Contributions to Panel Discussions at the June 1973 ASME Symposium on Film Lubrication: Turbulence and Related Phenomena

Approximation in Analysis: Opinions

D. Dowson, Chairman
J. H. Vohr, Vice-Chairman

D. DOWSON[1]

Professor R. A. Burton of the host Institution opened the discussion with a clear exposition of his view of the relationship between current theoretical and experimental studies of turbulent lubrication. He stressed the fact that the simple "Wall-Law" approach to turbulent lubrication problems appeared to give quite good predictions of bearing behavior, although he recognized that there were still some problems to be considered. He gave an account of an experiment in which inlet pressure build-up appeared to exercise an important influence upon bearing performance and he indicated that errors resulting from inaccuracy in the various turbulent lubrication theories might not be as important as the effect of the inlet pressure build-up in many cases. He also drew attention to the importance of boundary-layer carry-over effects in thrust bearings operating in the turbulent regime.

R. A. BURTON[2]

I will keep my remarks brief, because there will be a chance to expand this in some detail elsewhere in the meeting.

First I wish to note that the pivotal work of Smith and Fuller [1][3] on a "turbulent-film" bearing was actually in the Taylor-vortex-flow regime throughout. The fact that turbulent film analyses have served to predict the pressure distribution in such a flow should be confidence inspiring. It indicates that there is no great difference between the bulk behavior of turbulent flow and vortex flow. Each involves an energy or momentum transport mechanisms which has a dominant effect on the flow except very near the walls. Apparently, even though the appearance of the two types of flow is quite different, the overall effect of momentum transport is quite similar.

More recent work [2] has established that velocity profiles for these two types of flow are quite similar and that for Couette flow the vortex regime has a friction coefficient blending smoothly with that for turbulent flow. For more confidence in our calculations, however, we should develop a theory which will show clearly when vortex flow and turbulent flow may be dealt with in the same way, and when they may not.

As a second point I note that in short bearings and stepped pads there are large inertia-related pressure jumps. In many cases the effects of these on the flow are predominant. Although they were demonstrated to occur several years ago [2] they have not been generally recognized as important effects until recently. It is important that we learn to deal with inertia effects at steps and at the entrance to bearing pads. The latter are undoubtedly conditioned by aspects of the flow external to the film and this must also be understood if we are to be able to design with confidence.

References

1 Smith, M. I., and Fuller, D. D., "Journal Bearing Operation at Super Laminar Speeds," TRANS. ASME, Vol. 78, 1956, p. 49.

2 Burton, R. A., and Carper, H. J., "An Experimental Study of Annular Flows With Applications in Turbulent Film Lubrication," JOURNAL OF LUBRICATION TECHNOLOGY, TRANS. ASME, Series F, Vol. 89, No. 3, July 1967, pp. 381–391.

Professor S. Ostrach of Case Western Reserve University presented an excellent review of the development of analytical studies of turbulent lubrication. He pointed out that many so-called turbulent lubrication problems in journal bearings could not be analysed on the basis of data obtained from studies of fully developed duct flows owing to the occurrence of a laminar secondary flow in the form of vortices. The problem was further complicated by abrupt changes of geometry at lubricant entry points and at the edges of thrust pads of finite length.

Concern was expressed for the neglect of inertia terms, a recurring topic throughout the Symposium, and this thoughtful review ended with a warning about the unnecessary difficulties introduced by the dichotomy between fluid dynamicists and fluid-film bearing designers.

SIMON OSTRACH[4]

Where Do We Go From Here?

From most all the published literature on high-speed film lubrication one sees that the first attempts to deal with the deviations from classical lubrication theory were ad hoc and based on the premise that turbulence was the cause of the differences. The more recent papers are more complicated versions of the same approach or comparisons of various versions of these semi-empirical or curve-fitting techniques. No basically new concepts seem to have been introduced in the many ensuing years and no direct appreciation has been shown of the fact that the complex physical phenomena encountered are essentially fluid dynamical in nature and that the insights, knowledge, and techniques of that field should be utilized in attacking the problem. However, from talking to a number of the key workers in the field at this meeting and hearing some of the papers presented it is encouraging to find that there is, in fact, an awareness that careful

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fluid dynamics research is essential to resolve the many problems that still exist.

If we accept the primacy of fluid mechanics to the problem of high-speed lubrication we can provide some answers to the question which is the theme of this panel. Let me say, first of all, that if the ad hoc approaches had led to information that would have permitted high-speed bearings to be designed with confidence then there would have been no need for further work. Obviously, this was not the case. Therefore, more detailed understanding of the physics of the problem is necessary. In other areas of modern technology the key to many significant advances was the intimate relation between "doers" and "knowers" such as I propose herein.

In order to know where to go one must first clearly delineate what information is essential to solve the problem at hand. No such clear definition seems to have been made and that question has been raised by others at this meeting. If only gross quantities, like load-carrying capacity or torque, are needed it may well be that nonlinear or inertia effects may not be important. However, for so complex a problem as confronts us I believe that more detailed information is essential and the nonlinear effects will have to be dealt with. Essential physical characteristics, such as secondary flows, are discarded with the neglect of the inertia terms.

For further progress we must all recognize that the high-speed journal bearing is fundamentally different from other kinds of bearings in that the dominant aspect of the flow is the vortex structure. Therefore, basic fluid mechanics research is essential to understand the prevortex, the vortex, the wavy vortex, and vortex plus turbulence regimes. Di Prima and Stuart recognize this well and their work on this problem is far ahead of others and gives indications of supplying much of the information required to solve that problem. For other types of bearings there is need for work on the effects of abrupt changes in geometry and the like on both transition and turbulence structure. However, for this work the use of fully developed turbulence concepts must be abandoned.

Other speakers have already mentioned an aspect which I also think merits serious study. That is the effect of time-dependent boundary conditions. It is intuitively clear that shaft orbiting, slider oscillations, and the like will affect the flow transition. It may well be that such large-scale forced disturbances may also play a significant role in determining the structure of the flow regimes. Thus, the coupling between the shaft and fluid-film dynamics may well be more important than in classical low Reynolds number theory.

Finally, I have heard many leading workers in the field express concern about the existing experimental data. It seems rather remarkable that so many people who claim to be practically oriented have spent so much time on curve-fitting methods rather than in doing definitive experiments. To do definitive experiments, however, one has again to return to fluid mechanics and determine the fundamental dimensionless parameters that describe the phenomena and then utilize similitude principles. In lubrication theory a number of specialized dimensionless parameters have been developed and have been in vogue. Their meaning, however, is both vague and confusing. If the experiments were designed according to similitude principles based on the fundamental parameters no one would have to be nervous about large gap or large scale experiments. Even if large scale experiments could be designed to simulate real bearings properly I think that the use of hot-wire anemometers are limited only to yielding gross data. They cannot, for example, distinguish between vortices and turbulence. Therefore, for more detailed information other techniques will have to be used. G. I. Taylor and Coles have used particle suspensions for flow visualization in order to obtain qualitative information. For such a technique bearing clearances have to be large to accomodate the particles. I would like to suggest that consideration be given to the use of macromolecular solutions for the experiments and to utilize their birefringence properties for obtaining the data. If proper simulation is not possible with large clearances it would certainly seem worthwhile to investigate this technique.

