International Journal of Rotating Machinery
1999, Vol. 5, No. 3, pp. 155–165
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Two-Phase Air/Oil Flow in Aero-Engine Bearing Chambers – Assessment of an Analytical Prediction Method for the Internal Wall Heat Transfer

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(Received in final form 11 April 1997)

The present paper gives a theoretical outline on liquid film flows driven by superimposed effects of interfacial shear and gravity forces and discusses related heat transfer processes which are relevant for lubrication oil systems of aero engines. It is shown that a simple analytical approach is able to predict measured heat transfer data fairly well. Therefore, it offers scope for improvements within the analysis of bearing chamber heat transfer characteristics as well as for appropriate studies with respect to other components of the lubrication oil system such as vent pipeline elements.

Keywords: Bearing chamber, Heat transfer, Film thickness, Velocity profile, Shear driven, Gravity

INTRODUCTION

An efficient secondary air/lubrication oil system is an important demand for further improvements of performance characteristics and capabilities of modern jet engines. In order to reduce pressure losses in the secondary air system and to meet the requirements given by increasing thermal loads, sufficient knowledge on discharge characteristics and heat transfer coefficients are necessities within the design process. As a consequence, these subjects are currently investigated in worldwide research activities (Wittig and Schulz, 1992; Owen, 1992) and satisfying results have already been elaborated for many of these components. However, this holds solely as far as single phase air flows are concerned. Although it has a strong impact on the secondary air flows required for cooling and sealing purposes (Zimmermann *et al.*, 1991; Kutz and Speer, 1992), very little knowledge is available with respect to the flow phenomena at the interface between secondary air and lubrication oil systems, namely the bearing chamber. As a consequence, the designer has to deal with several uncertainties within the calculation

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scheme which cause performance losses and lead subsequently to the demand of additional time and cost consuming engine tests.

To overcome these problems, a test facility has been developed and built at the Institut für Thermische Strömungsmaschinen, University of Karlsruhe, which allows for the first time a detailed investigation of air/oil flow phenomena and related heat transfer processes in bearing chambers simulating aero-engine conditions for the whole flight envelope. As reported recently by Wittig *et al.* (1994) and Glahn and Wittig (1996), it has been found that the flow pattern inside a bearing chamber is dominated by a rotating oil film at the radial housing which is interspersed with small air bubbles. It is exposed to the turbulent gas flow in the core region of the chamber which carries oil droplets atomized by the rolling elements of the bearing as a dispersed phase. As indicated by the scheme in Fig. 1, the oil film flow parameters are dominated by the momentum transfer from the rotating shaft to the core flow and subsequently to the gas/liquid interface. Consequently, for a given geometry and at constant film Reynolds numbers an increase of the shaft velocity results in higher film surface velocities and in decreasing film heights. Therefore, gravity which has been identified as another important parameter for the liquid transport becomes less dominant. However, it has been demonstrated by Glahn and Wittig (1996) that it must not be neglected and has to be considered in the oil film flow analysis even for high shaft velocities and small chamber heights typical for modern jet engines. Another important effect of increasing interfacial shear forces is given by its influence on the film flow stability, i.e. interfacial shear forces can cause supercritical film flow conditions which are characterized by the onset of droplet removal from the film. As shown e.g. by Ishii and Grolmes (1975) and Himmelsbach (1992), droplet inception is not only caused by increasing gas velocities but also affected by the film Reynolds number. This is due to the fact that liquid films act upon the gas flow as a rough wall. A thick film is characterized by a high equivalent sandgrain roughness (Sattelmayer and Wittig, 1990; Himmelsbach, 1992) and, as a consequence, the interfacial shear increases. Summerizing this briefly given phenomenological description and considering a large amount of work documented in literature on liquid film flows (surveys are given by Himmelsbach, 1992; Hewitt, 1978; Sattelmayer, 1985; Mudawwar and El-Masri, 1986) it has to be stated that a detailed description of oil film flows in bearing chambers covering all interaction processes at the gas/liquid interface including unsteady wave

