The performance of journal bearings with a single axial groove

J C P CLARO and A A S MIRANDA, MSc, PhD
The University of Minho, Portugal

SYNOPSIS A computer program has been developed that can be used in the analysis of steadily loaded journal bearings with a single groove at any location with respect to the load line. The program incorporates Elrod's cavitation algorithm which ensures mass conservation in both the full film and the cavitated regions enabling rupture and reformation boundaries to be located automatically. Computer results are plotted and checked against experimental measurements for the cases of a single axial groove located at the load line and at 90° to the load line. A comparison of results of this method of analysis with the performance predictions of ESDU Item No.84031 (1984) for bearings with a single axial groove at 90° to the load line is presented.

NOTATION

a	groove length
b	bearing length
c	radial clearence
c_d	diametral clearence
d	shaft diameter
H	power loss
$\bar{H}$	dimensionless power loss $= H\cdot(c_d/d)/b\eta U^2$
N	speed of shaft rotation
p_f	supply pressure
$\bar{p}_f$	dimensionless supply pressure $= p_f\cdot(c_d/d)^2/\eta\Omega$
Q_s	side flow rate
$\bar{Q}_s$	dimensionless side flow rate $= Q_s/cbU$
t_i	lubricant inlet temperature
U	shaft surface velocity
w	groove (circumferential) width
W	bearing load
α	angular coordinate measured from the position of maximum film thickness in the direction of rotation of the journal
α_c	angular location of the cavitated boundary
α_g	angular location of the supply groove
ε	eccentricity ratio
η	dynamic viscosity
Ψ	attitude angle
Ω	angular velocity

1 INTRODUCTION

Accurate performance predictions of mechanical components at the design stage are crucial if an optimized design is required or when operating conditions are expected to be close to the limits. For oil lubricated hydrodynamic journal bearings flow rate and power loss in the bearing are important performance parameters which determine the lubricant temperature rise and hence its viscosity, which in turn affects the minimum film thickness and the bearing load carrying capacity. In particular, for a correct prediction of lubricant flow rate, reformation of the lubricating film in the vicinity of the supply groove should be taken into account. The film of lubricant is not continuous all around the bearing. Film rupture usually occurs soon after the film profile begins to diverge. The disrupted film will reform in the vicinity of the supply groove, depending on the lubricant supply conditions. Figure 1 shows the location of the reformation boundary (for a given groove position and geometry) for two limiting values of supply pressure: (a) zero; and (b) a high value. Reformation considerations make theoretical analyses based on computer solutions complex and time consuming. That appears to be the reason why most of the available data for the design of hydrodynamic journal bearings is based on solutions of Reynolds' equation which adopted simplified reformation boundary conditions (sketched on Figure 1-c), usually not satisfaying mass flow continuity. This is the case of the data given in the design procedure described in ESDU Item No.84031 (1), a commonly used procedure for the design of steadily loaded hydrodynamic journal bearings. It has been shown (2,3,4,5) that this might result in inaccurate predictions of flow rate in the bearing.

In a previous work (6) a numerical analysis technique for the study of journal bearings with a single axial groove located at the position of maximum film thickness has been developed which incorporated Elrod's cavitation algorithm (7), enabling rupture and reformation boundaries to be located automatically and allowing the introduction of the lubricant supply conditions. This technique has now been extended by the authors to allow for the analysis of bearings with a single axial groove at any location with respect to the load line or two diametrically opposed axial grooves. The lubricant was considered to be isoviscous.

The aim of this paper is to present and discuss the correlation observed between the performance predictions of ESDU Item No.84031 (1) for bearings with a single axial groove located at 90° to the load line, and those obtained using a computer program which takes full account of the lubricant supply conditions. Computer results were checked against experimental measurements available.

2 COMPARISON OF COMPUTER RESULTS WITH EXPERIMENTAL MEASUREMENTS

Published experimental data available were mainly for journal bearings with a single axial groove on the load line (2, 8, 9, 10). Experimental data for other groove locations (11, 12) is scarce, especially for the case of a single groove located at 90° to the load line. Clayton (8) has presented a few experimental results for flow rate and friction coefficient for this case, but they were not suitable for comparison over a range of variation of parameters.

