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# ELASTOMERIC ROD END BEARINGS: A SOLUTION FOR IMPROVING RELIABILITY AND MAINTAINABILITY

BY

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# ELASTOMERIC ROD END BEARINGS: A SOLUTION FOR IMPROVING RELIABILITY AND MAINTAINABILITLY

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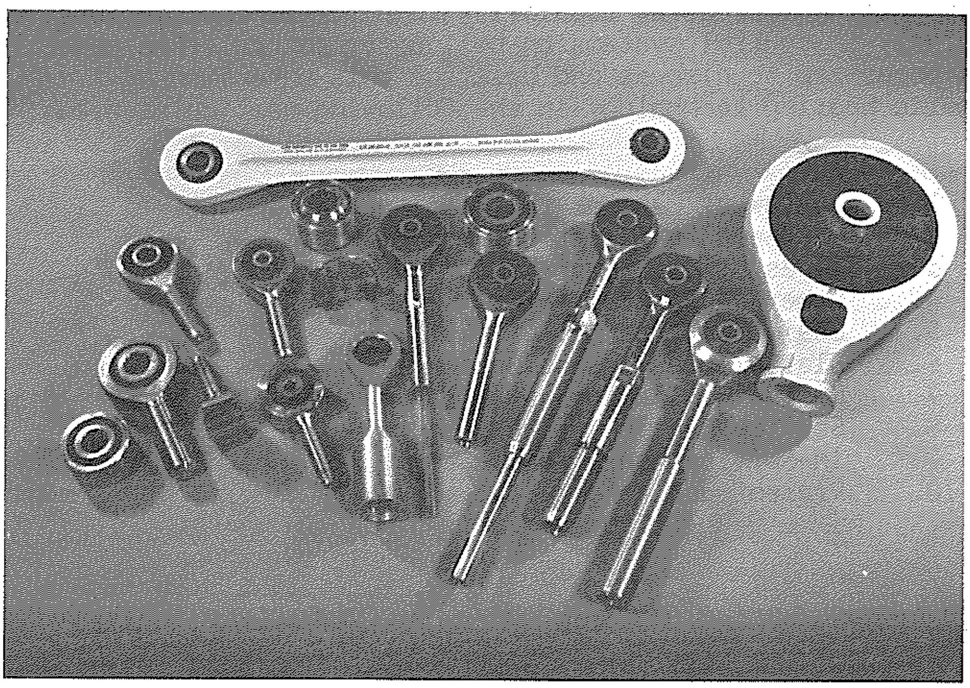
## ABSTRACT

The incorporation of elastomeric bearings into helicopter main and tail rotors, as well as other aircraft applications over the past 15 years, has demonstrated significant advantages. Of primary significance is the demonstrated improvement in product reliability which leads to improvement in overall system reliability, as well as improvement in maintainability.

The technology dealing with the design and manufacture of flight critical blade retention elastomeric bearings for helicopters has been further developed. This paper discusses the application of elastomeric bearing technology to the design of elastomeric rod ends. These bearings are gaining increasing popularity in aircraft applications, particularly in the rotating controls area of helicopter rotor systems. The benefits offered by elastomeric rod ends relative to conventional "dry" or fabric lined bearings are discussed. Significant increases in service life, reliability, and decreases in maintenance time and cost for the inspection and maintenance has been shown as a result. Overall benefits which have traditionally been available through the use of elastomeric bearings are now available for an increasing number of aircraft applications.



## 1.0 INTRODUCTION

An Elastomeric Rod End is defined as the flexible connection between a rod and a clevis which by use of an elastomeric bearing is capable of transmitting rod tension or compression force into the clevis while allowing nearly restraintless rotational motion about one, two, or all three axes. The elastomeric bearing may be any concept utilizing an elastomer (more commonly called "rubber") to separate inner and outer "races", but is more typically optimized as laminations of metal or composite shims and rubber layers to obtain maximum compression stiffness and minimum shear stiffness. It should also be noted that the elastomeric bearings may be designed and made without threaded rod ends for assembly with other mating parts, such as assembly into major structural components or for insertion into a threaded rod end as a component of an elastomeric rod end assembly.

The introduction of elastomeric bearings into helicopter main and tail rotors began more than 15 years ago, and they are now an established and proven design concept for rotorcraft. Performance has been spectacular in that service lives of over 5,000 hours, without lubrication or maintenance, have been demonstrated with no significant service failures known at this time. It may be said that the elastomeric bearing has been a design breakthrough comparable in performance improvement to the introduction of turbine engines in place of reciprocating engines. Not only have major reductions in inspection and maintenance time and cost been achieved, but this improvement was obtained with a significant reduction in complexity and number of components. Further details on elastomeric bearings may be found in References 1 and 2.

Continued development of the design and manufacture of elastomeric bearings has more recently been applied to Rod Ends where they are gaining increasing popularity in aircraft applications, particularly in control systems on helicopter rotors. The objectives of this paper are to discuss and describe this extension of elastomeric bearing technology in the following particular areas:

1. Review of Elastomeric Bearing Development.
2. The need for Elastomeric Rod Ends.
3. Types of Elastomeric Rod Ends.
4. Other Design Variables.
5. Design/Service Advantages.
6. Elastomeric Rod End Design.
7. Design Refinement.
8. Service Considerations.
9. In-Service Applications.
10. Present and Future Status.

## 2.0 ELASTOMERIC BEARING DEVELOPMENT

Elastomeric bearings are an extension of Lord Dynafocal R engine mountings or similar designs as used in most aircraft with piston or turboprop type engines, where some laminating shims were used to restrict bulge and increase compression stiffness while maintaining low shear stiffness. The full value of such parts as bearings was realized when larger numbers of shims were used to obtain many very thin rubber layers.

Compression spring rates such as 1.5 million pounds per inch were obtained with a capability of supporting loads of 40,000 pounds or more, yet a torsional spring rate of about 25 inch-pounds per degree was independent of compression loads. The worth of such bearings was recognized immediately for applications such as helicopter blade retention bearings where centrifugal force was high, where pitch changes caused oscillatory motion with localized uneven wear in conventional bearings, and where maintenance and lubrication were a continuous problem and expense.

Many elastomeric bearings have been designed and tested. Initially all such parts were qualification tested at actual (unfactored) loads, motions, and frequencies, since accelerating such tests sometimes distorted the results. The accumulation of such testing results, combined with more accurate and sophisticated finite element analyses have now made designing more scientific, allowing at least preliminary sizing to be done by designers with only basic experience in the field. However, final design optimizing still requires experienced engineering judgment if minimum size and cost are to be obtained with acceptable service life. Unusual design configurations or environments, or severe combinations of loads and/or motions continue to require at least partial qualification testing to assure that performance objectives are met.

Production rotorcraft applications for elastomeric bearings are primarily rotor blade support or hinge bearings, and bearings for pylon/transmission isolation systems. For fully articulated rotors, as on the Sikorsky UH-60A Black Hawk and the S-76, the elastomeric bearings must support very high axial compression load, smaller radial shear loads, but still allow large oscillatory pitch, lead-lag, and flap rotation. For semi-rigid rotors, as on the Bell 412 and the 400 series, elastomeric bearings react large centrifugal force and shear loads in compression with cyclic pitch change in torsion, but accommodate smaller angles of lead/lag or flap. Rigid teetering rotors as on most Bell helicopters, or Bell Nodal beams, utilize elastomeric Radial Journal Bearings which carry major radial loads and allow significant oscillatory torsional motion. Representatives of each of the above bearings have been subjected to Qualification tests of up to 2500 hours or actual flight service lives equal to or greater than qualification tests results.

