TID/SNA--1712

September 1971

MASTER

GIMBAL PIVOT ASSEMBLY DESIGN STUDY

**ENGINEERING OPERATIONS REPORT
N8610R:71-005**

NERVA Program, Contract SNP-1

AEROJET NUCLEAR SYSTEMS COMPANY
A DIVISION OF AEROJET-GENERAL

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COMPONENT DESIGN REPORT
(INTERIM)

N8610R:71-005

MASTER

GIMBAL PIVOT ASSEMBLY

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SEPTEMBER 1971

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1.0 INTRODUCTION

This report documents the concept design and analysis for the gimbal pivot assembly of the 75K NERVA Engine.

The primary objective of these design studies was to examine the effect of materials properties on the size and stiffness of the gimbal pivot assembly. The specific effect of Young's modulus and material strength are discussed.

A secondary objective was to present a conceptual introduction to a backup design suitable for installation in the space allotted to the gimbal pivot assembly in the engine concept.

This document also references and discusses the nuclear, thermal, stress, weight, and dynamic analyses made for the gimbal pivot assembly as a unit or as a part of the engine assembly. Finally, fabrication methods are discussed briefly.

The level of maturity of the gimbal pivot assembly design presented in this report does not provide a sufficiently sound basis for preliminary design of this component. It does, however, provide sufficient evidence to proceed with development of the analytical justification of the gimbal pivot concept described.

2.0 SUMMARY, CONCLUSIONS, RECOMMENDATIONS

2.1 SUMMARY

The gimbal pivot assembly provides a load transmitting universal joint function which provides the engine with sufficient degrees of freedom to meet the requirements for thrust vector control.

The gimbal pivot assembly consists of a cruciform structure upon which are mounted four bearings contained within trunnions.

Loads are transmitted from the lower thrust structure through the two pitch axis trunnions of the gimbal pivot to two flexure cores (bearings), from the cores through the cruciform to reacting flexure cores and trunnions on the yaw axis, then to a upper thrust structure.

A flexure core, simply stated, consists of a short tube fitted closely within another short tube and joined by a number of blades aligned in two crossed planes within the tubes.

When a load is applied to the outer tube, normal to the geometric axis of the assembly, it is transmitted through the blades to the inner tube which in turn transmits the load to an adjoining structure. The blades are thin (in tension) and provide relative motion of the tubes by flexing. As the load is increased, the resistance to flexing in the blades decreases to a point where no resistance is present; a friction free bearing with no moving parts requiring lubrication. In the event that a failure of a flexure core occurs, the gimbal pivot assembly design provides a backup bearing surface; a loose but workable plain bearing, to permit continued control of engine thrust vector.

This study is concerned largely with those factors that can be varied as the gimbal pivot assembly is advanced toward design maturity. The goal of this study was to delineate those material and construction choices which will optimize the component with respect to performance, size, and weight.

For the cross-flexure gimbal pivot concept chosen for design development, the cylindrical flexure pivot subcomponent shows wide variability in performance, size and weight characteristics depending on the physical properties of the material employed. A low modulus of elasticity is shown to be favorable by permitting larger vector angles for a given size pivot, with little compromise of axial spring constant. The study indicates also that high material strength is required to produce designs compact enough to permit inclusion in the space allotted in the reference engine design (P/N 1137400).

Those areas are indicated within this report where more extensive study should be conducted to determine optimum compromises within restrictions presented by space, weight, available materials, and the physical laws of the flexure device.

A backup design is presented in concept layout form. This backup design has dimensions allowing it to fit interchangeably into the engine assembly, replacing the flexure concept and requiring very little change in either upper or lower thrust structures at attachment points.

A design study of a flexure pivot for the NERVA engine gimbal pivot was performed by a specialist in the field, the Bendix Corporation, Fluid Power Division. This work was done subsequent to the ANSC analytical efforts discussed in this report.

The objective of the study was to produce a flexure pivot design within the envelope of the gimbal pivot design of Figure 1, 2 and 3 with data on torsional spring rate (no load and full load), stiffness in the direction of three principal engine axes, and weight of the flexure pivot assembly. The choice of material was the responsibility of Bendix.

2.2 CONCLUSIONS

High material yield strength is needed for the flexure pivot assembly in order to gain a low weight and size; if a material of lower strength than 18% Ni maraging steel (250 ksi yield) were used there would be a weight penalty; however, the size penalty should not significantly affect the engine configuration. Practically no envelope increase would be anticipated in the axial direction; therefore, no effect on overall engine length. Any size increase would be expected laterally, or in the radial direction, where the engine configuration can easily accommodate the growth.

Based upon the results of a flexure pivot design study by the Bendix Corporation, the "X" configuration flexure blade orientation of the reference gimbal pivot assembly is incapable of functioning at the specified loads and service life provided that flexure pivot size is maintained within the envelope of the reference design. Acceptable operating characteristics can be achieved with a "+" configuration flexure blade orientation without violating the envelope restraints.

The present gimbal assembly design employing flexure pivots and load relief bushings offer a secondary mode of operation in event of failure of the flexure blades and therefore offers a measure of reliability not available in other designs.

2.3 RECOMMENDATIONS

It is recommended that low temperature material tests on 18% nickel maraging steel be made to supplement the present library information on elongation.

An aggressive program should be launched to gather information on very high strength materials to permit evaluation of their suitability as alternative materials. In those cases where library research yields insufficient data, supplementary material tests should be made.

In designing the flexure pivot one should study a pivot design such that $M_R/\theta = 0$ occurs at 1.5 to 2.0 times the engine nominal thrust. This will increase the resisting moment the pivot offers the actuator, but will produce a smaller, lighter pivot.

A group of small computer programs should be written to perform least work stress analysis of the combination of flexure blade with the support arches, taking into account moments around 3 axes in the blade, with shear, moment, and torsional deflections in the support arch. This will permit accurate recheck of the design.

Deflection analysis of the combined upper thrust structure, gimbal pivot assembly, and lower thrust structure under design loads should be made to permit accurate determination of the loads in six components applied to the gimbal trunnions, together with the spring rates of the structures forming the load path.

An alternative design should be developed, as a backup, to the point that preliminary stress analysis, weight analysis, and flexibility matrix can be computed.

Materials research should be conducted to locate surface coatings suitable for preventing corrosion of candidate materials in launch pad environment. These finishes should also be selected to produce low absorptance/emmissibility ratio for solar heat.

The disproportionate deflection characteristics inherent within a gimbal pivot design of the type described in this report should be reviewed in consideration to the stiffness requirements as they develop for the engine.

3.0 REQUIREMENTS

3.1 REQUIREMENTS LISTING

The gimbal must perform to the requirements of NERVA Detail Specification No. EC-90299 Performance/Design and Qualification Requirements for Gimbal Pivot, which are, in part, delineated below.

The gimbal pivot (GP) must transmit an operational engine thrust load of 77,000 lb plus reactions occurring from the gimbal actuators producing the necessary thrust vectoring stated below:

Maximum Vector Angle limited by Mechanical Stops	<u>+4°</u>
Gimbal Angle Capability	<u>+3°</u>
Gimbal Rate	0.25°/sec
Gimbal Acceleration	0.5°/sec ²
Operating Life	21,950 cycles
Temperature Range*	-100°F to 250°F

*NOTE: In the absence of applicable temperature limits formally defined by currently available documentation, the limits stated above are an engineering judgement by the writer, taking into account thermal emission properties of finishes available for contiguous thrust structures, using thermal analysis studies, Reference 3 (for lower limit), and Reference 4 (for upper limit).

During space operation, the GP must withstand vibrations of nuclear stage operation superimposed upon the loads and conditions stated above. The GP must operate in the presence of nuclear radiation heating as specified by References 11 and 12. The above can be summarized as the ability to function in the space-vacuum environment after non-operating coast periods while withstanding the accumulated effects of load, motion, vibration, and nuclear radiation for the service life of the engine. Also, the GP must have

sufficient stiffness that, in conjunction with associated engine components, the dynamic response of the engine will be acceptable. Minimum stiffness and acceptable response characteristics for the gimbal pivot have not yet been defined.

Target weight for the GP = 170 lb.

Requirements for a design study of the flexure pivot for the gimbal pivot assembly performed by the Bendix Corporation under contract from ANSC are as follows:

Tension load, 62,500 lb.

Side load, 1700 lb.

Service life,

100,000 cycles @ $\pm 2^\circ$ deflection

10,000 cycles @ $\pm 3^\circ$ deflection

100 cycles @ $\pm 3.6^\circ$ deflection

6 cycles @ $\pm 4.5^\circ$ deflection

Temperature range, -65° to 250°F

Bendix was provided the authority for selection of the material for the flexure pivot, but were required to provide additional flexure pivot design data for an alternate material, Lescalloy 718.