Computer results were checked against experimental measurements of Dowson et al (10) for the case of a single groove on the load line (Figures 2 and 3), and of Claro and Miranda (13) for the case of a single groove at 90° to the load line (Figure 4). The experimental conditions for each case are shown on Table 1.

Table 1 – Experimental conditions for Figures 2, 3 and 4

Operating parameters	Figures 2 and 3 (10)	Figure 4 (13)
d (m)	0.1016	0.05
b/d	0.75	0.606
c_d/d	$1.25\cdot10^{-3}$	$5.23\cdot10^{-3}$
a/b	0.875	0.406
w/d	0.047	0.210
W (N)	2200 to 15 600	392.5
N (r/min)	1500	672 to 1900
p_f (kN/m²)	275	100
η (Ns/m²) at: 40°C	0.0250	0.0684
100°C	0.0049	0.0081
Groove at	load line	90° l.l.

The correlation observed was good for both flow rate and power loss. The maximum discrepancies on flow rate were of 16.6 per cent and 21 per cent for the cases represented on Figures 2 and 4, respectively. The maximum discrepancy observed on power loss (Figure 3) was 11 per cent.

3 COMPARISON OF COMPUTER RESULTS WITH THE PREDICTIONS OF ESDU ITEM No.84031

Predictions of flow rate and power loss of ESDU Item No.84031 (1) for bearings with a single axial groove at 90° to the load line, were compared with computer results. Various bearing and groove geometries, and supply pressures, were considered.

For non-zero values of supply pressure a direct comparison on dimensionless terms was not possible. Therefore, dimensional values of side flow rate and power loss were calculated for supply pressures of 50, 100, 150 and 200 kN/m² with bearing geometry and operating conditions as follows:

d = 0.05 m
N = 800 r/min
b/d = 0.5
$c_d/d = 1.5\cdot10^{-3}$
a/b = 0.8 and 0.4 ('square'hole)
w/d = 0.2
t_i = 40°C
oil viscosity at:
40°C = 0.0390 Ns/m²
100°C = 0.0058 Ns/m²
(ISO VG 46)

3.1 Flow rates

The comparative results for dimensionless flow rate at zero supply pressure are shown on Figures 5 and 6. For fixed (b/d) and (a/b) the flow rate predictions of Reference (1) were always higher than those of the computer analysis. The discrepancies observed increased with increasing values of (ε), (a/b) and (b/d). At a/b=0.8 and ε=0.9, these discrepancies were of 15 and 26 per cent for values of (b/d) of 0.4 and 0.8, respectively.

A more detailed study of these discrepancies was undertaken. Computer results of dimensionless flow rate for a/b=1 and w/d=0 ('line'groove with axial length equal to the bearing length) showed to agree within 2 per cent with the predictions of Reference (1). Equally good correlation was observed for values of (a/b) smaller than unity with w/d=0. Therefore, it was concluded that the discrepancies mencioned above for pratical groove geometries were mainly due to the consideration, or not, of the hydrodynamic pressure over half the width of the groove. The computer analysis assumed a constant pressure inside the groove and on the groove boundaries equal to the lubricant supply pressure. On the other hand, the data used on Reference (1) to calculate side flow rates at zero supply pressure does not take into account the width of the groove, originating higher predictions of flow rate.

It is worth noting the effectiveness of the introduction of the correction factor (a/b) by the procedure of Reference (1) to calculate side flow rates, accounting for the length of the groove. This approach has also proved to give good results for bearings with a single groove located at the position of maximum film thickness (14).

The predictions of side flow rate for values of supply pressure of 50 and 100 kN/m² are shown on Figures 7 and 8, respectively. For low values of (a/b) significant discrepancies between the computer results and the predictions of Reference (1) were observed, which decreased with increasing values of eccentricity ratio. At a/b=0.8, a commun practical value, with low to moderate supply pressures (p_f<150 kN/m²) and values of eccentricity ratio of practical interest (ε>0.5), the correlation observed was acceptable. At p_f=100 kN/m² (Figure 8) the maximum discrepancy under those conditions was 9 per cent.