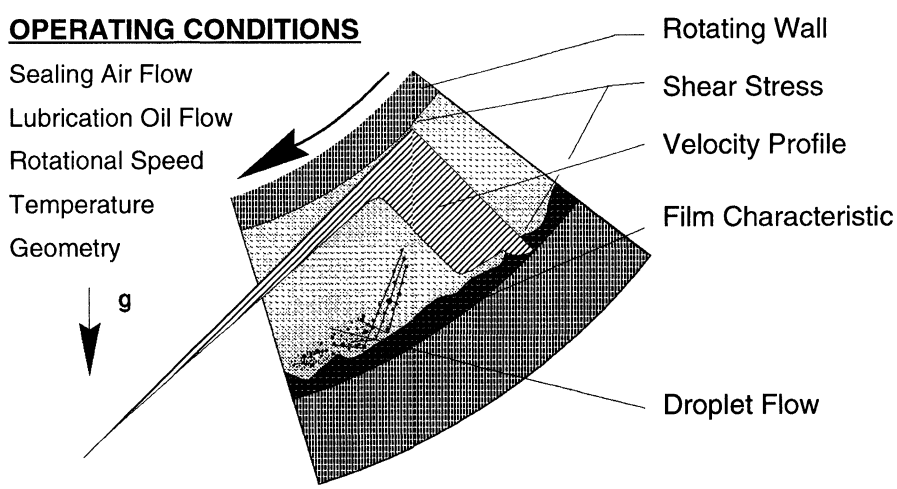

FIGURE 1 Oil film flow in bearing chambers and tasks for its analysis.

characteristics and supercritical conditions is not possible yet.

However, it can also be seen from literature that the main features of specific film flows can be predicted fairly well if the time averaged parameters are used for the calculation of both the liquid film flow and the interaction processes. The wavy film is treated as a liquid layer of constant height which is exposed to a time averaged shear force resulting from the air flow at its surface. As will be seen in some detail in the next section, applications of such approaches require the knowledge of the time averaged local film thickness and the averaged film velocity profile. The latter has to be taken from an assumption for the turbulence model of the film flow which is highly affected by the solid wall and under certain conditions by damping effects due to surface tension at the gas/liquid interface. Furthermore, if special interest is directed towards a heat transfer analysis, a correlation for the turbulent Prandtl number is required. Consequently, these topics are the subject of the theoretical outline given next.

THEORETICAL OUTLINE – MOMENTUM AND HEAT TRANSFER IN SHEAR DRIVEN LIQUID FILM FLOWS

Neglecting pressure forces and any acceleration of the fluid and based on the assumption, that the time averaged film height $\bar{h}_F$ is small in comparison to the curvature of the housing wall ($\bar{h}_F/r_i \leq 0.02$), the velocity u_F of a fluid element at a distance y from the wall (see Fig. 2) can be obtained from the momentum equation

$$1 - \sigma y_F^+ = \left(1 + \frac{\nu_t}{\nu_F}\right)\frac{du_F^+}{dy_F^+}, \tag{1}$$

whereas its temperature T_F is given by the heat flux distribution across the film

$$\frac{\dot{q}}{\dot{q}_W} = \left(\frac{1}{Pr_F} + \frac{1}{Pr_t}\frac{\nu_t}{\nu_F}\right)\frac{dT_F^+}{dy_F^+}. \tag{2}$$

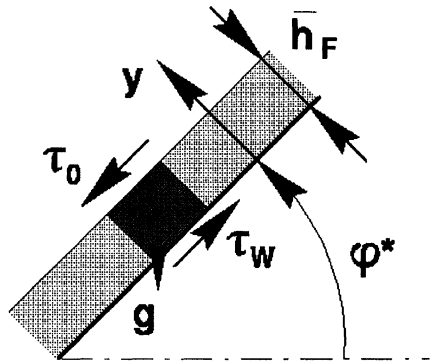

FIGURE 2 Force balance.

Velocity, wall distance and temperature have been transferred to non-dimensional values by use of the shear velocity

$$u_{\tau,F} = \sqrt{\tau_W/\rho_F}. \tag{3}$$

Thus, velocity and coordinate are

$$u_F^+ = \frac{u_F}{u_{\tau,F}}, \tag{4}$$

$$y_F^+ = y\frac{u_{\tau,F}}{\nu_F}, \tag{5}$$

and the dimensionless film temperature equals

$$T_F^+ = \frac{(T_W - T_F)\rho_F C_{P,F} u_{\tau,F}}{\dot{q}_W}. \tag{6}$$

The parameter σ in Eq. (1) is given by

$$\sigma = \frac{\nu_F g \sin\varphi^*}{u_{\tau,F}^3} \tag{7}$$

and expresses the superimposition of gravity and shear forces. Assuming zero interfacial shear ($\tau = 0$), this parameter becomes $\sigma = 1/\bar{h}_F^+$ and Eq. (1) is equal to the approaches known from freely falling films (e.g. Mudawwar and El-Masri, 1986). The other extreme case given by horizontal flow or extremely high interfacial shear is characterized by

TABLE I Eddy viscosity correlations for liquid film flows driven by superimposition of interfacial shear and gravity

