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Torque Transmission Device at Zero Leakage

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TORQUE TRANSMISSION DEVICE AT ZERO LEAKAGE

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ABSTRACT

In a few critical applications, mechanical transmission of power by rotation at low speed is required without leakage at an interface. Herein we examine a device that enables torque to be transmitted across a sealed environmental barrier. The barrier represents the restraint membrane through which the torque is transmitted. The power is transferred through elastic deformation of a circular tube into an elliptical cross-section. Rotation of the principle axis of the ellipse at one end results in a commensurate rotation of an elliptical cross section at the other end of the tube. This transfer requires no rigid body rotation of the tube allowing a membrane to seal one end from the other. Both computational and experimental models of the device are presented.

INTRODUCTION

In many applications where low leak sealing is required, buffer fluids are used to separate the product from the environment (fig. 1).

While these classes of seals leak both effluent and buffer fluid, the leakage across the buffer fluid interface is minimized. For these types of seals, proper disposal of the product-buffer-fluid mixture is required and in most of these cases, the effluents are either highly toxic (e.g., chemicals or biocides), contagious (e.g., biological or viral substances) or radioactive wastes. And in some instances, disposal of combinations of such vial effluents is required.

The challenge to transfer torque or rotating power across such an interface with zero leakage can be overcome through the use of parallel disks with close coupled magnetic fields; however another interesting method is through distortion of a tubular element (fig. 2), introduced in the 1960's by Eugene B. Zwick in George C. Szego's course on Space Power Systems at UCLA in Los Angeles, CA.

In a model made from paper (which the readers can make for themselves), the tubular shell, representing the shaft, is constrained at midspan by a circular membrane interface. When the drive-end is distorted into the form of and ellipse, the opposite end (driven-end) will distort into an ellipse of opposite phase, albeit the distortion somewhat less than the driver-end. The distortion is dependent on material as well as structural parameters. While a practical device would require a set of cam-followers or rolling elements on both the driver and driven ends, in principal one could transfer torque across a membrane interface forming a zero leakage seal.

In this presentation, using FEA we investigate and characterize the constraints, limitations and materials necessary to construct such a torque transmitting tubular device forming a zero leak seal.

PHYSICAL MODEL

Several torque transmission models were made of paper, aluminum and steel. All were thin-wall cylindrical tubes with wall thickness/diameter = 0.002 to 0.004 and approximately 0.6 m (2 ft) in length with a constrained interface membrane. The position of the restraint varied, yet usually fixed near the center of the length for convenience of using similar drive and driven rotors.

In each model the torque was transmitted through the tube by opposed rollers 5 cm (2 in.) in diameter affixed to a centerline shaft comprising the drive and driven rotors. These rotors placed in each end of the tube forced it into an elliptical-like interface with (a/b) approximately 1.8.

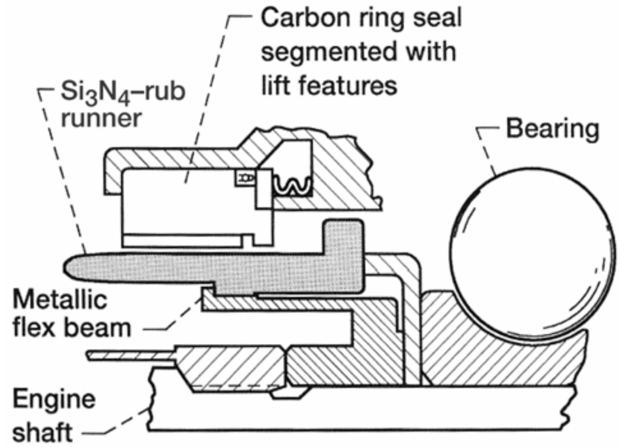

Figure 1.—Typical buffer fluid seal for an oil-air bearing compartment.

