$$\frac{\partial \mathrm{M_a}}{\partial \dot{m}}=\frac{\dot{m}\left(\dfrac{1}{PAg_c}\right)^2\left(\dfrac{Rg_cT_t}{\gamma}\right)}{\mathrm{M_a}\sqrt{\mathrm{VALUE}}} \tag{12}$$

$$\frac{\partial \mathrm{M_a}}{\partial T_t}=0.5\frac{\left(\dfrac{\dot{m}}{PAg_c}\right)^2\left(\dfrac{Rg_c}{\gamma}\right)}{\mathrm{M_a}\sqrt{\mathrm{VALUE}}} \tag{13}$$

where

$$\mathrm{VALUE}=1+0.8\left(\frac{\dot{m}}{PAg_c}\right)^2\left(\frac{Rg_cT_t}{\gamma}\right)$$

As a result, the estimated maximum uncertainty in Mach number and friction-factor calculation are 0.2 percent and 0.1 percent, respectively.

Concluding Remarks

A flat plate tester has been used to investigate the friction-factors of honeycomb surfaces. Four honeycomb cell widths, three honeycomb cell depths, and three clearances were used. Five inlet pressures and a Reynolds number range of 5,000 to 130,000 are also used for the test parameters.

Although the measurements made here are not complete enough to describe all aspects of the friction-factor for honeycomb surfaces, they do bring out some of the prominent features. The comparisons in the preceding discussion support the following conclusions:

(*a*) Generally, honeycomb surfaces provide a much larger friction-factor than a smooth surface.

(*b*) For 1.57mm cell width with 3.81mm cell depth and the 0.25mm clearance, the friction-factor is smaller than a smooth surface. A possible explanation is that the flow area is enlarged by expansion into the honeycomb cell. Therefore, while honeycomb surfaces generally reduce seal leakage, consideration must be given to the operating clearance and the honeycomb cell dimensions.

(*c*) Two distinct friction-factor patterns are resulted. One is that the friction-factor is nearly constant or decreases slightly as the Reynolds number increases, a common characteristic of turbulent flow in pipe. The other is that the friction-factor shows a great dependency on the Reynolds number (viz., the "friction-factor-jump" phenomena). Forty two percent of the present tests (viz., "dot" marked in Table 1) illustrated this phenomenon.

(*d*) The increase of inlet pressure affects the Mach number and Mach number gradient to decrease the friction-factor, especially, for the "friction-factor-jump" cases. Unlike incompressible pipe flow, the friction-factor for this flow condition can not be defined solely by the Reynolds number and the relative roughness.

(*e*) The effect of honeycomb material in reducing seal leakage is a function of the cell width, cell depth, and clearance. The ratio of honeycomb cell depth to honeycomb cell width (d/b) and the ratio of clearance to honeycomb cell width (H/b) are important parameters for the friction-factor. The data obtained from these tests indicate the maximum friction-factor results when d/b is 3.87 and H/b is 0.48.

The authors obtained most of these data several years ago, but delayed their submission for publication because of their confounding nature, particularly in regards to the "friction-factor-jump" phenomena. Dynamic pressure measurements, Ha et al. (1991), demonstrate that this phenomena is explained by acoustic excitation of a large coherent flow structure which obstructs the main flow. While these measurements explain the phenomenon, the authors are unable to predict when a "jump" will occur. In practical terms, the results remain confounding in that the friction factors for honeycomb surfaces are an erratic and largely unpredictable function of geometry and operating conditions. Moreover, recent limited tests with apposed smooth and honeycomb surfaces tend to discredit the idea that the resultant flow resistance can be obtained via separate measurements of the friction-factor characteristics of smooth and honeycomb surfaces. Specifically, friction factors for apposed smooth and honeycomb surfaces do not always lie between separately measured friction factors for smooth and honeycomb surfaces.

With respect to rotordynamic predictions, the friction-factor phenomenon would be expected to yield a reduction in the cross-coupled stiffness coefficient and a possible reduction in direct stiffness due to a loss of the "Lomakin" effect. Some recent limited test results show an abrupt drop in the cross-coupled stiffness coefficient with increasing Reynolds number, but no accompanying change in the direct stiffness or damping coefficient.

As yet, the present friction-factor measurements have not increased the accuracy of predictions for rotordynamic coefficients.

A complete set of friction-factor data is provided by Ha and Childs (1991).

Acknowledgments

The present research was supported by NASA Lewis Research Center under Grant No. NAG-3-181, Technical Monitor: Robert Hendricks. Honeycomb test specimens were provided by Kentucky Metals, Inc., New Albany, Indiana.

References

Elrod, D. A., Nelson, C. C., and Childs, D. W., 1989, "An Entrance Region Friction Factor Model Applied to Annular Seals Analysis: Theory vs. Experiment for Smooth and Honeycomb Seals," ASME JOURNAL OF TRIBOLOGY, Vol. 111, pp. 293-301.