Type	Authors	Model		Eq.
Laminar	Himmelsbach (1992)	$\nu_t/\nu_F = 0$	$0 \le y_F^+ \le \bar{h}_F^+$	(8)
Empirical correlations	Deissler (1954)	$\nu_t/\nu_F = n^2 u_F^+ y_F^+ \{1 - \exp(-n^2 u_F^+ y_F^+)\}$	$0 \le y_F^+ \le \bar{h}_F^+$	(9)
	Dukler (1960)	$n = 0.109$	$\bar{h}_F^+ \le 27$	
	Hubbard *et al.* (1976)	$\nu_t/\nu_F = -1/2 + 1/2\sqrt{1 + 4\kappa^2 y_F^{+2}\{1 - \exp(-y_F^+/A^+)\}^2(1 - \sigma y_F^+)}$	$0 \le y_F^+ \le y_i^+$	(10)
		$\nu_t/\nu_F = 8.13 \cdot 10^{-17}/Ka\,\sigma^{2/3} Re_F^{2m}\{1 + b(1 - \sigma\bar{h}_F^+)\}^2(\bar{h}_F^+ - y_F^+)^2$	$y_i^+ \le y_F^+ \le \bar{h}_F^+$	
		$\kappa = 0.4,\ A^+ = 26,\ m = 6.95 \cdot 10^2 \nu_F^{0.5},\ b = 0.9 + 1.73 \cdot 10^{12}\nu_F^2$		
	Mudawwar and El-Masri (1986)	$\nu_t/\nu_F = -1/2 + 1/2\sqrt{1 + 4\kappa^2 y_F^{+2}(1 - \sigma y_F^+)^2[1 - \exp\{-y_F^+/A^+(1 - \sigma y_F^+)^{0.5}(1 - \sigma h_{F,krit}^+)\}]^2}$	$0 \le y_F^+ \le \bar{h}_F^+$	(11)
		$\kappa = 0.4,\ A^+ = 26,\ h_{F,krit}^+ =$ "transition film height"		
Fully turbulent	Wurz (1971)	$\nu_t/\nu_F = \kappa y_F^+$	$0 \le y_F^+ \le \bar{h}_F^+$	(12)
		$\kappa = 0.4$		

$\sigma = 0$, i.e. the shear stress is constant over the film height as reported by Sattelmayer and Wittig (1990), Wurz (1971), and Wittig *et al.* (1991) for shear driven liquid films in airblast atomizers.

As it can be seen rapidly from Eqs. (1) and (2) a characterization of the momentum and heat transfer in bearing chamber oil film flows requires a proper matching of the eddy viscosity term ν_t/ν_F and, in addition, an expression for the turbulent Prandtl number Pr_t. The first item has been subject of numerous papers on all kinds of liquid film flows. For typical aero-engine bearing chamber oil films, Glahn and Wittig (1996) performed a fundamental study on the question of whether the time averaged oil film flow behaviour can be expressed in terms of an analytical approach or not. As it was the first investigation into bearing chamber oil films, they covered the entire range of possible flow conditions (see also Table I) from laminar (Eq. (8)) to fully turbulent (Eq. (12)). Besides that, the classical approach of Dukler (1960) has been included in the analysis, since it was the first which considers fluid transport due to interfacial shear as well as due to gravity. As demonstrated in Fig. 3, measured values are represented fairly well if the strategy proposed by Dukler (1960) – initially a two-equation model – is slightly modified to an approach using the eddy viscosity correlation of Deissler (1954) throughout the film. It should be pointed out, that applications of this model are limited to film heights $\bar{h}_F^+ \leq y_{F,max}^+ = 27$. However, oil films ($Pr_F \simeq 70$) observed at our test facility met this requirement and based on the agreement shown in Fig. 3, it is obvious to use Eq. (9) as a basis for a further investigation into heat transfer phenomena of bearing chamber oil film flows. Furthermore, it seems to be worthwhile to consider additionally some of the more sophisticated models which have been derived especially for heat transfer problems. As typical representatives, eddy viscosity correlations of Hubbard *et al.* (1976) and Mudawwar and El-Masri (1986) have been included in Table I. Both have been derived mainly for evaporating falling water films, i.e. low Pr_F fluids. Hubbard *et al.* (1976) considered superimposed interfacial shear due to a cocurrent vapor flow and, therefore, their approach can be transferred directly to bearing chamber conditions. Mudawwar and El-Masri (1986) developed their correlation for freely-falling films. Thus it has to be modified and Eq. (11) has been obtained by replacing the shear stress distribution for freely-falling films by the general case given in the left hand side of Eq. (1). As the most important difference to the other correlations summarized in Table I, Hubbard *et al.* (1976) as well as Mudawwar and El-Masri (1986) take damping effects due to surface tension at the free interface into account, since it has been identified to strongly influence heat transfer phenomena in falling films. The non-dimensional parameter group which comprises viscosity and surface tension as the most important parameters which influence the turbulent motion, namely the Kapitza number, is defined by

$$Ka = \frac{\rho_F^3 \nu_F^4 g}{\sigma_F^3}. \tag{13}$$

However, comparing Eqs. (10) and (11) with the simple correlation given by Eq. (9) for typical aero-engine bearing chamber conditions, it is shown in Fig. 4 that the differences with respect to the velocity and the eddy viscosity profiles are little due to the dominating influence of viscosity.