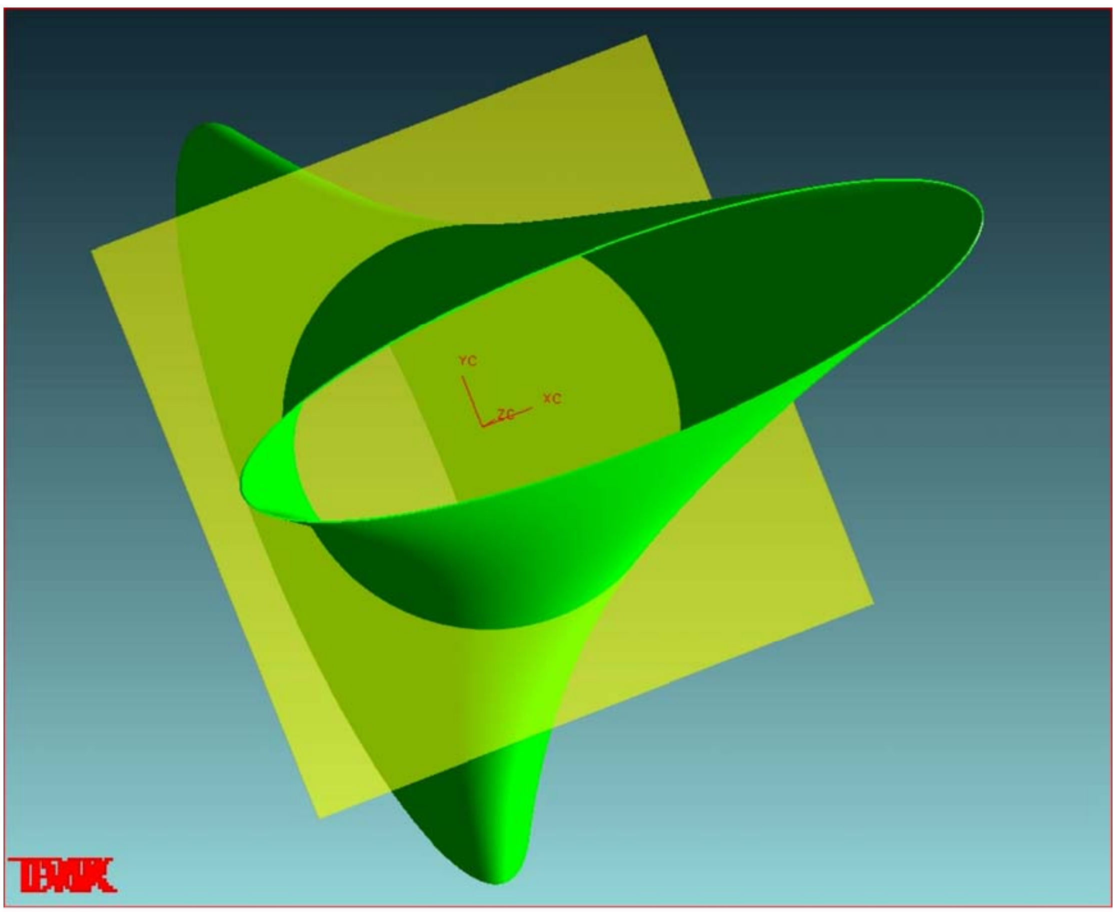

Figure 2.—Sketch of a torque-tube transmission device.