At higher supply pressures (p_f>150 kN/m²) significant discrepancies were observed for all values of (a/b) and (ε). For instance, at p_f=200 kN/m² and ε=0.75 the computer results for side flow rate were 26 per cent and 30 per cent greater than the predictions of Reference (1), for values of (a/b) of 0.4 and 0.8, respectively.
The reason for these discrepancies

was not completely understood and the matter is being investigated further. It was thought, however, that they could be the result of a combination of two effects: (a) the discrepancies which occur at zero supply pressure (Figures 5 and 6); and (b) the hydrostatic effect of the supply pressure on the groove region which was accounted for in the computer analysis, resulting on higher attitude angles and, therefore, higher clearences at the groove location than those determined according to Reference (1). Figure 9 shows the effect of supply pressure on the attitude angle. A similar effect has been reported for the case of a single groove located at the position of maximum film thickness (14).

3.2 Power loss

Predictions of power loss were compared using results for: (i) zero supply pressure, with b/d = 0.4 and 0.8; and (ii) supply pressures of 50, 100, 150 and 200 kN/m², with b/d = 0.5.

Figures 10 and 11 show comparative results for zero supply pressure (with b/d=0.4) and for p_f=100 kN/m², respectively.

The correlation observed was acceptable for both zero and non-zero values of supply pressure.

At p_f=0 the discrepancies only had some significance for high values of eccentricity ratio (ε>0.8), increasing with increasing values of (a/b). Maximum discrepancies were observed for ε=0.9 and a/b=0.8, being of 10 per cent for b/d=0.4 and 15 per cent for b/d=0.8.

At non-zero values of supply pressure, the observed discrepancies were mainly affected by (ε) and (p_f). A maximum discrepancy of 15 per cent was observed for p_f=150 kN/m², at ε=0.9.

4 CONCLUSIONS

Results of side flow rate and power loss for hydrodynamic journal bearings with a single axial groove at 90° to the load line provided by a computer analysis taking full account of the lubricant inlet conditions, have been compared with the predictions of ESDU Item No.84031 (1).

(i) Computer results have been checked against available experimental measurements. Good correlation was observed.

(ii) Results of the computer analysis for dimensionless side flow rate have been compared with those of Reference (1). At zero supply pressure, the correlation observed was good for low values of (ε) and (a/b). An increase of these parameters has resulted in increased discrepancies which were higher for high values of (b/d). For a/b=0.8 and ε=0.9, the discrepancies were of 15 per cent and 26 per cent for values of (b/d) of 0.4 and 0.8, respectively. Such differencies have been explained by the fact that the procedure of the ESDU document uses a correction factor to account for the length of the groove but neglects its width. Greater values of (w/d) will originate higher discrepancies on side flow rate.

(iii) For non-zero values of supply pressure and low values of (a/b), the correlation on side flow rate was generally not good. At a/b=0.8, a commun practical value, with low to moderate supply pressures (p_f<150 kN/m²) and values of (ε) of practical interest (ε>0.5), the correlation observed was acceptable. The maximum discrepancy under these conditions was 9 per cent, at p_f=100 kN/m². At higher supply pressures significant discrepancies have been observed for all values of (a/b) and (ε). The discrepancies occurring at non-zero values of supply pressure appeared to be due to the combination of two effects: (a) the conditions originating the discrepancies mentioned in (ii) above; and (b) the hydrostatic pressure on the groove region, which has been accounted for in the computer solution, resulting on higher attitude angles and, therefore, higher clearences at the groove location than those predicted by Reference (1).

(iv) The predictions of power loss of Reference (1) showed good agreement (within 10 per cent) with the computer results for commun values of bearing operating parameters. However, at very high eccentricity ratios (ε>0.8) higher discrepancies have been observed, with a maximum of 15 per cent at ε=0.9.

The study presented in this paper is one part of a broader research project concerning the analysis of hydrodynamic journal bearings fed via axial grooves, which includes an experimental programme for validation of the theoretical results.

ACKNOWLEDGEMENTS

The authors wish to thank Dr.C.M.Taylor of Leeds University for his interest on this project, in particular the helpful discussions had at various stages of the work and for providing access to published works.

The present work is part of a research project financially supported by JNICT (Portugal) under contract No.87.70/MATR.