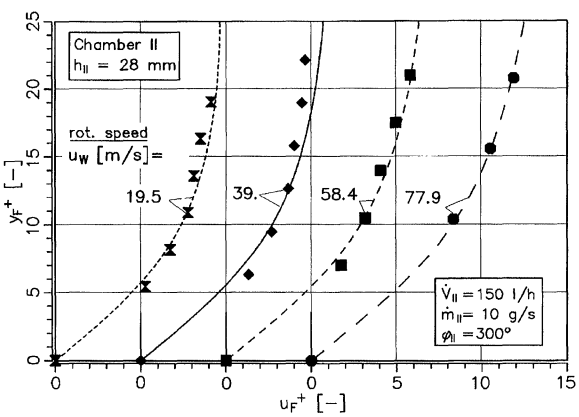

FIGURE 3 Oil film velocity profiles in bearing chambers – Measured data in comparison to a theoretical approach using Eq. (9) (Glahn and Wittig, 1996).

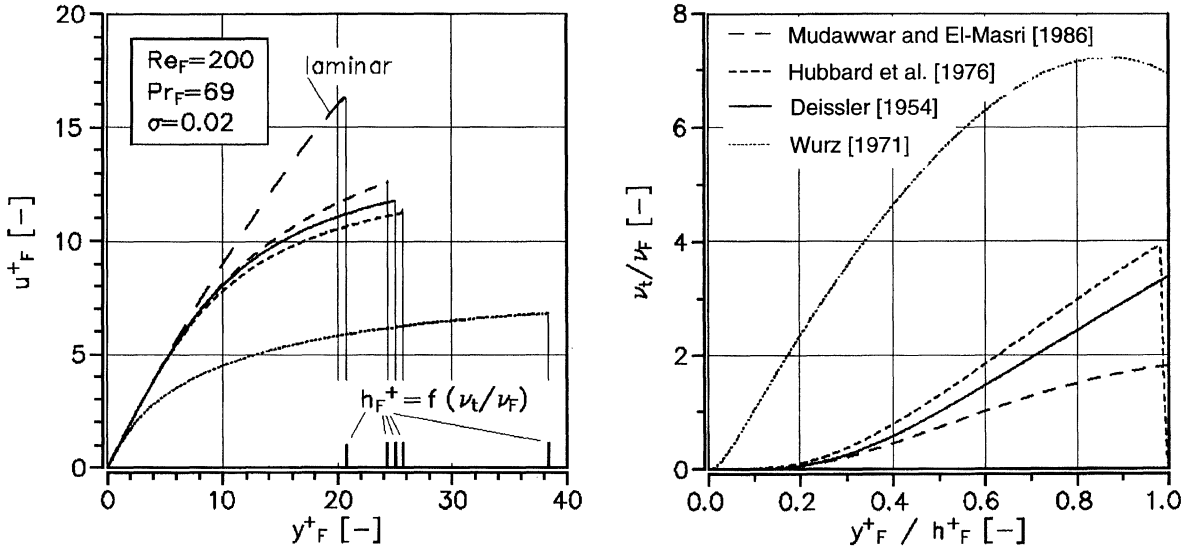

FIGURE 4 Profiles for velocity and eddy viscosity at aero-engine bearing chamber oil film flows.

As it has already been expressed by Eq. (2), a further analysis of heat transfer processes has to be based on an assumption for the heat flux in radial direction (i.e. in this study: $\dot{q}/\dot{q}_W = 1$) and requires a correlation for the turbulent Prandtl number. Jischa (1982) discusses data gained from experiments and from semi-empirical approaches for the definition of Pr_t. He demonstrates that this quantity depends on the Reynolds number and the molecular Prandtl number. Existing correlations show misrepresentations of measured data especially for decreasing values of Pr_F and Re_F. However, a constant value of $Pr_t \simeq 1$ has been found if molecular Prandtl number and Reynolds number are high enough. Although the limit is not defined exactly, for the turbine oil used in the present study ($Pr_F \geq Pr_F$ ($T_F = 100°$) $\simeq 70$) all available correlations indicate that turbulent momentum and heat transfer are dominated by the same mechanism. Other approaches documented in literature for liquid film heat transfer often use a value of $Pr_t \simeq 0.9$ and, finally, Mudawwar and El-Masri (1986) reported on a correlation which would result in $Pr_t \simeq 0.667$ for fluids with Prandtl numbers similar to oil.

