The prediction of liquid film journal bearing performance with a consideration of lubricant film reformation

Part 1: theoretical results

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Analyses of liquid film journal bearings rarely consider the reformation of the lubricant film. This reformation normally takes place in the vicinity of a supply groove and is influenced by, amongst other parameters, the lubricant supply pressure. In a previous paper the authors have described in detail the implementation of an algorithm to locate automatically and efficiently the locus of film rupture and reformation boundaries using a digital computer. In the present paper results are presented for a wide range of important variables. In particular, the prediction of lubricant supply rate is studied carefully and compared with data presented in a widely used design aid which does not account in detail for the influence of film reformation.

NOTATION

a	groove width
b	bearing width
c	radial clearance
d	bearing diameter ($= 2r$)
p_f	supply pressure
$\bar{p}$	dimensionless supply pressure $[= p_f(c/r)^2/\eta\Omega]$
q_a	groove length flow factor
q_w	groove width flow factor
Q_s	side leakage flow rate
$\bar{Q}_c$	dimensionless flow into cavitation region
$\bar{Q}_s$	dimensionless side leakage flow $[= Q_s/cbU, = \bar{Q}_{gr}]$
Q_p	pressure induced flow rate (4)
$\bar{Q}_p$	dimensionless pressure induced flow rate (4)
Q_v	velocity induced flow rate (4)
$\bar{Q}_v$	dimensionless velocity induced flow rate (4)
U	shaft surface velocity
w	groove (circumferential) length
W	load capacity
$\bar{W}$	dimensionless load capacity $[= W(c/r)^2/b\eta\Omega]$
ϵ	eccentricity ratio
η	dynamic viscosity
ψ	attitude angle
Ω	angular velocity

1 INTRODUCTION

Most plain journal bearing design procedures incorporate data which have been generated with an inadequate consideration of lubricant flow effects in the vicinity of the supply grooving. For example, it has been common in analysis to consider the lubricant to be supplied at ambient pressure along a 'line' groove extending the full width of the bearing. Only a few research workers have given attention to the phenomenon of film reformation which takes place in the vicinity of the supply grooving.

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It is now widely accepted that a more careful analysis of flow conditions at lubricant inlet to a bearing is required if acceptably accurate predictions of lubricant flowrate are to be obtained. A consideration of such inlet flow conditions is also important in the wider context of the prediction of the thermal operating characteristics of a bearing.

The incorporation of film reformation effects in the numerical analysis of a journal bearing is not an easy matter. In a previous publication (1) the authors have highlighted the difficulties and identified the contributions of a number of researchers. Specifically the suggestion of a novel algorithm to enable analytical consideration of both lubricant rupture and reformation was examined in detail. This cavitation algorithm was proposed by Elrod and Adams (2) and refined by Elrod (3), and invoked a technique which did not require explicit recourse to the boundary conditions on the cavitation region although a correct solution was in accord with them. A detailed study of the algorithm was undertaken encompassing a wide range of factors which might influence the numerical analysis process.

The implementation of the Elrod and Adams cavitation and algorithm in an automatic, reliable and robust software package (1) presented the opportunity to generate design data. Such data are presented in the present paper and compared with those already available in a widely used journal bearing design scheme (4).

Experimental studies were also carried out to enable a correlation to be made with the analytical data and the results of such work are reported in Part 2 of the paper.

2 THE ESDU DESIGN ITEM No. 66023

The plain journal bearing design document marketed by ESDU International Limited (4) is one of the most comprehensive of its type currently available. The Engineering Sciences Data Unit first published the Item

in 1966. Substantial aspects of the technological information contained in the Item still remain acceptable and up to date. However, it is widely recognized that the thermal balance procedure adopted to determine the effective and outlet lubricant temperatures, and in particular to predict the maximum temperature of the bearing material, is not adequate in its detail. In considering the data generated in the present paper it was felt important to compare and contrast it with other available design information. The data presented in ESDU Item No. 66023 provided a reliable and convenient comparison. Whereas the results to be presented by the authors were obtained with a consideration of lubricant film reformation, the data presented in ESDU Item No. 66023 did not have such a deliberation in their basis. It is important, therefore, to identify clearly the important assumptions and approaches of this Item to enable an effective and meaningful comparison to be made.

