THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS
345 E. 47th St., New York, N.Y. 10017

95-GT-114

TWO-PHASE AIR/OIL FLOW IN AERO ENGINE BEARING CHAMBERS – CHARACTERIZATION OF OIL FILM FLOWS

A. Glahn and S. Wittig
Lehrstuhl und Institut für Thermische Strömungsmaschinen
Universität Karlsruhe (T.H.)
76128 Karlsruhe, Germany

BREAK

ABSTRACT

For the design of secondary air and lubrication oil systems a sufficient knowledge on two-phase flow and heat transfer phenomena under bearing chamber flow conditions is required. The characterization of oil film flows at the bearing chamber walls is one of the major tasks for a better understanding of these processes and, therefore, a necessity for improvements of the efficiency of aero engines. The present paper gives a contribution to this subject.

Utilizing a fibre-optic LDV-setup, measurements of oil film velocity profiles have been performed in our high speed bearing chamber rig simulating real engine conditions. All data have been compared with different theoretical approaches which have been derived from a force balance at a liquid film element, including geometrical conditions and temperature dependent fluid properties, and by approaches for the eddy viscosity available in the literature.

NOMENCLATURE

Symbols

b	width	m
d	diameter	m
g	acceleration of gravity	m/s^2
h	height	m
l_t	turbulent length scale	m
$\dot{m}$	mass flow	kg/s
n_W	rotational speed	min^{-1}
n	constant	–
r, ϕ, z	cylindrical coordinates	$m, ^\circ, m$
T	temperature	K
u	velocity	m/s
u_τ	shear velocity	m/s
u^+	nondim. velocity	–
$\dot{V}$	volume flow	m^3/s
y	distance from the wall	m
y^+	nondim. distance	–
φ^*	inclination angle	$^\circ$
κ	Kármán constant	–
ν	kinematic viscosity	m^2/s
ν_t	eddy viscosity	m^2/s
ρ	density	kg/m^3
σ	nondimensional parameter for the film velocity profile	–
τ	shear stress	N/m^2

Subscripts

F	film/liquid
L	air
l	laminar
t	turbulent
W	wall
I, II	chamber I, II

INTRODUCTION

Performance characteristics and capabilities of modern jet engines are highly influenced by an efficient design of the secondary air system. Therefore, air has to be taken as early as possible from the compressor and has to be expelled back into the main stream at

Presented at the International Gas Turbine and Aeroengine Congress & Exposition
Houston, Texas - June 5-8, 1995
This paper has been accepted for publication in the Transactions of the ASME
Discussion of it will be accepted at ASME Headquarters until September 30, 1995

the highest possible pressure. To meet this requirement, sufficient knowledge on discharge characteristics and heat transfer phenomena of all secondary air system components, i.e. rotating orifices, labyrinth seals and annular gaps, are necessities within the design process. As a result, these subjects are currently investigated in comprehensive studies and worldwide research activities (Wittig and Schulz, 1992). As far as single phase air flows are concerned, satisfying results have been elaborated for many of these components. Examples are given by the work of Waschka et al. (1990)-(1992), Wittig et al. (1994a), and Jakoby et al. (1994) who have investigated influences of high rotational speeds on the performance of various labyrinth seal geometries and orifices, respectively. Contrary to this, little knowledge is available on the flow phenomena at the interface between the secondary air and the lubrication oil system, namely the bearing chamber.

Based on the fact that rolling element bearings cannot be substituted in aero engines within the near future, special effort has to be directed to the problem of heat generation due to bearing friction and, therefore, these bearings have to be lubricated by oil. As a consequence, chambers have to be built separating the lubricant from the hot zones of the engine in order to prevent oil fires. Bearing chamber seals (e.g. labyrinth or brush seals) have to be pressurized by air which is supplied by the secondary air system described above. Entering the chambers, the air mixes with the lubrication oil generating extremely complex two phase flow conditions. Finally, the air is discharged through vent lines carrying a certain amount of oil with it.

Due to contradictory tasks in designing the flow elements upstream and downstream of the bearing chambers, a proper matching of the appropriate sealing air flows is a highly iterative process. On one hand, the amount of sealing air flow has to be kept as small as possible, because it represents an engine power loss. On the other hand, small air flows discharged by vent pipes result in small pressure differences across the seals which can cause oil leakage and in consequence oil fire (Suter and Boyman, 1978). The amount of lubrication oil supplied to the bearing affects the pressure loss downstream of the bearing chamber and, therefore, it is an indirect design parameter for the sealing air flow. The oil flow necessary for a reliable engine operation is also influenced by the problem of heat generation to the oil which becomes greater with increasing amounts of hot air flow into the chamber. Other problems are given due to the fact that the oil scavenged back from the bearing chamber into the reservoir has to be cooled. High oil flows require larger pumps, filters, and cooling units

