Evaluating the Performance Parameters for the Journal Bearings by using the Graphical and Analytical Methods

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Abstract
This paper illustrates evaluating performance parameters for the journal bearing design by using three different methods. In the first approach, Raimondi and Boyd Charts (Graphical) method will be used. The second approach represents the first analytical method called Reynold equations tables, which can be used by utilizing familiar equations connected by other values. The second analytical method by Reason and Narang (Combined Solution Method) requires using the empirical equation and tables. Finally, the error percentage for all performance parameters for the analytical methods will be compared with the graphical to show which method is more precise. The main goal of obtaining the precise procedure for the journal bearing performance parameters evaluation is to determine the exact temperature rise in journal bearing and the average temperature of the oil film inside the journal bearing. By knowing these temperatures, the designers can select the best type of lubricant oil and bearing material to avoid bearing failure.

Keywords : *Journal Bearings, Lubricant oil, Sommerfeld No., Reynolds Equation, Temperature Rise, Combined Solution, and Leakage Flow Rate.*

I. INTRODUCTION

Clearly, the lubricant film performance illustrates the key factor in the journal bearings design. So, many equations and theories govern the design and operational conditions of the lubricant film. The aim of its design is to produce a journal bearing working in all operational conditions without any interruptions due to some error in design performance parameters. There are two basic aspects of the journal bearing design analysis [1]:

1. The first aspect refers to the basic analysis of the journal bearing load capacity, friction, and lubrication flow rate as a function of load, speed, and any other controlling parameters. This aspect can be found when the lubricant film geometry is defined and then the Reynolds equation is applied to find pressure filed, load capacity, and other parameters.
2. The second aspect relates to practical or operational problems, such as the method of lubricant supply and bearing design to avoid the vibration and cavitation or to allow misalignment and frictional heating of the lubricant.

One of the most important equations governing the pressure in the lubricant film is Reynolds Equation, which represents the simplification of the Reynolds's paper in 1886 [2]. To apply this equation to the lubricant film, many assumptions should be considered [3]–[5]:

1. The lubricant flow between the rotating surfaces should be laminar.
2. The bearing and the journal surfaces should be parallel to neglect the film curvature.
3. Pressure variation across the lubricant film thickness should be zero.
4. The internal force and inertial force should be neglected because they are very small compared with the viscosity and pressure forces.

Now, the equation can be written:

$$\text{Viscosity force} + \text{pressure force} = 0 \quad \text{……. (1)}$$

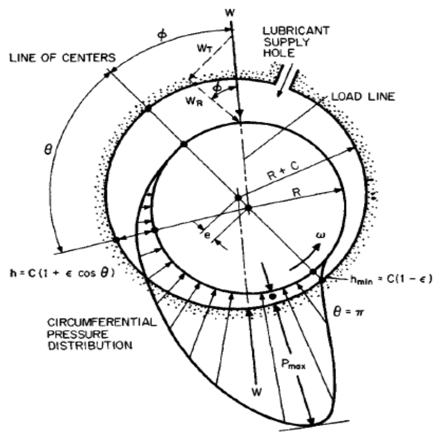
Many works and methods have been examined and designed over centuries from theories, experiments, and practices to manufacture the optimum design for the journal bearings. By studying the key parameters affecting journal bearing performance, Researchers have tried to improve models leading to a more perfect performance during the operation of the journal bearing [2].

In 1883, Towers was sponsored by the Institution of Mechanical Engineering to experiment with the friction in journal bearings. Towers examined many materials in his experiments such us cork and wooden plugs by placing these materials in the loaded zone of the journal bearing crown to stop up the lubricant hole. As a result of his experiment, he found that the oil under considerable pressure removes the plugs from the oil path (hole). This experiment showed the first attempt to investigate the hydrodynamic film pressure of the lubricant oil [2].

In 1886, Reynolds submitted his papers to the Royal Society. Reynolds described the principle by showing that the converging wedge-shaped for the lubricant film was very essential to develop the pressure within the film layer. Equation (2) represents the simplified version of Reynolds equation in a 2-D analysis. These papers represent the classical method for the journal bearing design especially after many.