These elastomeric bearings, plus many others of varying design, have been analyzed by computer programs, and the predicted test lives were compared with laboratory (and field service) test lives. This data bank has resulted in the development of proprietary techniques for predicting test life based on a load and motion spectrum established by the helicopter manufacturer. Most elastomeric rotor bearings, therefore, are now designed to the established load/motion spectrum with confidence that the desired service life will be obtained. Laboratory specimen tests have even provided important information in predicting performance at elevated temperatures up to 70°C (158°F) or cold temperatures down to -54°C (-65°F). More recently, valuable test data has been obtained specifically on Elastomeric Rod Ends so that manufacturing and performance aspects peculiar to such configurations, or size, by itself, could be evaluated.

## 3.0 THE NEED FOR ELASTOMERIC ROD ENDS

Over the years, conventional fabric-lined spherical rod ends have been developed to a significant degree so that excellent performance, <u>within</u> <u>wear</u> <u>limits</u>, has been obtained. This development required innovative manufacturing techniques to obtain close tolerance, smooth finish, spherical ball inner "races", and diligent development of the assembly operation to obtain close conformity between the ball and liner. The development of the liner material has involved many tests to assure low friction, low wear, and high durability. Careful application of this type of rod end has provided improved fatigue and static ultimate strength of this component. Conventional rod end designs have been useful for applications with continuous dynamic torsion or cocking motion, within certain PV (Pressure-Velocity) limits, with or without any static mean angle and/or rated radial load in the bearing.

However, there are other features which have at times ranged from minor disadvantages to major disqualifications. Among these are:

1. Undesirable "stick" or breakaway friction which is not easily or consistently controlled in manufacturing or installation.

2. Undesirable levels of wear, requiring periodic replacement.

3. Tendency for liners to "pound out" under dynamic reversing loads causing liner-chipping or set.

4. Severely worsening clearances initiated by liner wear and accelerated by pounding loads.

5. Increased friction and wear from sand, dust, or elevated temperature.

6. Significantly increased friction as a function of radial load.

7. Possible bolt wear if excessive liner friction causes inner race rotation relative to the bolt.

Most, if not all of these problems, are eliminated with Elastomeric Rod Ends. The low elastomer shear spring rate insures a predictable torque that is a direct function of angular motions. At typically small amplitude, that torque will be less than friction "breakaway", especially since it is totally independent of radial load. Since fatigue life can be increased as desired by increased size (number of layers), a design is always an optimized "balance" between size and life.

## 4.0 TYPES OF ELASTOMERIC BEARINGS FOR ROD ENDS

What types of elastomeric bearings can be incorporated into an Elastomeric Rod End? Almost any of the typical bearing designs can be utilized, with the selection depending primarily on the loads and motions to be imposed. Shim orientation will obviously be arranged to carry high loads or to obtain high spring rates perpendicular to the bonded surface. Major dynamic motion or low spring rates should be in pure elastomer shear parallel to the bonded surface. There are many applications of Elastomeric Rod Ends where combinations of the following types have been utilized to obtain maximum performance with minimum cost.

Spherical Tubular Bearing

The most common configuration is the spherical tubular bearing as shown in Figure 1. The laminations of rubber and metal form a zone of a sphere.

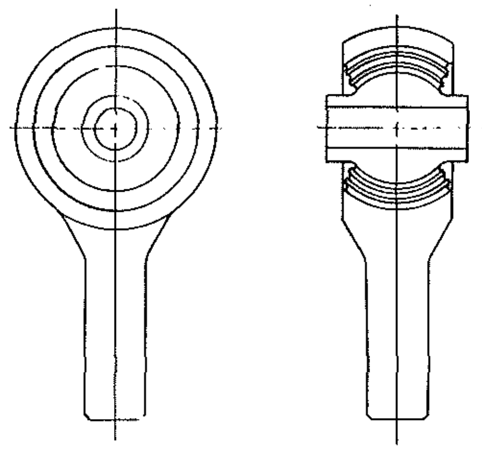


Figure 1

Spherical Tubular Rod End

This design provides maximum capability for rotational motion about any axis, combined with high spring rate for rod tension or compression loads and a relatively high spring rate for axial (along through bolt) loads. Spring rates for the elastomeric bearing of a typical part in the bearing radial, axial, torsion and cocking directions are shown in Figures 2 and 3.

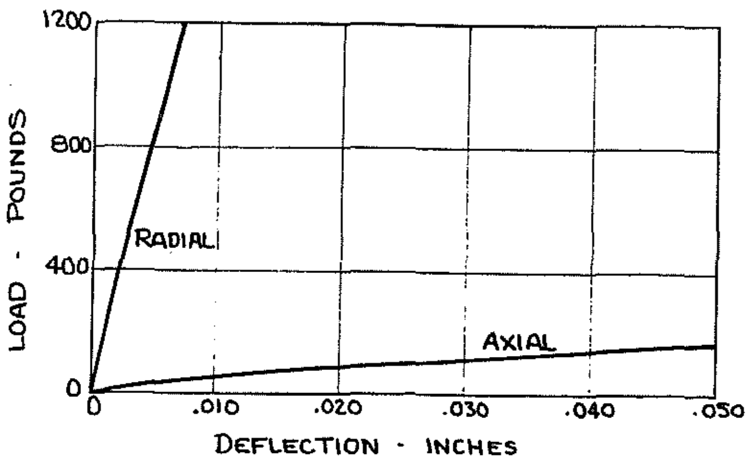



Figure 2

Typical Axial and Radial Stiffness
For a Spherical Tubular Rod End

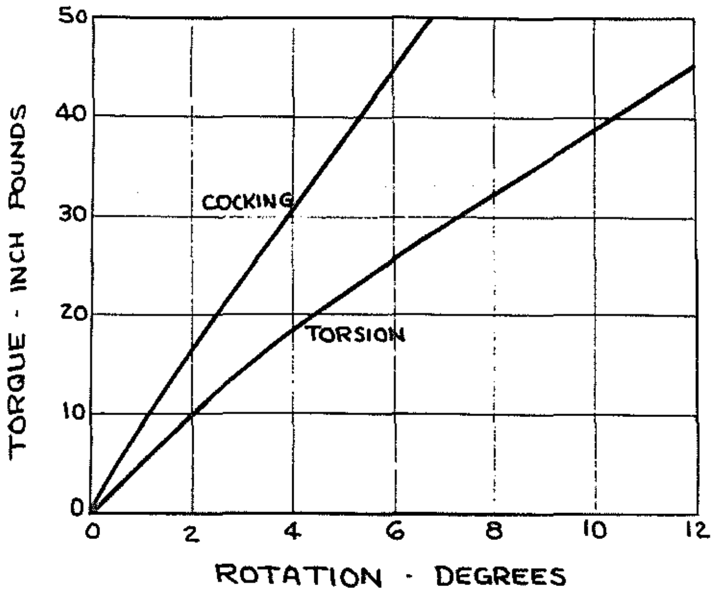



Figure 3

Typical Torsion and Cocking Stiffness
For a Spherical Tubular Rod End

Cylindrical Bearing

Where little or no cocking motion is imposed and axial loads are small, the lower cost cylindrical bearing may be considered, as shown in Figure 4.

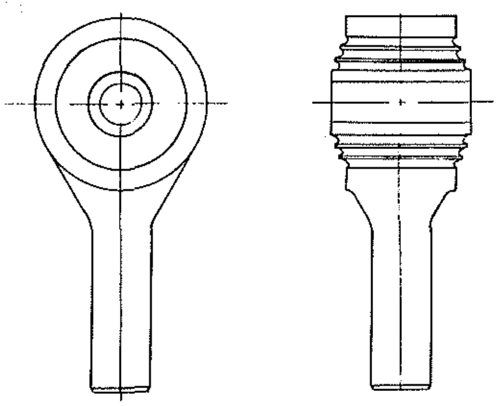


Figure 4

Cylindrical Rod End

Because of a significantly lower axial spring rate, rod tension or compression could cause excessive axial deflections if large cocking angles are imposed.