In conclusion, I want to say that in my judgment this Symposium has been a big step in the right direction of where to go because it has established a dialogue among people of diverse approaches and background and has led to an interchange of viewpoints in a constructive and helpful manner.

Dr. G. G. Hirs of Centec, Gesellschaft fur Centrifugentechnik, MBH, enlivened the evening with his amusing yet sincere interpretation of the merits of his bulk-flow approach to the analysis of turbulent lubrication. His analogy between lubricating films operating in the turbulent regime and a train in which the resistance (shear stress) between the wheels and the track was a function of speed (Reynolds number) was presented with clarity.

The merit of this approach in which the wall-shear stress is related to a Reynolds number for the flow based upon the mean velocity relative to the wall by a simple expression was well taken. Furthermore, Dr. Hirs stressed the point that the empirical constants in the theory could be obtained from experimental studies and did not call for a detailed knowledge of the cross-film velocity distribution. His analogy between turbulent films and trains could be said to have taken the term "transportation phenomena" to its logical conclusion.

G. G. HIRS[5]

A Transportation Phenomenon

After having listened to the opinions of fluid mechanics experts on Taylor vortices and turbulence, I feel slightly scared and I would like to start talking about a totally different transportation phenomenon. I would like to start talking about trains before entering the more controversial subject. In Fig. 1, I have sketched a very simplified train model. The train runs on two rails, it is infinitely long and it carries passengers. The train is slowly increasing its speed (u). Frictional forces are acting on the rails and these can be averaged over the rail area yielding a rail surface shear stress (τ). The relation between τ and u for this particular train has been measured for this particular train and is shown in Fig. 2. The figure clearly shows that there is a slow speed regime where the frictional stress rises slowly and a high speed regime where it rises quickly.

It is typical of mechanical engineers that they are able to live with the sudden increase of friction and adapt their train design if necessary.

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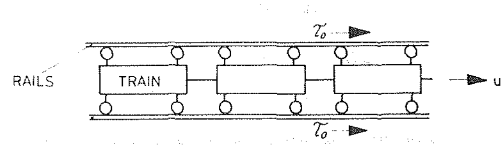

Fig. 1

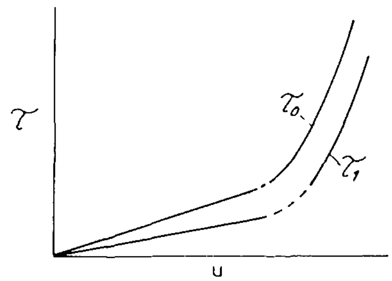

Fig. 2 Surface Shear stress-train speed

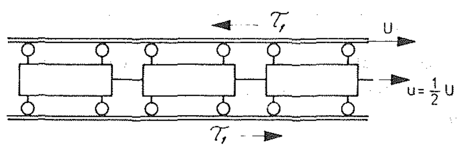

Fig. 3

Nevertheless, they are curious why this phenomenon occurs and they can not suppress looking through the windows of their passenger train. These passengers sit quietly in their seats and read papers at the slower speeds but there appears to exist a critical speed at which they lay down their papers and become excitable. As soon as the high speed regime has been entered they really start jumping and running about. It is clear that we are confronted here with a train load of fluid mechanics experts discussing whether they have entered the Taylor vortex regime or the turbulent frow regime.

Before leaving my train model, I would like to mention another peculiar property of it. In this train model it is possible to slide the rails with respect to each other, see Fig. 3. In this figure, it is shown that the frictional stresses on the surfaces are equal and opposite for the special case that the train speed (u) is half the sliding speed of the upper rail (U). This is in contrast with Fig. 1 where the frictional stresses have the same direction. In experiments, it has been found that different conditions in Fig. 1 and 3 do not greatly influence the frictional stress-train speed relation. Indeed, when plotting τ_0 and τ_1 in a graph shown in Fig. 2, the lines for τ_0 and τ_1 are parallel and within 20 percent in the high speed regime.

Comparable properties exist for turbulent film flow and the author has thoroughly exploited these in his bulk flow theory. This approach is generally applicable to all turbulent bearings including inertia effects. However, I will restrict myself to an example dealing with self-acting, inertialess bearings with smooth surfaces.

For such a bearing the following combination of Sommerfeld and Reynolds number can be formed for the turbulent flow regime.

$$\frac{\mu U r}{p_m h^2}\left(\frac{\rho U h}{\mu}\right)^{0.75} = f_1\left(\epsilon, \frac{l}{r}\right) \tag{1}$$

By comparison, the following has been found for the laminar flow regime

$$\frac{\mu U r}{p_m h^2} = f_2\left(\epsilon, \frac{l}{r}\right) \tag{2}$$

By extrapolating (1) toward lower Reynolds numbers, a critical Reynolds number can be found were laminar and turbulent Sommerfeld number are equal. This critical Reynolds number is about 1000 for wide bearings and 2000 for short bearings.

A relationship can also be found for the friction factor and the combination of Sommerfeld number and Reynolds number for the turbulent flow regime.

$$\frac{\tau_m}{p_m}\frac{l}{r} = f\left\{\frac{\mu U r}{p_m h^2}\left(\frac{\rho U h}{\mu}\right)^{0.75}\right\} \tag{3}$$

By comparison, the following has been found for the laminar flow regime

$$\frac{\tau_m}{p_m}\frac{\mu}{h} = f\left(\frac{\mu U r}{p m h^2}\right) \tag{4}$$

In both cases, the friction factor rises with an increasing Sommerfeld number and Reynolds number.

The main reason why I have introduced (1)–(4) is to explain why temperature and power loss problems must occur in turbulent bearings. In the laminar flow regime the value of the Sommerfeld number shows moderate variations only over a great speed regime. The sliding speed forces the Sommerfeld value upwards but the power loss, the increase in operating temperature and the associated decrease in viscosity force this value downwards. I know many bearings where the Sommerfeld value is roughly constant over most of the speed range and where power consumption rises linearly with speed.

Conditions are different in the turbulent flow regime. The combination of viscosity and sliding speed μU in the laminar Sommerfeld number changes into

$\mu^{0.25} U^{1.75}$ in the turbulent Sommerfeld/Reynolds number.

It is clear that the increase in operating temperature and the associated lower viscosity will not introduce a leveling off of the S/R number with speed. According to (3) this means a steady rise of the friction factor with speed.

I would like to mention the following remedies for this problem of increased power loss and operating temperature.

1 design changes leading to a smaller wetted area both inside and outside the lubricant film

2 oil development leading to the use of oils with steeper viscosity-temperature characteristics or oils with additives that suppress turbulence.

Professor V. N. Constantinescu of the Polytechnic Institute of Bucharest presented a succinct account of the approximations involved in current analytical studies of turbulent lubrication—both in relation to the reduction of basic equations and the evaluation of turbulent stresses. He remained reasonably optimistic about the present status and future progress in the field. Calculations based upon current methods appeared to give good predictions of actual bearing behavior in many cases.