II. EVALUATING PERFORMANCE PARAMETERS FOR THE JOURNAL BEARING

The graphical method represents the way to calculate the performance parameters for the journal bearing. This method had been developed by Raimondi and Boyd which illustrates the most famous method by using the relationships between the Sommerfeld No. (The dimensionless bearing characteristic number S) and minimum film-thickness variable, the temperature-rise variable, the friction variable, the flow variable, the flow ratio, and the attitude angle. This method stayed convenient till the middle of the last century when the analytical and empirical methods have been found.



Journal Bearing and Bearing Notation [4], [6]

$$\text{Bearing Characteristic No.}, S = \left(\frac{R}{c}\right)^2 \frac{\mu N}{P} \text{.... (4)}$$

Where,

c :- Clearance (in)

R :- Journal Radius (in)

μ:- Lubricant Viscosity (reyn)

N :- Revolution per sec.

P :- Load per unit projected area (lbf\in^2 or Psi)

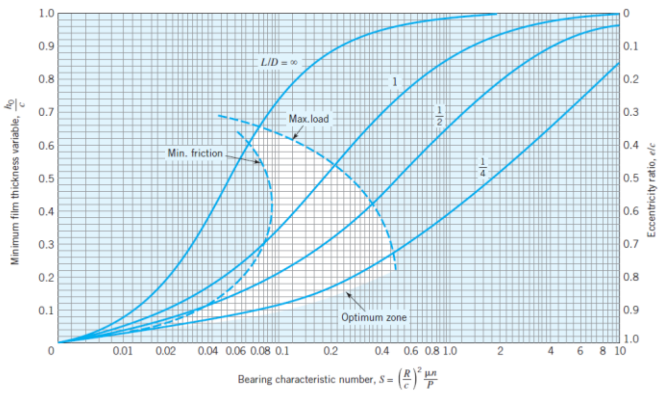
$$P = \frac{W}{L*D} \text{............(5)}$$

W: Load (lbf)

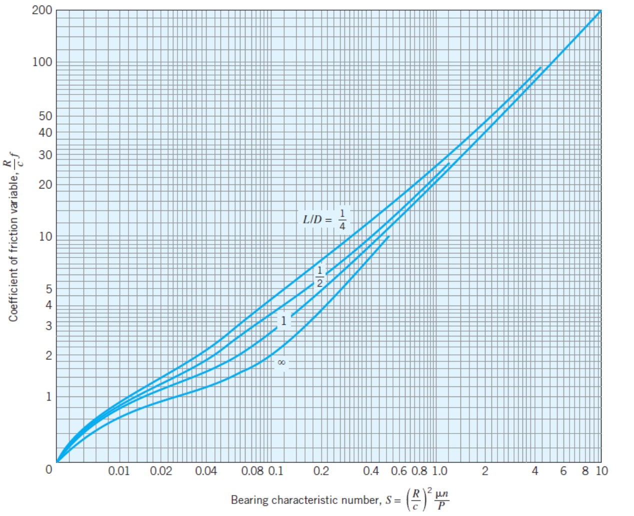
L: JB length (in)

D: JB Diameter (in)

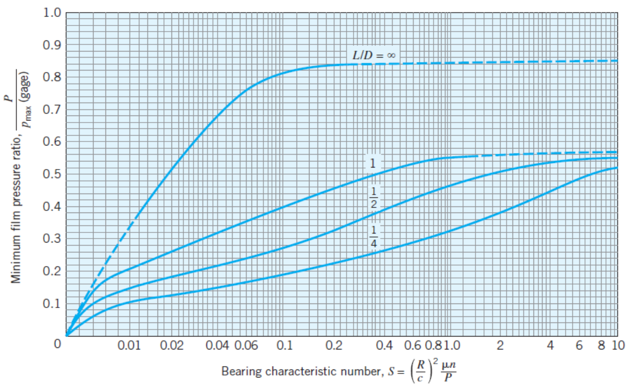
By using the charts below, journal bearing performance parameters can be evaluated:



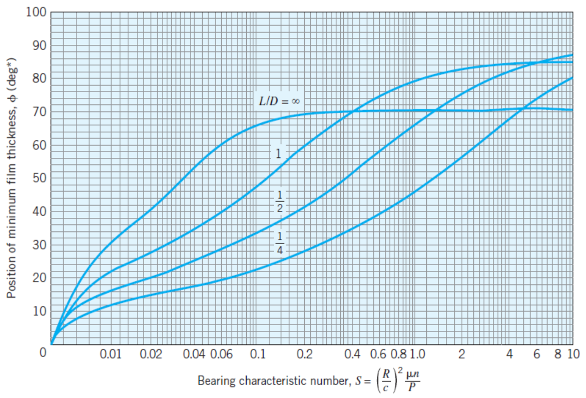
Relationship Between Sommerfeld No. and the Minimum Film Thickness [3], [4], [6]



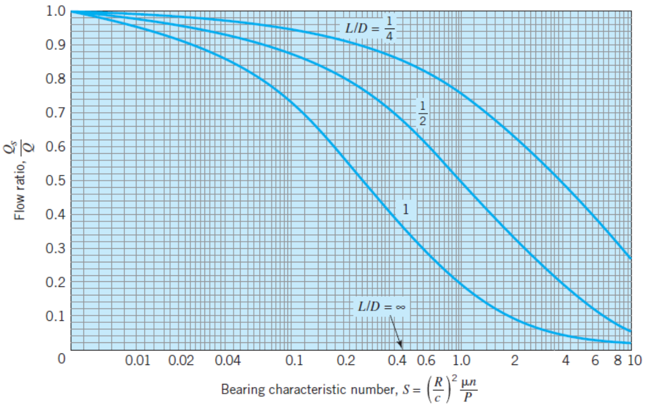
Relationship Between Sommerfeld No. and the Coefficient of Friction [3], [4], [6]



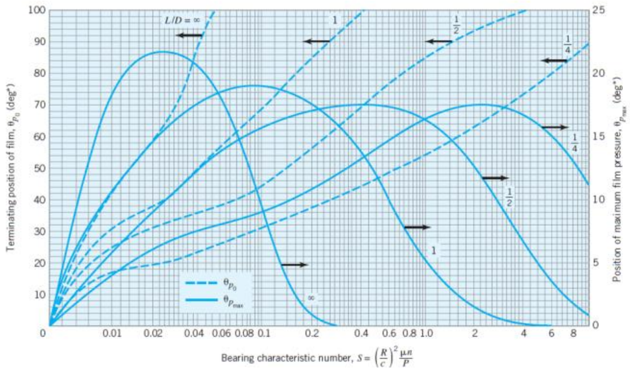
Relationship Between Sommerfeld No. and the Minimum Film Pressure Ratio [3], [4], [6]



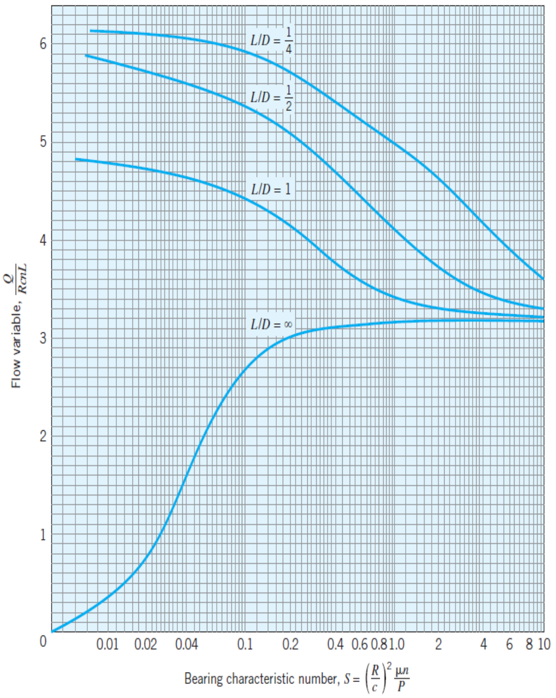
Relationship Between Sommerfeld No. And The Position Of Minimum Film Thickness [3], [4], [6]



