# DISCUSSION

## I. Etsion[1]

This paper is a valuable contribution to the seal literature. It may be of great help to both designers and users in selecting a seal concept, i.e., FMR or FMS, for some given application and operating conditions. Comparisons such as in Table 1 are very useful for this purpose. This comparison could be generalized by evaluating the corresponding transmissibility equations of the FMR and the FMS, i.e., equations (11) and (13*a*) in the present paper and equations (31*a*) and (37), respectively, in Green and Etsion (1985). Assuming $D_s \ll D_f$ (which is normally the case) and searching for the condition that makes the transmissibility of the FMS less than that of the FMR one obtains

$$D_s \neq 0 \tag{1}$$

for the static transmissibility, and

$$D_s^2 < \frac{(K-I)^2 + (D_f/2)^2}{(K+I)^2 + (D_f/2)^2} K_s^2 - I^2 \tag{2}$$

for the dynamic transmissibility (at $c = 1/2$).

From (1) is is clear that the static transmissibility of the FMS is always smaller than that of the FMR. Condition (2) can be easily met at low speed, $\omega$, when $I \ll K$ and $D_s < K_s$ (see Nomenclature for definition of dimensionless parameters). As the shaft speed, $\omega$, increases the dimensionless moment of inertia, $I$, increases too. Eventually at a certain speed $\omega = \omega^*$ the right-hand side of (2) vanishes. If $\omega$ is further increased so that $\omega > \omega^*$ condition (2) no longer holds and the dynamic transmissibility of the FMS exceeds that of the FMR. Since the static transmissibility is speed independent one may conclude that below a certain speed $\omega^*$ the relative misalignment of the FMS is always smaller than that of the FMR and, hence, the FMS is preferable. Above a certain critical speed (higher than $\omega^*$) the relative misalignment of the FMS becomes larger than that of the FMR and the FMR concept is preferable. There may, however, be other problems resulting from high speed rotating flexible support components that have to be resolved before the full benefit of the FMR concept at high speeds can be realized.

It is interesting to note that the relative misalignment of the FMR decreases at high speeds. Intuitively one may think that centrifugal forces will tend to align the FMR with the rotating shaft and, hence, prevent alignment with the tilted stator, causing increasing relative misalignment with speed. However, the special kinematic constraints of the FMR prevent any such centrifugal effects. The author should elaborate more on this point to enhance the clarity of this valuable paper.

## Author's Closure

The author wishes to thank Dr. Etsion for his interest in the paper and for his constructive remarks.

The author completely concurs with Dr. Etsion's observation that there is a speed, $\omega^*$, above which the benefits of the FMR seal are perceived. However, analytical determination of $\omega^*$ is not feasible. Therefore, a comparison between the FMR and FMS seals, based on the criteria in Table 1, can be performed numerically or graphically as shown in Fig. 8. This figure presents the static, dynamic, and total transmissibilities as a function of shaft speed, $\omega$, for the typical seal where $c = 1/2$. (Table 1 provides numerical data for this seal at a speed of 1000 rad/s.) The total transmissibility equals the maximum relative misalignment under the conditions of the comparison

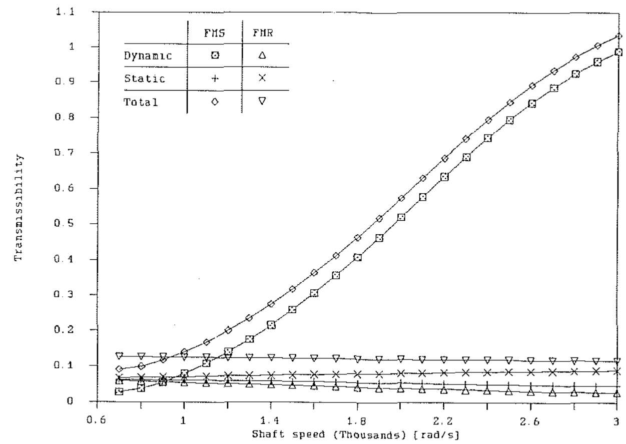


**Fig. 8 Transmissibility comparison between the flexibly mounted rotor (FMR) and the flexibly mounted stator (FMS) seals**

of Table 1. As noted in the paper, and also by Dr. Etsion, Fig. 8 confirms that the FMS seal always has a lower static transmissibility than the FMR seal. Nevertheless, the two static transmissibilities are of the same order of magnitude. However, as speed increases, the FMR seal substantially outperforms the FMS seal. The dynamic response of the FMR seal decreases as speed increases due to the gyroscopic effect (see section "Discussion of Results"), while the dynamic response of the FMS seal increases with speed under the same conditions. At 3000 rad/s there is an order of magnitude difference between the dynamic and the total transmissibilities of the two seals. The values of $\omega^*$ for the dynamic, and total transmissibilities are 903 rad/s, and 938 rad/s, respectively. It is worthwhile emphasizing that the FMR seal is unconditionally stable when $c = 1/2$, as opposed to the FMS seal.