The data in the ESDU Item No. 66023 is presented in a dimensionless form. Some important features in its derivation and implementation are:

1. The data for dimensionless load capacity, $\bar{W}$, are taken to be independent of the supply pressure (i.e. for zero gauge supply pressure) and the grooving configuration. The data for load capacity (and attitude angle, ψ) is presented as a function of eccentricity ratio ϵ for varying length to diameter ratio, b/d .
2. The dimensionless lubricant flowrate required theoretically to ensure a fully flooded bearing once again is presented as a function of ϵ and b/d . However, its magnitude is dependent upon the grooving arrangement.
3. The actual lubricant flowrate is assumed to be the sum of two individual components:
 - (a) A velocity induced flowrate for which the lubricant supply pressure is not considered. Lubricant is envisaged to be drawn into the clearance space by the shearing action of the shaft motion over the groove. In its non-dimensional form this component, $\bar{Q}'_s$, is actually the normalized side leakage determined for fully flooded lubrication. The actual side leakage component, Q_s , is determined by multiplying the value obtained from the analysis considering fully flooded lubrication by the factor a/b , the ratio of the width of the groove to the width of the bearing. For a bearing with no supply pressure such a correction can be construed as an attempt to allow for the finite width of the groove or alternatively the formation of the reformation interface (Fig. 2a). The effectiveness of this correction factor will be considered later.
 - (b) A pressure induced flow rate, $\bar{Q}'_p$, in its dimensionless form, for which shaft rotational effects are neglected. This component enables the determination of the flow forced into the clearance space from the groove by virtue of the supply pressure.

Although the determination of the total flowrate by the sum of these two components can only be an approximation, since the boundary conditions for the two solutions differ, it is a technique which has been widely adopted since first proposed by McKee (5). The experimental evidence presented in Part 2 of this

paper has been found to lend support to the approach.

4. In establishing a thermal balance approach to enable the calculation of an effective operating temperature and hence viscosity, it is assumed that any recirculating lubricant is ejected after traversing once around the bearing. Such an assumption may clearly lead to error in the prediction of lubricant temperatures. This aspect of journal bearing design will not receive consideration in the present paper. It is, nevertheless, intimately bound up with the prediction of lubricant flow components.

3 RANGE OF VARIABLES CONSIDERED

In developing the numerical analysis scheme to implement the cavitation algorithm of Elrod and Adams (1) the authors chose initially to consider the case of a bearing with an axial groove located at the position of maximum film thickness. Normally this is an impractical location for the supply, although solutions based upon this configuration have previously been adopted as the basis for journal bearing design and are an acceptable approximation for a number of practical grooving situations. However, the adoption of such a groove position considerably reduced the computational time necessary to develop the analytical scheme, since iterations to achieve a required attitude angle were unnecessary. Since the study was designed to provide a firm basis on which improved predictions of lubricant flowrate in plain journal bearings could be evolved, it was felt important to consider this groove location and to develop a firm understanding before moving on to examine more complicated grooving arrangements. Thus both the theoretical results of Part 1 of this paper and the experimental data of Part 2 are related to a full 360° bearing with a square-ended axial groove at the maximum film thickness position.

Figure 1 details the range of the variable parameters covered in generating predictions of bearing performance. The parameters considered were; the bearing width to diameter ratio, b/d ; the ratio of the groove to bearing width, a/b ; the ratio of groove circumferential length to bearing diameter, w/d ; the eccentricity ratio, ϵ ; and the non-dimensional supply pressure, $\bar{p}_t$. All combinations of the parameters indicated in Fig. 1 were covered and also specific additional cases were computed when necessary e.g. for the development of the design charts presented later.

