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DISCUSSION

Nicole Zirkelback[1] and Luis San Andres[1]

A detailed analysis for the solution of the zeroth- and first-order turbulent flow field equations modeling the motion of an incompressible fluid in the annular region between a rotating journal describing a conical whirl motion about a point X_o located within the seal is presented. The time averaged Navier-Stokes equations describe the flow and the turbulence closure is provided by the κ-ϵ model. Solution of zeroth- and first-order flow equations render the equilibrium and perturbed pressure and velocity fields, from which the seal static and force/moment coefficients due to rotor dynamic angulations (α) are calculated. The method extends that of Dietzen and Nordmann (1987) to include perturbations of the components of the turbulent stress tensor.

The original title of the paper is misleading since there is actually no rotor misalignment in the seal studied. The equilibrium flow field corresponds to that of a centered rotor concentric within its stator rather than a static misaligned position. It must be assumed then that the misalignment to which the authors refer is related to the small amplitude conical whirl motions of the rotor about X_o.

The equilibrium (zeroth-order) flow equations are solved along with transport equations for the turbulent kinetic energy (κ) and dissipation (ϵ) defining the turbulent viscosity (Eq. (6)). On the other hand, the first-order equations for the perturbed flow fields include perturbations of the turbulent (eddy) viscosity based on the Baldwin and Lomax (1978) model. Why have the authors used two turbulent viscosity models for the same problem? Please provide a sound rationale (other than reasons of simplification) for this consideration. In addition, have the authors considered the use of the Baldwin and Lomax model for the zeroth-order equations as well? If so, how are these results different from the results using the κ-ϵ model?

The correlation of force/moment coefficients for a misaligned straight seal ($L/D = 3$) with test data of Kanemori and Iwatsubo (1994) validates the author's complex computational model and the simpler bulk-flow models. The authors also indicate while discussing these results that, "The strong simplifying assumptions embedded in the bulk flow methods are well verified for long straight seals." Could the authors elaborate on this argument? The authors also state in the Introduction section that "the limited validity of the mathematical model" used in bulk-flow models cannot account for "flow separations and recirculation zones occurring in labyrinth seals." While it may be true that the bulk-flow model simplifies the flow field in seals, there is no mention of the relative importance and influence that (including) the more complex aspects of the flow have on the results given for labyrinth (groove) seals. The authors may choose to add further arguments to support their claims.

As is well known in the literature (Kanemori and Iwatsubo, 1992, 1994; Childs 1993), force/moment coefficients due to conical whirl motions are important in long seals (typically $L/D > 2$). This is also acknowledged in the authors Introduction section. The grooved seal example presented has a L/D ratio just equal to 0.50. Are the computed force/moment coefficients due to shaft angulations of importance in the application presented?

A more thorough discussion of the numerical predictions for the grooved seal should be included. Please provide values of the calculated leakage as part of the seal study. As the groove depth increases the radial force (F_r) and moment (M_r) decrease. However, (from Fig. (5a)) the claim that the mass coefficient ($M_{\epsilon\alpha}$) increases is not obvious since the predictions for $H_g/C_0 = 0$, 1, and 10 show (practically) no curvatuve. Furthermore, both theory and tests for labyrinth seals show that these mass coefficients are inversely proportional to the mean clearance, contrary to the authors' assertion.

The results shown in Figs. 5 (c–d) indicate that the tangential force (F_t) and moment (M_t) decrease as the groove depth increases indicating a destabilizing effect. More importantly, it appears that the whirl frequency ratio determining the threshold from stable to unstable conical seal motions is less than 0.50 as the groove depth increases. This then means that grooved seals are more stable than straight (smooth) annular seals, a fact that should be taken into serious consideration when selecting a grooved seal. The tangential moment (M_t) shows a peculiar behavior since for positive whirl frequency ratios it decreases as H_g/C_0 increases and becomes negative for $H_g/C_0 = 5$. The

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trend is reversed for $H_g/C_0 = 10$ with results almost identical to the $H_g/C_0 = 2.5$ case. The computations for the tangential force (F_t) apparently show the same peculiar behavior, but it is difficult to visualize since the figure is not clear. Could the authors provide more information on the nature of these results?

The ultimate test of a computational program is the correlation of numerical predictions to experimental measurements. In the absence of relevant, actual test data the computed results for the example must be validated with a sensitivity analysis. To this end, it would be of interest to know the variation of force/moment coefficients due to changes in the input parameters, namely the entrance and discharge loss coefficients (ξ), and in particular, the profile of the inlet velocity used.

Finally, the execution time for the annular seal is reported as 174.5 CPU seconds. Please provide the time for the grooved seal case analyzed.

The comments provided in this discussion are of interest to the research community and stem from our desire to better comprehend the extent of this skillful piece of work.

M. Athavale[2]

I would like to congratulate the authors for a pair of well-written papers that describe a 2-D N-S based perturbation model for seal rotordynamics and flow-field simulations.