REFERENCES

(1) Calculation methods for steadily loaded axial groove hydrodynamic journal bearings, Engineering Sciences Data Unit (ESDU International Ltd.) Item No.84031, 1984

(2) COLE,J A and HUGHES,C J; Oil flow and film extent in complete journal bearings, Proc.I.Mech.Engs., 1956, 170, pp 499-510

(3) McCALLION,H, LLOYD,T and YOUSIF, F B; The influence of oil supply conditions on the film extent and oil flow in journal bearings, Proc.I.Mech.Engs., Tribology Conv., Isle of Man, 1971, pp 31-37

(4) ETSION,I and PINKUS,O; Analysis of short journal bearings with new upstream boundary conditions, Trans.A.S.M.E, J.Lub.Tech., 1974, 96,3, pp 489-496

(5) ETSION,I and PINKUS,O; Solution of finite journal bearings with incomplete films, Trans.A.S.M.E., J.Lub.Tech., 1975, 97, 1, pp 89-100

(6) DOWSON,D, MIRANDA,A S and TAYLOR, C M; Implementation of an algorithm enabling the determination of film rupture and reformation boundaries in a liquid film bearing, Proc. 10th. Leeds-Lyon Symposion on Tribology, 1984, pp 60-70 (Butterworths)

(7) ELROD,H G; A cavitation algorithm, Trans.A.S.M.E., J.Lub.Tech., 1981, 103, pp 350-354

(8) CLAYTON,D; An exploratory study of oil grooves in plain bearings, Proc.I.Mech.Engs., Appl.Mech. 1946, 155, pp 41-49

(9) McKEE,S A; Oil flow in plain journal bearings, Trans.A.S.M.E., 1952, 84, pp 841-848

(10) DOWSON,D, HUDSON,J D, HUNTER,B and MARCH,C N; An experimental investigation of thermal equilibrium of steadily loaded journal bearings, Proc.I.Mech. Engs., 1966-67, 181, part 3B, pp 70-80

(11) DOWSON,D, MIRANDA,A S and TAYLOR, C M; The prediction of liquid film journal bearing performance with a consideration of lubricant film reformation, Part II - experimental results, Proc.I.Mech.Engs., Part C, 1985, 199, No.C2, pp 103-111

(12) WOOLACOTT,R G and MACRAE,D; The performance at high speed of complete plain bearings with two axial oil-inlet grooves, NEL Report No.326, 1967

(13) CLARO,J C P and MIRANDA,A A S; umpublished experimental work

(14) DOWSON,D, MIRANDA,A S and TAYLOR, C M; The prediction of liquid film journal bearing performance with a consideration of lubricant film reformation, Part I - theoretical results, Proc.I.Mech.Engs., Part C, 1985, 199, No.C2, pp 95-102

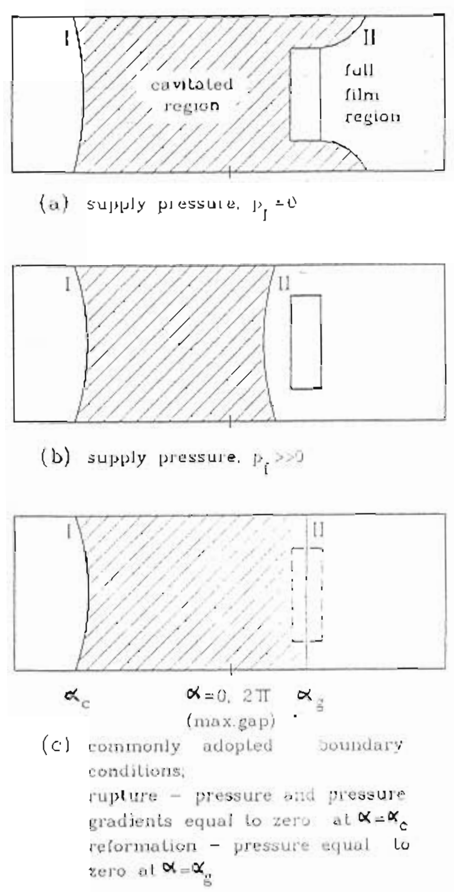

Fig 1 A sketch of the extent of the cavitated region in a journal bearing with a single axial groove. Direction of shaft rotation from left to right (I—cavitation boundary; II—reformation boundary)