3.2 PREDICTED CHARACTERISTICS COMPARED WITH REQUIREMENTS

The gimbal pivot assembly containing flexure pivots of the type recommended by the Bendix Corporation can be expected to meet the requirements set forth here for nuclear stage operation with the exception of the vibratory effects of the engine. This condition has not yet been considered.

The scope of this study has been restricted to a rather elementary investigation of trends regarding the latent possibilities in the crossed-flexure pivot device. The design of this device is properly the province of

vendors skilled in this specialty. The form of crossed-flexure pivot device considered for this study was limited to tension loading of the "X" configuration ligament system, based upon the tenet that this was the most versatile form and that acceptable strength characteristics could be demonstrated.

Subsequent studies by a flexure pivot specialty vendor, the Bendix Corporation, identified dimensional envelope restrictions which precluded the use of the "X" configuration. They were successful, however, in producing an acceptable design utilizing the "+" configuration ligament system.

The maximum overall space available for the GP has been determined within the envelope resulting from the dual propellant feed concept. While the envelope dimensions of the GP are a major influence on the optimization of the design, these dimensions were considered inviolate for this study.

However, within this envelope it was possible to achieve considerable variation in pertinent characteristics by the use of various materials. To illustrate the trends produced by the choice of materials, this study covers such trends produced by varying the width and thickness of the flexure blades for materials of a wide range of yield strength and elastic modulus.

A concept design study, included herein as Appendix A, examined various known gimbal pivot designs. This study concluded that the optimum choice is a cross-flexure pivoted gimbal with a "medium" diameter. Subsequent design analysis has confirmed that the roll strength and stiffness (around the engine longitudinal axis), of the smallest design feasible, are adequate. The design shown by Figure 1 therefore uses the smallest overall dimensions permitted by the cylindrical envelope of the cross-flexure pivot assembly dimensions used, together with the concept of a hollow shell cruciform load transfer member.

4.0 CONCEPT DESCRIPTION

The gimbal pivot design illustrated in Figures 1, 2 and 3 provides a load-carrying universal joint pivoting action centered half-way between the UTS and LTS mounting faces. The material of construction throughout, except for bronze bushings, is high strength 18% nickel maraging steel.

Thrust loads are transmitted through the two yaw axis trunnions (Figure 2) to two flexure cores (Figure 3); these cores transmit the load to the movable hollow cruciform (shown in Figure 1) to the reacting flexures and trunnions located on the pitch axis, and thence to the upper thrust structure.

The flexure core shown by Figure 3 is a device which permits relative rotation of the ends in a fashion similar to a journal rotating within a bushing or bearing. However, there are no sliding or rubbing surfaces. The relative rotation is provided by bending in the flexure blades (sometimes called ligaments or springs in the literature). The blades are loaded in tension during operation of the engine.

Within the trunnion housings (Figure 2) there are bronze bushings; each bushing having an inner diameter about 0.012-0.015 larger than the outside diameter of the moving core housing within it. This clearance is carefully controlled during assembly. In the event of abnormal surges of thrust from the engine, the bushings contact the core O.D. and provide a relief path for the overload. The clearance specified provides thrust relief at a stress well below the yield strength of the blade material.

In the unlikely event of complete failure of all the flexure blades, the bushing-core combination provides a loose but workable bearing to allow the gimbal action to continue. The bearing surfaces are impregnated with molybdenum disulfide to reduce friction and eliminate seizing or space-welding.

The dimensions employed for the X-flexure pivot cylindrical envelope were taken from an exploratory sales sketch provided by a vendor. The feasibility of these dimensions were checked by use of published flexure design equations (Reference 1). Further work was done by the Bendix Corporation, Fluid Power Division, in response to a contractual agreement with ANSC to design the flexure pivot to the load requirements stated here earlier within the outside dimensional envelope of Figure 3.

During the initial phase of the design study Bendix concluded that the conventional "X" configuration flexure blade orientation for a cantilever pivot design to the above criteria resulted in excessive length of the assembly. Under the above design conditions a "+" configuration flexure blade orientation was recommended where the width of the vertical engine thrust loaded blade is many times greater than the width of two horizontal stabilizing blades. Bendix proceeded with preliminary design on this basis which led to attainment of all requirements for this study. The material selected by them for the flexure pivot was 18% Nickel Maraging Steel produced by a vacuum melt process to conform to a Bendix material specification number E-18A,V.

5.0 DESIGN ANALYSIS

5.1 MECHANICAL

5.1.1 Prime Design

Mechanisms achieving relative motion by the use of simple flexure mechanics are legion in number. The following illustrations show two common applications.

SIMPLE FLEXURE PIVOT
AS IN SCALES

*SIMPLE FLEXURE
AS IN MINIATURE ACCELEROMETER*

This type flexure is permissible where a fixed center is unimportant and where resistance to side loads is not needed.

By using crossed blades as shown below it is possible to construct a hinge capable of carrying side load.

CROSSED-FLEXURE PIVOT

The effective center of rotation moves slightly during angular deflection. However, for the angles required in the NERVA gimbal, the movement is only a few thousandths of an inch. Figure 4 lists the applicable formulas and defines the design quantities used later in this discussion.

The formulas shown by Figure 4 express only those stresses and deflections arising in the blade portion of the assembled flexure pivot. This fact is mentioned at this particular point in the discussion, since there are stresses, linear deflections, and torsional rotations in the semi-annular cross section of the cantilever internal overhang within the cylindrical assembly proposed for use in the NERVA GP. These stresses and strains are beyond the purview of this design study, and will be the problem of the detailed design of the flexural pivots. For concept purposes, the sketch below illustrates the assembly construction assumed. The investigatory calculations following are applications of the published theory (Reference 1) to the blades made to reveal the space and weight trends produced by varying the properties of materials being considered.

CANTILEVER TYPE X-FLEXURE PIVOT
WITH DIRECTIONS RELATIVE TO THE
PIVOT'S FRAME OF DIRECTION

The form chosen for concept study is the symmetrical "X" configuration sketched above loaded to produce tension in the blades for engine operation. The tension-loaded configuration was chosen for two reasons: first, this permits the weight of the engine to cause compression in the blades while on the launch pad; and second, use of the tension-load mode permits the pivot to be designed for a zero rotational spring rate at full power operation of the engine. For explanation for this phenomenon, see Reference 1, Pages 26 and 27. For pivots crossed at the blade centers ($R = L/2$) the zero rotational spring rate ($M_R/\theta = 0$) is achieved by proportioning the blades so that the blade length

$$L = 2.4 \sqrt{\frac{EI}{P}} \quad \text{Eq. (1)}$$

where: E: Young's modulus, lb/sq in.
 I: moment of inertia of blade cross section, in.⁴
 P: axial load in the blade, lb

For concept designs and throughout this study, the flexure lengths were set to produce $M_R/\theta = 0$ at 75,000 thrust.

Examination of the equations shown by Figure 4 indicate that iterative methods must be used to calculate the flexure sizes suitable to permit a stipulated required angle of deflection with any stated combination of yield stress and Young's modulus, while also satisfying the previously stated requirement for approximately zero rotational spring rate at 77,000 lb thrust load.

The data presented in Figures 5, 6, and 7 were calculated by computer solution of the equations of Figure 1.

The design is required to withstand $\pm 4.0^\circ$ motion by applicable specification (Reference 2). However, to allow for the lower applicable fatigue stress suitable for operating life, a criterion of $\pm 8.5^\circ$ was adopted for estimating design proportions. This method also provides flexure design diameters and lengths which correlate with those received from an experienced vendor as a sales engineering estimate.

Figure 5 shows the flexure blade length variation as a function of material strength for two categories of material modulus of elasticity over a range of blade widths.

(Flexure blade width is the dimension parallel to the central axis of the flexure pivot, and length corresponds to the diameter of the flexure pivot, but is the working portion of the blade.)

Examining Figure 5 shows that the blade length (and therefore the pivot diameter and weight) increases hyperbolically as the F_{ty} is decreased. On restating the inference in a different way, the overall dimensions of a gimbal employing flexure pivots, and its weight, follow a law stated as

$$L \text{ and } W \text{ vary proportional to } \frac{1}{(F_{ty})^N}$$

For the design assumed, N has a value of about 2.7 for the variation in blade length with both values of Young's modulus used. It can be seen that very high yield strengths are most important to achieve minimum size and weight.