Inclusion of the complete flow equations and boundary condition treatment will make this paper a good starting point for anyone wishing to develop similar methods and codes. There are a few comments I would like to make:

1. The concept of an exit loss coefficient is interesting, and seems to represent the flow physics better by allowing a circumferential variation at the exit plane. The authors have used a value of 1 for the loss coefficient, simulation cases. Did the authors make any comparison runs with a smaller exit loss coefficient, and if so, what was the extent of its influence on the rotordynamic coefficient values?
2. Perturbations in the wall shear using the log law will hold for coarse grids and relatively slow flows. Given the narrow clearances in the seals, it is very easy to get the first computational cell too close to the wall and in the laminar sublayer. Is there any safeguard against such a case in the model?
3. Limitation of a smooth rotor is rather restrictive. Do the authors have any plans to extend their model to grooves on the rotor, as well as rotor walls that are not parallel (in cross-section) to the axis? These capabilities would allow the model to cover a much wider class of problems.

T. Staubli[3]

Annular seals are with respect to fluid dynamic interactions between rotor and stator the most sensitive elements of turbomachinery and are a favorite research topic since many years. Missing is still a method which allows to model flow phenomena and forces for general cases which come close to situations occurring in turbomachinery. All theories and models base on approximations which usually are rather far away from actual technical problems. In this context the paper of Arghir and Frene certainly is a step toward more realistic cases without complicating numerical procedures to an extent where only supercomputer may be used. The rather sophisticated application of perturbation theory, presented here, allows calculation of circular orbits and conical orbits for misaligned shafts. Comparison with well established theories give confidence that the developed theory provides good results for classical cases and that the physics are also well predicted for small misalignments. On the other hand, one also has to be aware of the limitations of this approximation. Further generalization of the method seems possible and certainly is worth being undertaken. *More weight should be put on more modelization of the inlet and outlet boundary conditions.*

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Authors' Closure

Answer to Comments of N. Zirkelback and Dr. San Andres

The discussers' interest in our paper is appreciated. According to the first comment, the previous title of the paper was adjusted and we are grateful to the discussers for their helpful remark.

In answer to the other questions, one should point out that the present work is the second part of an activity aimed to develop computational methods for predicting the dynamic characteristics of labyrinth seal. The first part, dealing with displacement vibrations of stator grooved seals is presented in another paper (Arghir and Frêne, 1996). Some of the questions are answered in that paper, for example the computing time, which is of the same order of magnitude for cylindrical and conical vibrations.

In order to answer the question concerning the algebraic turbulence model, one should observe that perturbed Navier-Stokes methods usually consider that the effective viscosity field is not affected by eccentricity or misalignment. The necessity of introducing this additional effect was determined when analyzing the displacement vibration of annular seals working at moderate or high axial Reynolds numbers. Its neglection entrains a moderate (20–30%) increase of the radial force which in some cases is difficult to distinguish from the influence of ξ_{in} and ξ_{ex}. Using the differential two equation k-ϵ model for the centered (zeroth-order) flow and the algebraic model of Baldwin and Lomax for estimating the perturbed effective viscosity might be an unusual solution. The reason of this approach is that the system of the perturbed equations is stiffer than the zeroth-order one, namely the under relaxation coefficients needed in the solution of the first order problem are smaller. The eventually perturbed differential equations of the k-ϵ model would certainly render the first order mathematical model more difficult to integrate. On the other hand, the use of the algebraic model of Baldwin and Lomax was never considered for the zeroth-order flow. Algebraic turbulence models could be used for straight annular seal (if the flow is developed and local equilibrium between turbulent production and dissipation prevails), but their recognized inability to describe turbulent transport phenomena makes them unsuitable for labyrinth seals analysis. In this context, the approach of using an algebraic model for estimating only the perturbed turbulent viscosity introduces the assumption that the additional turbulent transport effects induced by vibrations (cylindrical or conical) are negligible compared to zeroth order turbulent transport effects.

Bulk flow methods give good results for flows in straight seals and bearings, provided that concentrated inertia effects are well estimated. Nevertheless, one should not forget the main simplifying assumptions used in deriving the model. Due to the presence of two length scales (one normal to the flow and the second in the flow direction) the diffusion effects in flow direction are negligible compared to the same effects in normal direction. The second assumption, the neglection of the velocity profiles and the introduction of film thickness averaged velocities is less severe. Due to the first simplifying assumption, the streamlines must be parallel or with reduced curvature. Each time when the streamlines curvature is important, the bulk flow model entrains errors. This is the situation of the entrance flow development, when the streamlines might be curved and a recirculation bubble is located on the stepped wall just after the inlet section. The bulk flow model is rigorously valid only after the cancellation of the entrance effects. The perturbed Bernoulli's equation introducing ξ_{in} doesn't take into account these effects but for small clearance seals (reduced C_0/r_R) or for long seals