Conical Bearings

To increase the axial spring rate, conical bearings are frequently used, per Figure 5.

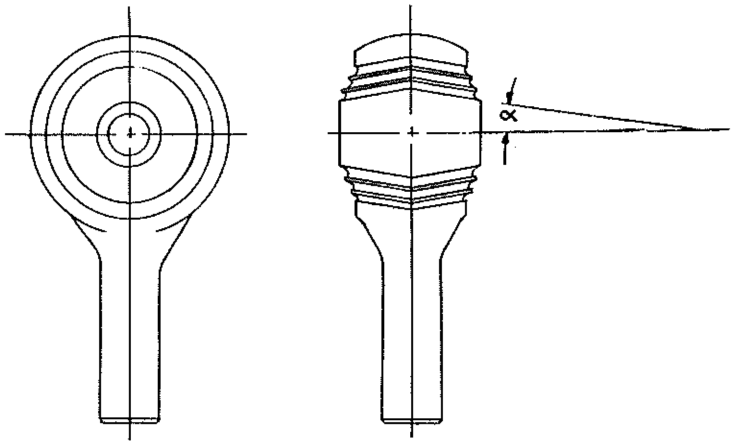



Figure 5

Conical Rod End

Since the cone angle α can be varied from 0° to 90°, considerable changes in spring rates are possible, high angles greatly increasing axial spring rate but also reducing load capability along the rod. In addition to the torsion motion capability, small cocking motion may also be accommodated in some of these designs.

Spherical Thrust Bearings

Although not common, several options for using spherical thrust bearings are available. Figure 6 shows one configuration using two small spherical bearings.

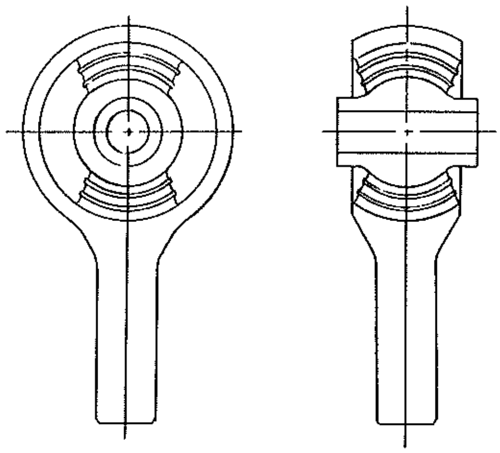


Figure 6
Spherical Thrust Rod End

## 5.0 OTHER DESIGN VARIABLES

In addition to basic geometric configuration, other design features affect performance. Total rubber wall thickness between inner member and outer member may be as little as one thin layer or up to a thick stack of many layers when large rotational motions will occur. The number of shims used to sub-divide the rubber wall depends on the Shape Factor needed to increase radial spring rate or obtain radial load capacity. If the static and dynamic radial loads are low, a thick rubber wall, omitting some or all shims, provides maximum angular motion capability, with minimum size. However, if radial loads are severe, using the correct number of shims to obtain the necessary Shape Factor may be critical, and careful Quality Control in manufacturing is essential to assure precise shim positioning. Various materials including composites have been used for the shims, but stainless or alloy steel is generally preferred when stresses are very high.

For the elastomeric layers, several compounds may be acceptable. The choice is usually based on endurance, low temperature flexibility, amount of hysteresis damping desired, and manufacturing processability. The SPE ™ II and SPE ™ IV blends developed by Lord Corporation have proven ideal because low temperature flexibility and long life are superior to natural rubber. Flexibility is excellent down to -65°F (-54°C), whereas natural rubber may be as much as 50 times normal stiffness on a first rapid cycle after a cold soak. The stiffening factor for the SPE ™ blends is only 1/4 as high. Although not impervious to some fuels, lubricating oils, and cleaning solvents, the elastomers show little or no effect from occasional splashes or minor leaks. In applications where low damping is desirable, such as in isolation systems, the SPE ™ IV offers extremely good performance.

Serrations are used on the ends of the inner member of some elastomeric rod end bearings to insure that torsion motions cause shear of the rubber rather than sliding on the bolt.

## 6.0 DESIGN/SERVICE ADVANTAGES

Designers of all types of aircraft are always pressured to find better designs which provide much longer service lives than have been seen in the past. In combination with this, there exists a trend to provide lower levels of maintenance, or preferably no maintenance at all. Should maintenance be required in some form or another, the trend is toward reduced level of effort, and therefore a reduction in the tools required, maintenance/inspection time required, all which leads to a reduction in maintenance costs over the long run. Emphasis on maintainability and reliability in the industry today has created new challenges for design and materials engineers.

These trends are especially apparent in the design and development of dynamic helicopter systems, most particularly control systems in helicopters but also in engine/transmission isolation systems. Several methods of design in combination with newer engineering materials now available are leading to achieving goals for decreased maintenance and increased reliability and service life. A primary objective of the designer is to reduce the number of components needed to form a working system which results in greater simplicity from a maintainability and reliability standpoint. Elimination of the many components associated with complex lubrication systems has provided a significant increase in reliability over the past 15 years. Reductions in the number of components is now possible more than ever through the use of composites, for example, which carry structural loads and also accommodate motion.

Because of the many design variables, attempts to apply one standard Elastomeric Rod End to a wide variety of applications are not always cost effective. In particular, using a design with more shims than necessary will be functionally acceptable but may add unnecessary size and cost. A custom design, when justified by at least a small production quantity, is usually the smallest, lightest, and most cost effective part.

Elastomeric Rod Ends may not be justified in some applications where small loads allow fabric-lined Rod Ends to provide adequate service life. Furthermore, Elastomeric designs will not be practical if angular motions are too high, such as above 45° statically or above $\pm$ 15° for the equivalent dynamic condition. But, they are an ideal solution for problem applications, where they may offer the following advantages:

1. No metal sliding friction, wear, or seizing.
2. Not affected by sand or dust.
3. No breakaway torque, even under high interference press fits or high radial loads.
4. Selectable spring rates for use in absorbing shock or vibration.
5. Small "return force" is available for self-centering.
6. In many applications, designs may provide "infinite" life.
7. For finite life designs, replacement "on-condition" is acceptable.

From the preceding discussion, as will become more apparent as

Elastomeric Rod End design procedures are summarized, there are several basic design principles which should always be observed when considering the use of these Rod Ends.

Basic Design Principles

1. Major loads must be supported by compression of the elastomeric (perpendicular to shim surface) layer.

2. Rotational motion should be accommodated as pure shear (parallel to shim surface).

3. The number of layers (total rubber wall thickness) is primarily determined by the equivalent (effective average) continuous dynamic oscillatory motions.

4. The size (effectively starting with inner member outside diameter) is determined by equivalent continuous, oscillatory radial force on the bearing.

5. The peak Limit or Ultimate Design loads are usually significant only in the design of the metal parts.

6. Large dynamic rotational motions are better accommodated as torsion (about the inner member bolt axis) than as cocking.

## 7.0 ELASTOMERIC ROD END DESIGN

For the aircraft designer to determine if an elastomeric rod end bearing is feasible for his installation he must have knowledge of how these bearings are sized. Sizing is based on the loading conditions along the thread axis of the rod end and the torsional and cocking angle conditions about the bolt axis and any axis perpendicular to the bolt axis. If cocking angles exist, spherical tubular elastomeric elements or a combination of spherical and cylindrical elastomeric elements are required. If cocking angles do not exist, cylindrical elastomeric elements would be chosen since they are less expensive to produce.