The performance characteristics computed included

- The dimensionless pressure distribution
- The dimensionless load-carrying capacity, $\bar{W}$
- The attitude angle, ψ
- The dimensionless flowrate from the groove, $\bar{Q}_{gr}$
- The dimensionless flowrate from the sides of the bearing ($\bar{Q}_s$ should equal $\bar{Q}_{gr}$)
- The dimensionless flowrate into the cavitation region, $\bar{Q}_c$
- The location of the rupture and reformation boundaries

Figure 2 shows typical graph plotter outputs for two solutions identifying the rupture and reformation boundaries defining the cavitation region. The results

Developed bearing area indicating notation

Parameter	Values considered
b/d	0.5, 0.75, 1.0
a/b	0.3, 0.5, 0.8, 1.0
w/d	0.2
Eccentricity ratio, ϵ	0.2, 0.4, 0.6, 0.8
Non-dimensional supply pressure, $\bar{p}_f$	0.0, 0.1, 0.25, 0.5, 1.0

Fig. 1 Range of variables considered

(a) Shaft rotation from left to right
 Normalized supply pressure = 0.000
 $b/d = 0.500$ $\epsilon = 0.600$
 $a/b = 0.500$ $w/d = 0.200$

(b) Shaft rotation from left to right
 Normalized supply pressure = 0.500
 $b/d = 0.500$ $\epsilon = 0.600$
 $a/b = 0.800$ $w/d = 0.200$

Fig. 2 The location of the reformation and the rupture boundaries for two different sets of operating parameters

presented here will be limited to a consideration of predicted load-carrying capacity, attitude angle and side leakage rate and a comparison with the data presented in reference (4). Fuller details of the solutions undertaken may be found in reference (6). In addition to the consideration of these results design charts for predicting load-carrying capacity and flowrate will be presented.

4 RESULTS AND DISCUSSION

4.1 Load capacity and attitude angle

Results for dimensionless load capacity and attitude angle from the present analysis were compared with the data of ESDU Item No. 66023 (4) for all combinations of the values of the parameters shown in Fig. 1. For non-reformation conditions ($\bar{p}_f = 0$, $a/b = 1$) the correlation observed for both $\bar{W}$ and ψ was excellent at all values of b/d and ϵ considered. For more realistic lubricant inlet conditions the agreement was dependent upon the values of the operating parameters. Figures 3, 4 and 5 highlight the important features of the comparison.

Figures 3 and 4 show the comparative results for dimensionless load capacity and attitude angle as a function of eccentricity ratio for varying dimensionless supply pressure, with ($b/d = 0.5$, $a/b = 0.8$ and $w/d = 0.2$). The discrepancies are more significant at low values of eccentricity ratio, ϵ , and increase with increasing normalized supply pressure, $\bar{p}_f$. This may be interpreted as a consequence of the hydrostatic pressures in the region of the groove, the significance of which as regards load capacity will depend upon their magnitude in comparison with the hydrodynamically generated pressures. The latter will clearly be smaller at lower eccentricity ratios. The pattern of results as identified in Figs. 3 and 4 was unchanged at higher values of b/d , the discrepancies being slightly smaller as b/d increased.

Fig. 3 Comparison of results for load capacity with the predictions of ESDU Item No. 66023

$$b/d = 0.5, a/b = 0.8, w/d = 0.2$$

Fig. 4 Comparison of results for attitude angle with the predictions of ESDU Item No. 66023

$$b/d = 0.5, a/b = 0.8, w/d = 0.2$$

Figure 3 serves to emphasize that for realistic operating eccentricity ratios, say 0.6–0.8, the effect of the supply pressure upon the load-carrying capacity would invariably be minimal. However, for lightly loaded bearings care must be exercised as regards its influence. This is true not only in regard to the static load capacity but also with regard to the dynamic characteristics of a bearing. The potential effect of lubricant supply pressure upon the whirl characteristics of lightly loaded bearings is widely recognized. This is evidenced by the substantial effect of $\bar{p}_f$ upon $\bar{W}$ at low ϵ in Fig. 3, an effect consequential upon the changed pressure pattern around the journal which in turn will influence vibrational behaviour.

