth journal bearing

2. Mathematical Formulation

2.1. Governing equation

   In this study, incompressible, unsteady, and isothermal flow is assumed. Based on these assumptions, the governing equations are as follows:

$$\frac{\partial}{\partial x_j}(\rho \bar{u}_i) = 0 \qquad (1)$$

$$\frac{\partial}{\partial t}(\rho \bar{u}_j) + \frac{\partial}{\partial x_i}(\rho \bar{u}_i \bar{u}_j) = \frac{\partial}{\partial x_i}\left(\mu\left[\frac{\partial \bar{u}_i}{\partial x_j} + \frac{\partial \bar{u}_j}{\partial x_i}\right]\right) - \frac{\partial \bar{p}}{\partial x_j} + \rho g_j + \frac{\partial}{\partial x_i}\tau_{ij} \qquad (2)$$

where u is the velocity component in x_i direction, ρ is the density and p is the static pressure. In the case of RANS modeling, the over-bar denotes time averaging. The stress tensor τ_{ij} is an unknown term representing Reynolds stress

tensor ($\rho \overline{u_i' u_j'}$) in the case of RANS. This additional term in the governing equations is the result of averaging of time dependent Navier-Stokes equations and needs to be modelled in order to achieve closure. Details about transport equation and closure approximation are presented in reference [4]. In this section the three turbulence models used in the numerical simulation are described briefly.

2.1.1. Standard k-ε Model

The standard k-ε model is a two equation eddy viscosity turbulence model [5]. In this model, the eddy viscosity is computed based on the turbulence kinetic energy k, and the turbulence dissipation rate ε via:

$$v_t = C_\mu \frac{k^2}{\varepsilon} \tag{3}$$

Each of these two turbulence scales has its transport equation. The turbulence kinetic energy equation is derived from the exact momentum equation by taking the trace of the Reynolds stress. This equation can be expressed as:

$$\frac{\partial k}{\partial t} + \bar{u}_i \frac{\partial k}{\partial x_i} = v_t \left[\frac{\partial \bar{u}_i}{\partial x_j} + \frac{\partial \bar{u}_j}{\partial x_i} \right] \frac{\partial \bar{u}_i}{\partial x_j} + \frac{\partial}{\partial x_i} \left(\frac{v_t}{\sigma_k} \frac{\partial k}{\partial x_i} \right) - \varepsilon \tag{4}$$

The dissipation rate equation, on the other hand, is obtained using physical reasoning. The equation is:

$$\frac{\partial \varepsilon}{\partial t} + \bar{u}_i \frac{\partial \varepsilon}{\partial x_i} = C_{\varepsilon 1} \frac{\varepsilon}{k} v_t \left[\frac{\partial \bar{u}_i}{\partial x_j} + \frac{\partial \bar{u}_j}{\partial x_i} \right] \frac{\partial \bar{u}_i}{\partial x_j} + \frac{\partial}{\partial x_i} \left(\frac{v_t}{\sigma_\varepsilon} \frac{\partial \varepsilon}{\partial x_i} \right) - C_{\varepsilon 2} \frac{\varepsilon^2}{k} \tag{5}$$

The standard k-ε has five empirical constants C_μ, σ_k, σ_ε, $C_{\varepsilon 1}$ and $C_{\varepsilon 2}$ with values of 0.09, 1.0, 1.3, 1.44 and 1.92, respectively. These values were obtained from experiments and computer optimization. It is worthy noting that these values are not universal and the *k-ε* model requires a certain amount of fine tuning in order to obtain correct results.

2.1.2. The Realizable k-ε Model

Shih and coworkers have proposed a new k-ε model in order to improve the performance of the standard model [6]. In the new model, the eddy viscosity formulation satisfies the mathematical constraint of the positivity of the normal Reynolds stresses. This is achieved by making C_μ variable and sensitizing it to the mean flow and the turbulence (i.e. k and ε). The new dissipation rate equation is:

$$\frac{\partial \varepsilon}{\partial t} + \bar{u}_i \frac{\partial \varepsilon}{\partial x_i} = C_{\varepsilon 1} S \varepsilon + \frac{\partial}{\partial x_i} \left(\frac{v_t}{\sigma_\varepsilon} \frac{\partial \varepsilon}{\partial x_i} \right) - C_{\varepsilon 2} \frac{\varepsilon^2}{k + \sqrt{v \varepsilon}} \tag{6}$$