Professor Constantinescu did, however, draw attention to a number of problem areas. In particular the neglect of inertia terms posed an important question in relation to both the dynamic head at entry to the bearing and their influence upon the pressure distribution within the bearing. Additionally, the influence of inertia effects upon cavitation within journal bearings and film stiffness in high-speed bearings called for further consideration.

In concluding his useful summary Professor Constantinescu called for further studies of transition in both journal and thrust bearings.

V. N. CONSTANTINESCU[6]

The present results obtained in turbulent lubrication theory include obviously a large number of approximations. Several approximations are simply taken from the classical laminar lubrication, although some are questionable for large Reynolds numbers. Some are new approximations, necessary in order to correlate in reasonably simple ways, the new effects and phenomena involved. A number of the most important approximations included in various analyses may be listed as follows:

1 Basic Equations. The basic equations are the momentum equations (Navier-Stokes equations), continuity and energy equations. The Reynolds method of assuming each variable φ of the form

$$\varphi = \bar{\varphi} + \varphi'$$

is commonly used, where φ' is a fluctuation. Averaging term by term each of the equations is employed and the assumption of small film thickness is then used in order to neglect a series of terms. This includes or leads to some approximations which may not be valid under certain circumstances. Thus:

(*a*) Convective inertia forces (other than turbulent stresses) are neglected. In laminar flow, the ratio of the inertia forces against viscous forces is

$$\frac{\rho V h^2}{l} = \mathrm{Re}\frac{h}{l}.$$

This ratio is no longer small where Re > 10^3. In turbulent flow, tangential stresses are increased in a certain ratio $\bar{\tau}_c$ ($\bar{\tau}_c = 1$ in laminar flow, $\bar{\tau}_c > 1$ in turbulent flow). Therefore, a criterion similar to that used in laminar flow should be

$$\frac{1}{\bar{\tau}_c}\,\mathrm{Re}\,\frac{h}{l}.$$

This ratio again can no longer be considered small at large Reynolds numbers.

(*b*) The pressure is variable across the film according to

$$\frac{\partial p}{\partial y} = -\frac{\partial}{\partial y}\,(\overline{\rho v'^2}).$$

This does not seem to introduce a significant effect on the pressure distribution on the two surfaces, but some details of the flow may be altered, for example $\partial\tau_{xy}/\partial y$ is no longer constant across the film (but assumes the same values on the two surfaces).

(*c*) Steady-state calculations are extended to nonsteady regimes by using the same type of averaging. Then, obviously, a question of the validity of the averaging method may be raised when the frequency of the mean motion becomes of the same order of magnitude as the frequency spectrum of turbulent fluctuations.

2 Evaluation of Turbulent Stresses. The following approximations may be subjected to criticism:

(*a*) The extension of the existing information for parallel flow to lubrication conditions (almost but not rigorous parallel flow).

(*b*) Isotropy of the apparent (eddy) viscosity. It is possible that the reciprocal influence of the two walls may alter isotropic behavior of turbulent stresses.

(*c*) Separate and quasi-independent behavior near each wall is assumed together with some rather arbitrary joining conditions in the mid-channel region.

(*d*) Additional approximations are used according to each model. Thus for example, the analytical calculations performed by using the mixing length model considered the laminar sublayer close to the wall but neglected the transition (buffer) layer. In order to compensate for the error introduced, the mixing length was considered as

$$l = k^* y$$

where k^* is no longer $k^* = 0.4$ but a function of the Reynolds number. In the law of wall model, the original wall stress τ_w was replaced by local stress τ in order to remove the difficulties arising for situations of zero wall stress and extend the model to variable stress situations. Finally, the linearized approaches consider only small perturbations of a Couette flow.

It is probable that most of the mentioned approximations do not alter qualitatively the description of the phenomenon, neither are they likely to alter the overall performances of turbulent bearings. There is however one exception, namely the influence of inertia forces. Inertia forces in the film itself alter the pressure distribution, as dramatically emphasized in step bearings. In addition, the pressure at the inlet edge of a bearing pad is changed by a partial conversion in static pressure of the dynamic head of the incoming flow, together with a boundary layer development starting from the stationary edge. The question of whether or not inertia forces alter the conditions of film rupture, cavitation or separation is also of importance. Finally, time dependent inertia forces ($\rho\ \partial u/\partial t$) lead generally to a reduction of the film stiffness which may be important at high rotational speeds.

The problem of transition regimes in various bearings is still unsolved. Laminar stability criteria for full journal bearings with zero or small eccentricity ratios exists (inner member rotating). Very little information concerning transition in thrust bearings is available. Finally no comprehensive procedures are available for evaluating the performances of bearings operating in the transition region.

An open debate followed these presentations from the Panelists and several speakers contributed to the discussion. Two of the more substantial contributions are included in this Review.

In the first, Professor Morkovin of the Illinois Institute of Technology introduced a timely note of caution which is recommended for all persons interested in furthering studies of turbulent lubrication. Many of his points drawn from wider studies of turbulent flow can be detected in the current state of turbulent lubrication analysis. He voiced his concern for the lack of basic physics support for many current theories and he correctly noted that agreement between theories based upon different approaches did not necessarily provide cause for optimism. It is interesting that the same point was to be made in a more formal session the following morning by Dr. C. M. Taylor.

1 Panel on Approximations in Analysis[7]

(D. Dowson, Chairman; J. H. Vohr, Vice-Chairman)

MARK V. MORKOVIN[8]

Since serious concern about turbulence effects in lubrication

[6] Polytechnic Institute of Bucharest, Bucharest, Romania.

[7] The background effort and preparation for this cross-discipline discussion was supported under Themis Project, USAF F 44620-69-0022, AFOSR (NAM).

[8] Illinois Institute of Technology, Chicago, Ill.

has arisen relatively recently perhaps I could share with you some relevant prior experiences with turbulence theories for incompressible boundary layers and for compressible free shear layers. A number of us—Steve Kline, Don Coles, Gino Sovran, myself and others—had felt in 1966 that time had come to subject the computer-spawned theories to a *systematic* confrontation with carefully selected experiments, i.e., with the most reliable physical evidence at hand. Thus Sympo 68 [1][9] tested some 25 distinct prediction methods on 16 "certified" mandatory experimental boundary layers (compulsory figure-skating) and 17 additional (free-skating) flows. The relative consistency and merits of individual methods and of groups of methods[10] were vigorously analyzed and recorded in the *Proceedings*. A year ago the "Son of Sympo 68," [2], was organized by a group of NASA Langley researchers—Stan Birch, Dennis Bushnell, the late Mitch Bertram, and others—to shed in similarly organized fashion some light on experiments and prediction methods in Free Turbulent Shear Flows. Since my name appears as a member of the Organizing as well as the Evaluation Committees of both of these cooperative efforts, I should make clear that the following observations, hopefully useful to turbulent lubricators, represent purely my personal interpretations. The turbulence oriented specialists among you are urged to go to the *Proceedings* [1, 2] themselves.