Generally load/motion spectra are available. If a load/ motion spectrum is not available a high speed flight condition can be used. A typical load/motion spectrum for a main rotor pitch link upper rod end is shown in Table I. The important considerations for preliminary design are the equivalent and limit conditions for each loading direction. An equivalent dynamic condition is a single load or motion which theoretically imposes the same fatigue damage as the entire spectrum of conditions for that load or motion. A limit load is the largest peak load, usually considered as only one cycle which must be endured without measurable damage. An equivalent dynamic condition can be determined using Miner's rule. Miner's Rule can be expressed by the following formula:

$$\theta_{EQ} = \left[\theta_1^5 \frac{\%}{100}_1 + \theta_2^5 \frac{\%}{100}_2 + \ldots + \theta_n^5 \frac{\%}{100}_n\right]^{1/5}$$

$\theta_{EQ}$ - Equivalent Angle (or load)

$\theta_1$ - Angle (or load) from Condition 1

% - percent occurrence for applicable angle (or load)

The equivalent static mean loads and motions are calculated by the same equation except with an exponent of 1 instead of 5. The power factor of 5 for dynamic amplitudes is used for preliminary sizing purposes since this is approximately the slope of the S-N (strain-cycles) curve for elastomers. The equivalent conditions are shown tabulated under the established spectrum in Table I. Note that the equivalents are usually comparable in magnitude to the load or motion which applies for the highest percent of the time in flight.

Table I
Typical Load/Motion Spectrum for an Upper Pitch Link

| Cond. No. | Flight Condition | Time % | Radial Load (Pounds) | Torsion (Deg.) | Cocking (Deg.) |
|---|---|---|---|---|---|
| 1 | Hover | 10 | 200 ± 600 | -6.1 ± 2.8 | 0 ± 3.5 |
| 2 | Transition | 2 | 200 ± 600 | -5.8 ± 2.8 | .1 ± 5.4 |
| 3 | 80% speed | 10 | 550 ± 1500 | -6.5 ± 5.4 | .5 ± 1.0 |
| 4 | 90% speed | 30 | 730 ± 1800 | -4.9 ± 7.2 | .5 ± 1.1 |
| 5 | 100% speed | 5 | 900 ± 2300 | -2.6 ± 9.1 | .4 ± 1.4 |
| 6 | 115% speed | 1 | 1200 ± 3000 | 1.5 ± 10.9 | .4 ± 3.8 |
| 7 | Dive | 3 | 1300 ± 1300 | 1.5 ± 10.9 | .4 ± 3.8 |
| 8 | Misc. | 39 | 400 ± 1200 | -7.6 ± 3.5 | .5 ± 1.8 |
| 9 | Limit | - | 10,000 | ± 30 | ± 10 |
| | Equivalent (calculated) | 100 | 550 ± 1675 | -5.88 ± 6.91 | .43 ± 2.84 |

Frequency 5 Hz
Torsion motion 90° out of phase relative to cocking motion and radial load.

The preliminary size of an elastomeric rod end bearing can be determined by obtaining the minimum acceptable load area of the elastomer on the inner member, and then determining the thickness of the elastomer necessary from the applied motions. This is done as follows:

1) The inside diameter and length of the first elastomeric layer is based on obtaining a minimum required area to carry the equivalent and limit radial load. Figure 7 shows the maximum stresses allowed for these two conditions. The allowable stress for an equivalent alternating load condition is presented with the assumption that dynamic radial loads are the primary fatigue contributor. However, if the elastomer thickness required in Step 2 by the equivalent dynamic angle is greater than the thickness required by the limit condition, then shear strain is the primary fatigue contributor. In this case, the necessary load area must be increased by a factor of 1.3.

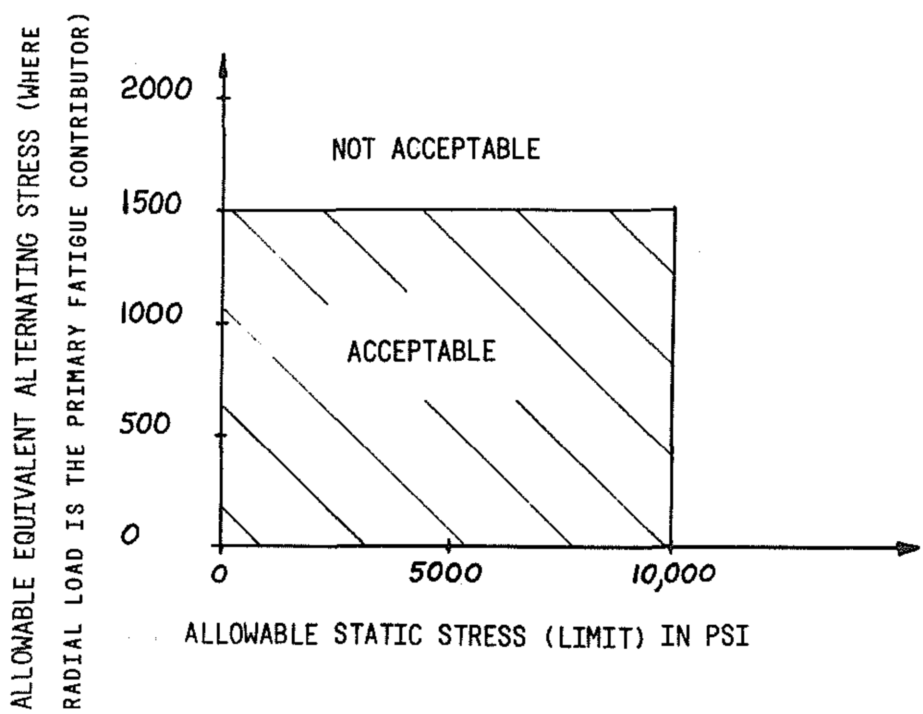
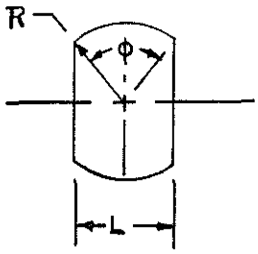



Figure 7

Acceptable Stress Loading for Lord SPE ™ II and SPE ™ IV
Elastomeric Rod End Bearings

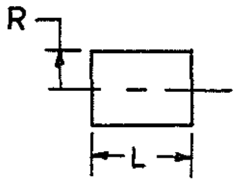
The area referred to on this figure is the projected area on the inner member. These areas are determined for a cylindrical and a spherical rod end bearing as follows:

Projected area, A, of spherical tubular bearing




| | | |
|---|---|---|
| $A$ | = | $2\,R\,L\cos\left(\frac{\phi}{2}\right) + R^2\left(\frac{\phi\Pi}{180} - \sin\phi\right)$ |
| $\phi$ | = | $2\sin^{-1}(L/2\,R)$ |
| L | - | Bonded length of inner member along bolt axis |
| R | - | Radius of inner member |

Projected area, A, of cylindrical rod end bearing




| | | |
|---|---|---|
| A | = | $2\,R\,L$ |
| R | - | Radius of inner member |
| L | - | Bonded length of inner member |

For the equivalent condition of $\pm$ 1675 pounds from Table I, a minimum area of 1.3 (1675/1500) = 1.45 square inches is required per Figure 7. To support the limit load of 10,000 pounds, a minimum of 10,000/10,000 = 1.0 square inch is required. In this case, a spherical radius of .725 inches (spherical tubular elastomer elements were chosen due to presence of cocking) and a length of 1.18 inches were selected resulting in an area of 1.5 square inches.