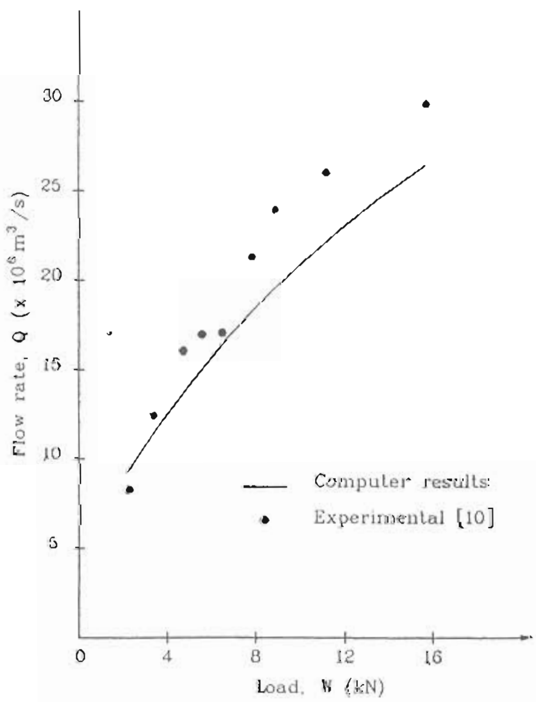

Fig 2 Comparison of predictions of flow rate with experimental results for a single axial groove on the load line

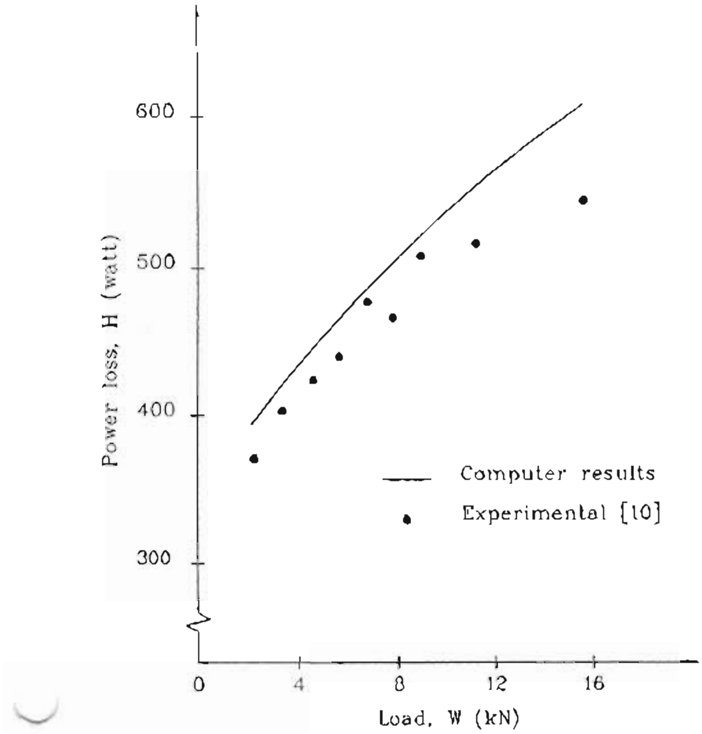

Fig 3 Comparison of prediction of power loss with experimental results for a single axial groove on the load line