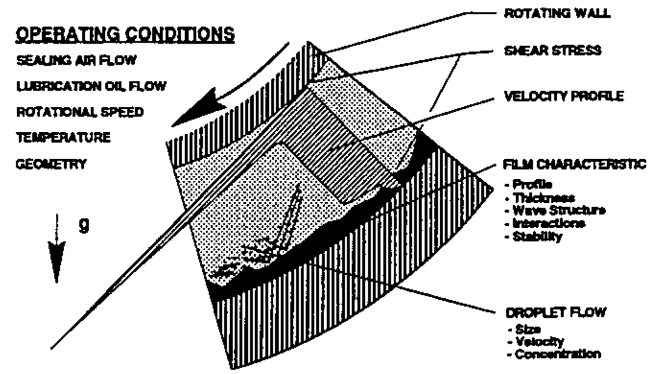

FIG. 1: TASKS IN BEARING CHAMBER FLOW ANALYSIS

with the consequence of higher energy requirements for accessory drives and additional weight. Besides that, improvements in aero engine efficiency have reduced the fuel consumption considerably, thus reducing the cooling capacity for the oil heat exchangers. Therefore, to design lubrication oil flows a sufficient knowledge of all heat transfer phenomena involved in bearing chamber flow conditions is required. Additional heat sources are the bearing friction and the heat transfer from the bearing chamber walls to the oil which is bounded to a substantial part in a rotating film at the inner side of the housing.

The characterization of oil film flows and the associated heat transfer processes is one of the major tasks for a better understanding of bearing chamber flow and, therefore, a necessity for improvements of the efficiency of secondary air systems.

The present paper gives a contribution to this subject. Based on a literature survey on film flows and an experimental determination of oil film velocity profiles an analytical approach is given for the characterization of shear-driven oil film flows under gravity.

TWO PHASE AIR/OIL FLOW IN BEARING CHAMBERS

Phenomenological description – modelling of film flows

Two-phase air/oil flow patterns inside aero engine bearing chambers can be expressed simplistically as shown in Fig. 1. The complex geometry of a real engine application has been abstracted by an annular gap between two concentric cylinders, the inner one rotating. It has been shown in comprehensive preliminary flow visualization studies that the flow field within the annular gap is characterized by a rotating oil film interspersed with small air bubbles, and by oil droplets dispersed in the turbulent air flow over the wall film.

Obviously, the interaction between air flow and wall film affects both the velocity profile of the gas phase as well as the properties of the oil film. The oil film acts upon the air like a rough wall, thus increasing the shear stress at the wall and the interface, respectively. The latter results in an increase of the surface velocity and can cause supercritical film flow conditions characterized by the onset of droplet removal from the film. As shown for instance by Himmelsbach (1992) and Wittig et al. (1992), the incorporation of a detailed analysis of all interaction processes at the gas/film interface including unsteady wave characteristics and supercritical conditions into practical applications is not possihle yet. However, due to the importance of film flows for many technical applications, a considerable amount of work on the relevant topics exists. Based on these investigations models have been developed which predict the main features of the specific film flows fairly well. On the assumption that characteristic time-averaged parameters can be used for tbe description of both tbe film flow and the interaction processes, the wavy film is treated as a film of constant height exposed to an averaged shear force resulting from the air flow at its surface. Applications of sucb an approach require the knowledge of the time-averaged local film thickness and the averaged film velocity profile or the mass flow rate, respectively. Furthermore, a relationship for the coupling between film and gas phase is necessary and, therefore, the research activities have heen focussed to these topics.

Literature survey

A survey on efforts on the above-mentioned research topics has been presented by Wurz (1971), Hewitt (1978), Sattelmayer and Wittig (1989) and Himmelsbach (1992).

The fundamental problem within the calculation procedure of film flows is the question of whether the flow regime is laminar or turbulent. The investigations documented in literature do not show a uniform strategy on this topic. One reason may be given by the fact that most of the investigators could not measure velocity profiles directly. Therefore, a comparison of measured data with the appropriate values calculated from the individual theory has been performed on the basis of data which can be determined much more easily (e.g. Dukler and Bergelin, 1952, Dukler, 1960, Lee, 1965, Wurz, 1971, and Sattelmayer and Wittig, 1989). Most of the investigators mentioned above, have taken the time averaged film thickness for this comparison. Other researchers (Ueda and Tanaka, 1975) have generated thick ($\overline{h}_F \simeq 4-5\ mm$) viscous film flows, in order to measure velocity profiles utilizing conventional prohe techniques. However, it is questionable whether their results showing laminar flow conditions can be transferred to bearing chamber flows, because their investigation covers only falling liquid films without shear forces at the interface or droplet interaction.