2) The elastomer thickness is based on the equivalent shear motions which the bearing needs to accommodate. At the maximum motion which may occur at a control check condition, 150% to 170% shear strain is allowed. The equivalent condition predicts the endurance life of the bearing. The maximum strain allowed for the equivalent condition can be determined from the following formula:

| | | |
|---|---|---|
| N | = | $(1100/\varepsilon)^5$ |
| $\varepsilon$ | - | Shear Strain in Percent |
| N | - | Number of cycles until elastomer failure |

If the example described above has a required life of 2000 hours and operates at 5 Hz, the total number of cycles required before failure is $3.6\ (10)^7$. Therefore, the maximum shear strain allowed is 33.9% using the above formula.

Using the allowable shear strain, the required elastomer thickness can be determined by the following formula which includes the effect of metal shim thickness. Figure 8 shows an adjustment factor F used to estimate the total thickness required for shims plus elastomer:

$$t = \frac{R}{\left(\frac{1.8}{\pi}\frac{\varepsilon}{\theta}\right) - \left(\frac{F}{2}\right)}$$

R - the inside radius of the elastomer/shim section
t - elastomer thickness
$\varepsilon$ - allowable shear strain in percent
$\theta$ - equivalent angle in degrees

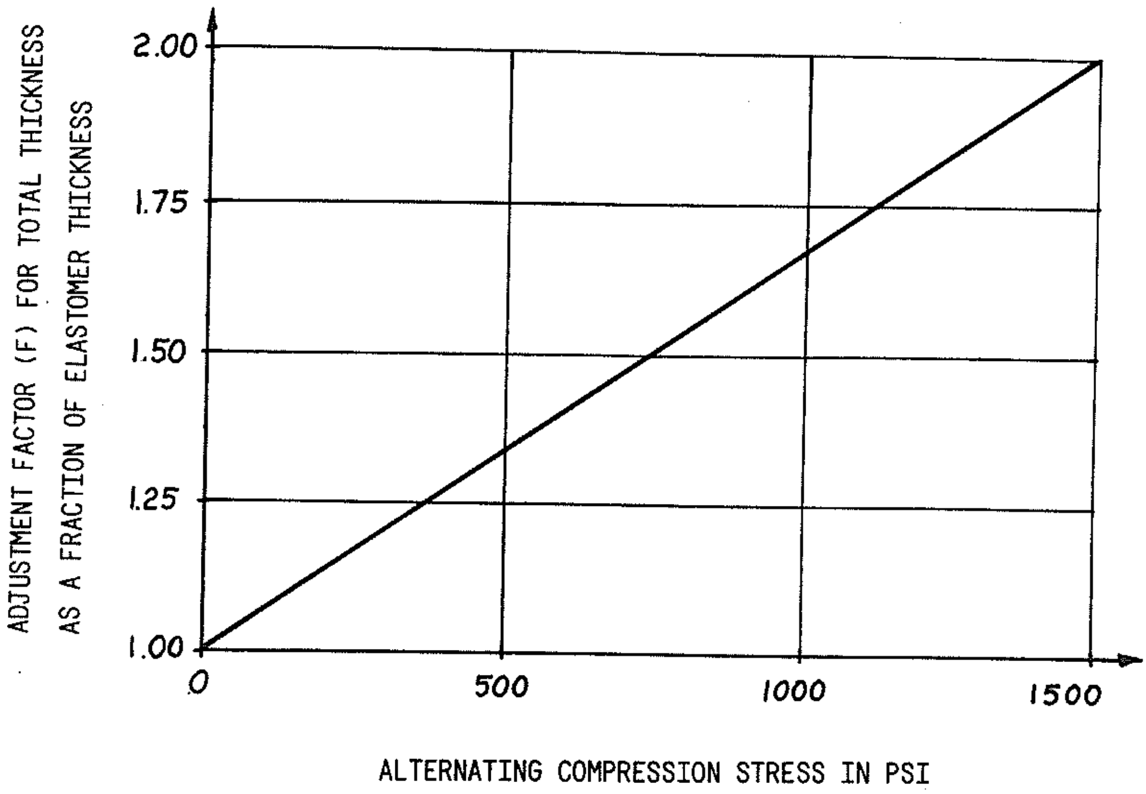



Figure 8
Required Total Thickness Relative to Elastomer Thickness versus Alternating Compression Stress

The example operates at 1675/1.5 = 1117 psi alternating compression stress and, therefore, has a wall thickness factor of 1.70. Calculating the required rubber thickness from the above formula yields .370 inches and the total shim/elastomer thickness is .629 inches. The required outside rubber radius is now .629 + .725 = 1.354 inches.

There is a practical limit to the amount of motion that elastomeric rod end bearings can accommodate. These limits are affected by the motion, frequency and by life requirements, but are typically as shown below:

LIMITS TO MOTION ACCOMMODATION

| Direction | Limit Motion (Degrees) | Equivalent Motion (Degrees) |
|---|---|---|
| Torsion | ± 45 | ± 15 |
| Cocking | ± 18 | ± 7 |

Limit motion may be an aircraft control check condition which occurs not more than four times per hour.

3) The metal wall thickness of the rod end is determined by conventional stress analysis using loads imposed. Typical metal wall thicknesses range from .120 inch up to .50 inch.

Spring rate requirements for rod end bearings do not normally determine the design configuration. That is, rod ends designed to achieve a required endurance life at minimal cost will usually have acceptable spring rates. When necessary, specific spring rates can be achieved, but size or cost may be increased. The following table lists typical spring rate ranges that have been designed at Lord in various designs of elastomeric rod ends.

| | |
|---|---|
| Torsion | 1.3 to 100 inch-pounds/degree |
| Cocking | 4 to 120 inch-pounds/degree |
| Radial | 4,000 to 1,500,000 pounds/inch |
| Axial | 3000 to 500,000 pounds/inch |

## 8.0 DESIGN REFINEMENT

Completing and optimizing the design of an elastomeric rod end bearing is largely an iterative process because of a high degree of parameter interaction. This task is best left to the bearing manufacturer to insure that all parameters are fully optimized. Previous manufacturing and design experience are also relied upon, to design a functional as well as producible bearing.

When the basic sizing and the number of layers are established, each elastomeric layer is optimized such that it will have essentially the same fatigue life as every other layer. First, the optimum shear modulus for each layer is established to achieve equal shear strains and then the thicknesses of the layers are adjusted to equalize compression bulge strains. The shear and compression strains are analyzed and the designs optimized at Lord using a finite element computer program called "HCL". This program has automatic gridding features allowing quick input of geometry and other design parameters, yet results comparable to other more costly and time consuming finite element analysis programs. It also has capabilities to model rigid and flexible shims. The analysis assumes infinitely rigid inner and outer members. A typical geometry plot from this program is shown in Figure 9.