A comparison of different Pr_t assumptions can be performed by an integration of the film temperature distribution in radial direction

$$T_F^+(y_F^+, Pr_F) = \int_0^{y_F^+} \left(\frac{1}{Pr_F} + \frac{1}{Pr_t} \frac{\nu_t}{\nu_F} \right)^{-1} dy_F^+. \tag{14}$$

As it has been expected from eddy viscosity profiles discussed above, it is demonstrated in Fig. 5 that the Pr_t impact on the turbulent heat transfer is suppressed by the fluid viscosity, i.e. the calculation of heat transfer processes in bearing chamber oil film flows is less sensitive to the accuracy of the turbulent Prandtl number correlation and depends much stronger on the eddy viscosity distribution ν_t/ν_F and the molecular Prandtl number Pr_F. This is shown in a more expressionable manner in Fig. 6 which summarizes some sample calculations for the Nusselt number Nu_F obtained from the temperature distribution

$$Nu_F = \frac{\alpha \bar{h}_F}{\lambda_F} = \frac{\dot{q}_W \bar{h}_F}{(T_W - \bar{T}_F)\lambda_F} = \frac{Pr_F \bar{h}_F^+}{\bar{T}_F^+} = \frac{Pr_F Re_F}{\bar{T}_F^+ \bar{u}_F^+} \tag{15}$$

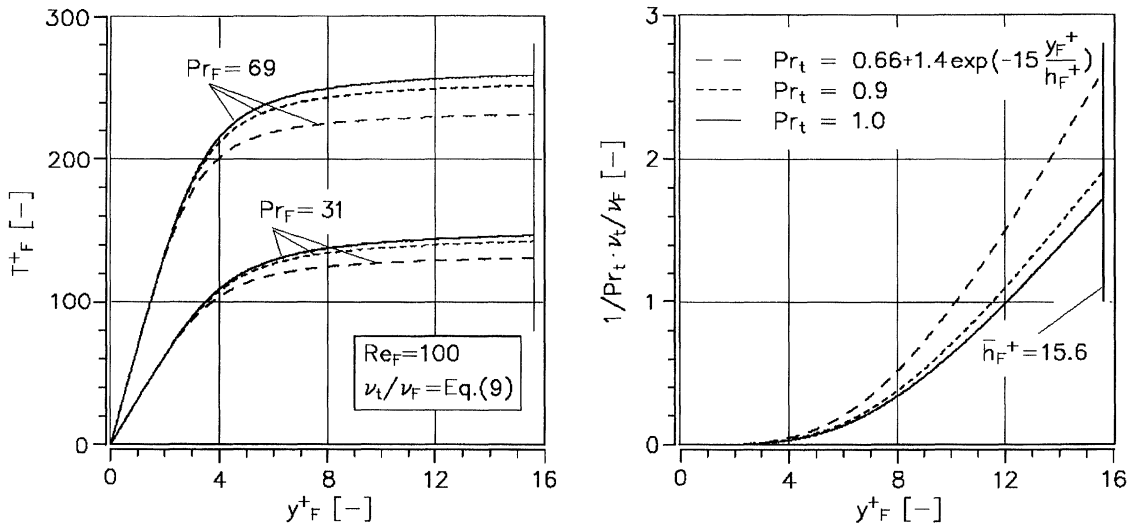

FIGURE 5 Profiles for temperature and turbulent heat transfer at oil film flow conditions typical for aero-engine bearing chambers.

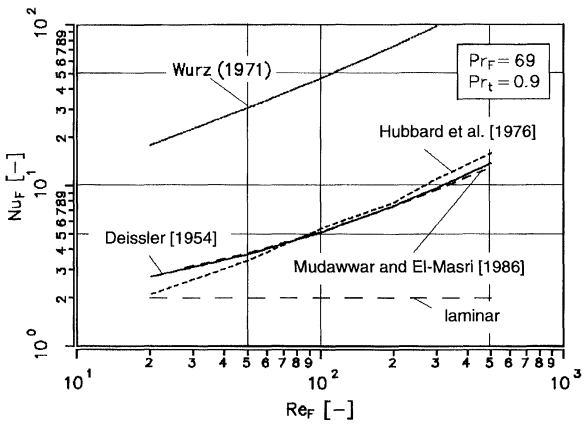

FIGURE 6 Nusselt numbers for oil film flows.

with

$$\bar{T}_F^+ = \frac{1}{\bar{h}_F} \int_0^{\bar{h}_F^+} T_F^+(y_F^+, Pr_F)\, dy_F^+. \qquad (16)$$