In recent years the development and improvement of optoelectronical measuring devices has given a much better insight into film flow behaviour. Wittig et al. (1992) have reported on the assessment of different approaches for the calculation of shear driven thin ($\overline{h}_F \leq 200\mu m$) water films based on measurements utilizing a newly developed optical measuring device for the simultaneous determination of film thickness and velocity. In a parameter range typical for the conditions in prefilming airblast atomizers, the best fit between predicted and measured film surface velocities has been found for the assumption of a laminar film velocity profile. Besides that, they have noticed a turbulent structure of the film for a decrease in the interface shear stress and an increase in the liquid flow rate, both leading to a larger film thickness. This statement has been confirmed by the investigation of Plimon (1991) who used a modified LDV-setup for the analysis of shear driven liquid films. For water films of $\overline{h}_F \simeq 500\mu m$ thickness and comparatively low interface shear forces he has observed a turbulent structure.

However, a generalized statement for the liquid film flow behaviour can not be found in the literature, because experiments transferable to bearing chamber flow conditions have not been performed so far. Therefore, our own investigations on this subject have to be done. For the measurements described in this paper our compact high speed bearing chamber test rig (Wittig et al. 1994b) has been adapted to this task. The discussion of oil film velocity profiles measured under real engine conditions at our test facility has to be based on a theoretical outline on film flows and the question of whether the measured data can be expressed in terms of an analytical approach or not. Some of the theoretical approaches available in literature have been used in this paper as a reference to the measurements. These theories are presented next.

Theoretical outline on film flow

Neglecting pressure forces and any acceleration of the fluid, a force balance for a liquid film flowing on an inclined plane (inclination angle φ^*) results in

$$\tau = \tau_W - \rho_F\ g\ \sin\varphi^*\ y\ . \qquad (1)$$

With the shear velocity

$$u_{\tau,F} = \sqrt{\tau_W/\rho_F} \qquad (2)$$

this can be rewritten in a dimensionless form

$$\frac{\tau}{\tau_W} = 1 - \sigma\ y_F^+ \quad . \tag{3}$$

The parameter σ is given by

$$\sigma = \frac{\nu_F\ g\ \sin\varphi^*}{(\tau_W/\rho_F)^{3/2}}$$

and the dimensionless distance from the wall has been derived utilizing Eq. (2)

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

The total shear stress in the film results from a superimposition of the molecular and the turbulent shear stress, respectively. The latter can be expressed following the Boussinesq-approach. Thus,

$$\frac{\tau}{\tau_W} = \left(1 + \frac{\nu_t}{\nu_F}\right) \frac{du_F^+}{dy_F^+} \quad , \tag{4}$$

where ν_t is the eddy viscosity and u_F^+ is the film velocity parallel to the wall based on the shear velocity given by Eq. (2). Substituting Eq. (3) into Eq. (4) results in

$$1 - \sigma\ y_F^+ = \left(1 + \frac{\nu_t}{\nu_F}\right) \frac{du_F^+}{dy_F^+} \tag{5}$$

as a universal velocity profile for shear driven liquid films under gravity. This equation can be solved for a given relationship for the eddy viscosity. In the present study three approaches for ν_t are compared. The first one represents laminar flow conditions

$$\frac{\nu_t}{\nu_F} = 0\ . \tag{6}$$

Next we have adapted an approach discussed by Wurz (1971) for horizontal flow to inclined film flows resulting in additional gravity forces. Here, the eddy viscosity is expressed in terms of the turbulent length scale:

$$\frac{\nu_t}{\nu_F} = \frac{1}{\nu_F}\ l_t^2\ \frac{du_F}{dy_F} = \frac{1}{\nu_F}\ \kappa^2\ y_F^2\ \frac{du_F}{dy_F}\ . \tag{7}$$

Finally, we have followed Dukler (1960) who proposed an empirical correlation given first by Deissler (1954):

$$\frac{\nu_t}{\nu_F} = n^2\ u_F\ y_F\ \frac{1}{\nu_F}\{1 - \exp(-n^2\ u_F\ y_F\ \frac{1}{\nu_F})\}\ . \tag{8}$$

The Kármán constant has been set to $\kappa = 0.4$ and the empirical constant in Eq. (8) to $n = 0.109$, respectively. It should be noted, that the model proposed by Dukler (1960) utilizes two approaches for the eddy viscosity, namely the correlation given in Eq. (8) for film heights in the range of $0 \le y_M^+ \le 20$, and another correlation in the range $y_M^+ > 20$. Based on the fact, that Deissler (1954) has set $y_M^+ = 26$ as the matching point between the two approaches and our investigation covers the range of $0 \le y_M^+ \le 27$, for the sake of simplicity we have solely taken Eq. (8) into account.