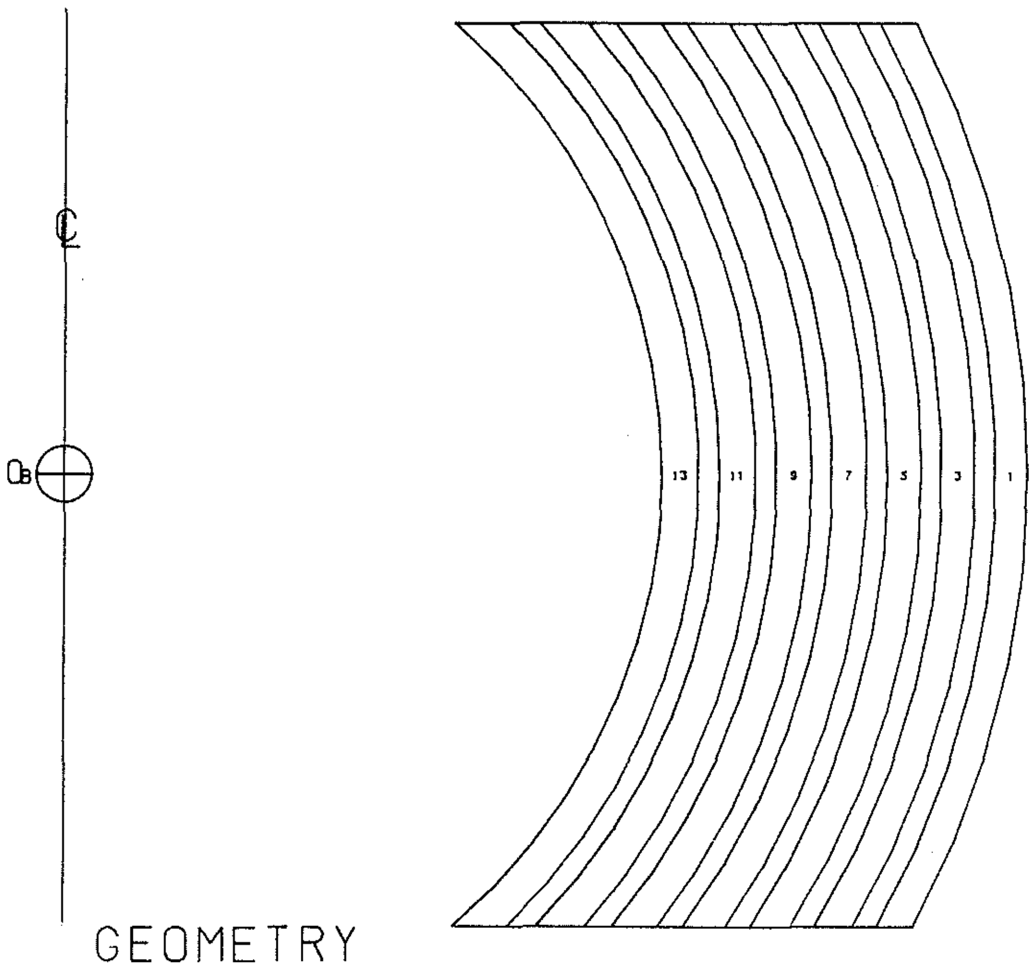



Figure 9

Typical "HCL" Computer Program Geometry Plot

The most accurate stress analysis for an elastomeric rod end is obtained by finite element computer programs which models the entire bearing, including inner and outer members. All new designs which will have significant dynamic radial loads are so evaluated at Lord using a finite element program called SARLAS to insure that elastomer, shim, and outer member stresses are within acceptable limits. Both 2-dimensional and 3-dimensional analysis programs are available, and "Patran G", a graphics pre and post processing program, is used for quick analysis of complex results. A typical SARLAS finite element deformed plot is shown in Figure 10.

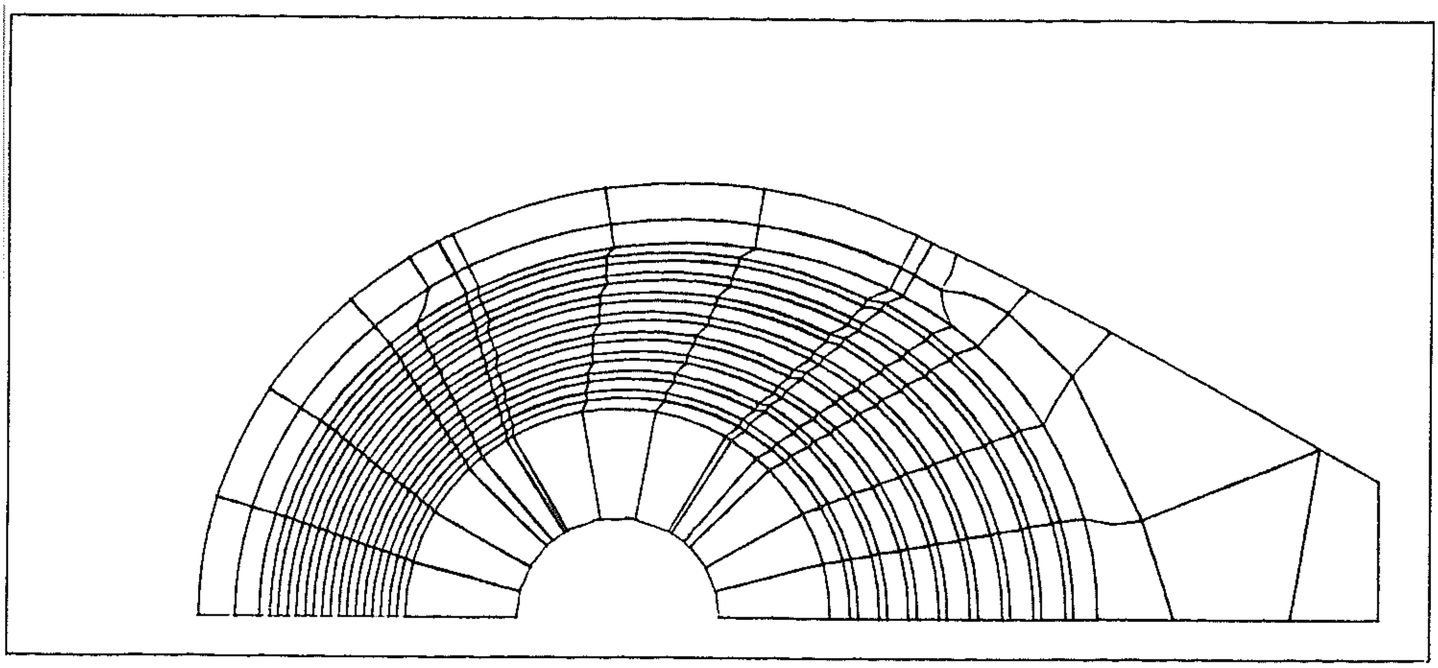


Figure 10
Typical Rod End Bearing SARLAS 3D Deformed Grid

## 9.0 SERVICE CONSIDERATIONS

Typically applications involving oscillatory motions in combination with an applied load have utilized fabric lined bearings to eliminate the need for external lubrication. These types of bearings have been utilized in helicopter control system applications, such as upper and lower pitch link bearings, inboard and outboard damper bearings, and many others. But, service history has proven these types of bearings do not always supply the service life the bearings were originally designed to provide. Service history has also proven elastomeric rod ends can provide longer service life and many other service related advantages over the previously accepted hardware. The following summarizes the most important service advantages:

## Longer Component Service Life

Laboratory test data in which conventional bearings were tested beside elastomeric rod end bearings has proven longer service lives can be obtained. In-service experience has also shown this to be true. In many cases this can be accomplished by direct retrofit into existing hardware with elastomeric rod ends. Longer component life can and should result in longer times between inspections, the frequency of which can easily be determined through laboratory testing during development.

## Increased Life of Other Components

The elastomer/shim package, being permanently bonded together between inner and outer "race" will not "wear" to the point where clearance or backlash can exist as in conventional bearing designs. The elimination of system backlash in dynamic reverse loading applications eliminates shock loads in the system. Elastomeric rod ends, therefore, do not invite shock loading to occur, and furthermore cushion any shock loading which may originate externally to the bearing, extending the life of other components in the system.