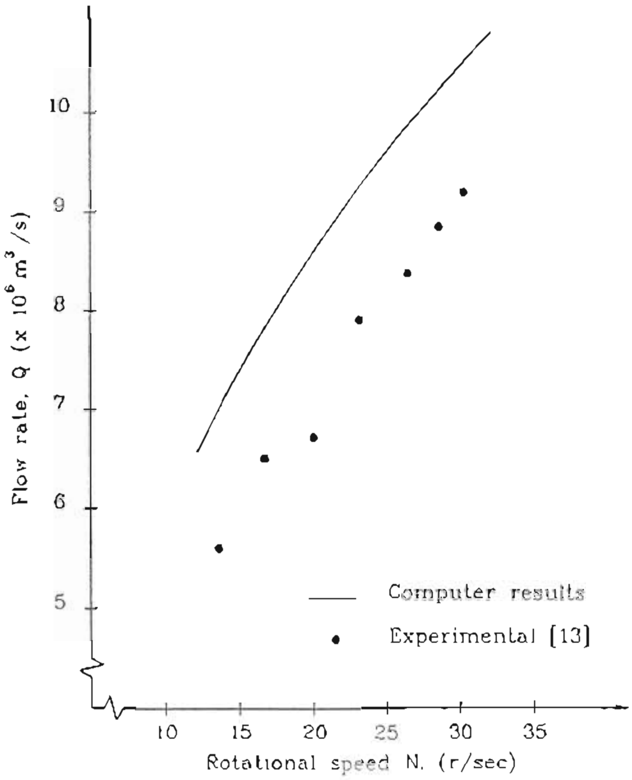

Fig 4 Comparison of predictions of flow rate with experimental results for single axial groove at 90° to the load line

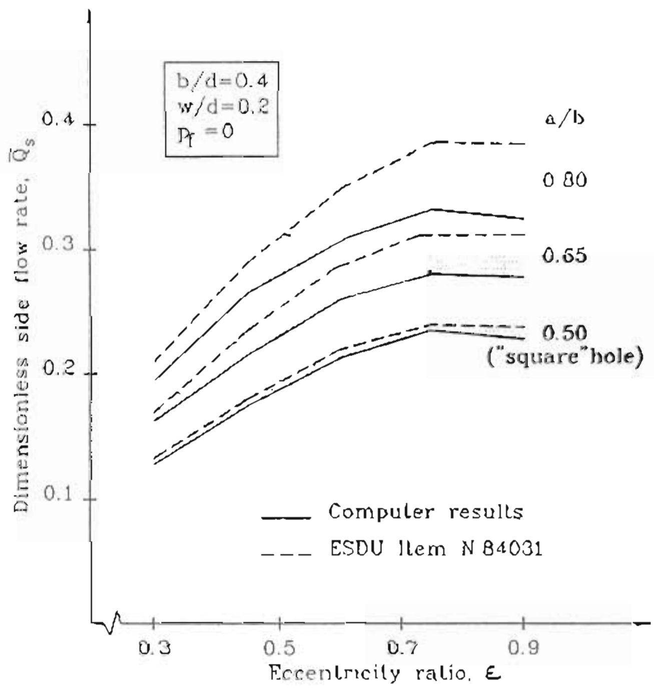

Fig 5 Comparison of predictions of dimensionless side flow rate (b/d = 0.4)

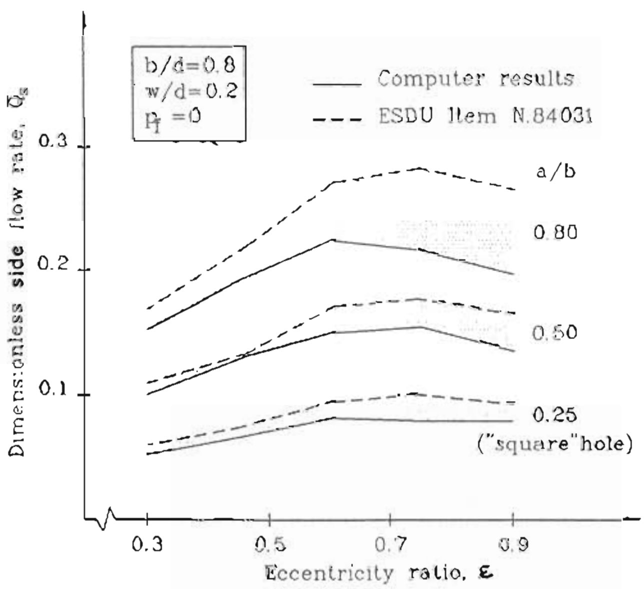

Fig 6 Comparison of predictions of dimensionless side flow rate (b/d = 0.8)