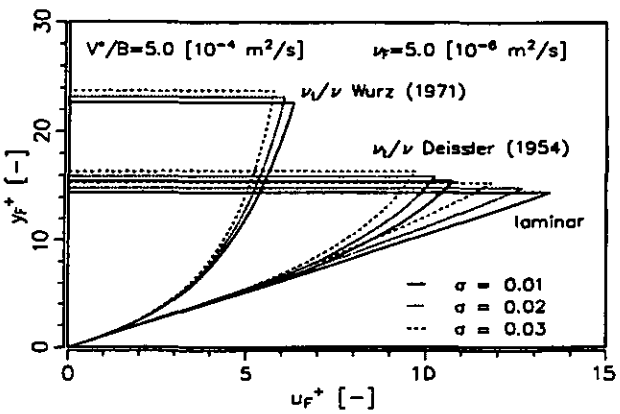

FIG. 2: COMPARISON OF FILM VELOCITY PROFILES AT BEARING CHAMBER FLOW CONDITIONS

By use of Eq. (5) the following equations can be written for the laminar profile

$$u_F^+ = y_F^+ - \frac{1}{2}\ \sigma\ y_F^{+^2} \quad , \tag{9}$$

the approach following Wurz (1971)

$$\frac{du_F^+}{dy_F^+} = -\frac{1}{2\kappa^2 y_F^{+^2}} + \frac{1}{\kappa y_F^{+^2}} \sqrt{\frac{1}{4\kappa^2} + y_F^{+^2} - \sigma y_F^{+^3}}\ , \tag{10}$$

and the empirical correlation following Deissler (1954)

$$\frac{du_F^+}{dy_F^+} = \frac{1 - \sigma\ y_F^+}{1 + n^2\ u_F^+\ y_F^+\{1 - \exp(-n^2\ u_F^+\ y_F^+)\}}\ , \tag{11}$$

where Eq. (10) and Eq. (11) have to be integrated numerically. Differences in the resulting film thickness, velocity, and velocity profile for the approaches given by Eq. (9) – Eq. (11) are demonstrated by Fig. 2.

The equation of continuity

$$\int_0^{\overline{h_F^+}} u_F^+ dy_F^+ = \frac{\dot{V}_F/b}{\nu_F} \tag{12}$$

has been used for the calculation of film thickness and surface velocity at a given film flow rate and temperature, representing typical bearing chamber flow conditions. In addition, the parameter σ has been varied,

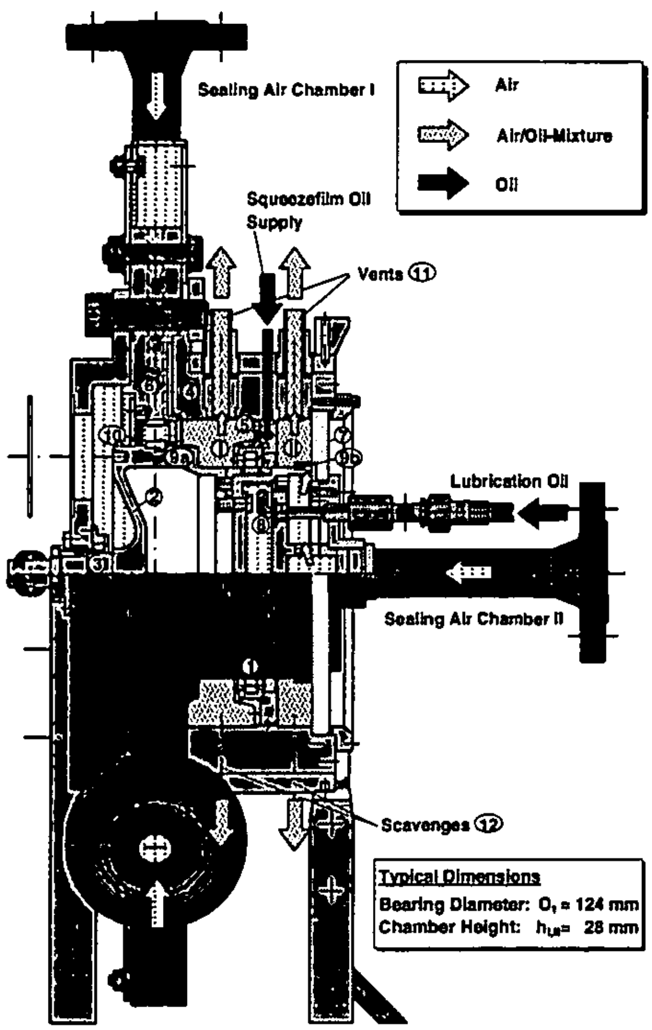

FIG. 3: HIGH SPEED BEARING CHAMBER TEST RIG

showing the significance of gravity forces. It can be easily seen from this example, that the uncertainty in the calculation of the film thickness is about 60 %, whereas differences in the film surface velocity are in the range of 55 %. A more detailed discussion is given together with the presentation of measured data.