Figure 5 shows the effect of groove width, expressed through the ratio a/b , upon dimensionless load-carrying capacity with zero gauge supply pressure and ($b/d = 0.5$, $w/d = 0.2$). For a/b equal to or greater than 0.8 the agreement of dimensionless load capacity with the data of ESDU Item No. 66023 is excellent. For values of groove to bearing width ratio less than 0.5 significant discrepancies are apparent, particularly at lower eccentricity ratios. This is as a result of an incomplete lubricant film downstream of the groove as the reformation of the film occurs. This observation highlights the importance of a correct consideration of film reformation when lubricant is supplied through a hole.

The comparison of the data generated with that presented in reference (4) may be summarized as follows.

1. For zero gauge pressure ($\bar{p}_f = 0$) the agreement in values of dimensionless load capacity is good with $a/b \geq 0.5$ for all ϵ and b/d .
2. At non-zero values of $\bar{p}_f$ the discrepancy in $\bar{W}$ is dependent on $\bar{p}_f$ and ϵ , being largest with high dimensionless supply pressure and low eccentricity. With $\epsilon > 0.3$ and $\bar{p}_f \leq 0.25$, a high value of dimensionless supply pressure, the maximum difference in the values of normalized load capacity was about 10 per cent.
3. The correlation in attitude angle is good with all b/d and ϵ for $\bar{p}_f = 0$ and $a/b \geq 0.8$. At non-zero values of $\bar{p}_f$ significant discrepancies can occur. However, for practical values of eccentricity ratio and dimensionless supply pressure ($\epsilon \geq 0.6$, $\bar{p}_f \leq 0.25$) the maximum difference in attitude angle was 8° .

It will be noted from the dimensionless form of supply pressure [$\bar{p}_f = p_f(c/r)^2/\eta\Omega$] that high values will tend to be caused by large supply pressure and clearance ratio, and low dynamic viscosity (e.g. process fluid lubrication) and angular velocity. The table below gives three examples of situations resulting in a dimensionless supply pressure ($\bar{p}_f = 0.25$):

Supply pressure p_f kN/m ²	Clearance ratio c/r	Dynamic viscosity η Ns/m ²	Angular frequency r/min	Dimensionless supply pressure $\bar{p}_f$
100	0.001	0.025	153	0.25
200	0.003	0.025	2750	0.25
100	0.001	0.001	3820	0.25

Fig. 5 Comparison of results for load capacity with the predictions of ESDU Item No. 66023

$$b/d = 0.5, w/d = 0.2, \bar{p}_f = 0$$

Fig. 6 Comparison of results for dimensionless side flow from the present analysis with the predictions of ESDU Item No. 66023 at zero supply pressure

$b/d = 0.5$, $w/d = 0.2$

4.2 Side leakage flow

The approach of the ESDU Item No. 66023 to the determination of actual flowrate has been outlined in a previous section. A pressure induced flow and a velocity induced flowrate are summed. A comparison of these data with the results obtained by the authors is most readily undertaken in the first instance when the supply pressure p_i is taken to be zero. In reference (4) the total flow is then equal to the velocity induced flow, and as has already been pointed out, this is actually equal to the side leakage. In fact the dimensionless side leakage of ESDU Item No. 66023, Q'_s , is that which would be determined by an analysis assuming the bearing to be fully flooded. In determining the actual flowrate a correction factor accounting for the width of the groove was incorporated. Thus in terms of the non-dimensionalization adopted here for flowrate $\bar{Q}_s$, it may readily be demonstrated that the data of reference (4) may be represented by:

$$\bar{Q}_s = \frac{a}{b} \cdot \frac{2}{\pi} Q'_s$$

Thus for the case of zero supply pressure a direct comparison of the authors' data for side leakage flow with that predicted by reference (4) is possible. The comparison is shown in Fig. 7 where dimensionless side flow, $\bar{Q}_s$, is plotted as a function of eccentricity ratio, ϵ , for varying a/b with $b/d = 0.5$ and $w/d = 0.2$. The correlation at a value of $a/b = 0.8$ is almost perfect. Even at the condition of worst agreement ($a/b = 0.3$, $\epsilon = 0.8$) the discrepancy is only about 12 per cent. A similar trend was observed for the other values of b/d considered,

however, there was a slight tendency for the discrepancies to increase with increasing b/d for given values of a/b and ϵ .