Relationship Between The Sommerfeld No. and the Flow Ratio [3], [4], [6]



Relationship Between The Position Of Maximum Film Pressure And The Film Terminating [3], [4], [6]



Relationship Between The Sommerfeld No. And The Flow Variable [3], [4], [6]

The results which evaluated by using the Raimondi and Boyd Charts have an error percentage. Moreover, using charts for calculating the journal bearings performances required many interpolations and extrapolations process which makes this method very tedious. The journal bearings performance parameters have been coded, tabled and become easy to calculated for both short and long journal bearings [7]. Table (1) shows the analytical method for evaluating the performance parameters for the journal bearings by solving Reynold's Equation.

Table (1)

Performance Parameters	Equation
Sommerfeld No. S	$S = \frac{\mu_i N_s LD}{W} * (\frac{R}{C})^2$
Lubricant Oil Leakage Flow Rate Q_{Leak} (in^3/sec)	$Q_{Leak} = \bar{Q}_{Leak} * \frac{\pi}{2} * N_s * LCD$
Lubricant Oil Inlet Flow Rate Q_{Inlet} (in^3/sec)	$Q_{Inlet} = \bar{Q}_{Inlet} * \frac{\pi}{2} * N_s * LCD$
Minimum Lubricant Oil Film Thickness $\bar{h}$	$\bar{h} = 1 - \varepsilon$
Friction Force F (lbf)	$F = f * W$
Power Loss E (Btu/sec)	$E = 2\pi * F * RN_s$
Temperature Rise ΔT (^{o}F)	$\Delta T = \frac{E}{J\rho c_p * (Q_{Inlet} - \frac{Q_{Leak}}{2})}$
$P_{max}, \left(\frac{R}{C}\right) f, \theta_{max}, and\ \emptyset$	From the table directly

Reason and Narang have found a new technique which can be used for journal bearing design in both long journal bearings and short journal bearings. This way can be utilized to design the journal bearings which are effected by steady load. According to the new technique, pressure and other parameters can be evaluated by using the combined solution approximation table (2) which includes several equations which depend on the value of the Sommerfeld No., I_c, I_s, ε, and $\frac{L}{D}$ (Values of empirical table[4].

Table (2)

Performance Parameters	Equation
Sommerfeld No. S	$S = \frac{1}{6\pi * \sqrt{I_s^2 + I_c^2}}$
$\emptyset$	$\emptyset = tan^{-1}(-\frac{I_s}{I_c})$
$\left(\frac{R}{C}\right)f$	$\left(\frac{R}{C}\right)f = 6\pi S * (\frac{\varepsilon I_s}{2} + \frac{\pi}{3 * \sqrt{1-\varepsilon^2}})$
$\frac{Q_{0,\pi}}{RCLN_s}$	$\frac{Q_{0,\pi}}{RCLN_s} = \pi\left[1 \pm \varepsilon \mp \varepsilon\gamma\left(1 - \frac{2\gamma}{\sqrt{1+2\gamma}} tanh^{-1}\frac{1}{\sqrt{1+2\gamma}}\right)\right.$ Where $\gamma = \frac{(1\pm\varepsilon)(2\pm\varepsilon)}{(2+\varepsilon^2)}\left(\frac{D}{L}\right)^2$ $\theta = 0$, use top sign for max. flow $\theta = \pi$, use bottom sign for min. flow
$\frac{Q_s}{Q_0}$	$\frac{Q_s}{Q_0} = 1 - \frac{Q_\pi}{Q_0}$
$\frac{J\rho C^* LD\Delta T}{W}$	$\frac{J\rho C^* LD\Delta T}{W} = \frac{1}{1 - \frac{1}{2}\frac{Q_s}{Q_0}} * \frac{4\pi\left(\frac{R}{C}\right)f}{\frac{Q_0}{RCLN_s}}$

III. CASE STUDY

Determine the performance parameters of a steadily loaded full journal bearing.