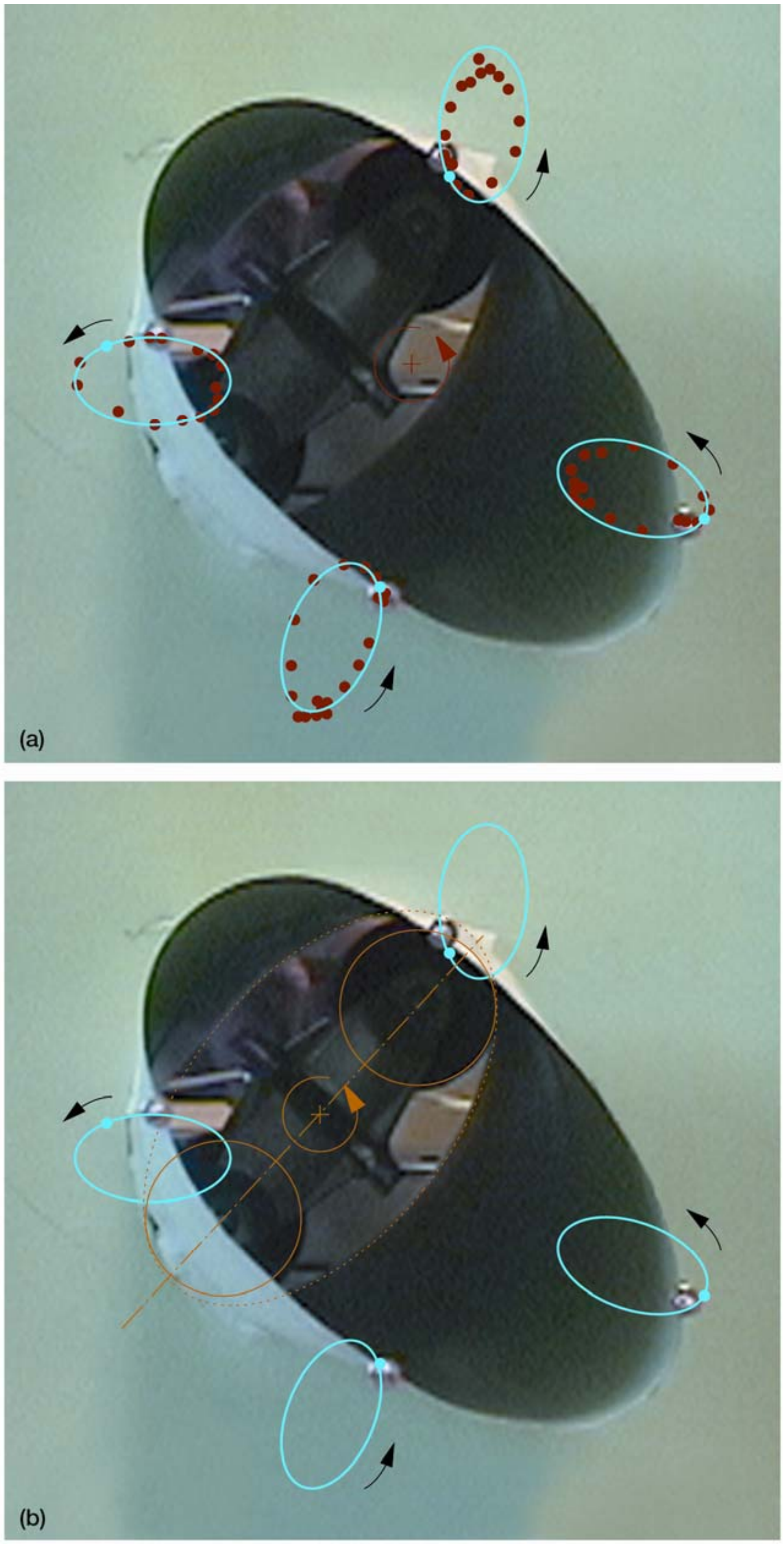

Figure 3.—Near elliptical rotary motion of points on the surface of a paper model torque transmission geometry. (a/b) = 1.8 with amplitude about a fixed point of 2.95 cm (1.16 inch). Undistorted rotor diameter 10.7 cm (4.2 inch.) (a) Surface point trajectories. (b) Driver end surface features.

The phase relation between the driven and drive rollers is nearly 90° and while rotating, points at the interface or elements of the surface move in conjugate elliptical-like paths in the direction of rotation. Figure 3(a) illustrates the motion of the interface at four pin-locations of the paper model, where local distortions are attributed to the simplicity of the model. Figure 3(b) shows the drive-end geometry and drive rollers.

The amplitude, phase and direction of rotation of the surface element is the same as that of the tube itself and connecting elements from one end to the other appear to pass through the interface membrane in straight lines.

The steel tube model was 0.051 cm wall thickness by 15 cm diameter by 61 cm length and 51 cm to 60 cm axial roller positions (0.020 by 6 by 24 in. (20 to 23.5 in. axial distance between rollers). Inserting the drive and driven rollers, the static torque limit of the steel model was then determined by restraining the elliptical-like surface using a threaded rod with nuts adjusted to the internal surface, fixing the geometry. The driver rollers were spaced at 15.9 cm (6.25 in.) distorting the cylinder into an elliptical end of 16.2 by 14.3 cm (6-3/8 in. by 5-5/8 in.). With 60 cm (23.5 in.) between the driver and driver rollers and a centered interface membrane, the driver end was then torqued until slippage which occurred at an estimated 15 N-m (11 ft-lb_f). With a driver roller spacing of 17.8 cm (7 in.) it was difficult to reach a torque limit with the equipment at hand, without severely distorting the torque-tube.

The failure of most models occurred at the membrane interface where transition to the circular geometry occurred and motion is restrained. For small shifts in the position of this interface along the axis of motion, the amplitude on one end of the tube decreases and increases on the other in accordance with the line segment ratio.

FEA MODEL

A parametric finite element study of the proposed seal were constructed using 21-node three dimensional hexahedra continuum elements based on a conventional displacement formulation. A typical mesh contained 12688 nodes and 1769 elements (fig. 4).