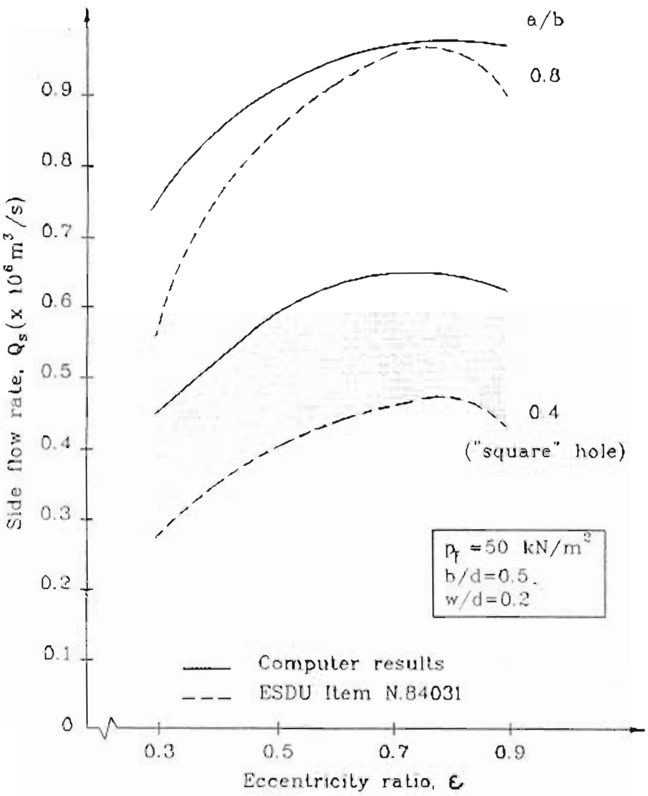

Fig 7 Comparison of predictions of side flow rate (p_f = 50 kN/m²)

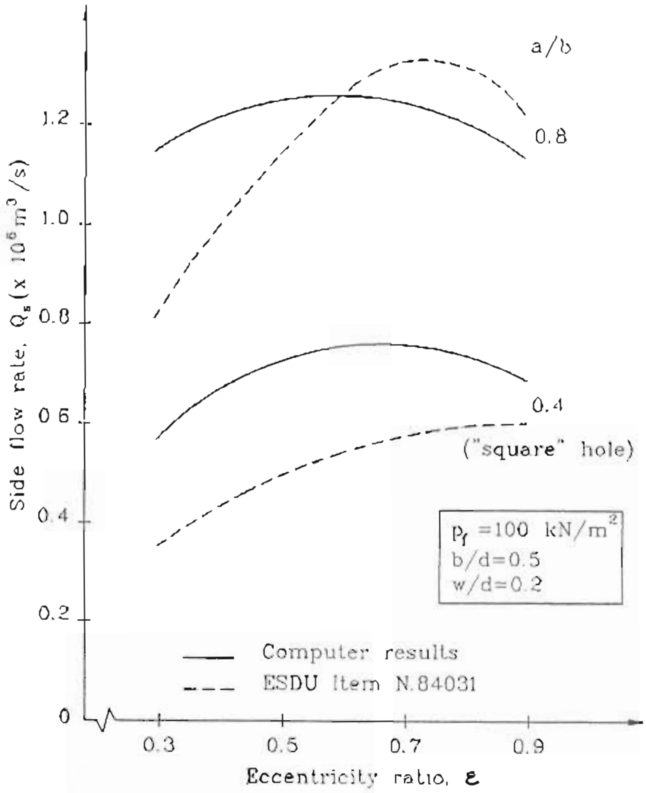

Fig 8 Comparison of predictions of side flow rate (p_f = 100 kN/m²)