1 A large majority of the methods agreed significantly *less well* with the widely ranging mandatory experimental data than the authors had believed before the exercises. As the samples of data broaden, correlations become fuzzier, and the so-called absolute empirical constants in various theoretical models become slowly varying parameters for which additional modeling often becomes necessary. In turbulent lubrication the amount of experimental information appears to be much more scarce than in either [1] or [2] and a controlled systematic comparison is yet to be made. Thus at this stage turbulent lubrication data should be much easier to "fit" and we should expect a similar deterioration in the "excellent agreement between theory and experiment" as broader experiments accumulate.

2 Distinct prediction methods have occasionally been in excellent agreement among themselves but in relatively poor agreement with experiments. Thus agreement between theories alone does not provide sufficient cause for optimism.

3 We should understand that the designation "theory" refers to rather gross modeling of the exceedingly complex physics of turbulence which was described to you this afternoon in Kovasznay's survey and in Stewart's movie through which I gave you a conducted tour [22]. The solution for the so-called closure problem of always having more unknowns than basic equations is not in sight. The consequent numerous possibilities of assumptions whether in theoretical formulations with one averaged equation or with multiple, coupled partial differential equations have indeed led to a population explosion in competing theories. Since much of the basic physics is missing, I view these "theories" as more or less complex means of generating families of functions which can then be fitted to the experimental results. As such, they appear then as *sophisticated interpolations between available experimental data* and must be viewed with caution when extrapolated to previously untested ranges of parameters. As the enthusiastic authors compute more and more cases, they tend to endow various ingredients of their models with a sense of reality which of necessity conflicts with the sense of reality of the competing theorists. They are all forced to adjust their empirical constants or functions when they generalize beyond the original data, say to compressible flows or to very high Reynolds numbers. Part of the explanation of observation (2) is that the detailed structure of the theories often causes little difference between predictions as long as these have been originally fitted to essentially the same data.

4 The other part of the explanation is that most theories conform to basic constraints of mass and momentum conservation and to asymptotically valid constraints such as the (viscous) similarity law of the wall and the (inviscid) defect or "core" *similarity* law [3, 4]. *Quasi-parallel flow theories which obey all* these should be dimensionally correct and the functions they generate should approach the wall or the "free stream" properly once their adjustable parameters are fitted to the data.

For turbulent lubrication theorists a rather reassuring observation emerged from the Langley Working Conference [2]: similar methods seem to yield more divergent results for free shear layers than for boundary layers, presumably because of the *dominant role played by the law of the wall* in the latter cases. In other words, disparate behavior away from the walls, allowed by the differences between methods, apparently tends to be overwhelmed by the uniformizing effect of the wall which is absent for free flows. In lubrication problems, the constraint of the law of the wall is present from both sides so that the average flow fields allowed by the basic and asymptotic constraints may be much more limited and therefore more easily "predictable," at least for two-dimensional cases. On the other hand, for the not infrequent three-dimensional flows in lubricating passages all these constraints allow more latitude and will require more careful modeling and experimentation.

5 Concern has been expressed here that for *journal bearings* the *presence of quasi-organized Taylor vortices* at the higher Reynolds numbers makes the flow *rather unlike normal turbulent flows.* However, as observed under (3), the turbulent prediction methods *do not really describe the details of the physics anyway*, so that these methods might still yield acceptable prediction accuracy if the appropriate constants or functions were adjusted to the journal data. The desired output: wall shear, pressure distribution, wall temperature, etc., all represent *averages* which are unlikely to hide *dangerous local peaks* due to the vestigious cellular structure. For practical purposes, the cells, which probably dance and jitter at the higher Reynolds numbers, would be difficult to distinguish from "turbulence with an adjusted mixing length." Since the issue is not one of understanding or detail flow description, one would wish to have clear experimental evidence that the above outputs are very poorly predicted by such approximations before attempting more sophistication. The presence of a force field similar to that in journal bearings is known to have an effect on the structure of turbulence in boundary layers with concave streamlines. The up-to-date account of the associated work of Rotta, Bradshaw and others [5] may possible suggest an improved treatment for "turbulent" journal bearings.

In the foregoing five observations I have tried to convey cross-disciplines impressions of several strictly practical problems with turbulent prediction theories. In Part 2 of this discussion I will address myself to broader problems.

2 Panel on New Directions

(R. A. Burton, Chairman)

As a cross-disciplines member of this Panel, I may be an unknown to the lubrication community so that perhaps I should first identify my priorities—my biases if you wish. My primary urge is to understand phenomena deeply, in my guts as well as in my head, and then to use the understanding to manipulate or control the phenomena if possible [6]. I have therefore been listening, asking and learning at this Symposium and the remarks

[9] *Numbers in brackets designate References at end of discussion.*

[10] The uninitiated would do well to read first W. C. Reynolds' "Morphology of Turbulent-Boundary Layer Prediction Methods." in [1]. P. Bradshaw's 1972 critical survey [3] is another must, even though some of his projections are considered controversial in some quarters.

below represent hopefully constructive reactions to your needs as they come across to a sympathetic "fluid neighbor."

(*a*) Having spent thirteen years as an internal fluid consultant in the aerospace industry I have been especially impressed by *three figures* presented here which seem to *epitomize the character of a number of your basic problems at higher Reynolds numbers.* Fig. 1 of N. H. New (73-LubS-5) underscores the increasing importance of "churning losses" *on top* of the useful, load-associated shear losses (presumably turbulent-like at the higher design speeds). Fig. 10 of Gardner and Ulschmid (73-LubS-7) presents a sharp contrast between the losses and transition speeds in the same 19-inch sleeve bearing with only the difference in single versus double oil inlets. Diagnostics of the subphenomena associated with these overall performance curves would presumably include secondary flows (churning), three-dimensional flows shifting with speed, temperature rise, intermittency of turbulence, etc., in addition to "honest turbulence." The substantial difference in transition shaft speeds in Fig. 10 and the peculiar reversal of temperature rise in Garner-Ulschmid's Fig. 9 (also mentioned by R. S. Gregory in his presentation of 73-LubS-16) emphasize the empirical lack of determinism and/or our ignorance of the instability and transition mechanisms. There seems little question that the phenomena presented in the three figures are both practically important and difficult to pin down.

(*b*) Since these fluid phenomena are exceedingly complex we need to employ all viable tools at hand—with minimum of intergroup impatience or snobbishness—to gain empirical or conceptual insight. Thus, we need to understand the mechanisms and concepts lurking in *pure analytical approaches*, e.g., in the singular-perturbation results for the instability of eccentric journal bearings of DiPrima and Stuart (73-LubS-4) and their relation to the simple but possibly oversimplified earlier concept of local instability. The related concept of the *rate at which vorticity fields forget their genesis and past* as they are convected along the film in presence of pressure gradients and of new sources (or sinks) or vorticity at the walls of the bearing underlies the physics of many approximations including those in the turbulent regime.