EXPERIMENTAL SETUP

Compact high speed bearing chamber rig

As mentioned above, the test rig developed at our institute for the investigation of air/oil flow and heat transfer phenomena in bearing chambers has been adapted to film velocity profile measurements under real engine conditions characterized e.g. by high rotational speeds up to $n_W = 16000\ rpm$. Design considerations and a description of all test facility components are given elsewhere (Wittig et al.,1994b). However, for a better understanding of air and oil flows inside the bearing chamber rig, a short description of the general arrangement is presented next. A co-axial sectional view of the test rig is shown in Fig. 3.

At each side of squeezefilm-damped roller bearing (1), separate chambers (I,II) are formed. The axial location of the rotor (2) is realized by use of a conventional groove-ball bearing (3). The chambers are bounded by a thick-walled housing (4), the roller bearing support (5), the rotor, a flange (6) realizing the sealing air supply of chamber I and the support of the housing, and a transparent cover (7) for chamber II representing the optical access for the film velocity measurements. It can be readily concluded from Fig. 3 that the bearing chamber geometries of our test rig have been abstracted from the very complex arrangements given by a real engine to a more or less rectangular shape. Thus, two different aspect ratios of $(b/h)_I = 1.62$ and $(b/h)_{II} = 0.71$ are realized, both expected to give transferable results.

However, air and oil flows have been arranged in the same way as in the real engine. An under-race lubrication (8) supplies the roller bearing with preheated oil. To prevent oil leakage the chambers are sealed using three-fin labyrinth seals (9a,b). Both labyrinth seals are pressurized by air which is preheated to the same temperature level as the lubrication oil. Air/oil mixtures are discharged through the vent lines (11) at the top of each chamber, while the oil sump at the bottom is dropped out via radial scavenges utilizing the pressure difference from the bearing chamber atmosphere to the environment. Thus, another simplification is given to a real engine application, where scavenge pumps are used. This circumstance has been considered by selecting the

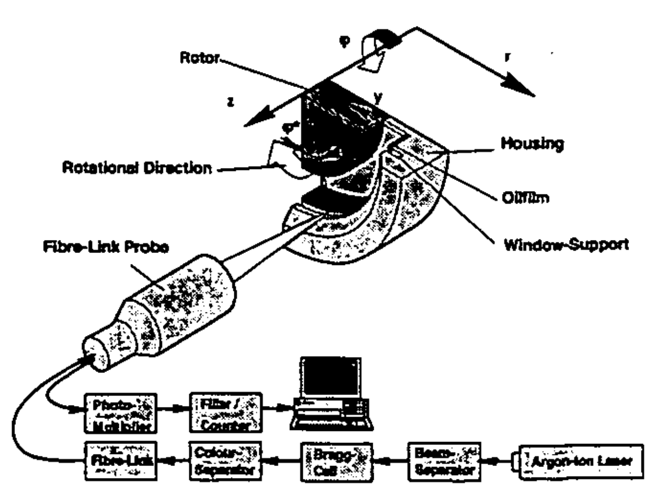

FIG. 4: LDV-SETUP (SCHEMATIC)

location of film velocity measurements and, therefore, the results gained with the setup described above will not be limited to the present investigation.

Fibre-optic LDV-setup

The determination of all relevant operating parameters and the control of the test facility are done by use of the computer based data acquisition described by Wittig et al. (1994b). Film velocity measurements have been performed using the setup shown schematically in Fig. 4. The system presented in detail by Jakoby et al. (1994) has been used in the present investigation for the determination of oil film velocities parallel to the housing wall. It comprises a 4 W Argon-Ion laser, a standard optic with 40 MHz Bragg-cell, color and beam separators, and a 2D fibre probe. The availability of a fibre-probe LDV-system guarantees a high degree of flexibility which is necessary for applications under real engine conditions as given in this bearing chamber flow investigation. The signals are detected in a backscatter mode by a fibre link and photomultiplier setup in combination with filters and the counter processors. Working as a data link between the counter processors and the micro computer (PC 80386), the interface card writes all data into the computer memory.

Due to the thin oil films in the range of $1\ mm \leq \overline{h}_F \leq 2\ mm$ measurements have to be performed close to the wall and, therefore, only one component of the LDV could be used. Additional seeding of the flow is not necessary here, because small gas bubbles interspersed in the oil film as mentioned above can be used as tracer particles. All velocity profiles and measured data presented in the next section have been determined at an axial distance from the transparent cover (7, Fig. 3) large enough to avoid any effects of wall friction. Preliminary studies have shown, that an axial distance of $\Delta z = 2\ mm$ from the cover has been sufficient for this assumption. The circumferential position of all measurements has been chosen to be $\varphi^* = 30°$ upstream of the scavenge port (Fig. 4). Temperature measurements necessary for the calculation of oil properties have been performed 5° downstream from this position by use of NiCr-Ni thermocouples mounted at a radial distance of $\Delta r = 1\ mm$.