It is apparent also, that short blade lengths can be achieved with lower strength materials providing that the modulus of elasticity is low.

Figure 5 provides a variety of dimensional combinations for flexure blade design with no specific limiting characteristics. An inference can be drawn that as material strength improves into the 150 KSI range, blade length savings in size and weight are less significant. With high modulus materials then, a lower limit in yield strength should be set about there. Space limitations for the gimbal pivot limit the working length of flexure blade (but not width) to a maximum of about 5 inches. Inspection of the curves illustrates an attraction to wider blades regardless of the modulus of the material.

Further evidence in support of short wide blades is illustrated below.

← High L/W ratio blade is low in strength and stiffness when loaded in shear. This type loading occurs in pitch axis pivot when engine vectored in yaw produces a load component along pitch axis.

Low L/W ratio has high shear strength and stiffness, but produces pivot cylinder of abnormal length. →

Figure 6 plots the same family of pivots as Figure 5, but shows the thrust-wise spring rate of the blades related to the yield strength of the material in the blades. In both Figures 5 and 6, the family of designs produce ± 8.5 degrees permissible deflection when the combined tensile and bending stresses reach the yield stress indicated by the abscissa. The thickness of the blades is a dependent variable changing in value with the combinations of blade width and material strength in the same relationship as blade length.

Examination of Figure 6 indicates that: (1) the stiffness available in the flexure blades of a practical design is very high and will not compromise the dynamics of a rocket engine, and (2) that the stiffness increases proportionally

to the available yield strength of the material used.

In Figure 7, the length and thickness are held constant as parameters, and the resultant permissible vector angle, θ is plotted as a function of yield strength, F_{ty} .

Figure 7 indicates, that as one would expect, the permissible angulation increases almost proportionally with the yield stress, and that material of low Young's modulus permits higher angulation.

5.1.2 Alternate Design

In the event that the flexure pivot concept can not be adopted to the NERVA engine application an alternative concept is offered for consideration. See Figure 8.

The selection of an alternate to the flexure pivot was based upon the concept design study, Appendix A, where the second ranking design was chosen.

This alternate design employs a plain pin-type bearing as a hinge. Friction is not considered to be a prime problem, since the actuator is designed by inertial loads of much greater magnitude than the resistance of the gimbal.

To provide lubrication while in space vacuum environment, the hinge is sealed in a flexible bellows capsule filled with lubricant.

No further details are available on this alternative design, since it was not included in the project program plan or funding.

5.1.3 Fabrication

During design reliability analysis and Quality Engineering review of the gimbal pivot assembly and the material of construction, Maraging Steel, 18% Nickel, two areas of concern were noted:

a. Maraging Steel can attain a high degree of magnetization during fabrication processing.

b. Maraging Steel has a relatively low hardening temperature.

The magnetic properties can create problems with cleanliness control during fabrication and assembly through entrapment of foreign material. Special attention will be required to eliminate this item as a problem area. For instance, it was noted that a process specification used by one Aerojet affiliate for inspection of Maraging Steel forgings employs the magnetic particle principle. It may be that a different method be employed for gimbal pivot materials, or steps be taken to demagnetize the materials after inspection.

The low hardening temperature properties of Maraging Steel can also create problems, particularly when ductility in the end product is required. Fusion welding could harden local areas adjacent to the weldment. The basic design of the flexure blades, a highly stressed constituent of the gimbal pivot assembly contains enlarged ends, Figure 3, details -5 and -6 of Sheet 2, where fusion welding to the outer housing is performed, which keeps the weld affected zone out of the higher stressed areas.

After weld assembly of the flexure pivot, the unit will be annealed to relieve stresses then maraged at 900°F in a controlled atmosphere furnace.

5.1.4 Materials

The maximum stress at which a flexure blade can be worked has been identified as a critical parameter from the standpoint of allowable angular deflection of the gimbal pivot.

Low modulus of elasticity also increases the permissible angular deflection for any chosen set of other design parameters. However, low modulus of elasticity is exhibited usually by materials having low strengths relative to the ferrous/nickel alloys generally used for flexures. Low modulus materials which have high strengths generally exhibit unfavorable welding properties and/or questionable toughness at sub-zero temperatures.

For the above reasons, the search for suitable materials has been concentrated on ferrous/nickel alloys. However, certain titanium alloys (low modulus materials) have been considered for comparison with the stronger steels. The following tabulation lists those materials that have been examined for suitability for a flexure design.

	<u>MATERIAL</u>	<u>X 10⁶</u> <u>E</u>	<u>R.T.</u> <u>KSI</u> <u>F_{ty}</u>	<u>R.T.</u> <u>% ELONG</u>	<u>-320°F</u> <u>% ELONG</u>	<u>REF</u>
1.	A286 Forging	28.8	92.4	15.2	26.0	(6)
2.	A286 Cold W.	28.8	159	10.9	?	(7)
3.	18 Ni Maraged Steel	27.0	208*	12.0	8.0	(8)
4.	Lescalloy 718	29.4	152*	16.0	26.0**	(10)
5.	Titanium A-110-AT	16.0	95.0	16.0	12.5	(9)

*The published "typical" data were multiplied by 0.8 to make these values comparable with the values given for other materials.

**RT Elongation is 2" for .250 sheet. At -320°F the figure given is for .062 sheet over 0.5".

A-286 is a material whose mechanical properties are largely dependent upon the amount of cold working employed in processing the material or shaping the part from the material. The data of items 1 and 2 provide a clear indication of this characteristic. It is doubtful, however, that the higher strength cold worked material can be used in a flexure pivot design. Assembly of the flexure pivot requires welding; stress relief heat treatment following welding will remove most of the strengthening effects of the cold working. However, this is a matter which should be explored further with a subcontractor having experience in crossed-flexure design and fabrication.

Materials with low tensile yield strength, such as Number 1 above, produce designs with long blades, and become impractical from the viewpoint of space required and weight. It has been established that the size of the pivot varies inversely as the strength of the material. The relation between the yield strength and the flexure length is shown in Figure 5.

Item 3 in the above chart, 18% Nickel maraging steel, is commercially available in formulations such as Vascomax 250, marketed by Vandadium Alloy Steel Co., Latrobe, Pennsylvania. It is known that the titanium nitride impurity content affects the elongation. For use in flexure blades, one user of this material purchases the steel to his own specification in which the titanium nitride content and distribution is regulated more closely than is the supplier's practice.

Regarding suitability of 18 Ni Maraged Steel, it was stated in the discussion of thermal analyses that the minimum expected temperature of the assembly is -200°F , therefore, it is not expected that elongation of less than 8%, except from the statistical point of view, needs to be considered.

In response to a need for a protective coating for Maraging steel in the sea coast environment of the launch pad, a pure, dense, aluminum coating, 0.5 to 1.5 mils thick, produced by a process which will not embrittle the base metal with hydrogen or soften the base metal with process heat, is offered as a candidate.

Three coating processes were evaluated which include (1) ion plating by Battelle Memorial Institute, (2) alkyl plating by General Technologies Corporation, and (3) mechanical plating by 3M. Ion plating looks the most promising with rigorous precoating cleaning, uniform process control for good adhesion, and capability of handling parts to 1 ft x 2 ft x 6 ft size. Alkyl plating is less attractive since 500°F heat is required and cleanliness control is questionable. Aluminum coating is not deposited by the 3M mechanical plating process.

It is recommended that a representative specimen be made available to coat utilizing the ion plate process. The part could be tested for adhesion and corrosion resistance by accelerated salt spray cabinet exposure for further process evaluation.

5.2 NUCLEAR RADIATION

5.2.1 Summary and Conclusions

Thermal heating rates due to nuclear radiation have been estimated for specification extremes for both the flight condition and the ground test condition. References (11) and (12) show these rates.

The materials being considered for gimbal use are all of such chemistry that no degradation is expected from the nuclear field.

5.2.2 Method of Analysis

The heating rates were estimated for the gimbal pivot assembly at selected points radiation doses from unperturbed gamma KERMA rate and neutron flux maps, adjustment of the dose rates by estimating perturbations due to secondary radiation sources and PVARA radiation leakage, and conversion of radiation dose to heating rate by application of the energy absorption coefficient for 18% Nickel Maraging Steel.

5.2.3 Description of Input Data

The heating rates are based upon specification extreme conditions with secondary gamma radiation from other components which influence gimbal pivot assembly heating. The shielding effect of the engine external disk shield is not included in this analysis. For the case of ground test conditions, the facility contributions were estimated. No consideration was given to self shielding of the gimbal pivot assembly.

The engine model for which the analysis was performed was 1137400-A.