How we can learn about the nonlinear inertial effects from *numerical experimentation with Navier-Stokes equations* was demonstrated by C. H. T. Pan this afternoon (73-LubS-21) for the example of the step bearing of Putre [7]. These difference techniques do have their problems with convergence and costs at the higher Reynolds numbers and with spatial resolution near abrupt changes of geometry so that ensuing compromises, e.g., the neglect this morning of convective derivatives in the momentum equation by K. H. Huebner (73-LubS-14) or the vorticity approximations of Dailey and Geiger [p. 7 of 8], do introduce some errors in the results. Nevertheless, our engineering intuition can extract reliable conceptual building blocks from the indicated parametric developments in monitored details of the computed fields as the Huebner and Dailey-Geiger contributions well exemplify. How otherwise can we get as convincing a picture of the thermal field, including the upstream influence through heat conduction in the wall? And the Dailey-Geiger paper includes revealing comparisons with experimental pressure drops for the same geometries.

In view of the aforementioned numerical difficulties the prospects of including among "New Directions" cost-effective computer solutions of NS equations in *three-dimensional problems* to help us with the understanding of the role of *secondary flows* and of *churning losses*, observation (*a*), do not appear very promising. However, we should keep an eye on the optimistic prognoses of the Los Alamos group, e.g. [9], and on the interesting pressure-velocity approach of D. B. Spalding's group at Imperial College, e.g. [10].

As I explained in Part 1 of the discussion in observations (3) and (4) the approximations in turbulent theories are not comparable in quality to those of the more basic theoretical approaches above. Here a computer-bred intuition may foster a false sense of security when extrapolations beyond the range of the original empirical information are attempted. Hence additional experimentation appears essential.

(*c*) Broadly speaking *three types of experimenters* can well peacefully and patiently supplement each others efforts. The experimenters who aim primarily at design data usually face problems of accessibility of instrumentation, overabundance of geometrical parameters, time pressure, etc. In presence of a large number of parameters, scaling and clear assignation of causes present considerable difficulties. In fact, unsuspected parameters in the form of departures of surfaces from presumed shape and smoothness, of vibrations, of imperfectness of axes, etc. do at times affect the measurements because of the close tolerances.

The more academic experimenters choose simplest geometries with instrument accessibility, which can help to build detailed basic concepts. Hopefully, our engineering intuition can group such concepts and with the aid of scaling laws can infer lubricating performance under realistic design conditions.

Experiments aimed at checking turbulent theories as such may well straddle the two sets of conditions above and strive for multiple, preferably redundant measurements on more complex geometries closer to realistic sizes. As per the discussion of this afternoon the various measurements should be *designed to relate* in sufficiently quantitative detail the *pressure gradients, the bulk flow, and the wall shear stress* for typical geometries and "boundary conditions," including spanwise flows. From what I understand of the flow categories which Dr. Hirs intends to present to us tomorrow [11] these categories would indeed represent the type of *systematic experimentation* which was found essential in turbulent predictions in other fields. The proposal covers the field well enough (including roughness and grooving) so as to merit industry-wide support which should be expeditiously translated into governmental and private funding—the key to success and utilization.

Insofar as the financial boundary conditions will permit, the above objectives should be broadened to include some information on *velocity gradients* (and hence on the local *dissipative sources of heat*) and on *wall temperatures.* At the speeds under consideration such information will be important for proper "fitting" of the turbulent theories and for their correct application to high-performance bearings.

While the choice of the measurements is understandably circumscribed by the severity of the experimental environment both Dr. Kovasznay and I were surprised by the apparent absence of at least qualitative *diagnostics*[11] based on *time-dependent gauges* (simply or in arrays): hot-films, piezoelectric transducers, wall-deflection indicators, and possibly on Laser-velocimeters. In other fields frequency (including intermittency) and relative-amplitude information turned out to be most valuable not only for understanding the true nature of the usually overidealized flow, but also for trouble-shooting, "cures" of poor performance, and development guidance.

(*d*) To my comments in Part 1 on turbulent predictive theories (3)–(5), I would like to add some speculations on the relevant physics in thin confined films. By 1954 Klebanoff [13] established that in a high-Reynolds number boundary layer approximately 50 percent of turbulent energy at a given station is produced between the wall and y^* of 60 and that approximately 80–90 percent of the total energy lost dissipates within $y^* < 30$. With our present knowledge of near-wall bursting discussed in this meeting by Kovasznay (see Kline, et al. [14] and Willmarth and Lu [17]) this implies that for the relatively low Reynolds numbers of the doubly walled-in lubricating films turbulence should be produced locally and should die almost immediately but somehow

[11] Subsequently, during the presentation of 73-LubS-13, Frêne and Godet illustrated with slides the type of surprising information they obtained in connection with [12]. The ensuing lengthy discussions established both the desirability and feasibility of using two hot-film gauges for determination of scales of the observed phenomena and for verification of various conjectured models.

should still trigger another self-regenerating turbulent birth. Also according to Leutheusser and Chu [15] (in agreement with Reichardt [16]) there is little trace of the larger-scale behavior associated with the "outer" defect law for Reynolds numbers (based on half-channel width b and mid-channel velocity) as high as 1500, corresponding to y^* of 60 at $y = b$. The consequent *short-length memory could well make the theory easier:* the layer may be in mean local equilibrium since the mean pressure changes slowly in distances measured in terms of film thicknesses.

On the other hand, turbulence requires concatenation of large numbers of stochastic events, i.e., large Reynolds numbers, to approach a sensibly universal behavior. Thus the lower Reynolds numbers may make the lubricating-film turbulence *more sensitive to local geometrical features* such as edges, grooving, and roughness. The sensitivity to controlled changes in local geometry should indeed form an *important part of the experiments discussed under* (*c*).

(*e*) Finally some remarks on the *onset of instability and transition to turbulence* are in order, especially in view of the last two figures discussed under (*a*). There is a tendency, especially among engineers, to want to believe that transition to turbulence is a deterministically correlatable process despite the unending series of surprises and unexplained paradoxes in the various transition phenomena [18]. These *surprises and paradoxes* undoubtedly spring from the nonlinear multi-instability nature of transition, a sequence of processes controlled by a large number of factors and by unknown, small but highly amplifiable input disturbances [18]. While we have some theoretical and physical appreciation for the initial instabilities in free shear layers and one-wall boundary layers [18], we find no equivalent physically relevant linear or nonlinear stepping stones for the baffling transition in wall-enclosed plane Couette and axisymmetric Poiseuille flows.

Two-dimensional free jets and wakes exhibit local maxima in mean vorticity which feed oscillatory vortical waves upon excitation by special disturbances in a sensitive frequency range. These unsteady vorticity waves amplify as they propagate downstream and through various secondary instabilities lead to turbulence—a conglomeration of randomized nonlinear three-dimensional vorticity waves—at relative low Reynolds numbers. When we remove these free mean vorticity maxima by introducing a wall, we modify the initial instability mechanism (from an inviscid type to one where viscosity causes the destabilization) and defer transition to very high Reynolds numbers. So we expect further stabilization *when we add a second wall* and form a Poiseuille channel but find that as yet an undiagnosed "sublinear instability" [19, Fig. 1] takes over and somehow in all experiments leads *to early turbulence* at Reynolds numbers at which the theoretically most unstable linear modes are highly damped! For the other wall-enclosed non-rotating flows, the plane Couette and axisymmetric Poiseuille flows, no linearly unstable modes have been discovered and yet they merrily turbulate past some Reynolds number, presumably that corresponding to "global instability," [19].