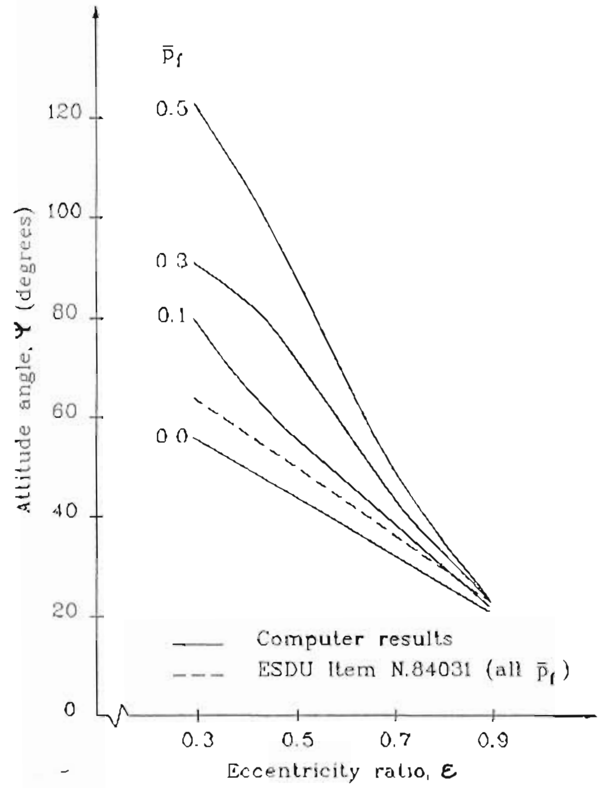

Fig 9 Influence of dimensionless supply pressure on attitude angle (a/b = 0.8; w/d = 0.2)

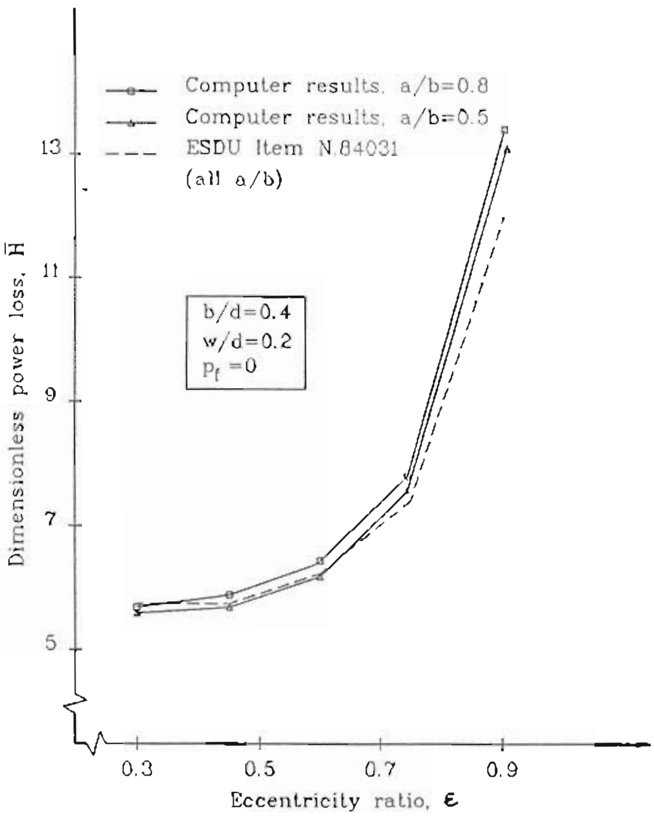

Fig 10 Comparison of predictions of dimensionless power loss (P_f = 0)

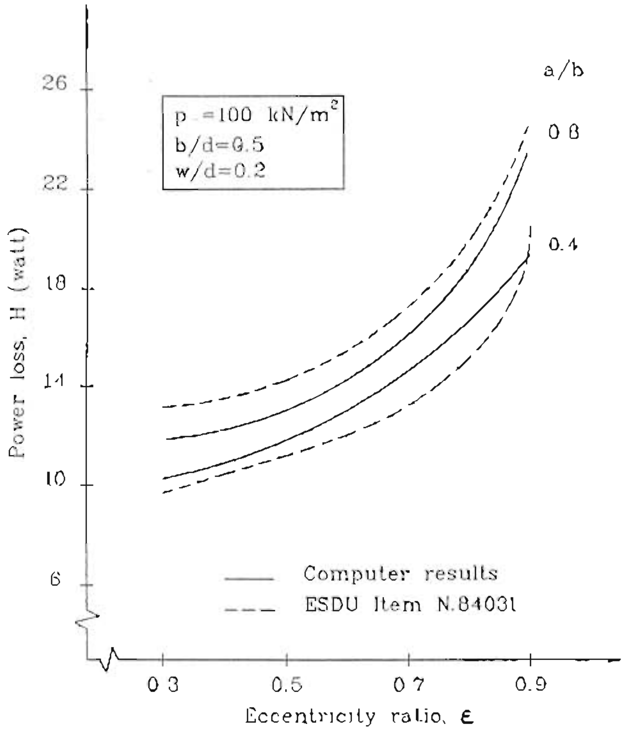

Fig 11 Comparison of predictions of power loss (p_f = 100 kN/m^2)