Assuming horizontal laminar film flow $\bar{T}_F^+ = \frac{1}{2} Pr_F \bar{h}_F^+$, thus Eq. (16) results in $Nu_{F,lam} = 2$. In the case of turbulent film flow, Nusselt numbers have been determined by numerical integration in terms of ν_t/ν_F, Re_F and Pr_F. It can be seen clearly from deviations of two orders of magnitude occurring in the calculation of non-dimensional heat transfer coefficients that a fundamental characterization of oil film flows as it has been performed by Glahn and Wittig (1996) becomes extremely important for the thermal analysis. In addition, further proof is given for the comparatively weak significance of free interface damping effects as considered by Hubbard *et al.* (1976) and Mudawwar and El-Masri (1986). However, uncertainties within the calculation of bearing chamber oil film flow heat transfer still exist and experimental data for an assessment of theoretical approaches are strongly required.

EXPERIMENTAL INVESTIGATIONS

A co-axial sectional view of the bearing chamber test rig used in the present investigation is shown in Fig. 7(a). Although similar configurations have already been subject to papers presented by Wittig *et al.* (1994) and Glahn and Wittig (1996) the general arrangement should be described briefly in order to ensure a better understanding of the test results. A squeezefilm-damped aero-engine roller bearing (1) lubricated by an underrace oil supply

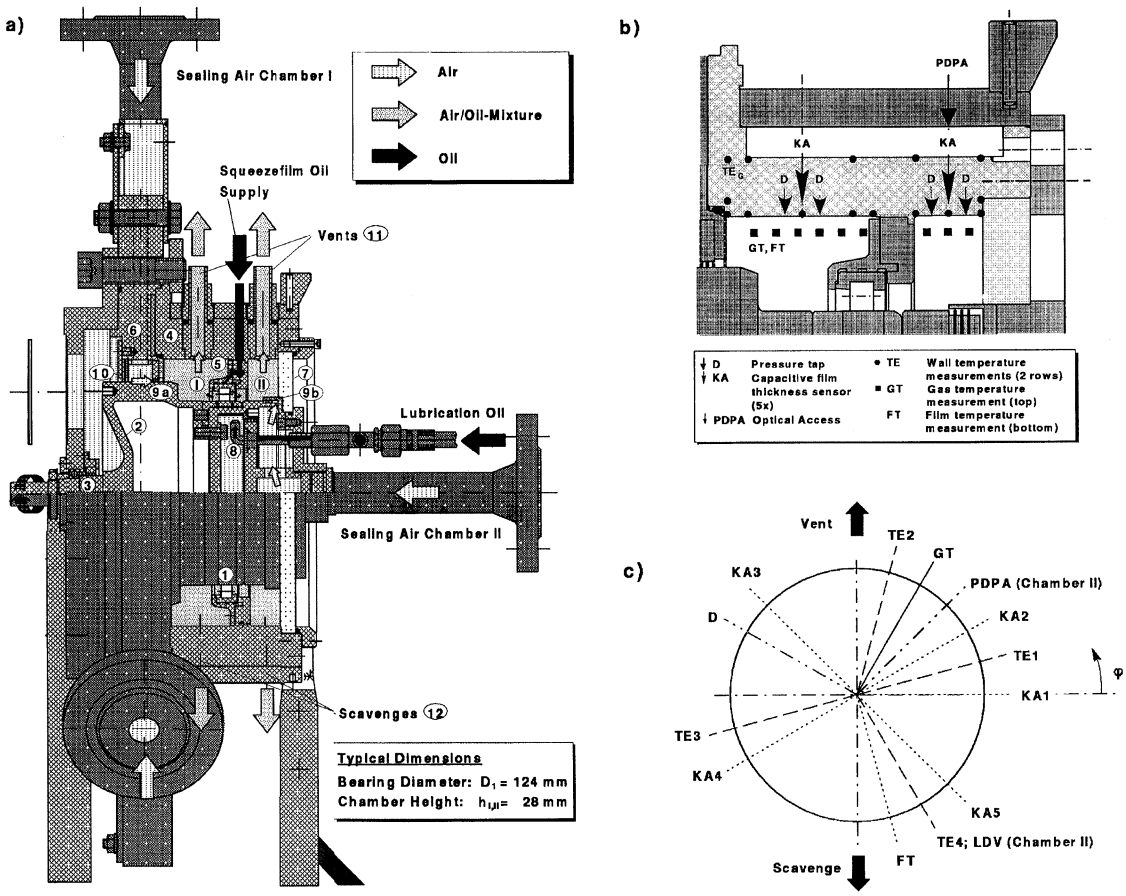

FIGURE 7 High speed bearing chamber test rig – General arrangement and instrumentation (a) Rig components: (1) roller bearing, (2) rotor, (3) location ball bearing, (4) thick-walled housing, (5) roller bearing support, (6) sealing air supply, (7) window, (8) lubrication oil jet, (9) labyrinth seals, (10) brush seal, (11) vents, (12) scavenges, (b) Instrumentation – co-axial section ($\varphi = 75°$), (c) Instrumentation – radial section.