In all of these flows *transition* generally occurs at progressively *lower Reynolds numbers as free-stream vortical disturbances increase.* However, subject to the difficult definition of what constitutes mean undisturbed flow in presence of very large disturbances in space and time, there presumably exists for each flow a Reynolds number R_g for global instability: below R_g all disturbances should die out no matter how large. We are all familiar with the magic Reynolds number of 2100 for pipe flow. Actually, at Reynolds numbers just past R_g, the physically realized flows are only intermittently turbulent and the magnitude of ΔR_g, the increment in Reynolds number to achieve 100 percent turbulence, depends on geometry (including roughness) and the nature of the shear layer.

The preceding cross-discipline briefing on our ignorance of transition leads up to *two questions* for our topic of New Directions:

(*A*) Are the disturbance levels in film lubrication so high that you must operate for practical purposes at R_g? and (*B*) In view of our ignorance are there design risks in assuming specific Reynolds numbers (and ΔR) for the onset of turbulence in prediction of performance? If the answer to (*A*) is no, would your design constraints allow for a meaningful attempt to decrease the disturbances? In other applications the deferral of transition to higher Reynolds numbers has paid off handsomely.

For plane Couette flow Leutheusser and Chu [15] suggest 280 for R_g (their lower stability limit) on the basis of their experiments in presence of a "disturbed" moving wall (really a "turbulent free-surface water flow"). The contrast with the transition Reynolds number estimated by Reichardt from his experiments [16], namely 750, illustrates the magnitude of possible pay-offs or risks in manipulating or predicting R_t. As speeds increase and internal heating looms more ominously, *changes in the onset of turbulence* should appear as *important design parameters and hence deserve more careful investigations.* The experimenters aiming at the turbulence effects discussed under (*c*) should probably include transition as part of their target. In particular, since one can't eliminate disturbances altogether, study of transition sensitivity to *added* on-purpose disturbances (including distributed and isolated roughness) should be revealing with respect to questions (*A*) and (*B*).

In closing I trust that you will kindly overlook where my naivete in your field showed and that you will accept the various remaining observations in the constructive spirit in which they are offered.

References

1 Kline, S. J., Morkovin, M. V., Cockrell, D. J., and Sovran, G., "*Proceedings of the 1968 AFOSR-IFP-Stanford Conference on Computation of Turbulent Boundary Layers*," Vol. 1 and 2, available from Mech. Eng. Dept., Stanford University.

2 Birch, S., and Bushnell, D., "*Proceedings, Working Conference on Free Turbulent Shear Flows at NASA Langley Research Center*," July 1972; NASA SP 321, June 1973.

3 Bradshaw, P., "The Understanding and Prediction of Turbulent Flow," *The Aeronautical Journal*, Roy. Aero. Soc., Vol. 76, No. 739, July 1972, pp. 403–418.

4 Tennekes, H., and Lumley, J. L., *A First Course in Turbulence*, MIT Press, 1972.

5 Bradshaw, P., "*Effects of Streamline Curvature on Turbulent Flow*," AGARDograph No. 169, AGARD, NATO, Paris, fall 1973.

6 Morkovin, M. V., "An Approach to Flow Engineering Via Functional Flow Modules," *Beiträge zur Strömungsmechanik, A. Walz 65th Anniversary Volume II*, Deutsche Luft-und Raumfahrt, pp. 72–27, May 1972; also AFOSR-TR-72-0908.

7 Putre, H. A., "Computer Solution of Unsteady Navier-Stokes Equations for an Infinite Hydrodynamic Step Bearing," NASA TN D-5682, Lewis Res. Center, Apr. 1970.

8 Dailey, G. F., and Geiger, G. E., "Analysis of Flow Separation in an Annular Expansion—Contraction With Inner Cylinder Rotation," ASME Paper 73-FE-7, June 1973.

9 Hotchkiss, R. S., "The Numerical Calculation of Three-Dimensional Flows of Air and Particulates about Structures," LA-DC-13071, Dec. 1971, also Hirt, C. W. and Cook, J. L. "Calculating Three-Dimensional Flows around Structures and over Rough Terrain," LA-DC-13289, 1972, Los Alamos Sci. Lab., Los Alamos, New Mexico, 87544.

10 Caretto, L. S., Curr, R. M., and Spalding, D. B., "Two Numerical Methods for Three-Dimensional Boundary Layers," Rept. EF/TN/A/40 Imperial College, Dept. Mech. Eng., London, England, June 1971.

11 Hirs, G. G., "A Systematic Study of Turbulent Film Flow," presented at the present Lubrication Symposium, but not preprinted.

12 Frêne, J., and Godet, M., "Detection of Taylor Vortex Transition in Very Small Clearances by Hot-Film Wall Anemometry," submitted to *Tribology*.

13 Klebanoff, P. S., "Characteristics of Turbulence in a Boundary Layer With Zero Pressure Gradient," NACA Report 1427, 1955.

14 Kline, S. J., Reynolds, W. C., Schraub, F. A., and Runstadler, P. W., "The Structure of Turbulent Boundary Layers," *Jour. Fluid Mech.*, Vol. 30, Part 4, 1967, pp. 741–773.

15 Leutheusser, H. J., and Chu, V. H., "Experiments on Plane Couette Flow," *Proc. Am. Soc. Civil Engineers*, Vol. 97, No. HY9, *Jour. Hydraulics Division*, Sept. 1971, pp. 1269–1284.

16 Reichardt, H., "Ueber die Geschwindigkeitsverteilung in einer gradlinigen turbulenten Couette-stroemung," *Zeit. f. Angew. Math. und Mech.*, Sonderheft, 1956, pp. S26-29.
17 Willmarth, W. W., and Lu, S. S., "Structure of the Reynolds Stress near the Wall," *Jour. Fluid Mech.*, Vol. 55, Part 1, 1972, pp. 65-92.
18 Morkovin, M. V., "On the Many Faces of Transition," pp. 1-31 of *Viscous Drag Reduction*, C. S. Wells and W. Spangler, Editors, Plenum Press, N. Y., Mar. 1969; also AFOSR TR70-0293, and DDC AD 701271.
19 Joseph, D. D., "On the Place of Energy Methods in a Global Theory of Hydrodynamic Stability," *Proc. IUTAM Sympo. on Stability of Continuous Systems*, Herrenalb, Germany, 1969, Springer Verlag 1971.
20 Tillmann, W., "Zur Turbulenzentstehung bei der Stroemung zwischen rotierenden Zylindern," *Zeit. f. Angewandte Physik*, Vol. 13, No. 10, 1961, pp. 468-475 (see especially Figs. 8-13 on onset of turbulence in Taylor cells).
21 Loehrke, R. I., Morkovin, M. V., and Fejer, A. A., "New Insights on Transition in Oscillating Boundary Layers," Keynote Address, *Proc. Inter. Fluid Dynamics Symposium, McMaster Univ.* Aug. 1970, Gordon & Breach Publ. Co., 1973, also AFOSR 70-1586 TR.
22 Stewart, R. W., "Turbulence," 20-minute color film by the Nat. Com. of Fluid Mechanics Films, available from the Encyclopaedia Britannica Educ. Corp., 425 N. Michigan Ave., Chicago, Ill. 60611; see also "Illustrated Experiments in Fluid Mechanics, the NCFMF Book of Film Notes, pp. 82-88, $3.—MIT Press., Cambridge, Mass.