The displacement boundary conditions for the static analysis were a zero displacement in all three directions along the outer circumference of the hub and a set of non-zero radial displacements at the inside of each end of the hollow shaft. The tangential and axial displacement at the ends of the shaft were unconstrained. These boundary conditions were selected to represent an elliptical journal bearing or cam set at each end of the seal. The radial displacements were prescribed to deform the end of the initially circular shaft into an elliptical geometry. At each node, the radial displacement was calculated from

$$U_r = \sqrt{\frac{1}{\cos(\theta+\phi)^2(r+\varepsilon)^2 + \sin(\theta+\phi)^2(r-\varepsilon)^2}} - r \quad (1)$$

where U_r is the prescribed radial displacement, r and θ are the cylindrical coordinate of the node, φ is the angle of the principle axis of the ellipsis to the global coordinate system and ε is the magnitude of the elliptical deformation. We kept ϕ zero at one end of the shaft and calculated the deformation in the seal over a 90° range of ϕ at the other end; therefore, the ϕ value represents the nominal phase difference between the ends of the shaft.

The FEA models were based on an initial set of material and geometric parameters which are given in table 1.

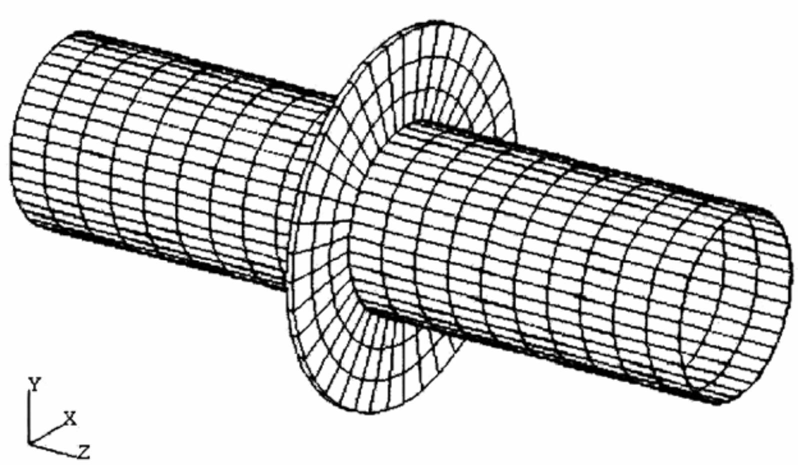

Figure 4.—FEA mesh isometric.

TABLE 1.—BASELINE MODEL GEOMETRY

Parameter	Value 1	Value 2
Shaft Length, mm	60.	60.
Shaft inside radius, mm	10.	10.
Shaft wall thickness, mm	1.0	1.0
Hub thickness, mm	1.0	1.0
Hub radius, mm	50.	50.
Young's modulus of shaft, MPa	1.0×10^6	1.0×10^5
Young's modulus of hub, MPa	1.0×10^4	1.0×10^5
Poisson's ratio	0.25	0.25

A parametric study of variations in the material properties, geometry, end deformation as well as phase angle was conducted using a static analysis in order to elucidate the basis of operation as well as identification of important seal design parameters and their impact of device performance. Parameters include shaft length, magnitude of elliptical deformation, shaft radius, wall thickness, and elastic modulus. In addition, the effect of the relative stiffness of the hub was studied. Our primary concentration was to determine the torque/phase angle relationships in the seal. In order to evaluate the total torque applied to the shaft, we used the finite element model to calculate the strain energy in the system for various phase angles. The torque was then calculated using Castigliano's First Theorem.

$$T = \frac{\partial U}{\partial \varphi} \qquad (2)$$

where U is the strain energy function.

The basic behavior of the torque transmission tube can be illustrated by examining the strain energy as a function of phase angle, which is shown for the base line model in figure 5.

The finite element data can be well represented in an analytic form as

$$U = \alpha + \beta \operatorname{Sin}^2 \varphi \qquad (3)$$

The computed strain energy of a seal, based on the parameters given in column 1 of table 1, was found to follow the equation

$$U = 81.6 - 27.6 \operatorname{Sin}^2 \alpha \qquad (4)$$

Scaling follows as:

$$U \;\; \alpha \;\; U\,[E, r, L^{-2}, (\varepsilon/t)^2, f_m] \qquad (5)$$

where E is Young's Modulus, r tube radius, L tube length between driver-driven ends, ε radial deflection, and t wall thickness. The function f_m relates to the membrane interface which we have not been able to scale.