5.2.4 Results

Because the gimbal is located a relatively long distance from the nuclear source, the heating rates are relatively low, 1.3 to 1.6×10^{-3} Btu/in³-sec for flight environment. For the ground test environment, the heating rates range between 1.9 to 2.2×10^{-3} Btu/in³-sec; still very low.

5.3 THERMAL

5.3.1 Summary and Conclusions

The gimbal pivot is located between and firmly attached to two aluminum structures. Since these structures represent a significantly larger surface area exposed to the sun than the gimbal pivot, the temperature of the gimbal pivot during periods when the engine is not running is largely dependent upon the temperature of these structures.

The thermal analyses performed for the NERVA Engine 1137400-C indicate that the range of temperatures expected for the gimbal pivot assembly for all natural and induced environments are well within the capabilities of the component.

5.3.2 Method of Analysis

The gimbal pivot assembly thermal data was obtained during an analysis of the total NERVA Engine assembly. Thermal interaction between the gimbal pivot assembly and mating components, the upper and lower thrust structures were provided for in the engine thermal model.

The finite difference method of thermal analysis was performed with the SINDA-3G (Systems Improved Numerical Differencing Analyzer for third generation computers) program. SINDA is a dimensionless multi-option compiler program for analyzing one, two, or three dimensional lumped parameter representations of physical systems which can be described by sets of diffusion equations with source terms.

5.3.3 Description of Input Data

The space coast solar heating case considered continuous solar exposure at one Au with the engine axis normal to the source of radiation and an orientation about the roll axis such that peak exposure occurs at $\theta = 135^\circ$.

The gimbal and thrust structures were considered to be coated with aluminum oxide having an absorptivity of 0.52 and an emissivity of 0.64. These are specification extreme values and include an allowance for degradation due to ultraviolet radiation and neutron exposure.

The full power 60 minute engine operation thermal results represent specification extreme conditions which represent the least favorable combination of limits.

5.3.4 Results

The range of temperatures for the gimbal pivot under the conditions of 5.3.3 above for the case of space coast with solar heating is 15° to 73°F . At the end of a 60 minute operating period at full power, the nuclear radiation heating effects have essentially no effect. The temperatures increase 5 or 6 degrees at each end of the range.

For a hypothetical case where no solar energy is applied to the thrust structures or gimbal, such as a condition where the stage is pointed at the sun with the engine in the shadow of the tank the gimbal pivot assembly temperature would decay, during a period of 350 hours of space coast, to about -200°F .

5.4 STRUCTURAL

5.4.1 Loads

5.4.1.1 Summary and Conclusions

Preliminary stress analyses of the gimbal pivot assembly were made for the condition of normal full power engine operation only. The nominal engine thrust value, 75,000 lb, was used rather than the specification extreme value of 77,000 lb, but the margins of safety are large enough to accommodate the difference.

Loads due to thermal environment, engine transient operation, engine ground testing, handling shipping, and engine gimbaling were not considered in this analysis.

The analysis of Reference 13 lists very high axial loads for launch conditions based upon the engine movable mass being supported against transverse acceleration by cantilever beam action, resulting in very high moment reactions in the gimbal and gimbal actuators. The recommended engine support method to accommodate the boost loads of Reference 14 was used to alleviate the high loading condition. The launch support method required lateral ties from the engine at the nozzle throat to the stage ascent shell and axial struts from the stage/engine interface to the lower level of gimbal actuator attachment (on the LTS) to isolate the gimbal and gimbal actuators.

5.4.1.2 Method of Analysis

Development of engine operation loads on the gimbal pivot assembly were the result of fundamental manual calculations considering only static accelerations and nominal thrust load due to engine operation.

The flexibility data resulted from hand calculations with gimbal pivot assembly design of Figures 1, 2, and 3. The preliminary design work by the Bendix Corporation demonstrated a need to change from the "X" configuration to a "+" arrangement for the flexure blades. This change is not represented in the flexibility matrix of this report.

5.4.1.3 Description of Input Data

Gimbal Pivot Assembly interface loads were determined at an engine thrust load of 75,000 lb with a lateral static acceleration of 1/8g acting upon the engine. No dynamic loading effects were considered, nor were loads due to engine gimbaling.

5.4.1.4 Results

Gimbal pivot assembly interface loads were determined for the nominal full power engine operating case with no thrust vector control activity. The reactive loads are:

103,000 lb axial
5,000 lb lateral

A flexibility analysis of the gimbal pivot assembly produced the following:

	<u>Δ_x</u>	<u>Δ_y</u>	<u>Δ_z</u>	<u>θ_z</u>
P_x	1.75	0	0	0
P_y	0	.4754	0	0
P_z	0	0	.4754	0
M_x	0	0	0	.0045

These flexibilities are based upon a 1,000,000 lb load and a 1,000,000 in-lb moment.

5.4.2 Mass Properties

5.4.2.1 Summary and Conclusions

Weight, moment of inertia and center of gravity analyses were conducted for the gimbal pivot assembly, the results reported in Reference 22. The weight of the gimbal pivot assembly was calculated to be 195 lb.

Target weights for all NERVA Engine components are listed in Reference 23. The target for the Gimbal Pivot Assembly is 170 lb.

5.4.2.2 Method of Analysis

Determination of Gimbal Pivot Assembly mass properties is a combination of computer solutions and manual calculations. Each constituent of the assembly is subjected to a manual weight analysis; the data input to an ANSC developed computer program for determination of C.G. and moment of inertia. The program provides these data for the assembly as well as each constituent.

5.4.2.3 Description of Input Data

Input data requirements for determination of mass properties is limited to an accurate description of the component via the design drawings and the materials of construction.

5.4.2.4 Results

The current calculated weight of the gimbal pivot assembly, described by Figures 1, 2, and 3 is 195 lb. Each flexure pivot weights 18.3 lb. The Bendix design study discussed elsewhere in this report identified the weight of a flexure pivot as 22 lb. The adjusted weight of the gimbal pivot assembly with Bendix design flexure pivots is 210 lb. This component, the gimbal pivot assembly is therefore, 40 lb heavier than the assigned target weight of 170 lb.

5.4.3 Dynamics Analysis

5.4.3.1 Summary and Conclusions

A dynamic analysis was conducted for the 1137400-B engine incorporating the flexibility matrix of the gimbal pivot assembly. The objectives of the analysis was to determine modal data for the engine, not individual components. The analysis was reported in Reference 24.

Free vibration analyses were conducted for the launch and boost phase and engine operational phase of flight. For each, three specific axes of lateral motion were considered in addition to longitudinal motion.

No conclusions can be drawn until a response analysis is conducted.

Further work was done, which relates to the gimbal pivot assembly with an engine model revised to the 1137400-D configuration. This work was reported in References 25 and 26.

5.4.3.2 Method of Analysis

The dynamic analysis of the 1137400-B engine was conducted using a dynamic computer code called SPRVIB (Spring Vib.) developed at WANL and converted to the IBM 360-65 at ANSC. It is identified as Number D13907. The program computes the natural frequencies and mode shapes with any particular problem input in matrix form using Jacobi's method of successive rotations.

The dynamic analysis as the 400D engine employed the NASTRAN program using the original scalar spring-mass model of the SPRVIB program.

5.4.3.3 Description of Input Data

A scalar spring-mass model of the 1137400-B engine was developed using flexibility matrices for all thrust train components. The model contained 92 degrees of freedom.

For the dynamics response analysis performed on the 1137400D engine the nuclear stage operation input consisted of a 1g input at each TPA and at the nozzle. For launch vehicle operation two time periods were considered, $T = 10$ seconds (liftoff) and $T = 160$ seconds (the period of maximum acceleration). For $T = 10$ seconds the spectral distribution of Reference 14 was combined into a single transient load for the lateral and longitudinal directions. For $T = 160$ seconds, the input was taken to be a steady state acceleration of 0.1g for all natural frequencies up to 36 Hz.

5.4.3.4. Results

The longitudinal analytical data from the free vibration analysis of the 1137400-B engine indicated that the lowest natural frequency, 6.5 Hz, was characteristic of the stage structure, not the engine.

The lateral data revealed disproportionate deflection characteristics in the vicinity of the gimbal pivot and uppermost part of the lower thrust structure. These deflections can be attributed in part to unsymmetrical design, but also indicate a potential problem area. That is, should a need arise to increase engine stiffness, then the gimbal pivot and the local area of the lower thrust structure should be approached first.

The results of the analysis of Reference 25 are presented at Table I.