Elrod, D. A., Childs, D. W., and Nelson, C. C., 1990, "An Annular Gas Seal Analysis Using Empirical Entrance and Exit Resion Friction Factors," ASME JOURNAL OF TRIBOLOGY, Vol. 112, No. 2, pp. 196-204.

Frössel, W., 1938, "Flow in Smooth Straight Pipes at Velocities Above and Below Sound Velocity," N.A.C.A. TM, no. 844.

Grashof, F., 1875, "Theoretische Maschinenlehre," L. Voss, Leipzig, Germany, pp. 593-597.

Ha, T. W., Morrison, G. L., and Childs, D. W., 1991, "Friction-Factor Characteristics for Narrow Channels with Honeycomb Surfaces," ASME Paper No. 91-Trib-21, accepted for STLE-ASME Tribology Conference, Oct. 13-16.

Ha, T. W., and Childs, D. W., 1991, "Friction-Factor Data for Flat Plate Tests of Smooth and Honeycomb Surfaces (Including Extended Test Data)," Texas A&M University, Turbomachinery Laboratories Report, TL-SEAL-1-91, Jan.

Holman, J. P., 1978, *Experimental Methods for Engineers*, McGraw-Hill, New York, NY, p. 45.

McGreehan, W. F., and Ko, S. H., 1989, "Power Dissipation in Smooth and Honeycomb Labyrinth Seals," ASME Paper 89-GT-220.

Schlichting, H., 1979, *Boundary-Layer Theory*, 7th Ed., McGraw-Hill, New York, NY, p. 656.

Stocker, H. L., Cox, D. M., and Holle, G. F., 1977, "Aerodynamic Performance of Conventional and Advanced Design Labyrinth Seals with Solid-Smooth, Abradable, and Honeycomb Lands," Detroit Diesel Allison, NASA CR-135307.

Stodola, A., 1927, *Steam and Gas Turbines*, New York, N.Y., McGraw-Hill, Vol. 1, p. 61.

Zeuner, G., 1900, *Technical Thermodynamics*, Leipzig, Germany, pp. 264-273.

DISCUSSION

J. K. Scharrer[2]

The authors are to be congratulated for an excellent experimental study on honeycomb seals. While this new information may seem confounding and unexplanable today, it will lead to advances in honeycomb seal analysis techniques in the near future. While I agree with the conclusions reached by the

---

[2] Rockwell International, Canoga Park, CA 91304.

authors and their comments concerning the applicability of this data to current seal analysis techniques, I believe that significant point was overlooked. The friction factor jump phenomenon presented in the paper implies that current analyses are conservative in nature (i.e., the assumed friction factor is less than actual in the vicinity of jumps). This means that the honeycomb seal will leak less and have better rotordynamic stability characteristics than predicted. From an application point of view, this is very good news. Even without the jump phenomena, the accuracy of most seal analysis codes is questionable. Therefore, the degree of conservatism of the prediction becomes an important factor in the design process.

The data presented in the paper are for gas flows only. However, pump manufacturers are investigating applications of honeycomb for incompressible flow seals. Would you recommend using the low Mach number, no jump phenomenon data for those applications?

The authors mention that recent tests with opposed smooth and honeycomb surfaces showed that the resulting friction factor did not lie between the friction factor for either one. However, no mention is made of the jump phenomenon. Did the opposed smooth and honeycomb surfaces exhibit the jump phenomenon?

Several papers have been published by the authors in the past with test data for the rotordynamic coefficients of honeycomb seals. Have the authors reviewed any of that data to determine whether or not a jump in the friction factor was possible based on the honeycomb geometry? If the answer is yes, how did the rotordynamic coefficients change with the jump in friction factor?

Authors' Closure

The interest expressed by Dr. Scharrer in this work is appreciated. Taking the questions in order, our response is as follows:

1) Even though the data presented in the paper are for air flow, the data should be applicable for incompressible flow seals of low Mach number condition. The authors are not aware of any honeycomb seal data for liquid flow.

2) The apposed smooth and honeycomb surfaces also exhibit the jump phenomenon. However, the jump was much attenuated.

3) One of the test conditions given by Kleynhans and Childs (1992) suggests a friction-factor jump result for a smooth rotor/honeycomb-stator seal with a 0.4 mm seal width and 2.29 mm seal depth. Specifically, at 16,000 rpm and 18.3 bar supply pressure, dropping the back pressure (increasing the ΔP) decreased the pressure ratio from 0.67 to 0.4 and resulted in a sharp drop in the cross-coupled stiffness coefficient k. Parallel tests with smooth seals showed no drop in k. So far, this is the only dynamic-seal test result which seems to demonstrate the effects of a friction-factor jump.

Reference

Childs, D. W. and Kleynhans, G. F., 1992, "Experimental Rotordynamic and Leakage Results for Short (L/D = 1/6) Honeycomb and Smooth Annular Pressure Seals," accepted for publication in the *Proceedings of 5th IMECHE International Conference on Vibrations in Rotating Machinery*, Sept. 1992, Bath, England.