It will be readily recognized that the correlation exhibited in Fig. 6 is a direct result of the incorporation of the correction factor a/b to account for the finite width of the groove in reference (4). The engineering judgement shown in the introduction of this factor has been demonstrated to be remarkably good by the more precise calculations of side leakage considering film reformation carried out by the authors. The excellent agreement at what is one of the most common values of groove to bearing width ratio ($a/b = 0.8$) gives confidence in the flow data of the Item.

For non-zero values of supply pressure comparison of the authors' flow data with those of reference (4) is not directly possible in normalized terms. For this reason a specific bearing was chosen to examine the correlation. The details of the bearing and lubricant were as follows:

$d = 50$ mm

$b/d = 0.5$

$c/r = 0.0015$

$a/b = 0.4$ and 0.8

$w/d = 0.2$

$p_i = 150$ kN/m² and 300 kN/m²

$\Omega = 83.78$ rad/s (800 r/min)

Kinematic viscosity of lubricant:

25.88 cSt (25.88×10^{-6} m²/s) at 40°C

4.64 cSt (4.64×10^{-6} m²/s) at 100°C

Density of lubricant:

867.5 kg/m³ at 20°C

For each combination of $\bar{p}_i$ and a/b and at a variety of loading conditions the side flowrate was determined

Fig. 7 Comparison of results for side flow with the predictions of ESDU Item No. 66023 at a supply pressure of $p_i = 150$ kN/m²

using ESDU Item No. 66023. The effective viscosity of the oil determined from this procedure for each case was then used to determine $\bar{p}_f$ for the authors' computations, which were, of course, carried out for the same eccentricity ratio.

A comparison of the results for a supply pressure of $p_f = 150 \text{ kN/m}^2$ is shown in Fig. 7. At a value of $a/b = 0.8$ the correlation is again excellent. For $a/b = 0.4$ the agreement is less good, discrepancies of about 10–16 per cent being observed. For $\bar{p}_f = 300 \text{ kN/m}^2$ good correlation was obtained for both values of groove to bearing with ratio, the maximum discrepancy being about 5 per cent.

The comparisons of side leakage rate determined by the authors with their film reformation analysis and that predicted by reference (4) with its more pragmatic approach have proved remarkably good for the range of conditions investigated. Whilst it must be recalled that the agreement is only for the case of a single, square-ended, axial groove located at the maximum film thickness position, the results do lend support to the approach adopted for flow determination in ESDU Item No. 66023.

5 CHARTS FOR PREDICTING DIMENSIONLESS LOAD CAPACITY AND SIDE LEAKAGE RATE

Data charts for predicting the dimensionless load capacity and side leakage rate for the range of conditions examined were developed.

5.1 Load capacity

There were five independent parameters involved: b/d , ϵ , a/b , w/d and $\bar{p}_f$. The parameter w/d was not expected to influence significantly the predicted load capacity if it was confined to the range 0.1–0.3. The charts were therefore developed from solutions with $w/d = 0.2$ and were subsequently checked against random parameter value solutions with w/d varying in the range indicated.

Attempts were made to incorporate the four remaining independent parameters in a single chart but these were unsuccessful. A series of charts were therefore developed for specific values of the normalized supply pressure, $\bar{p}_f$. The construction of a load capacity chart for $\bar{p}_f = 1$ proved impossible for acceptably accurate predictions at low values of a/b and ϵ (6). Thus Figs. 8, 9, 10 enable the determination of dimensionless load capacity for values of $\bar{p}_f = 0.0, 0.25, 0.50$ respectively for the following range of the other parameters:

$$\epsilon = 0.2-0.8 \quad a/b = 0.3-0.8$$

$$b/d = 0.5-1.0 \quad w/d = 0.1-0.3$$

The way the charts should be used is indicated on each one. The separate charts were tested for at least twenty-five random combinations of the independent parameters. The maximum errors in prediction of dimensionless load capacity compared with the computed values were approximately as follows:

Chart $\bar{p}_f = 0.0$: Error ± 2 per cent

Chart $\bar{p}_f = 0.25$: Error ± 3 per cent

Chart $\bar{p}_f = 0.50$: Error ± 3 per cent
(or ± 7 per cent for $\epsilon < 0.4$, $a/b < 0.5$)

Fig. 8 Dimensionless load capacity chart: valid for w/d in the range 0.1–0.3 and $\bar{p}_f = 0.0$

Fig. 9 Dimensionless load capacity chart: valid for w/d in the range 0.1–0.3, a/b in the range 0.3–0.8 and $\bar{p}_f = 0.25$

Fig. 10 Dimensionless load capacity chart: valid for w/d in the range 0.1–0.3 and $\bar{p}_r = 0.5$

It is interesting to note that for the $\bar{p}_r = 0.25$ chart the influence of a/b is insignificant within the error bounds indicated. For this reason a/b does not appear as a parameter on the chart.

5.2 Side leakage rate

The construction of data charts to predict dimensionless side leakage proved more difficult than for dimension-

less load capacity (6). After considerable trial and error the following charts were evolved. Figure 11 permits the determination of side leakage for a bearing with $a/b = 0.8$ and $w/d = 0.2$. The range of the other independent variables covered in the charts are

$$\varepsilon = 0.2-0.8$$

$$b/d = 0.5-1.0$$

$$\bar{p}_r = 0.0-1.0$$

The way the chart is to be used is clearly indicated.

It did not prove possible to incorporate the effect of varying a/b and w/d over the total range of these variables covered. However, the influence of varying w/d in the range 0.1–0.3 upon dimensionless side leakage for a fixed $a/b = 0.8$ may be estimated through determination of q_w , the groove width flow factor, from Fig. 12. Similarly, the effect of varying a/b in the range 0.3–0.8 for a fixed value of $w/d = 0.2$ may be considered through the groove length flow factor, q_a , shown in Fig. 13. Thus the side leakage for $a/b = 0.8$ and $w/d = 0.2$ may be determined from Fig. 11, $\bar{Q}_s$, CHART, and for values of a/b and w/d different from these but satisfying the limitations indicated above the dimensionless side leakage, $\bar{Q}_s$, will be given by:

$$\bar{Q}_s = \bar{Q}_{s, \text{CHART}} \times \text{flow factor } (q_w \text{ or } q_a)$$

The accuracy of prediction of side leakage flow using Figs. 11–13 was checked against computed conditions for more than sixty random combinations of the parameters involved. The error was normally in the range ± 2 per cent to ± 4 per cent, with a maximum error detected of ± 5 per cent.

6 CONCLUSIONS

1. The implementation (1) of a cavitation algorithm (2, 3) to enable film rupture and reformation in a journal bearing to be considered has permitted the predicted performance to be evaluated for a wide range of operating parameters.

Fig. 11 Dimensionless side flow chart: valid for $a/b = 0.8$ and $w/d = 0.2$

Fig. 12 Groove width factor q_w , valid for $a/b = 0.8$ only

Fig. 13 Groove length flow factor q_L , valid for $w/d = 0.2$ only

2. The analysis has been restricted to the case of a single, square-ended, axial groove located at the position of maximum film thickness. Data have here been presented for dimensionless load capacity, attitude angle and side leakage flow.
3. Comparison of the data obtained with that presented in a comprehensive design document (4) has revealed that:
 - (a) A consideration of more realistic flow conditions at lubricant inlet will, as would be expected, not normally influence the value of predicted load capacity significantly. However, particular care must be exercised in predicting load capacity without a consideration of film reformation for small values of eccentricity ratio, e , and groove bearing width ratio, a/b , and large values of dimensionless supply pressure $\bar{p}_t$.
 - (b) The introduction of a correction factor, a/b , in reference (4) when determining the side leakage flow has been shown to be an effective approach to account for film reformation effects if data for non-reformation conditions are employed.
4. Design charts enabling the accurate prediction of dimensionless load capacity and side leakage rate with a consideration of film reformation have been presented for a wide range of the independent variables.

5. The predictions of bearing performance evaluated in Part 1 of this paper have been investigated experimentally. The results are presented in Part 2 of the paper.

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