The model constants σ_ε and $C_{\varepsilon 2}$ have values of 1.2 and 1.9 respectively while $C_{\varepsilon 1}$ is computed from:

$$C_{\varepsilon 1} = max \left[0.43 \frac{\left(S \frac{k}{\varepsilon} \right)}{\left(S \frac{k}{\varepsilon} \right) + 5} \right] \tag{7}$$

where, $S = \sqrt{2 S_{ij} S_{ij}}$ and $S_{ij} = \frac{1}{2} \left(\frac{\partial u_j}{\partial x_i} + \frac{\partial u_i}{\partial x_j} \right)$.

The eddy viscosity is computed as in the standard model via equatian (3). However, in order to ensure the positivity of the normal Reynolds stresses C_μ is no longer a constant and it is computed from:

$$C_\mu = \frac{1}{A_o + A_s U^* \frac{k}{\varepsilon}} \tag{8}$$

where, $A_o = 4.0$, $U^* = \sqrt{S_{ij}S_{ij}\Omega_{ij}\Omega_{ij}}$, $A_s = \sqrt{6} \, cos \left(\frac{1}{3} arccos(\sqrt{6}W) \right)$, $W = \frac{\sqrt{8}S_{ij}S_{jk}S_{ki}}{S^3}$

and vorticity tensor, $\Omega_{ij} = \frac{1}{2}\left(\frac{\partial \overline{u}_i}{\partial x_j} - \frac{\partial \overline{u}_j}{\partial x_i} \right)$.

2.1.3. The Reynolds Stress Model

The Reynolds stress model involves calculation of the individual Reynolds stresses $\overline{u_i' u_j'}$ using differential transport equations [7, 8]. The individual Reynolds stresses are then used to obtain closure of the Reynolds-averaged momentum equation, thus, avoiding the use of the eddy viscosity approximation which was proven to perform poorly in many types of flows. The exact form of the Reynolds stress transport equations can be derived by taking moments of the exact momentum equation. This is a process wherein the exact momentum equations are multiplied by a fluctuating property, the product then being Reynolds-averaged stress. Unfortunately, several of the terms in the exact equation are unknown and modeling assumptions are required in order to close the equations.

The exact transport equations for the transport of the Reynolds stresses, is:

$$\frac{\partial}{\partial t}\left(\rho\overline{u_i'u_j'}\right) + \frac{\partial}{\partial x_k}\left(\rho u_k \overline{u_i'u_j'}\right) = -\frac{\partial}{\partial x_k}\left[\rho\overline{u_i'u_j'u_k'} + \overline{p\left(\delta_{kj}u_i' + \delta_{ik}u_j'\right)}\right] + \frac{\partial}{\partial x_k}\left[\mu \frac{\partial}{\partial x_k}\left(u_i'u_j'\right)\right] -$$
$$\rho\left(u_i'u_k'\frac{\partial u_j}{\partial x_k} - u_j'u_k'\frac{\partial u_i}{\partial x_k}\right) - \rho\beta\left(g_i\overline{u_j'\theta} + g_j\overline{u_i'\theta}\right) + \overline{p\left(\frac{\partial u_\iota}{\partial x_J} - \frac{\partial u_J}{\partial x_\iota}\right)} - 2\mu\overline{\frac{\partial u_\iota}{\partial x_k}\frac{\partial u_J}{\partial x_k}} 2\rho\Omega_k\left(\overline{u_j'u_m'}\varepsilon_{ikm} +$$
$$\overline{u_\iota'u_m'}\varepsilon_{jkm}\right) \tag{9}$$

Some terms in these exact equations, namely the convection, molecular diffusion, stress production, and production by system rotation do not require any modeling. However, the turbulent diffusion, buoyancy production, pressure strain, and dissipation need to be modeled to close the equations. Many approximations have been made to close these equations [8].

3. Simulation

3.1. CFD model description

A 3-dimensional simulation model was developed using the ANSYS Fluent software package. The pre-processor Gambit 2.3 is used for the grid generation. One of the problem during the modeling the model is the the clearance size is very small compared to journal diameter and length. Therefore, a hexahedral cell was used in meshing the model since the tetrahedral cells leads to an enormous number of cells. The number of cells is 86965 and 120 divisions were used in the circumferential direction initially, giving acceptable results for the angle of the maximum pressure.