## On-Condition Visual Inspection

The ability of a component to be "on-condition" has proven to be the most desirable from a maintainability standpoint. To be on-condition with only visual inspection required for determining flightworthiness is a clear benefit. The gradual process of fatigue in elastomeric rod end bearings is easily monitored through simple visual periodic inspections. There is generally no need for any removal of components, or any disassembly, providing the surfaces of the elastomeric bearing can be seen. No mechanical inspection, such as dimensional checks or measurement of "wear" or backlash is necessary during any part of the bearing's life.

## Less Sensitive to Fluids, Solvents

Elastomeric rod end bearings are permanently bonded to metal components and form their own seal. Resistance to accidental splash or spillage of such fluids as MIL-H-5606, MIL-L-7808, and MIL-T-5624 and cleaning solvents is good, with only wiping with a clean dry rag required to prevent soaking.

### Not Affected by Harsh Environments

Elastomeric rod ends are inherently resistant and are virtually not affected by harsh service environments, such as sand, dust, snow and ice.

### Refurbishment Option Available

Most Lord designs of elastomeric rod end bearings can be returned for refurbishment of the major metal part or parts. In this case, the metal parts are stripped of rubber, and must pass original component inspection criteria before being rebonded to again form a zero hour bearing.

## 10.0 IN-SERVICE APPLICATIONS

Lord has participated with nearly all major helicopter manufacturers over the past five years looking at applications for elastomeric rod end bearings. Several designs have resulted in worthwhile increases in service life, reliability and maintainability and have been in production service for several years.

In service since 1979, the first Lord production application of an elastomeric rod end bearing was on the Bell Model 412 helicopter in the upper and lower main rotor pitch link rod end locations. This rod end, built in left handed and right handed thread configurations but otherwise the same, is shown in Figure 11. The bearings have achieved service lives of more than 800 hours without failure. Failure of elastomeric rod end bearings is a gradual process allowing visual inspection for "on-condition" determination of replacement. Figure 12 shows a Bell 412 pitch link rod end bearing with an equivalent of 1000 hours of service. The slow degradation typical of elastomeric bearing fatigue provides a warning of replacement time, usually hundreds of hours before performance is affected.

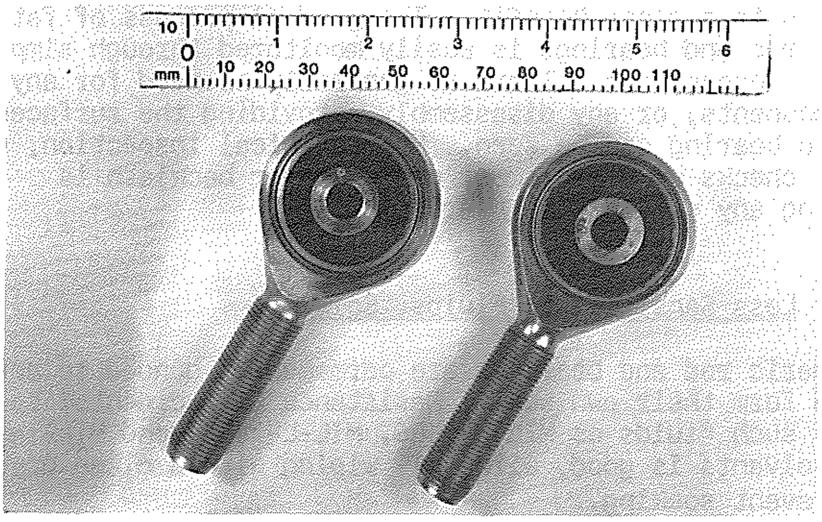


Figure 11
Bell 412 Pitch Link Rod Ends

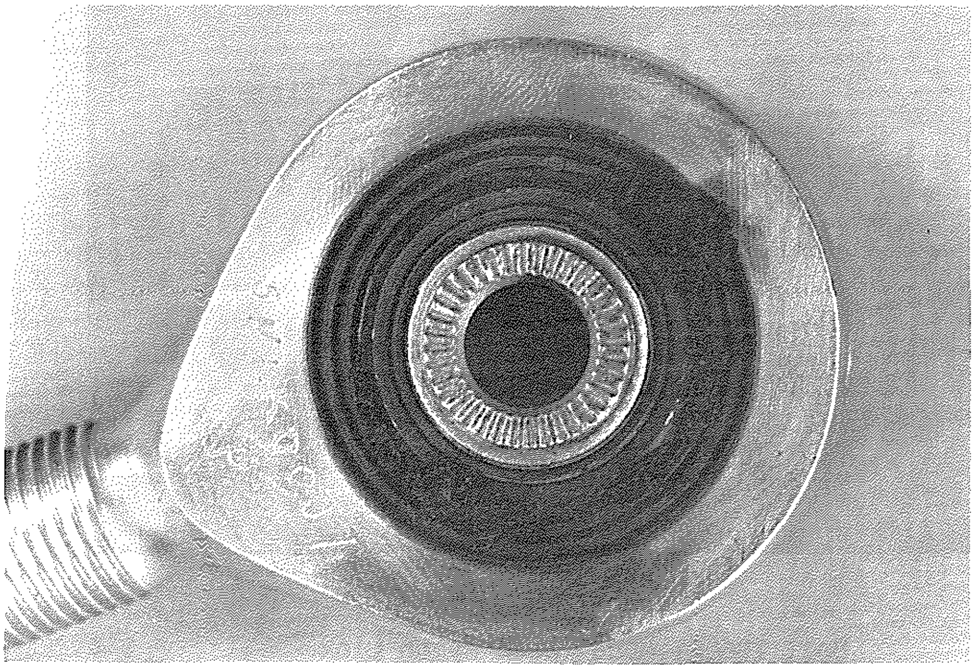


Figure 12

Bell 412 Rod End with Equivalent of 1000 Hours of Service

Currently in production for the Bell Model 206L-1 helicopter is the nodal beam pylon link which contains an elastomeric rod end bearing at each end. In service since 1980, the pylon link attaches the Nodamatic [R] nodal beam isolation system to the gearbox. Originally designed to use fabric lined spherical bearings at each end, the links were revised to retrofit elastomeric rod end bearings for reduced bearing wear. In initial flight tests, it was found that elimination of the friction damping greatly improved the vibration isolation. Ultimately a retrofit program was established to convert pylon links by removing the dry bearings and bonding in the replacement elastomeric bearings. Concurrent with longer bearing life, even greater total link service time was obtained by a refurbishment program established to rebond the outer member with new rubber, converting it to a zero hour bearing, when required. This part with its assembly in the nodal beam system for the 206L-1 is shown in Figure 13.

A more recent introduction into production is a set of inboard and outboard bearings for the hydraulic lead-lag damper on the Boeing Vertol Model 234 commercial Chinook. Shown in Figure 14, with the outboard damper bearing on the right, and the inboard bearing cartridge on the left, these bearings are now in service following laboratory testing at Boeing Vertol which substantiated a 3 to 1 increase in service life.