The second observation from the floor to be recorded here came from Dr. A. J. Smalley of Mechanical Technology Incorporated. He ended on a more optimistic note from the designers point of view by drawing attention to the evidence of close agreement between theoretical predictions and the measured performance of bearings in the turbulent regime. However, the important role of secondary flows in grooves adjacent to bearing pads and the influence of a dynamic head of some magnitude at entry to the lubricating film was in accord with similar observations made throughout the Symposium.

A. J. SMALLEY[12]

Contrary to some earlier discussion, I believe that, from the bearing designer's point of view, existing empirically based turbulent film models are capable of providing consistent, satisfactory predictions of important fluid film phenomena (pressure, total flow, and friction) under turbulent conditions. The Ng-Pan "wall law;" Constantinescu "mixing length;" and Hirs "Bulk Flow" theories are the major examples of these models. There exists a growing body of experimental support for such predictions, both in hydrodynamic bearings, and in hydrostatic bearings where the pressure flow Reynolds numbers may reach up to 50000 and above. Similar success may be claimed for models of fluid film inertia effects as evidenced by papers presented in this meeting.

What is clear from the experimental work in large journal and thrust bearings is that more attention must be devoted to the entire fluid film bearing system. The secondary flows which exist in the grooves between pads of a bearing represent a significant source of energy dissipation, and complicate the process of determining inlet temperature to a pad.

The work of Leopard [1][13] and New [2] has shown the noticeable benefits in temperature rise and film thickness which can be obtained by designing a lubricant supply system to minimize such secondary flows. It will be interesting to see if the same benefits can be demonstrated when similar approaches are applied to journal bearings.

In addition, the measurements of Burton [3] and Smalley Vohr, Castelli, and Wachmann [4] indicate that the dynamic head of low viscosity fluids entering a fluid film can be significant relative to the hydrodynamic pressures generated in the film itself. Even here the empirical models obtained from these independent measurements show reasonable consistency- Burton shows that fluid enters the film with an average of 17 percent of the dynamic head based on the shaft speed—Smalley, et al., show that fluid enters the film with an average 15 percent of this reference head.

To summarize, then, the approximations which are now available to handle turbulent and inertial effects in fluid films are adequate. More effort should be devoted to developing models and understanding of the phenomena which occur outside the bearing film but directly influence its behavior.

[12] Mechanical Technology Inc., Latham, N. Y.
[13] Numbers in brackets designate Reference at end of discussion.

References

Leopard, A. J., "Directed Lubrication for Tilting-Pad Thrust Bearings," *Tribology*, Nov. 1970, pp. 206-210.
New N. H., "Experimental Comparison of Flooded, Directed, and Inlet Orifice Type of Lubrication for a Tilting Pad Thrust Bearing," Journal of Lubrication Technology, Trans. ASME, Series F, Vol. 96, No. 1, Jan. 1974.
Burton, R. A., Carper, H. J., and Hsu, Y. C., "An Experimental Study and Analysis of Turbulent Film Tilted Pad Bearings," Journal of Lubrication Technology, Trans. ASME, Series F, Vol. 96, No. 1, Jan. 1974.
Smalley, A. J., Vohr, J. H., Castelli, V., and Wachmann, C., "An Analytical and Experimental Investigation of Turbulent Flow in Bearing Films Including Convective Fluid Inertia Forces," Journal of Lubrication Technology, Trans. ASME, Series F, Vol. 96, No. 1, Jan. 1974.

A remarkable feature of this valuable exchange of views under informal conditions was the measure of unanimity among research workers and designers with widely differing backgrounds and interests. This was all the more remarkable because the subject of turbulent lubrication can be described as an immature science.

What is the present situation? There is undoubtedly evidence that some fluid-film bearings are now operating in the turbulent regime, although in the case of journal bearings the distinction between Taylor Vortex flow and fully developed turbulence is still blurred in many minds. Indeed, the possibility that Vortex flow might imprint itself upon turbulent flow in journal bearings at Reynolds number well beyond "transition" was a point raised frequently in discussion and throughout the Symposium.

There was a general feeling that mathematical analysis of turbulent lubrication is still inadequate. Furthermore, and perhaps of greater significance, it was widely recognized that physical understanding of the phenomenon is far from complete. In several instances speakers echoed the view that mathematicians were stretching out toward more involved analysis while standing on very shaky physical foundations. The great need for basic experimental studies to alleviate the present uncertainty was clearly seen. Quite apart from the normal considerations in studies of turbulent flow it became apparent that scale effects, thermal action and the influence of film rupture due to cavitation called for careful consideration.

Notwithstanding these reservations by physicists, mathematicians and lubrication analysts it appeared that there was strong support for the view that the bearing designer could, even now, predict the major turbulent bearing operating characteristics with some confidence on the basis of current theories. This makes the subject somewhat unusual, but at the same time demanding, since the concern of the analysts appears not to be justified in the eyes of the designer in many instances.

An overriding observation based upon the Symposium as a whole and this Informal Session in particular is the inter-disciplinary nature of the subject of turbulent lubrication. Those working in the field include mathematicians, engineers, physicists and machine designers and it is clear that the approaches and attitudes of the various groups are often quite different. The lubrication specialists benefited greatly from the contributions to the meeting by specialists with a wide experience of turbulent flow in other and perhaps more conventional fields. The mathematicians present appeared to appreciate the opportunity to learn something of the special problems facing analysts and designers of fluid-film bearings for turbulent operation.

The Organizing Committee is to be congratulated on its vision and efforts to bring together persons with backgrounds in so many disciplines. Their wise judgment and sincere efforts have provided a firmer base for future studies of turbulent lubrication. We await the fruits of their labor with interest and hope.

New Directions

R. A. Burton, Chairman

V. N. CONSTANTINESCU[1]

In my opinion, this Conference was an extremely valuable and useful meeting for at least three reasons:

1 It brought together mechanical engineers (who actually design and build high speed and/or high Reynolds number bearings), specialists in lubrication (who are actually asked to provide adequate tools for the analysis of such systems) and specialists in fluid mechanics (who devote their time and energy in order to provide basic knowledge and understanding of viscous flow, particularly turbulence).

2 It provided an opportunity to summarize and up-date the existing information concerning the fluid film lubrication at large Reynolds numbers.

3 It provided an opportunity to emphasize what should be done from now on in order to improve the knowledge, the analysis and the design of what we call turbulent bearings.