The dimensions of the journal bearing used in this simulation are: length of the bearing is 133 mm, radial of shaft is 50 mm and radial clearance is 0.145 mm while the eccentricity ratio is 0.61. According to the data, other derived data were obtained which are: Radius of the bearing is 50.145 mm and eccentricity equal to 0.08845 mm. The viscosity of lubricant is 0.0127 Pa s while the density is 840 Kg/m^3.

For the boundary conditions, the operating pressure is set to 101325 Pa. To simplify the geometry, one side of the clearance is used as a lubricant inlet and the other as an outlet. The boundary condition at the lubricant inlet was set as pressure inlet with an appropriate value leading to the right-side lubricant flow rate. For the outlet, the boundary condition was set as pressure outlet with gauge pressure at zero Pascal. The bearing shell and the journal was set as a wall which is in the post processing, the journal was defined as moving wall while the bearing shell as stationary wall. The rotational axis origin is set to the value of eccentricity. When the problem is solved for a constant eccentricity, this value is known. When the problem is solved for a constant external force, the final position of the origin is computed automatically by the dynamic mesh technique. The dynamic mesh model in ANSYS Fluent is used to model flows where the shape of the fluid domain is changing due to motion on the domain boundaries.

4. Result and Discussion

Transient response of a dynamically loaded thin film lubricated journal bearing is carried out in for all turbulent models mentioned above in terms of several analyses which are a pressure analysis, wall shear stress and load carrying capacity.

4.1. Pressure and shear stress analysis

Static pressure variation on the walls of the bearing for all models of simulation is presented in the Fig. 2. From the figure, it can be concluded that all the turbulent models gave a good agreement for each other and here the different model of turbulent didn't give a much different result for the simulation. Similar results were shown for the shear stress, as in Fig. 3 where the results for all turbulent models look like quite similar for all models.
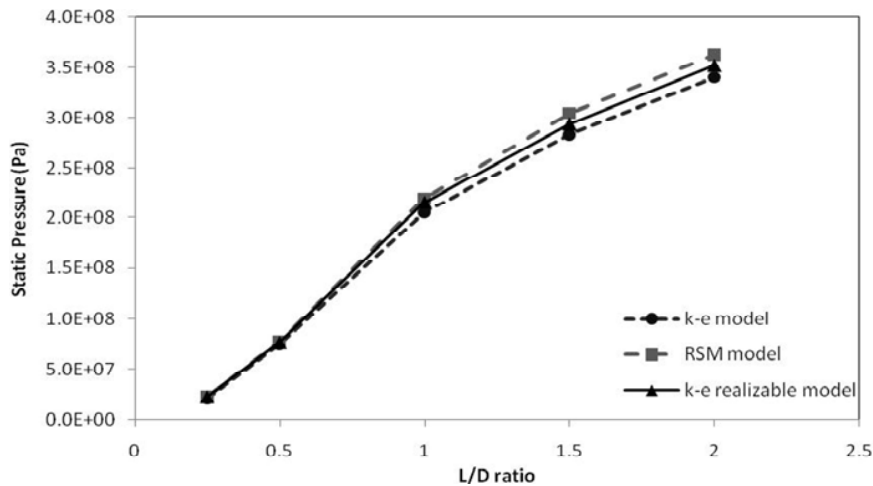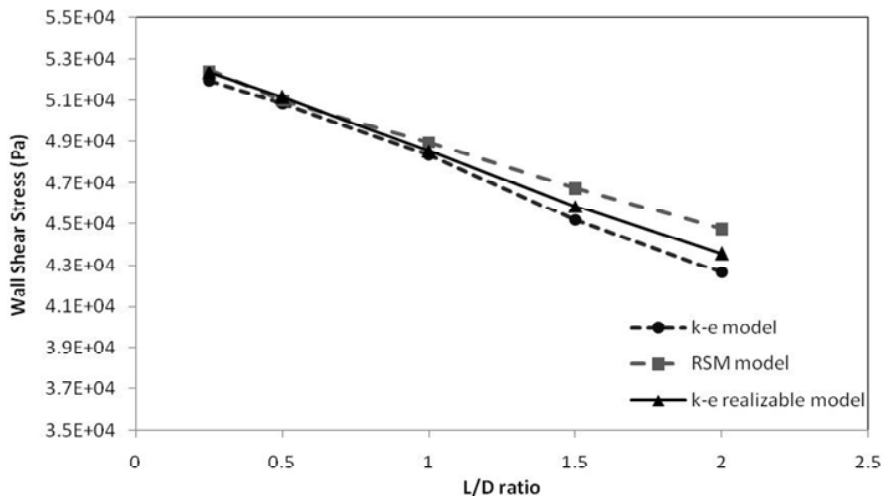