Given Data:

D=3 in , L=1.5 in

N=4000 RPM, W= 1000 lbf

Inlet Temp. = 120°F, C=1.5*10^{-3} in, Lubricant oil is SAE 20

ρ= 0.03 lbm/in^3, C^* = 0.40 Btu/lbm °F

JB performance parameters at initial condition when the Inlet Temp. = 120°F:

Table (3)

Parameters / Method	ε	$\emptyset$ rad	$\left(\frac{R}{C}\right)f$	$\frac{Q}{RCLN_s}$
Raimondi and Boyd charts	0.28	70	25.2	3.95
Analytical Method	0.2803 9	69.837	27.882 6	3.948
Combined Solution	0.2852	69.890 2	25.764 6	3.9678 3

Table (4)

Parameters / Method	$\frac{Q_s}{Q_0}$	ΔT (°F)	Error (%)
Raimondi and Boyd charts	0.42	208	------
Analytical Method	0.4188	198.4779	4.578%
Combined Solution	0.4179 3	204.587	1.64%

JB performance parameters at normal operation condition when the lubricant film temperature:

$$T_{ava.} = T_{inlet} + \frac{\Delta T}{2} \text{ .... (6)}$$

Table (5)

Parameters / Method	$T_{ava.}$ °F	$\mu_{ava.}$ Reyn *10^{-6}	S	ε
Raimondi and Boyd charts	224	0.67	0.201	0.718
Analytical Method	219.238 95	0.778	0.233 4	0.687
Combined Solution	220.793 5	0.7626	0.228 6	0.666

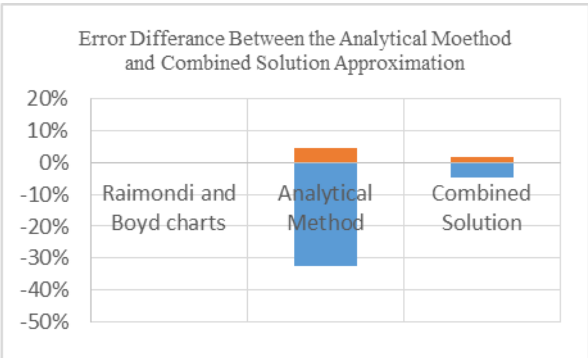
Table (6)

Parameters / Method	ΔT °F	Error (%)
Raimondi and Boyd charts	38.438	--------
Analytical Method	51.054	-32.68%
Combined Solution	40.177	-4.524%

Table (7)

Parameters / Method	$\frac{Q}{RCLN_s}$	$\frac{Q_s}{Q_0}$	$\emptyset$ rad	$\left(\frac{R}{C}\right)f$
Raimondi and Boyd charts	5.12	0.8	44	5.75
Analytical Method	5.097	0.795	41.73 8	6.293
Combined Solution	5.3367	0.537	43.33	6.2911

Chart (1)



IV. CONCLUSION

It is obvious that using different methods to obtain the journal bearing performance parameters have many advantages because it will show the researchers which method is very precise to use in journal bearings design. In this paper, the comparison between the two analytical methods and the graphical method, which it evaluation results represent the research guide reference, showed that the results which came from the combined solution approximation method have less error percentage than the analytical method. According to the table (4), the error percentage between the temperature rise at the initial conditions illustrate that the value of all methods are approximately close but the combined solution approximation method have the more precise evaluation due to the few error rate (1.64%) comparing with the guide reference results. As well as, table (7) and chart (1) show the same results which they proof that the combine solution approximation method is the perfect way to calculate the performance parameters for the journal bearings with the minimum error rate (-4.524%, 1.64%). While, the analytical method recorded error rate (-32.68%, 4.578%).

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