Consequently, the new directions for research and development in turbulent lubrication can be related to the areas of interest of the various specialists who participate in this conference, and can be briefly stated as follows:

For Mechanical Engineers. In spite of numerous uncertainties concerning the actual nature of the flow in the fluid film at large Reynolds numbers and of the accuracy of the various models used to describe the flow, there is already available a useful amount of information which can be employed in order to design bearings operating at high speeds and/or large Reynolds numbers.

Thus, several turbulent lubrication theories are available: phenomenological (law of wall, mixing length, energy model) nonlinear and linearized, global or bulk flow. All lead to the same main results:

1 Direct relationships between flow and pressure gradient. The relationship is quasi-linear for self-acting films, and can be related by using two parameters G_x, G_z which are functions of the Reynolds number based on the sliding velocity.

2 Linear dependence between friction stresses on the two lubricated surfaces and pressure gradients for self-acting films. The mid-channel tangential stress τ_c is almost the same as for Couette flow (parallel surfaces) which is a function of the Reynolds number.

3 The pressure field satisfies a differential equation almost identical to the Reynolds pressure equation for laminar lubrication, with the only difference that some constants (1/12) are now replaced by G_x, G_z.

4 Where most of the heat is carried out by the lubricant, an energy equation almost identical to the one used in laminar flow is obtained in which the mentioned parameters G_x, G_z, τ_c are again used.

On the basis of the aforementioned results, most of the existing design information and computer programs for laminar flow can be generalized in order to include turbulent effects. The obtained results are in a reasonably good agreement with the existing experimental information concerning the operation of bearings at large Reynolds numbers.

The main features of bearings operating at large Reynolds numbers are: higher friction and power loss, higher load-carrying capacity, no significant change in flow but less sensitivity of overall performance function of operating temperature.

The existing information concerning transition from laminar flow to super-laminar conditions and eventually to turbulence is still less accurate. For each type of bearings and operating conditions one may define a critical Reynolds number Re_c up to which the flow in the film remains laminar. In most cases, some kind of laminar vortex flow develops (when the value Re_c is exceeded) which gradually degenerates into fluctuaring flow (turbulence). Although one cannot define a precise Reynolds number for beginning of turbulent flow, one can, for engineering purposes, state that from a Reynolds number of Re^+ of a value roughly 2–4 times the critical one Re_c, the existing turbulent theories can be applied. However, it is to be pointed out that most of the information concerning the critical Reynolds number Re_c (e.g., Taylor's formula) refers to coaxial cylinders (zero eccentricity journal bearings) and thus such formulas cannot be extended directly to actual applications of bearings with nonzero eccentricity. For such situations one should use some more accurate stability analyses or, at least, define a Reynolds number with respect to the mean flow velocity in the film rather than, with respect to the sliding speed. Then, such effects as delaying the transition by diminishing the flow can be at least qualitatively explained.

In addition, a better understanding and experience with turbulent bearings should lead to a better design of such bearings in terms of proper design solutions and constructive types, by taking into account the peculiarities of their operation. For example, knowing that turbulent bearing performances are less sensitive to clearance ratio and temperature and that the flow is less influenced by the Reynolds number, different optimum criteria as compared to laminar bearings can be emphasized. Minimizing the power loss and maximizing the flow (larger clearance ratios) may then lead to a better design.

For Specialists in Fluid Film Lubrication. It can be emphasized that there are still numerous problems, both basic and applied, which need to be solved. Among these, one may quote:

Better transition criteria from laminar to superlaminar and eventually to turbulent flow on various types of bearings (journal, thrust, conical, spherical, self-acting, externally pressurized, hybrid, etc.). In addition, procedures to evaluate the performance of bearings operating in the transition region are needed.

Study of thermal effects in order to evaluate the operating temperature, temperature field and thermal distortions.

A more accurate knowledge of the influence of the inertia forces is perhaps the most important problem for the next years. In bearings operating with Reynolds numbers of the order of 10^4–10^6, changes in the kinetic energy of the flow may lead to pressure differences of the same order of magnitude or bigger than the one produced by the self-acting effect. The determination of the inlet edge pressure should be coupled to the effect of convective inertia forces in the film itself. Finally, time-dependent inertia

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forces may significantly alter the film stiffness at large frequencies (high rotational speed).

More basic type experiments are needed in order to check the validity of the existing methods of analysis. First, such experiments in which the global relationships between flow and shear stress versus pressure gradients be measured with a high degree of confidence would be very useful for a whole range of Reynolds numbers including transition. Eventually, more detailed information concerning the flow field will be useful.

For Specialists in Fluid Mechanics. As already pointed out, it is a very fortunate event that this Conference succeeded to point out the peculiarities of fluid film lubrication flow to some of the most distinguished specialists working in the field of viscous flow and turbulence.

The characteristic particularity of the lubrication flow is the fact that two solid walls limit the flow and that the gap thickness is very small as compared to the other characteristic dimensions of the flow. Consequently, a reciprocal effect of the two walls exists. In addition, even when small turbulent eddies are produced by the shear flow, their sizes are limited to the film thickness so that both production and dissipation of turbulent eddies take place on almost the same scale. Other nonlinear effects in lubrication problems, such as compressibility in gas films, point out a dependence on local or short-time conditions, for example frequency dependent properties due to nonlinear compressible effects. Most of the existing turbulent lubrication theories, as well as some stability theories, assume implicitly or explicitly that only local effects are dominant (local quasi-parallel flow, local stability, short-length or short-time memory, etc.). A confirmation (or infirmation) of such concepts as well as more basic knowledge concerning turbulent shear flow between two solid walls may indeed significantly help lubrication people in understanding and solving their problems.

J. H. VOHR[2]

Theories for predicting the resistance to motion of turbulent flows have all been based on rather crude models, but nonetheless have done a surprisingly good job when applied to simple flow configurations. For example, the logarithmic universal velocity profile for flow in circular pipes is derived from mixing length theory by means of the obviously incorrect assumption that shear stress remains constant across the pipe. In spite of this, the logarithmic law rather accurately predicts velocity most of the pipe cross section. I believe this "forging" situation continues to exist with regard to existing turbulent lubrication theories. For example, all existing theories of turbulent lubrication ignore the fact that secondary vortex flow continues to persist in turbulent flow in journal bearings. In spite of this, present theories, in my opinion, do a quite adequate job of predicting bearing behavior once turbulence is fully developed.

With regard to the desirability of developing a turbulent lubrication theory which takes explicit account of vortex flow, I think it would be a difficult task and probably not warranted for design purposes. I would be content to account for a preturbulent vortex flow regime by means of a semiempirical interpolation between classical laminar theory and present theories for fully developed turbulent flow.

From a design standpoint, it seems to me that the principle area in which turbulent lubrication analysis needs to be developed is in accounting for inertial effects in the bearing film and for effects of the flow field outside the film, e.g., power losses in feeding grooves. Under conditions when flow is turbulent in bearing films, these effects become quite significant and may overshadow inaccuracies associated with ability to predict the effect of turbulence and/or vortices on flow within the film.

[2] Mechanical Technology Inc., Chatham, N. J.