The relative misalignment, $\gamma_0$, which is a direct outcome from the static transmissibility (see equation (11)) is rather increasing with speed as can be seen in Fig. 8. For the typical seal under consideration, values for $K_s^*$, $D_s^*$, $K_f^*$, and $D_f^*$ have been determined to be 400 N•m, 0.24 N•m•s, 5791 N•m, and 4.082 N•m•s, respectively. Rewriting equation (11) using dimensional parameters, yields

$$T_s = \frac{\gamma_0{}^*}{\gamma_s{}^*} = \left[ \frac{K_s^{*2} + (D_s^* \omega)^2}{(K_s^* + K_f^*)^2 + \left(D_s^* \omega + \frac{1}{2} D_f^* \omega\right)^2} \right]^{1/2}$$

The derivative, $\partial T_s / \partial \omega$, for the above parameters results in positive values for a speed range 0 to 10,000 rad/s. This indicates that $T_s$ is monotonically increasing with $\omega$. But this monotonic behavior is attributable to the relative magnitudes of the parameters under consideration rather than to the gyroscopic effect ("centrifugal force") which does not exist for this static forcing function. To explain this phenomenon we resort to a system which is kinematically equivalent to the problem in hand, as it similarly responds to a static forcing function.

Consider the system of Fig. 9, where a disk is mechanically engaged to a rotating shaft by means of a universal joint. A stationary pin, supported by a spring, is brought into contact with the rotating disk, and then further pushed to cause the disk to tilt an amount, $\gamma$, measured between the axis of shaft rotation, $Z$, and axis $z$ which is normal to the disk. The system $xyz$ can only tilt about axis $x$. The angular velocity of $xyz$ is

$$\vec{\omega}_c = \dot{\gamma} \hat{x}$$

The angular velocity of the disk relative to $xyz$ is the spin $\dot{\phi}$. However, due to the kinematical constraint of the universal

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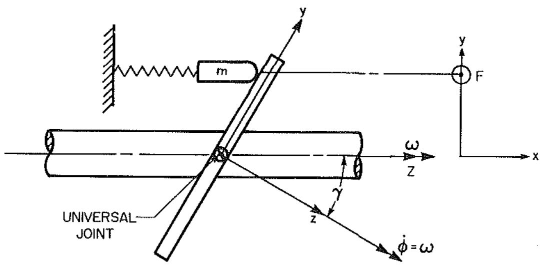


**Fig. 9 Pin and disk system**

joint (see Green and Etsion (1986a)) the transmissibility $T = 1$, or $\dot{\phi} = \omega$. The absolute angular velocity is then

$$\vec{\lambda} = \vec{\omega}_c + \dot{\phi}\hat{z} = \dot{\gamma}\hat{x} + \omega\hat{z}$$

The relative angular momentum is simply

$$\vec{L} = I\dot{\gamma}\hat{x} + I_z\omega\hat{z}$$

The dynamic moment for a nontranslating disk is given by

$$\vec{T} = \frac{\partial \vec{L}}{\partial t} + \omega_c \times \vec{L}$$

Hence,

$$\vec{T} = I\ddot{\gamma}\hat{x} - I_z\omega\dot{\gamma}\hat{y}$$

The equations of motion are obtained by equating the dynamic and applied moments. The only applied moment results from the contacting force, $F$, between the pin and the disk. Hence, the equations of motion are

$$I\ddot{\gamma} = FR$$

$$-I_z\omega\dot{\gamma} = 0$$

where $R$ is the radial distance to the contact point. From the last equation we see that the gyroscopic moment vanishes where we have $\dot{\gamma} = 0$. Therefore, $\ddot{\gamma} = 0$, which results in $F = 0$. This result indicates that although the disk is spinning the spring remains uncompressed, regardless of the shaft speed, $\omega$. To conclude, the reason for the vanishing dynamic moment originates with the kinematical constraint which enables the disk to spin about its own axis rather than the shaft axis.