5.4.4 Stress Analysis

5.4.4.1 Summary and Conclusions

The preliminary stress and flexibility analysis was reported in Reference 4 and 15. The gimbal pivot assembly is structurally adequate to sustain primary loads due to nuclear stage operation. If it is found that the stiffness requirements are greater than the current values, they can be increased considerably by an increase in thickness and radii of the cruciform.

Design data for the flexure pivots was submitted by the Bendix Corporation by References 18, 19, 20 and 21. See Figure 9 for a summary. From the work done with the alternate material, Lescalloy 718, it can be concluded that the increased diameter and length of the gimbal pivot assembly could require redesign of the cruciform and trunnions with a probable violation of the space allotment and weight target for the gimbal pivot.

5.4.4.2 Method of Analysis

The stress and flexibility analysis of the gimbal pivot assembly consists entirely of manual calculations to determine component sizes, flexibilities, and margins of safety, in adherence to the criteria of SNPO-C-1, "Structural Design Requirements for NERVA Program."

5.4.4.3 Description of Input Data

A preliminary stress analysis was performed for the gimbal pivot assembly configuration of Figure 1. Material properties were obtained for Vascomax 250 CVM from a Vendor Catalog, VASCO, Latrobe, Pa. The thermal environment was considered to be 300°F. Thermal analyses subsequent to the completion of the stress analysis indicate that the temperature is less than 100°F, tending to make material allowable strengths used in the analysis conservative.

The loads used for the analysis were obtained from Reference 13, nuclear stage operation only.

5.4.4.4 Results

A summary of the minimum margin of safety for each component and the mode and magnitude of stress is presented below.

<u>Component</u>	<u>Stress (psi)</u>	<u>Mode</u>	<u>M.S.</u>
Cruciform	138,500	Tension & Bending	+0.44
Trunnions	45,000	Compression & Bending	+3.45
Trunnion Bolts	101,000	Tension	+0.51*

*Assumed 5/8 x 18 UNF-3B Bolt with Fty = 180 KSI.

5.5 RELIABILITY AND SAFETY

5.5.1 Failure Mode Analysis

A cursory analysis was conducted for the Gimbal Pivot Assembly and reported in Reference 27. The modes and mechanisms presented below were expanded in Reference 27 with a discussion of a plan for attaining reliability.

<u>Mode Ident. No.</u>	<u>Component Mode of Failure</u>	<u>Mech. Ident. No.</u>	<u>Component Mechanism of Failure</u>
1.	TS Attachment bolt failure	1.0	Tensile failure
		1.1	Shear failure
		1.2	Bending failure
		1.3	Combination of 1.0, 1.1, 1.2
		1.4	Vibrating loose
2.	TS Attachment lug failure	2.0	Bending failure
		2.1	Tear out failure
3.	Failure of bolts attaching flexure core to trunnion or to cruciform	3.0	Vibrating loose
		3.1	Shear failure
4.	Failure of flexure blades	4.0	Compressive buckling
		4.1	Tensile failure
		4.2	Combined bending tensile failure
5.	Cruciform frame failure to transmit load	5.0	Failure in bending
6.	Failure to permit full motion	6.0	Entry of foreign body

The September 1971 issues of R-202 (Reliability Allocations, Assessments and Analysis Report) and S-103 (Flight Safety Contingency Analysis Report) reflect the currently defined failure modes and their allocated failure probabilities.

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2. ANSC Specification No. EC-90299, "Detail Specification, Part 1, Performance/Design and Qualification Requirements for Gimbal Pivot"
3. ✓ ANSC Memorandum JDS:7731:M1445 dated 27 April 1970, K. Sato to L. A. Shurley, "Terhmal Analysis of the NERVA Engine 1137529 Gimbal Concept"
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10. Lescalloy Book
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12. ✓ ANSC Memorandum 7831:6296M dated 20 May 1970, J. C. Courtney to W. E. Stephens, "Gimbal Assembly Heating Rate - Ground Test Environment"
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14. NASA Report No. SNPO-C-6, dated 22 June 1970, "NERVA Program External Environments Definition and Requirements"

15. ✓ ANSC Memorandum JHR:7733:1221, dated 30 April 1970, J. H. Robbins to J. Nevenzel, "Preliminary Stress and Flexibility Analysis of Gimbal Assembly 1137978"
16. ✓ ANSC Memorandum KET:7733:1229, dated 11 May 1970, G. S. Kaveney to A. L. Armitage, "Weight Analysis for the Gimbal Assembly"
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18. ✓ Bendix Letter to Lyle Lutton, ANSC, dated 9 December 1970, Subject: Purchase Order N00494, Purpose: Transmittal of Preliminary Layout of Gimbal Pivot Drawing LD6172 and Weight Estimate
19. ✓ Bendix Letter to Lyle Lutton, dated 14 December 1970, Subject: Purchase Order N-00494, Purpose: Transmittal of Installation Drawing of Bendix Gimbal Pivot No. 2479466 and Stiffness and Spring Rate Data
20. ✓ Bendix Letter to Lyle Lutton, dated January 5, 1970, Subject: Flex Pivot Layout Drawing LD6172, Purpose: Transmittal of "A" Change Drawing.
21. ✓ Bendix Letter to Lyle Lutton, dated 10 March 1971, Subject: Purchase Order N-00494, Purpose: Transmittal of Pertinent Design Data for a Gimbal Pivot of Alternate Material
22. ✓ ANSC Memorandum N4310:038, R. C. Rosser to H. J. Bronner, dated 12 February 1971, Subject: NERVA Engine Mass Properties
23. ✓ ANSC Memorandum N4110:0090M, H. J. Bronner to R. W. Froelich, dated 13 April 1971, Subject: NERVA Engine and Components Target Weight Update
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25. ✓ ANSC Engineering Operations Report No. N8120R:71-003, dated 30 June 1971, "Launch and Nuclear Space Operation Vibration"
26. ✓ ANSC Engineering Operations Report No. N8610:71-009, dated July 1971, "NERVA Engine Auxiliary Support for INT-21 Launch and Boost"
27. ✓ ANSC Memorandum 7750:M0860, L. A. Shurley to J. H. Ramsthaler, dated 14 April 1970, Subject: Failure Mode Analysis for Project 1.6.3, Gimbal Assembly and Actuator

M_R = RESTORING MOMENT OF FLEXURE PIVOT ABOUT ITS AXIS
 POSITIVE $\frac{M_R}{\theta}$ INDICATES MOMENT OPPOSING ANGULATION
 NEGATIVE $\frac{M_R}{\theta}$ INDICATES MOMENT AIDING ANGULATION

$$J = \sqrt{\frac{EI}{P}}$$

$$M_1 = \frac{EI\theta}{J} \left[\frac{\frac{L}{2J} - \frac{R}{J}}{\frac{L}{J} \coth\left(\frac{L}{2J}\right) - 2} + \frac{1}{2} \coth\left(\frac{L}{2J}\right) \right] \text{ FOR } R = \frac{L}{2} \text{ THIS BECOMES } M_1 = \frac{EI\theta}{2J} \coth\left(\frac{L}{2J}\right)$$

$$M_2 = \frac{EI\theta}{J} \left[\frac{\frac{L}{2J} - \frac{R}{J}}{\frac{L}{J} \coth\left(\frac{L}{2J}\right) - 2} - \frac{1}{2} \coth\left(\frac{L}{2J}\right) \right] \text{ FOR } R = \frac{L}{2} \text{ THIS BECOMES } M_2 = \frac{EI\theta}{2J} \coth\left(\frac{L}{2J}\right) = -M_1$$

$$S = P\theta \left[\frac{1 - \frac{R}{J} \coth\left(\frac{L}{2J}\right)}{\frac{L}{J} \coth\left(\frac{L}{2J}\right) - 2} \right] \text{ FOR } R = \frac{L}{2} \text{ THIS BECOMES } S = P\theta \left[\frac{1 - \frac{L}{2J} \coth\left(\frac{L}{2J}\right)}{\frac{L}{J} \coth\left(\frac{L}{2J}\right) - 2} \right]$$

$$\frac{M_R}{\theta} = \frac{EI}{J} \left[\frac{\frac{L}{2J} - \frac{R}{J} \left(\frac{L-R}{J}\right) \coth\left(\frac{L}{2J}\right)}{\frac{L}{J} \coth\left(\frac{L}{2J}\right) - 2} + \frac{1}{2} \coth\left(\frac{L}{2J}\right) \right] \text{ FOR } R = \frac{L}{2} \text{ THIS BECOMES } \frac{M_R}{\theta} = \frac{EI}{2J} \left[\coth\left(\frac{L}{2J}\right) - \frac{L}{2J} \right]$$

FIGURE 4
 SUMMARY OF FORMULAS FOR
 TENSION LOADED CROSSED FLEXURE PIVOTS

COMPARISON OF FLEXURE BLADE DIMENSIONAL RELATIONSHIPS
WITH MATERIAL PROPERTIES

FIGURE 5

EUGENE DIETZGEN CO.
MADE IN U. S. A.