EXPERIMENTAL RESULTS AND DISCUSSION

This investigation has been focussed on the question of whether the oil film flow in bearing chambers can be characterized by a theoretical approach for the velocity profile, which is the main presupposition for any further modelling of bearing chamber flow conditions. The analysis is based on measured data as given in Fig. 5. The operating conditions are given by a lubrication oil flow of $\dot{V}_{II} = 150\ l/h$ and a sealing air flow of $\dot{m}_{II} = 10\ g/s$ to chamber II, and by rotational speeds of $n_W = 3000\ rpm$ to $16000\ rpm$. As expected, an increase of the rotational speed results in an increase of the gas/liquid interface shear force and, as a consequence, in enhanced film velocities and decreasing film thickness.

According to the outline on film flow theories given before, transferable results can be derived by referring the data shown in Fig. 5 to the shear velocity u_τ and the kinematic viscosity ν_F, respectively. The wall shear stress, necessary for a dimensionless treatment of the oil film flow behaviour, can be determined from one measured data point $u_F = f\ (y_F)$ by use of Eq. (5) and a given approach for the eddy viscosity (Eqs. (9) to (11)). It has to be pointed out that each model for the eddy viscosity results in a different wall shear stress. Thus, different quantities have to be considered for the definition of the dimensionless parameters σ, u_F^+, and y_F^+.

A comparison between measurement and theory for each of the models given by Eqs. (9) to (11) is presented in Fig. 6. Each single frame shows one operating condition characterized by air and oil flows and rotational speed.

Assuming laminar flow conditions, wall shear stresses have been calculated iteratively from the measured values $u_F = f\ (y_F)$ next to the wall ($y_F = 0.5\ mm$) and by use of Eq. (9). With the wall shear stress and the circumferential location of the measurements ($\varphi^* = 30°$) the parameter $\sigma = f\ (\tau_l)$ and the shear velocity $u_\tau = f\ (\tau_l)$ can be evaluated. Therefore, theoretical velocity profiles as a function of $\sigma\ (\tau_l)$ can be compared with measured data referred to $u_\tau\ (\tau_l)$.

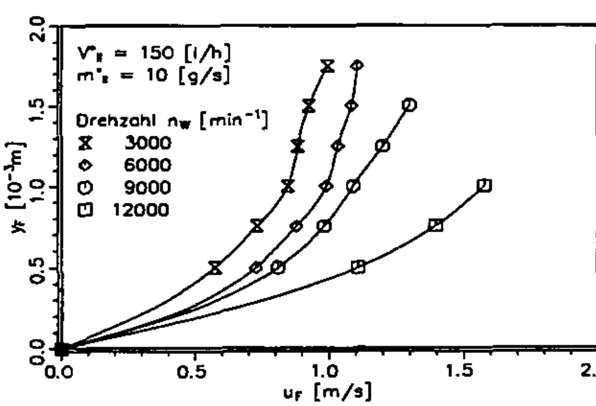

FIG. 5: FILM VELOCITY PROFILES IN BEARING CHAMBER II: φ^*=30°, $\dot{V}_{II}$=150 l/h, $\dot{m}_{II}$=10 g/s