(8) separates two chambers (I, II) which are bounded by the rotor (2), a thick-walled housing (4), the bearing support (5), the sealing air supply for chamber I and a transparent disc covering chamber II. To prevent oil leakage both chambers are sealed using three fin labyrinths (9a,b). Air/oil mixtures which are generated by the atomization of lubrication oil inside the bearing and due to droplet interaction with pressurized sealing air flows are discharged through vents (11) at the top of the chamber whereas the oil sump at the bottom is dropped out via radial scavenges (12). Operating conditions relevant for aero-engine applications are guaranteed by a broad range of air and oil flows ($5\,\text{g/s} \leq \dot{m}_{G_{I,II}} \leq 20\,\text{g/s}$ and $\dot{V}_{L_{I,II}} \leq 200\,\text{l/h}$) and by high rotational speeds up to $n_{S_{max}} = 16{,}000$ rpm. In addition, air and oil are preheated to chamber inlet temperatures of $T_G = T_L = 150°\text{C}$. Figure 7(b) and (c) demonstrates an extremely high degree of instrumentation necessary for an investigation into circumferential effects of the bearing chamber flow pattern and the internal wall heat transfer.

In the present study oil film flow characteristics are investigated in the bottom region of chamber II at a location 30° upstream of the scavenge port, where the film velocity measurements (LDA) shown in Figs. 3 and 8 have been performed. In addition, heat transfer instrumentation (TE4) described by

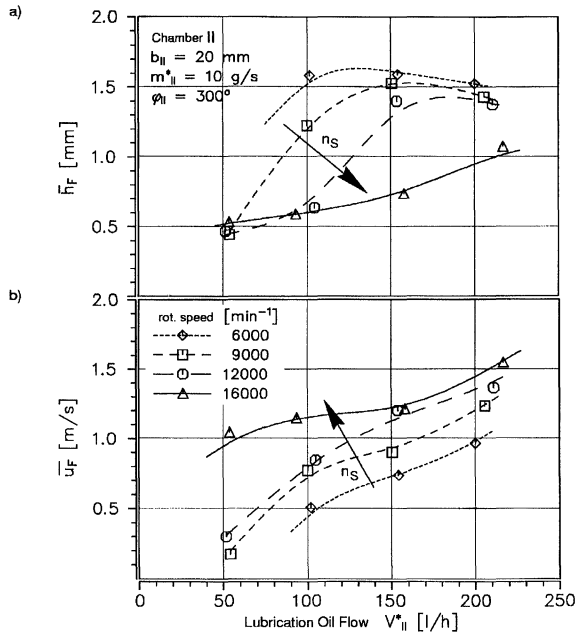

FIGURE 8 Oil film flow analysis inside chamber II: (a) film thickness, (b) film velocity.

Wittig *et al.* (1994) is provided to this location and oil film temperatures (FT) necessary for the non-dimensional treatment of oil film velocity and height have been measured next to it. For the non-reactive determination of film heights a capacitive gauge (KA5) has been mounted flush with the internal bearing chamber housing wall at a position $\Delta\varphi = 15°$ upstream of the heat transfer measuring plane.

Exemplarily results for film heights $\bar{h}_F$ and film velocities $\bar{u}_F$ measured for typical aero-engine conditions are shown in Fig. 8. An increase of rotational speeds causes an oil flow pattern distributed more homogeneously inside the chamber, i.e. for a given lubrication oil flow $\dot{V}_{II}$ to the chamber the local oil film mass flow $\dot{m}_F(\varphi)$ at a location in the bottom region decreases and the film height decreases as well. As it has been expected, higher lubrication oil flows give larger film heights. Limitations for a growth of the film thickness are given as a consequence of increasing interfacial shear due to a larger equivalent sandgrain roughness of thick films and due to its obstruction effect on the gas flow. As shown in Fig. 8(b), both effects as well as the momentum transfer from the shaft tends the film velocities to higher values with increasing rotational speed and enhanced oil flows. Using this oil film flow data Reynolds numbers can be obtained from

$$Re_F = \frac{\bar{u}_F \bar{h}_F}{\nu_F} = \int_0^{\bar{h}_F^+} u_F^+(y_F^+)\,\mathrm{d}y_F^+. \qquad (17)$$

With Eq. (15) and one of the proposals for the eddy viscosity film heat transfer coefficients α can be calculated and compared with appropriate data measured directly, e.g. as shown by Wittig *et al.* (1994).