NO. 340R-M DIETZGEN GRAPH PAPER
MILLIMETER

LB/IN STIFFNESS K (SPRING RATE) FOR BLADE OF X-FLEXURE PIVOT

5x10⁶ 10x10⁶ 15x10⁶ 20x10⁶

0 100,000 200,000
FTY LB/IN²

FIGURE 6

FLEXURE PIVOT GIMBAL ANGLE RELATIONSHIP
WITH MATERIAL PROPERTIES

For X-Flexure pivot with 60° included angle and zero rotational spring rate at 75000 lb. load.

FIGURE 7

SEALED PLAIN BEARING
GIMBAL PIVOT CONCEPT

FIGURE 8

PERTINENT DATA FROM THE BENDIX
DESIGN STUDY OF THE FLUEXURE PIVOT

	MARAGING STEEL, E-18V	LESCALLOY 718
DIA. (in)	5	7
LENGTH (in)	8	14
TORSIONAL $\left(\frac{lb \text{ in}}{deg}\right)$		
NO LOAD	663	878
FULL LOAD	5.4	48
AXIAL STIFFNESS $\left(\frac{lb}{in}\right)$		
X	1.5×10^6	3×10^6
Y	NA	17×10^6
Z	5.7×10^6	17×10^6
WEIGHT (lb)	22	75 *

* Estimated by Scaling $Dia^2 \times L$

FIGURE 9

APPENDIX A

15 January 1970

(Preliminary)

CONCEPT DESIGN STUDY OF GIMBAL
BEARING ASSEMBLIES FOR NERVA

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APPENDICES:

- A. "Forcing a Good Decision", M. J. Gelp;
- B. Decision Matrix for Gimbal Assembly

LIST OF REFERENCE DRAWINGS

1136835	Gimbal Bearing Concepts
1136836	Gimbal Bearing Concepts
1136750	Gimbal Study - Use of Shear Harness
1137057	Concept - Cylindrical Housed Flexure Gimbal Pivot
1137058	Concept - Channel Housed Flexure Gimbal Pivot
1137059	Concept - Plain Bearing Gimbal Pivot
1137060	Concept - Anti-friction Bearing Pivot
1137079	Concept - Flexure Housing with Overload Provision
1137083	Concept - Cross-cylindrical Gimbal Pivot
1137089	Concept - Gimbal Bearing
1137098	Concept - Compact X-flexure Gimbal Bearing
1137099	Concept - Compact X-flexure Gimbal Bearing
1137138	Concept - Cylinder Housed Flexure with Overload Provision
1137145	Disclosure - Elements of Overload Relief Device for Flexure Pivots

I. INTRODUCTION

This report records a study made (as of 15 October 1969) of the capabilities and limitations of several possible gimbal bearing concepts, and documents the method used in choosing the optimum candidate from these designs suitable for the reference NERVA hot-bleed cycle engine design configuration drawing number 1136339.

II. SUMMARY

The study surveys nine gimbal bearing concepts which appear able to meet the performance requirements of the NERVA engine within the space allotted in recent configuration layouts. Three frame diameters and three styles of bearing comprise the elements combined into the nine concepts studied.

Diameters considered fall into three categories, the first being those large enough to mount on the outer portions of the middle thrust structure and upper thrust structure, thereby avoiding large bending moments and deflections in the adjacent thrust structure transverse members to which the gimbal assembly attaches. In the second category there are two minimal-diameter "universal joint" concepts. Medium diameter concepts fall into the third category.

The study considers bearings of three types; these are the plain-bearing type, the rolling element anti-friction bearing, and the cross-flexure pivot.

Each of the concepts studied was evaluated on nine attributes. Using a methodology borrowed from value engineering, each concept is assigned a figure of merit based on a weighted scoring of its estimated relative excellence on each of the nine attributes.

This study rates highest the gimbal concept using a medium diameter frame combined with crossed-flexure pivots.

III. REQUIREMENTS

The gimbal bearing assembly must carry loads imposed upon it by the thrust structure under boost condition and under operational condition. The loads itemized below contain no multiplicative factors; they represent the nominal design loads before application of design factors or safety factors.'

A. BOOST CONDITION

1. Inertial loads caused by 6g acceleration of the vehicle axially.
2. Inertial loads due to transverse 1g acceleration of engine, acting in any transverse orientation.
3. Inertial loads caused by the response of the engine to vibration applied at vehicle/engine interface; $\pm 2.25g$ at 3 to 35 Hz axially and $\pm 1.5g$ at 0.1 to 15 Hz transverse, as specified by NERVA Engine CEI Spec. C002-CP090194-F-1 Nov. 1968.
4. During the application of the above loads the temperature was assumed to be from 400°R to 600°R.

5. Any or all of the loads stated above in any combination from zero to maximum.

B. OPERATIONAL CONDITION

1. Axial thrust load nominally 75K while vectoring $\pm 3.0^\circ$ nominal in either pitch or yaw or both combined (square pattern) at $0.75^\circ/\text{sec}$ maximum angular velocity.

2. Shear load caused by the application of $1/8g$ acceleration in any transverse direction to the vehicle axis.

3. Transmitted vibratory loads in transverse plane arising from pump abnormal operation. Magnitude and frequency range were not yet determined at time this study was made.

4. Additional axial component to balance TVS drive mechanism reaction to transverse acceleration. (Currently time, with present layouts, this amounts to adding roughly 20% to thrust).

5. The above loadings must be carried at temperatures assumed to be within the range 200°R to 800°R .

6. The gimbal must carry these loads at the greatest combination of above components, while being vectored $\pm 3.0^\circ$ for 10,000 cycles, and $\pm 3.5^\circ$ for 100 cycles, with one-half of the test conducted at 200°R and one-half at 800°R . In the absence of stated requirements for cycles of operation, these criteria were assumed to provide a reliable design.

IV. CONCEPTS STUDIED

A. UNIVERSAL JOINT

See Print 1136333. This print shows a compact unit using plain bearings lubricated by molybdenum sulfide surface treatment. Axial slack is eliminated as far as possible by using a flanged type of bearing bushing bolted tightly over laminum shims, permitting adjustment on assembly to eliminate end-play. Torsional rigidity is supplied by installing a harness per Drawing 1137083 sheet 2. The term "plain bearing" means the type in which primary motion is sliding.

B. UNIVERSAL JOINT VARIATIONS

The general mechanism of the universal joint can also be designed using anti-friction bearings, or using crossed-flexure pivots (referred to later in this document as X-flexure). A harness per Drawing 1137083 sheet 2 is required to furnish torsional rigidity.

The variation with anti-friction bearings was not drawn as a layout since it is very easy to mentally substitute an anti-friction bearing for the plain bearing shown on 1136333. However, suitable X-flexure cells have greater width and smaller diameter than plain or anti-friction bearings for the same load rating. Assemblies using the X-flexure cannot be classified as compact U-joints since their span classifies them under Paragraph B(2) below.

B. (1) Titan first stage engines used a crossed cylinder gimbal with needle bearings. Concept layout 1137083 illustrates the application of this type to NERVA design. Since this design is incapable

of carrying reverse-thrust loads, it incorporates lugs which would take the boost phase loads. On Titan, the assembly was arranged to pick up the reverse load after a slack motion of 0.002 inches. This arrangement offers little resistance to loads rotating the engine around the longitudinal axis. Therefore, the layout includes a harness arrangement to provide torsional rigidity and to carry transverse loads.

B. (2) Extended floating member gimbal bearings. Rings or cruciform frames or blocks are commonly used in gimbal design to carry loads between widely separated opposing gimbal bearing pairs. Toroidal rings have less rigidity (for a given weight) than rectangular or cruciform structures and therefore have not been considered in this study.

These extended floating members can be pivoted by any of the three pivots discussed, that is, plain bearing, anti-friction bearing, or X-flexure, as illustrated by 1137059, 1137060, 1137058, and 1137057.

One can design many combinations of structures and bearings, limited by space available between propellant feed lines and by the mounting locations available on the thrust structure, taking due account of the overall rigidity and evenness of load distribution of the combination.

The report is limited to the more promising combinations; several "starts" were discarded because early in the layout process spatial interference or other undesirable factors appeared.