Accordingly, the torque predicted for the steel tube based on the calculations for the tube in table 1 and equations (2) and (4) gives a maximum torque of (assuming $f_m = 1$; equivalent membranes)

$$\left(\frac{T_{max}}{f_m}\right) = 27.6\times\left(\frac{2\times10^5}{1\times10^6}\right)\left[\frac{152}{(2\times10)}\right]\left(\frac{600}{60}\right)^{-2} \times\left[\left(\frac{(5/0.51)}{(0.05/1)}\right)\right]^2 = 16.13 \text{ N-m } (11.9 \text{ ft-lb}_f) \qquad (6)$$

While the rotation angle was not measured directly it was estimated to be between 20° and 30°; the scaled torque becomes 10.4 N-m (7.6 ft-lb$_f$) and 14 N-m (10.4 ft-lb$_f$), respectively. The scale of the function f_m is unknown. However, for the second column of table 1, with shaft and hub

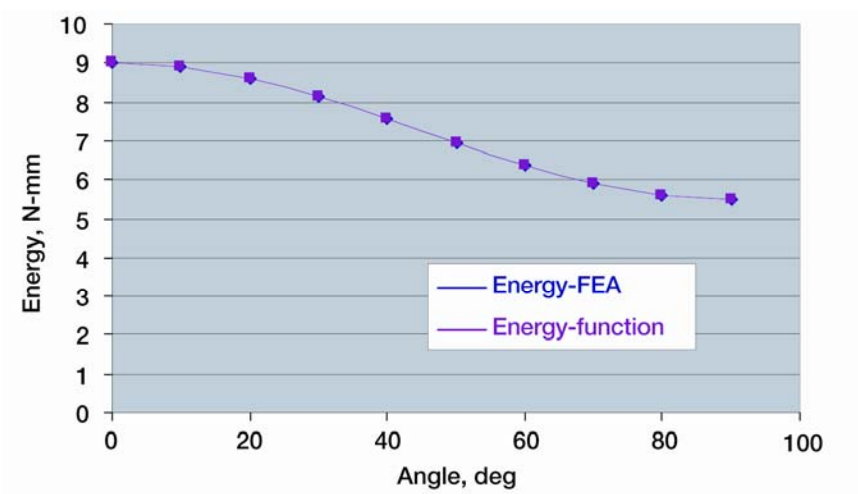

Figure 5.—Strain energy in baseline design as a function of phase angle. Blue curve finite element results, pink curve equation, $E_{tube} = E_{hub} = 10^5$ MPa.

modulus changed an order of magnitude, the strain energy is illustrated in figure 5. For this case the scaled torque becomes (again assuming $f_m = 1$)

$$\left(\frac{T_{max}}{f_m}\right) = 3.52\times\left(\frac{2\times10^5}{1\times10^5}\right)\left[\frac{152}{(2\times10)}\right]\left(\frac{600}{60}\right)^{-2} \times\left[\left(\frac{(5/0.51)}{(0.05/1)}\right)\right]^2 = 22 \text{ N-m } (16.2 \text{ ft-lb}_f) \quad (7)$$

The membrane has a significant impact on the transmitted torque (see also table 2).

TABLE 2.—STEEL TORQUE TUBE SCALED TORQUE FOR VARIOUS MODULI

E_{tube}	E_{hub}	T_{max}		$T_{30°}$		$T_{20°}$	
MPa		N-m	ft-lb$_f$	N-m	ft-lb$_f$	N-m	ft-lb$_f$
10^4	10^6	16.1	11.9	14	10.3	10.4	7.6
10^5	10^5	22	16.2	19.1	14.1	14.2	10.4
10^4	10^5	19.2	14.1	16.6	12.2	12.3	9.1

SUMMARY

Several physical models and a three-dimensional finite element small displacement model of the torque-tube seal were constructed.

From additional parametric studies it was determined:

1. The torque is proportional to the square of the magnitude of the ellipticity.
2. The torque is proportional to the modulus, radius, and inversely as the square of the length of the tube.
3. The membrane stiffness has an optimal value. If the membrane is too soft or too stiff, the torque decreases, but does not scale readily.
4. The phase and amplitude relations of points on the surface correspond to those followed by cross-sections of the overall geometry.
5. Plane lines of the tube parallel to the axis of rotation of the undistorted state remain plane in the distorted geometry
6. Distortion amplitude of the tube follows as the ratio of the line segment position of the interface-membrane constraint.
7. The scaled results compare favorably with data if one assumes correct modeling of the interface membrane.

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In a few critical applications, mechanical transmission of power by rotation at low speed is required without leakage at an interface. Herein we examine a device that enables torque to be transmitted across a sealed environmental barrier. The barrier represents the restraint membrane through which the torque is transmitted. The power is transferred through elastic deformation of a circular tube into an elliptical cross-section. Rotation of the principle axis of the ellipse at one end results in a commensurate rotation of an elliptical cross section at the other end of the tube. This transfer requires no rigid body rotation of the tube allowing a membrane to seal one end from the other. Both computational and experimental models of the device are presented.

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