Based on the excellent agreement between theoretical approach and measured data for the momentum transfer (Fig. 3) and considering that damping effects at the free interface (Eqs. (10) and (11)) apparently do not have a significant impact on oil film flows, it is reasonable to take into account the simple one-equation model Eq. (9) for an initial assessment of theoretical approaches on internal bearing chamber wall heat transfer processes. This has been done by assuming a turbulent Prandtl number of $Pr_t = 1$ and the comparison is shown in Fig. 9. The theory leads to lower predictions of the heat transfer coefficients taken from measurements but the qualitative estimate of the heat transfer characteristic is quite well. Considering the simplifications which have been made, the agreement between theory and experiment ($\Delta\alpha/\alpha_M \simeq 34\%$) is remarkable and could be further improved by correlations which give lower values for Pr_t (Fig. 5). However, it has to be kept in mind that the impact of hot oil droplets emanated from the bearing has yet not been included in the study. Therefore, a further analysis has to concentrate on this subject instead of any fitting with respect to the turbulent Prandtl number. The potential of other models than Eq. (9) can be assessed by use of Figs. 4 and 6. It is shown clearly and it has been stated above, that the more complex theories of Hubbard *et al.* (1976) and

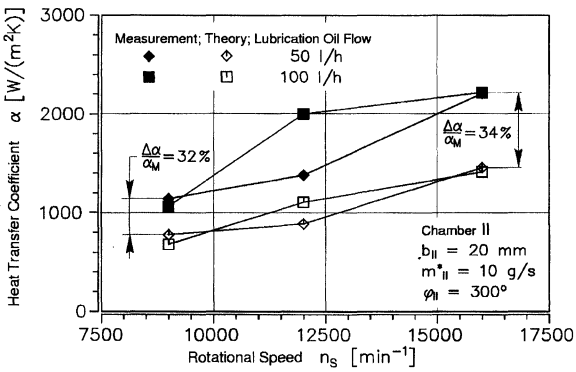

FIGURE 9 Comparison of an analytical approach based on the eddy viscosity correlation Eq. (9) and $Pr_t = 1$ with measured values.

Mudawwar and El-Masri (1986) do not give any benefit to an oil film flow heat transfer analysis in comparison to the simple approach used in Fig. 9. Studies based on laminar or fully turbulent models would fail for the present application since Nusselt numbers differ from those obtained by Eq. (9) of more than one order of magnitude (Fig. 6) which is not ruled out by differences in the film height (Fig. 4).

CONCLUSION

Based on a comparison with measured data, typical representatives of existing models for the calculation of liquid film flows have been assessed for their potential on heat transfer predictions across bearing chamber oil film flows. It has been demonstrated that a simple analytical approach using an empirical correlation proposed by Deissler (1954) offers some scope for this task. Typical deviations between measured data and predictions show values of $|\Delta\alpha|/\alpha_M \simeq 34\%$. As it has been expected due to the fact that the calculations have been performed without consideration of droplet bounded thermal energy transport from the bearing into the wall film, heat transfer coefficients obtained from theory show lower values than experimental data.

NOMENCLATURE

Symbols

b	width	m
d	diameter	m
g	acceleration of gravity	m/s^2
h	height	m
Ka	Kapitza number	—
$\dot{m}$	mass flow	kg/s
Nu	Nusselt number	—
n_s	rotational speed	min^{-1}
n	constant	—
Pr	Prandtl number	—
$\dot{q}$	heat flux	W/m^2
Re	Reynolds number	—
r, φ, z	cylindrical coordinates	m, °, m
T	temperature	K
u	velocity	m/s
u_τ	shear velocity	m/s
$\dot{V}$	volume flow	m^3/s
y	distance from the wall	m
α	heat transfer coefficient	$W/(m^2\ K)$
κ	Kármán constant	—
λ	thermal conductivity	W/(m K)
ν	kinematic viscosity	m^2/s
ν_t	eddy viscosity	m^2/s
ρ	density	kg/m^3
σ	non-dimensional parameter for the film velocity profile	—
σ_F	surface tension	N/m
τ	shear stress	N/m^2

Subscripts

F	film
G	gas
L	liquid
l	laminar
t	turbulent
W	wall
I, II	chamber I, II

Acknowledgment

We thank Rolls-Royce, Bristol and Motorenund Turbinen-Union, München for funding the

experiments documented in this paper. Special thanks are due to H. Zimmermann and R.M. Coleman for helpful discussions.

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