1. Drawing 1137089 illustrates the use of an extended floating member using aircraft style sheet-and-extrusion construction to

provide a light, rigid structure. Either anti-friction bearings or plain bearings can substitute for the X-flexure pivot used in the layout. This particular drawing was dimensionally scaled to fit an un-numbered middle thrust structure (MTS) layout which was superseded by something more promising. 11137089 will fit middle thrust structure 1136758 when the left and right arms are suitably shortened, producing a bearing of the type shown by 1136836 upper left view. By adding husky load distributing members to 1136758 the load can be equalized between the four main axial members. The attachments to the upper thrust structure (UTS) are more widely spread than those attached to the MTS.

Figure 1

2. Print 1137060 shows a concept for gimbal bearing assembly which makes use of forged beam members and spherical roller bearings. The assembly mounts on middle thrust structure assembly 1136755 at the upper periphery of the conical frustrum.

3. Prints 1137098 and 1137099 show medium-diameter assemblies with cruciform; a machined forging floating member transmitting loads between X-flexure pivot units mounted to upper and middle thrust structures. The span between pivots should provide sufficient torsional rigidity to avoid using a separate shear harness.

V. DISCUSSION

At this point note the applicable characteristics of the several devices, especially noting those attributes which may distinguish one or several as superior to the others.

A. LUBRICATION

These devices must function after relatively long periods of inactivity in space vacuum and cold environment. The lubricant used must be suitable to prevent contact welding in this environment, and further must be suitable for operation at temperatures up to TBD°R. Molybdenum sulfide is suitable for such use, and is planned for use on plain anti-friction bearing types. However, the lubrication problem does not exist on the X-flexure pivot, which has no sliding or rolling members.

B. AXIAL RIGIDITY

Together with the thrust structure, the gimbal bearing assembly must be sufficiently rigid to provide the engine with natural vibratory frequencies exceeding the minimums established by specification. At first glance, it appears advantageous to place the pivots at or near the thrust structure periphery to minimize bending deflections and stress produced by central loading. (see Figure 2).

Figure 2

This reasoning prompted the consideration of the concepts shown on 1136835 and 1136836.

However, as layouts of this type of extended center member progressed, using available space, it appears the flexural rigidity gained by widely separating the pivots will be offset by the deflection of the connecting center block. (See Figure 3).

Figure 3

C. TORSIONAL RIGIDITY

To give the engine reasonable strength and sufficiently high frequency in the mode vibrating about the engine axis, the gimbal bearing assembly and adjacent structure must have suitable rigidity. As the mountings of the gimbal are moved from the periphery toward the center of the thrust structure, both the forces involved and the deflection in the adjacent thrust structure increases hyperbolically as the center is approached. (See Figure 4).

Figure 4

Thus, the span between pivot points must be sufficient to keep the loading moderate and the rigidity high. Therefore, each of the very short span universal joint types (Paragraph IV-1, -2, E-3) will require the addition of a harness as shown by 1137083 sheet 2.

D. SLACK MOTION

To prevent hammering and/or non-linear load deflection characteristics, the slack motion due to clearances within the bearing components must be minimized. Preloading the anti-friction type bearing to eliminate slack for a wide temperature variation is impossible if aluminum housings are used. Use of steel alloy housing will preserve pre-load, but incurs a weight penalty. Similarly, it is possible to reduce, but not entirely eliminate, slack in plain bearings. The X-flexure has no component clearances to produce slack motion and resulting discontinuity in the load-deflection characteristic. (See Figure 5).

IDEALIZED LOAD DEFLECTION CHARACTERISTICS

Figure 5

E. RESISTANCE TO MOTION

Either plain bearings or anti-friction bearings have a friction effect opposing motion. This friction is a very small factor compared to other loads opposed by TVS drive mechanism. X-flexures exhibit virtually zero opposition to rotation when under their nominal design load; however, under lower or higher loading, the X-flexure exhibits a spring-like resistance to motion which increases with deflection, but is zero at zero angle. The spring rate is positive (stable) for loads from zero to nominal, and are negative (unstable) for loads above nominal. The magnitude of this spring rate is a non-simple function of the geometry and material characteristics of the flexure ligaments. For estimating purposes in this application, pending detail design, it may be taken as 100 lb. per degree of vector angle, or less, at the drive mechanism attachment. The result is a negligible load at the actuator drive point.

F. EXPECTED USEFUL LIFE

The plain bearing concept being considered shows low unit loadings and can be expected to perform for a very large number of cycles, depending on lubricant life. The expected useful life is therefore limited by the stability of the lubricant rather than bearing material or geometry. These remarks apply also to anti-friction bearings. An anti-friction bearing with manufacturer's dynamic rating sufficient to carry the design operational load lacks sufficient manufacturer's static capacity to carry

the boost phase load without detrimental distortion. Larger sizes do not permit ready adaptation to present space allotted for the gimbal assembly.

G. AXIAL LOAD CAPACITY

Several styles of anti-friction bearings have suitable axial capacity. Plain bearings as shown by 1136333 and 1137059 can be arranged to easily carry axial load. X-flexures inherently resist axial load, with a reduction in permissible angular deflection ratings in presence of the axial load.

H. TEMPERATURE TOLERANCE

All types of pivots here considered have load ratings that diminish with increasing temperature due to lowered material properties at higher temperatures. There is little to set one pivot apart from the other within the assumed required temperature range of 200°R to 800°R.

I. POST-FAILURE OPERATION

Depending upon the mode of failure, plain bearings or anti-friction bearings may become inoperable if there is a failure of a component or lubricant. The X-flexure assembly contemplated incorporates provisions that allow the pivot to continue operation as a plain bearing after the failure of one or all of the ligaments, thus providing a fail-safe feature.

VI. EVALUATION

Having presented the items being considered and their attributes, it remains to choose that combination which seems to be the highest overall reliability under existing circumstances.

With nine attributes to evaluate on nine candidate designs, the large number of size/attribute/design combinations creates a complex problem in deciding which seems most advantageous.

The solution to this decision problem is a logical methodology such as that described in Appendix A of this report. Appendix B records the application of this logical method to the present problem. Below is a list of the scores yielded by this method, in descending order of relative merit:

Medium-spread, X-flexure pivot, P/N 1137099	0.1778
Medium-spread, plain bearings	0.1516
Wide-spread, X-flexure pivot, P/N 1137089	0.1377
Medium-spread, anti-friction bearings	0.1271
Wide spread, plain bearings	0.1116
Wide spread, anti-friction bearings, P/N 1137060	0.1070
U-joint, plain bearing, P/N 1136333	0.0970
U-joint, anti-friction bearing	0.0738
Cross-cylinder pivot, P/N 1137083	0.0124

In accordance with the above logic, prints 1137098 and 1137099 were prepared as concepts of medium spread gimbal bearing assemblies using the X-flexure for pivot.

Assemblies of this concept can be arranged to mount on any of the thrust structure concepts currently available. Small adaptations in the gimbal assembly and in the thrust structures can be made to suitably mount the gimbal and distribute loads.

VII. CONCLUSIONS

1. Medium diameter gimbal assembly using X-flexure pivot is the optimum choice among the concepts considered.

2. The choice of the diameter to fulfill the "medium" concept must be made later when the magnitude of torsional loadings is known.

Forcing A Good Decision

This mathematical decision-making process differs from many others because it derives rather than assigns a relative ranking for each aspect of the problem being considered.

Good decisions don't "just happen." They are not the result of instinct, common sense, or experience—by themselves. Rather, good decisions result from weighing the relative values of each requirement against all possible solutions.

When one requirement is all-important, the right decision may be obvious. However, most decisions are not this simple. A combination of many requirements must usually be compared with several possible solutions. It becomes more difficult under these circumstances to make decisions with a high degree of confidence. The greater the number of variables, the greater the risk of neglecting an important factor, or assigning too little importance to some key aspect of the problem.

A number of means have been devised to aid the process of making good decisions. Comprehensive detailed statistical analyses can provide a sound decision-making basis. This may not be either practical or economical. Cost analyses, weight analyses, reliability analyses, and many others can be used to guide decision making. But, it is often necessary to make decisions weighting the merits of all of these requirements and more, in a short time, using a limited amount of

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data. It is also often necessary to make the best decision possible with information that is available. In any case, the objective of the decision maker is to be certain that the information that is available is evaluated in an orderly fashion. A simple mathematical process has been devised to aid in this evaluation.

Two basic steps are involved in this mathematical procedure for decision making:

1) Determine the relative importance of each performance or specification requirement;

2) Evaluate each possible solution against each of the above requirements.