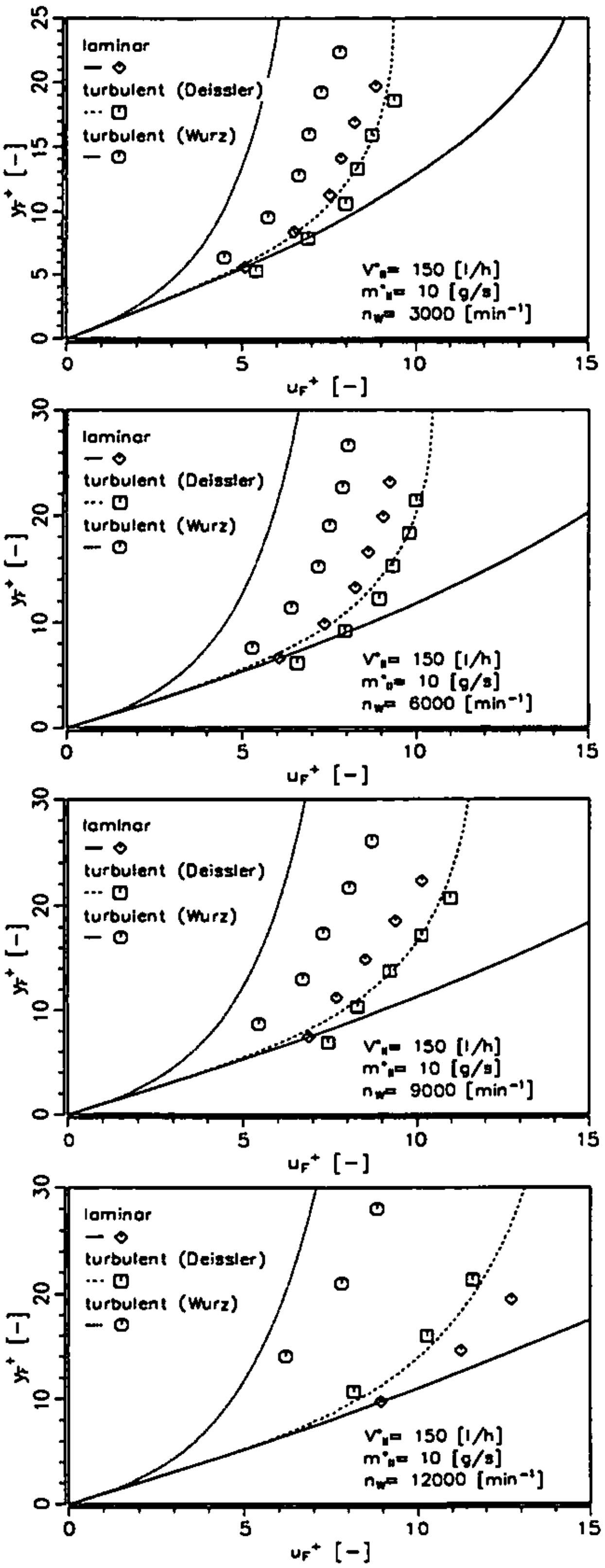

FIG. 6: NONDIMENSIONAL FILM VELOCITY PROFILES: COMPARISON WITH THEORY

In an analogous manner, but using data points $u_F = f\ (y_F)$ in the middle of the film, the values $\sigma\ (\tau_{t,Wurz})$, $u_\tau\ (\tau_{t,Wurz})$, $\sigma\ (\tau_{t,Deissler})$, and $u_\tau\ (\tau_{t,Deissler})$ have been determined. Consequently, measurements and theory can be compared as well for the approaches given in Eqs. (10) and (11).

It can be readily seen from Fig. 6 that the oil film flow in bearing chamhers is of turbulent character. All operating conditions considered here show poor agreement between measurement and theory when laminar flow is assumed. The discrepancies are larger for the low speed conditions reflecting a bigher influence of gravity forces as expressed in the parameter σ (see also Fig. 2).

Assuming turbulent flow conditions and following the approach for the eddy viscosity proposed by Wurz (1971), the deviations between measured and calculated velocity profiles are in the range of $\Delta u_F^+\ (y_F^+) =$ 17% to 30%. This approach results in an overpredicted total wall shear stress.

However, it is also demonstrated by Fig. 6 that an excellent agreement between measurement and theory for all rotational speeds can be obtained when the approach following Deissler (1954) is used for the eddy viscosity. The comparison shows maximum deviations of $\Delta u_F^+\ (y_F^+) = 5\%$ next to wall. Data points in the middle of the film and next to the surface are reflected with an excellent agreement by Eq. (11). Therefore, it can be concluded that oil films under the operating conditions described here can be characterized by an analytical approach utilizing the universal velocity profile for shear driven liquid films under gravity as given by Eq. (5) in combination with the empirical correlation for the eddy viscosity proposed by Deissler (1954). In addition, proof is given for the simplification of Dukler's (1960) proposal resulting in Eq. (11) which has been derived without consideration of a second empirical correlation for the eddy viscosity.

CONCLUSION

Based on LDV-measurements of oil film velocity profiles, an assessment of three different approaches for the characterization of oil film velocities in bearing chambers has been performed. Turbulent flow conditions have been found for all operating conditions. An analytical approach considering temperature dependent liquid properties, geometrical conditions, and a correlation for the eddy viscosity has been found to reflect the measured data excellently. This leads to several benefits with respect to an analysis of bearing chamber oil film flows. Due to the knowledge of the velocity profile a further modelling of the oil film behaviour is possible by reducing dimensional values for film height and velocity to nondimensional flow parameters as represented by the Reynolds number. Therefore, results can be expressed in data transferable to other applications. In addition, any numerical analysis of two-phase flow phenomena in bearing chambers requires a relation for the coupling between the phases. Based on the film velocity profile presented in this paper existing coupling approaches are expected to be adaptable to the computation of bearing chamber flows.

ACKNOWLEDGEMENTS

We wish to thank Rolls-Royce plc, Bristol and Motoren- und Turbinen-Union, München for funding the research described in this paper. Special thanks are due to H. Zimmermann and R.M. Coleman for helpful discussions.