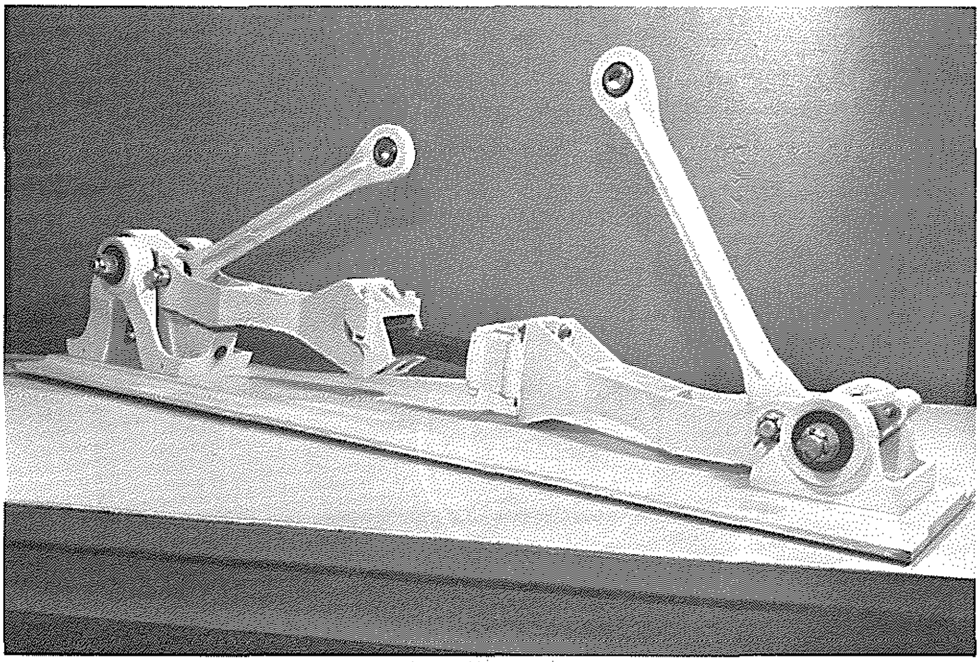


Figure 13

Bell 206L-1 Nodal Beam Pylon Link

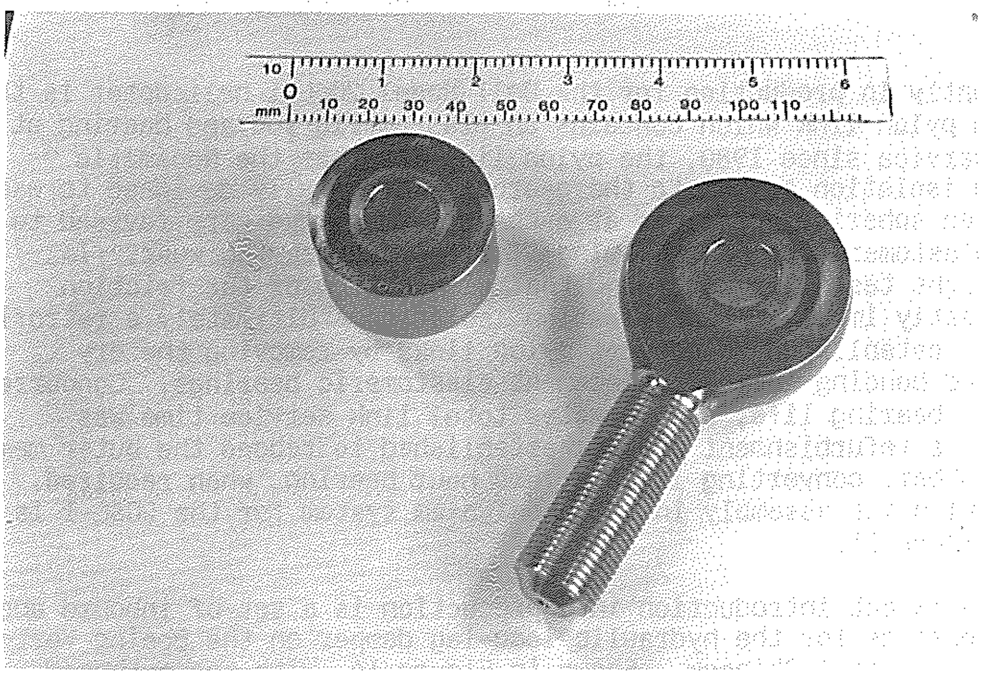



Figure 14

Boeing Vertol 234 Elastomeric Damper Bearings
(Inboard) (Outboard)

An elastomeric rod end is presently used in production on the Bell 214 S/T nodal beam system. The attachment of the helper spring to the nodal beam is made through an elastomeric rod end, allowing torsional and cocking motions yet carrying high dynamic radial loads. The use of the elastomeric rod end also eliminates any friction in the system. The location of the rod end in the nodal beam system installation is shown in Figure 15. Service lives of more than 1000 hours have already been achieved in this application.

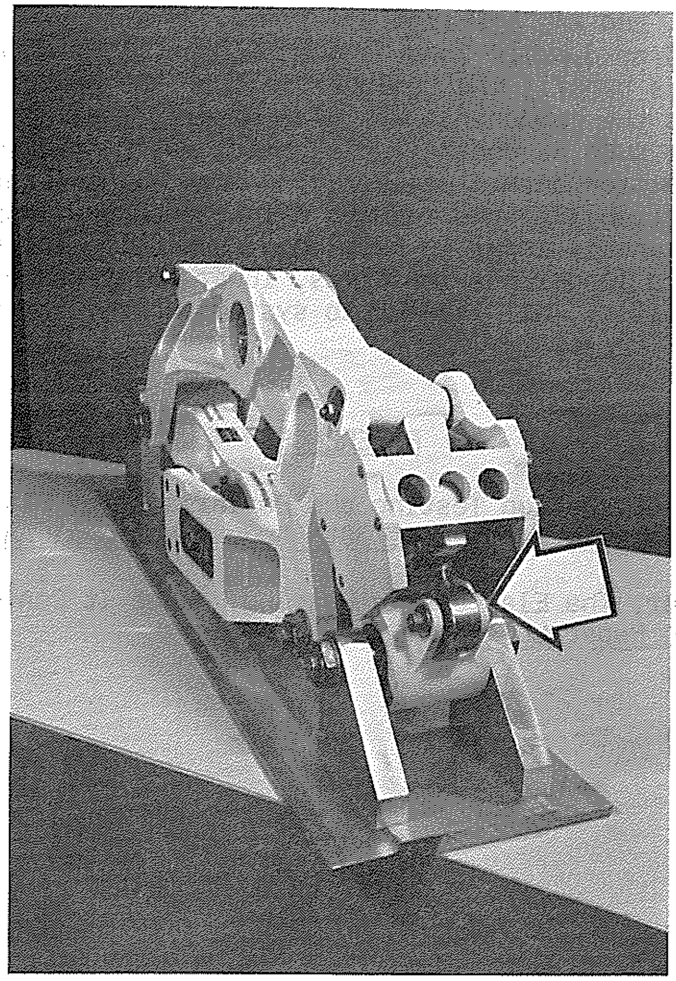


Figure 15

Bell 214 S/T Helper Spring
Rod End

Several prototype applications of elastomeric rod end bearings are currently being evaluated, and/or are being built by Lord for evaluation. Prototype elastomeric rod ends currently being considered are primarily for pitch link applications and damper bearings. However, many designs have been utilized in the past as attachment bearings to the prototype "LIVE" isolators for the Bell 206, 205 and 222 helicopters.

## 11.0 STATUS SUMMARY

Elastomeric rod end bearings offer many advantages in applications where static and/or dynamic loads and motions must be accommodated. Many of the advantages benefit the designer in that the number of components are reduced, and the components are also generally simpler. Performance advantages exist such as controllable, predictable stiffness, and good resistance to environments. However, the most significant advantages are the increased reliability and reduced maintenance and inspection. The elastomeric rod end is ideal in meeting or exceeding the challenging new objectives toward longer service lives, lower maintenance costs and increased reliability, all leading to system economy and customer satisfaction.

## REFERENCES

1) W. E. Schmidt, Design Testing and Performance of Elastomeric Bearings, Paper presented at 30th Annual National AHS Forum, May, 1974

2) J. L. Potter, The Elastomeric Bearing: What It Can Do for You, Lord Corporation Paper LL2125, 1978

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