Fig. 2. Static pressure for different L/D ratio

Fig. 3. Wall shear stress for different L/D ratio

4.2. Load carrying capacity

The dimensionless load carrying capacity of the bearing had been found out for different L/D ratio. Similar to the cases of static pressure and shear stress where all turbulent models were used in the simulation to analysis the thin film lubricated journal bearing. These results were presented in Fig. 4-6 for different turbulent models used. The results show that the standard *k-ε* models performed similar to other models which are the realizable *k-ε* and RSM. The unexpected superiority of the realizable *k-ε* over the RSM is justified by the inadequacy of implementing RSM for wall induced shear flows. This requires the grid size to be reduced substantially in order to have a successful RSM computation.
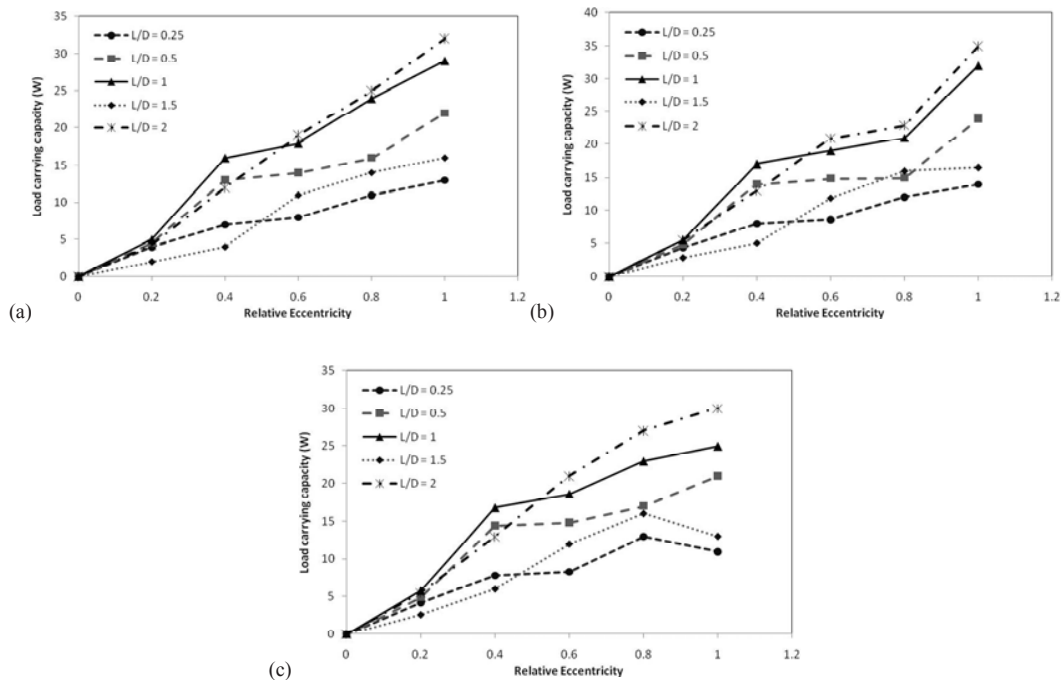

Fig. 4. Load carrying capacity for different turbulent models used in the simulation.(a) standard *k-ε* model; (b) realizable k-ε model; (c) RSM

5. Conclusion

A 3-D numerical investigation of the transient dynamics behavior of thin film lubricated journal bearing had been conducted. For the simulation work, three types of turbulent models were used in order to study the capability of the turbulent models in simulation of thin film lubricated journal bearing. The comparison showed that all models gave similar results in term of performance. Therefore, in this case study it was revealed that, the *k-ε* model was just enough to do the simulation since it was the simplest model compared to others. But, the advantage of *k-ε* model was it can converge faster than RSM and *k-ε* realizable.

Acknowledgements

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