REFERENCES

Deissler, R., 1954, "Heat Transfer and Fluid Friction for Fully Developed Turbulent Flow of Air and Supercritical Water with Variable Fluid Properties," *Transactions of the* ASME, Vol. 76, pp. 73–85.

Dukler, A., 1960, "Fluid Mechanics and Heat Transfer in Vertical Falling-Film Systems," *Chemical Engineering Progress Symposium Series*,Vol. 56, pp. 1–10.

Dukler, A.E., and Bergelin, O., 1952, "Characteristics of Flow in Falling Liquid Films," *Chemical Engineering Progress*, Vol. 48, pp. 557–563.

Hewitt, G., 1978, "Measurements of Two-Phase Flow Parameters," *Academic Press*, London.

Himmelsbach, J., 1992, "Zweiphasenströmungen mit schuhspannungsgetriebenen welligen Flüssigkeitsfilmen in turbulenter Heißluftströmung – Meßtechnische Erfassung und numerische Beschreibung," Dissertation, Institut für Thermische Strömungsmaschinen, Universität Karlsruhe, Germany.

Jakoby, R., Benz, E., Willmann, M., and Wittig, S., 1994, "Phase Resolved LDV-Measurements in High-Speed Rotating Disk Systems with Orifices," *Fifth International Symposium on Transport Phenomena and Dynamics of Rotating Machinery, May, 8-11*, Vol. A, pp. 573–591.

Lee, J., 1965, "Turhulent Velocity Profile of a Vertical Film Flow," *Chemical Engineering Science*, Vol. 20, pp. 553–536.

Plimon, A., 1991, "Velocity Profiles in Shear Driven Wall Films," *Experiments in Fluids*, Vol. 11, pp. 339–340.

Sattelmayer, Th., and Wittig, S., 1989, "Performance Characteristics of Prefilming Airblast Atomizers in Comparison with other Airblast Nozzles," *Encyclopedia of Fluid Mechanics*, Vol. 8, Golf Publishing Company.

Suter, P., and Boyman, T., 1978, "Transport Phenomena in Labyrinth-Seals of Turbomachines," AGARD *Conference Proceedings*, No. 237, pp. 8.1–8.10.

Ueda, T., and Tanaka, H., 1975, "Measurements of Velocity, Temperature and Velocity Fluctuation Distributions in Falling Liquid Films," *International Journal of Multiphase Flow*, Vol. 2, pp. 261–272.

Waschka, W., Scherer, Th., Kim, S., and Wittig S., 1992, "Study of Heat Transfer and Leakage in High-Rotating Stepped Labyrinth Seals," *4th International Symposium on Transport Phenomena and Dynamics of Rotating Machinery*, Honolulu, USA.

Waschka, W., Wittig, S., and Kim, S., 1990, "Influence of High Rotational Speeds on the Heat Transfer and Discharge Coefficients in Labyrinth Seals," ASME *Gas Turbine and Aeroengine Congress and Exposition*, Brussels, Belgium, 90-GT-330.

Waschka, W., Wittig, S., Scherer, Th., and Kim, S., 1991, "Leakage Loss and Heat Transfer in High-Speed Rotating Labyrinth Seals: An Experimental Verification of Numerical Codes," *Proceedings of the 1991 Yokohama International Gas Turbine Congress*, Vol. II, pp. 239–247.

Wittig, S., Glahn, A., and Himmelsbach, J., 1994b, "Influence of High Rotational Speeds on Heat Transfer and Oil Film Thickness in Aero Engine Bearing Chambers," ASME *Journal of Engineering for Gas Turbine and Power*, Vol. 116, pp. 395–401.

Wittig, S., Himmelsbach, J., Noll, B., Feld, H.J., and Samenfink, W., 1992, "Motion and Evaporation of Shear - Driven Liquid Films in Turbulent Gases," ASME *Journal of Engineering for Gas Turbine and Power*, Vol. 114, pp. 395–400.

Wittig, S., Kim, S., Jakoby, R., Weißert, I., 1994a, "Experimental and Numerical Study of Orifice Discharge Coefficients in High Speed Rotating Disks," *International Gas Turbine and Aeroengine Congress and Exposition*, The Hague, The Netherlands, Paper 94-GT-142.

Wittig, S., and Schulz, A., 1992, "A Survey on Efforts in Heat and Mass Transfer Analysis in Aero Engine Secondary Air/Oil Systems," *International Symposium on Heat Transfer in Turbomachinery*, Athens, Greece.

Wurz, D., 1971, "Experimentelle Untersuchung des Strömungsverhaltens dünner Wasserfilme und deren Rückwirkung auf einen gleichgerichteten Luftstrom mäßiger bis hoher Unterschallgeschwindigkeit," Dissertation, Institut für Thermische Strömungsmaschinen, Universität Karlsruhe, Germany.