are negligible. Further analysis is necessary to introduce inertial effects in the model (Simon and Frêne, 1992) and to establish a relation between the equivalent sand roughness and the roughness normalized characteristic R_a. Finally, it is necessary to validate the rough turbulent model through an experimental study.

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DISCUSSION

W. F. Hughes[1]

The authors have presented an interesting and informative paper which helps in understanding the effect of roughness on annular seals. However, it is not clear that inertia effects may be entirely neglected. The "inlet losses" are due to inertia effects and if the back pressure is low enough there is the possibility of anomolous liquid choking as the liquid flashes at the exit, again an inertial effect.

While it is true that the mean acceleration terms are identically zero in a uniform channel of fully developed flow in steady state, the inertia (i.e., kinetic energy) effects can be vitally important if the flow channel is not uniform as is the case if the shaft is misaligned or eccentric. These effects might indeed be small for low leakage seals, but they are generally important in high leakage turbulent seals such as those analyzed by P. A. Beatty and myself (1987, 1990).

Perhaps the authors could comment on any restrictions on leakage rate or the general validity of their model.

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L. San Andres[2]

A model of turbulent flow in annular seals taking into account the surface roughness of the stator and rotor is a welcome addition to the literature. The authors are complimented by their comprehensive work on the mechanics of turbulent flows with rough surfaces. The analysis complements and extends the classical work of Elrod and Ng, and soon, it will have well-deserved recognition by the fluid film lubrication dynamicists. Although the omission of land fluid inertia effects in annular pressure seal analysis is a major drawback of the work, I do agree with the authors' treatment that reduced the complexity of the problem, in order to give close attention to the mechanics of turbulent flow with rough surfaces.

Relevant questions which may improve the quality of this paper are:

(*a*) How are damping coefficients extracted from Eq. (3) since this does not have any squeeze film term, i.e., $d(\rho h)/dt$?

(*b*) As derived in the analysis, the turbulent coefficients Gx and Gz are strictly valid for planar flows $(dh/dx=0)$. How do the authors consider applying these results to seal geometries which as soon as rotor eccentricity varies, then dh/dx is nonzero.

(*c*) Could the authors provide a clear discussion as to the Reynolds number ranges (pressure and shear flow) defining the transition zone from laminar flow to fully developed turbulent flow? How is this accounted for in their analysis? Many annular seals handling light hydrocarbons appear to operate with shear Reynolds numbers between 800 to 2,000 and pressure flow Reynolds numbers on the same magnitude. The authors input on this issue would be most welcome.

(*d*) Is it possible for the authors to publish the polynomial expressions for kx and kz as outlined in Eq. (32)?

(*e*) Are Eqs. (40) for the force coefficients correct? Why is the total load WTo used on the calculation of the stiffness and damping coefficients and not its component on the X and Y direction? For example, the direct stiffness coefficient kxx should be equal to $kxx=(WTo\cos\phi - Wxx)/\Delta X$, where ϕ is the attitude angle between the load vector and the X axis.

(*f*) The results from the analysis are to be used for annular pressure seals where the axial pressure gradient is typically very large, i.e., axial pressure flow Reynolds no. of same magnitude or larger than circumferential flow shear Reynolds no. However, the results presented in Fig. 4 for the shear coefficients Gx and Gz are more applicable to plain journal bearings rather than pressure seals. Could the authors provide a similar figure but for $dP/dz \gg dP/dx$? Also, some results for $dP/dx>0$ (backflow) will be of extreme interest.

(*g*) I would very much appreciate the authors comments on their planned work to analyze seals with macroscopic rough surfaces such as honeycomb or knurled patterns. Is the present work applicable to such situations?

Authors' Closure

First, we want to acknowledge Electricity of France for its financial and technical support. The authors would like to express their appreciation of discussions written by W. F. Hughes and by Luis San Andress.

W. F. Hughes discussion:

Inertial effect may be very important; we have chosen, as

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said in the paper, to neglect these effects, because we wanted to separate the different phenomena's. These effects could be introduced in this study, however with an important assumption: the velocity profiles remain unchanged by inertial effects.

L. San Andres discussion:

(a) Equation (3) of our paper represent the static Reynolds equation; obviously we take into account squeeze effects when we compute damping coefficients.

(b) Gx and Gz are indeed strictly valid for planar flow, we have retaken an assumption done by Elrod and Ng (1967) called "adaptation of parallel flow analysis in lubrication" where the authors say: in lubrication the film thickness h changes only slightly in a distance of one film thickness. Under these circumstances, the local velocity distribution becomes the same as that for flow between parallel plates having the same surface velocity, film thickness and pressure gradient.

(c) In our study the flow is always turbulent because the axial pressure gradient is very important. The transition zone is very difficult to analyze; more work should be done in this respect.

(d) The polynomial expression for Kx and Kz is a classical Tchebycheff polynomial form (with a 3rd range).

(e) We have chosen the X axis aligned on the static load. Then the dynamic coefficients are those given in the paper. If there is an ϕ attitude angle between the load vector and the axe X, it is necesary, according to the reviewer question, to apply a geometrical base change to the coefficient matrix.

(f) Indeed, we have presented, in the paper, Gx and Gz comparisons for $dp/dz \ll dp/dx$ case. The same comparison can be done for $dp/dz \gg dp/dx$ (Figure 11). The agreement is very good.

(g) This analysis is done for classical rough surfaces; it is not adapted for macroscopic rough surfaces such as honeycomb. The objective of our work is to analyze the influence of manufacturing roughness. The equivalent sand roughness model is not adapted for macroscopic sand roughness; another study should be done for this case.

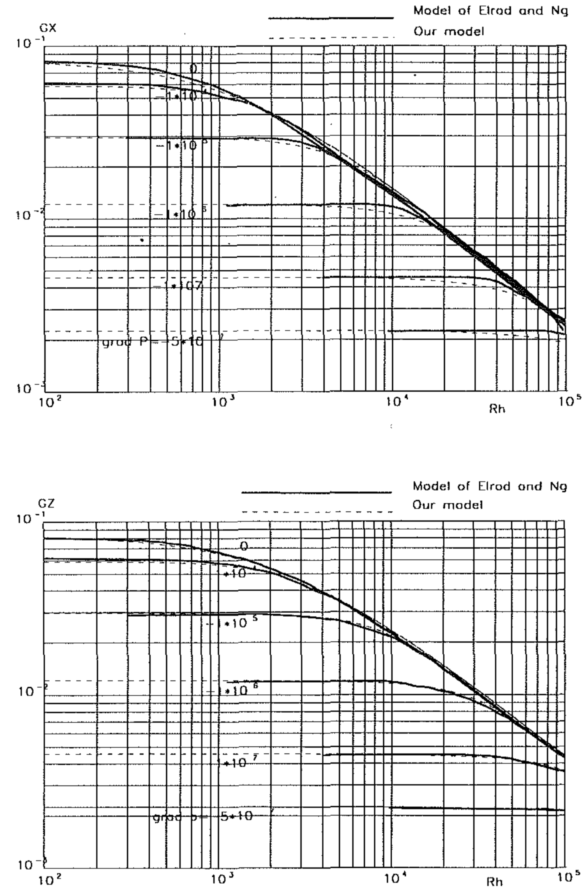

Fig. 11 Coefficients Gx and Gz for $dp/dx \ll dp/dz$