For both of these steps, evaluations are arranged so that only two alternatives are considered at a time. Every possible combination is submitted for comparison. In this way, no shades of choice are required—only a *yes* or *no* decision for each evaluation.

The decision-making process can be illustrated by applying it to a typical problem. Assume that the military services require a component of electronic equipment. In designing the equipment, the supplier attempts to develop a unit that will meet military specifications and at the same time have a minimum overall cost (initial cost plus maintenance) to the military services. In designing the equipment, a decision is required whether to use existing printed-circuit boards from a previously designed unit, new printed-circuit boards designed especially for the present unit, or a newly designed "standard universal" board. To determine the lowest overall cost to the military services, the alternatives are

evaluated for these requirements:

- 1) Low design cost
- 2) Low manufacturing cost
- 3) Low maintenance stock cost
- 4) Low maintenance work cost
- 5) Low shock-resistance capability cost
- 6) Low vibration-resistance capability cost
- 7) Low cost for future systems.

The procedure used to establish the relative importance of each requirement is illustrated in Table I. The seven requirements are listed in the left-hand column and comparisons are made in the columns to the right. In comparing any two items, numeral one (1) indicates the more important requirement and zero (0) indicates the lesser requirement. For example, when comparing design cost to manufacturing cost in the first column, manufacturing cost is the more important consideration for this particular project. Similarly, for this hypothetical example, design cost is more important than maintenance stock cost, and design cost is more important than maintenance work cost. Thus, each evaluation is made by comparing only two items at a time, and every possible combination of the seven items is considered.

The total positive decisions for each requirement is tabulated in column N. The summation of these total positive decisions equals 21, which checks with the total number of possible decisions.

A relative emphasis coefficient for each requirement is obtained by dividing the number of positive decisions for each requirement by the total number of possible decisions, as shown in the right-hand column of the table.

Table I—Weighting of Requirements

	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	Positive Decisions (N)	Requirement Emphasis Coefficient (E = N/21)
1. Design Cost	0	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	5	0.238
2. Manufacturing Cost	1	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	6	0.286
3. Maintenance Stock Cost	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	1	0.048
4. Maintenance Work Cost	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	4	0.190
5. Shock-Resistance Cost	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	3	0.143
6. Vibration-Resistance Cost	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	1	0.048
7. Future System Cost	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	1	0.048
																						21	1.000

Total number of possible decisions = $\frac{n(n-1)}{2}$, where n = number of items under consideration; i.e., $\frac{7(7-1)}{2} = 21$ total possible decisions.

Table II—Evaluation of Solutions for Each Requirement

	1	2	3	Positive Decisions	Solution Emphasis Coefficient
1. Design Cost					
Existing Boards	1	0		1	0.33
New Boards	0		0	0	0.0
Standard Universal Boards		1	1	2	0.67
2. Manufacturing Cost					
Existing Boards	1	1		2	0.67
New Boards	0		1	1	0.33
Standard Universal Boards		0	0	0	0.0
3. Maintenance Stock Cost					
Existing Boards	1	1		2	0.67
New Boards	0		0	0	0.0
Standard Universal Boards		0	1	1	0.33
4. Maintenance Work Cost					
Existing Boards	1	1		2	0.67
New Boards	0		1	1	0.33
Standard Universal Boards		0	0	0	0.0
5. Shock-Resistance Cost					
Existing Boards	1	1		2	0.67
New Boards	0		0	0	0.0
Standard Universal Boards		0	1	1	0.33
6. Vibration-Resistance Cost					
Existing Boards	0	0		0	0.0
New Boards	1		1	2	0.67
Standard Universal Boards		1	0	1	0.33
7. Future System Cost					
Existing Boards	1	1		2	0.67
New Boards	0		0	0	0.0
Standard Universal Boards		0	1	1	0.33

Table III—Decision Matrix

	Existing Boards	New Design	Standard Universal Package
Design Cost 0.233 X	0.33 = 0.078	0.0 = 0.0	0.67 = 0.160
Manufacturing Cost 0.233 X	0.67 = 0.190	0.33 = 0.094	0.0 = 0.0
Maintenance Stock Cost 0.043 X	0.67 = 0.0325	0.0 = 0.0	0.33 = 0.0159
Maintenance Work Cost 0.190 X	0.67 = 0.128	0.33 = 0.063	0.0 = 0.0
Shock-Resistance Cost 0.143 X	0.67 = 0.035	0.0 = 0.0	0.33 = 0.047
Vibration-Resistance Cost 0.043 X	0.0 = 0.0	0.67 = 0.0325	0.33 = 0.0159
Future System Cost 0.043 X	0.67 = 0.0325	0.0 = 0.0	0.33 = 0.0159
	0.233	0.190	0.233

Figure of Merit = (Existing Emphasis Coefficient X Solution Emphasis Coefficient) = Figure of Merit

The second step in the process is illustrated in Table II. For each requirement—design cost, manufacturing cost, etc.—the three possible solutions are evaluated, one against another. The number of positive decisions for each solution is converted to a *solution emphasis coefficient*, as shown.

The preferred solution is then determined by summarizing these emphasis coefficients in matrix form, as shown in Table III. The product of each requirement emphasis coefficient (Table I) and each solution emphasis coefficient (Table II) is found, and these quantities are summed for each of the three possible solutions. These sums become "figures of merit" for the solutions. In this example, the use of existing printed-circuit boards is the best decision.

By breaking the decision-making process down into a simple step-by-step application of logic, an orderly decision-making procedure is insured. Thus, the decision-maker knows that his solution is as good as the accuracy of his input information. In the example illustrated, the decision-maker can have a high degree of confidence in his choice because the cost of the various requirements can be either calculated or estimated rather accurately; thus, the yes-no evaluations of the various requirements can be made with high assurance. In other cases, a decision might be required where it is difficult or even impossible to assign direct costs or dollar values to all aspects of the problem. In these circumstances, the degree of confidence in the final decision must be tempered by the decision-maker's confidence in his yes-no evaluations. But by applying an ordered process in making the evaluations, the decision-maker knows that all aspects of the problem will be accounted for, and that the final decision will not be subject to inconsistencies in his analysis of the problem.

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January 1967

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This page and the following pages record the decisions made for evaluation of the nine gimbal bearing assembly candidates described in the report to which this is appended. The method used is described in Appendix A of this document.

[illegible]

		<u>Positive Decisions</u>	<u>Relative Emphasis Coefficient</u>
1. Lubrication			
Universal joint, plain bearing	1 1 1 1 0 0 1 0	5	0.139
Universal joint, a-f bearing	0 1 0 0 0 0 0 0	1	0.028
Crossed cylinder pivot	0 0 0 0 0 0 0 0	0	0.0
Widespread pivots, plain bearing	0 1 1 1 0 0 1 0	4	0.111
Widespread pivots, a-f bearing	0 1 1 1 0 0 1 0	3	0.083
Widespread pivots, X-flexure	1 1 1 1 1 1 1 0	7	0.194
Medium-spread pivots, plain bearing	1 1 1 1 1 1 1 0	6	0.167
Medium-spread pivots, a-f bearing	0 1 1 1 0 0 0 0	2	0.056
Medium-spread pivots, X-flexure	1 1 1 1 1 1 1 1	8	0.222
	36		
2. Longitudinal Rigidity			
Universal joint, plain bearing	1 1 1 1 1 0 0 0	5	0.139
Universal joint, a-f bearing	0 1 1 1 1 0 0 0	4	0.111
Crossed cylinder pivot	0 0 1 1 0 0 0 0	2	0.056
Widespread pivots, plain bearing	0 0 0 0 1 0 0 0	1	0.028
Widespread pivots, a-f bearing	0 0 0 0 1 0 0 0	1	0.028
Widespread pivots, X-flexure	0 0 0 1 1 1 0 0	2	0.056
Medium-spread pivots, plain bearing	1 1 1 1 1 1 1 0	7	0.194
Medium-spread pivots, a-f bearing	1 1 1 1 1 1 1 0	6	0.167
Medium-spread pivots, X-flexure	1 1 1 1 1 1 1 1	8	0.222
	36		
3. Torsional Rigidity			
Universal joint, plain bearing	0 1 0 0 0 0 0 0	1	0.028
Universal joint, a-f bearing	1 1 0 0 0 0 0 0	2	0.056
Crossed cylinder pivot	0 0 0 0 0 0 0 0	0	0.0
Widespread pivots, plain bearing	1 1 1 1 1 1 1 1	6	0.167
Widespread pivots, a-f bearing	1 1 1 1 1 1 1 1	7	0.194
Widespread pivots, X-flexure	1 1 1 1 1 1 1 1	8	0.222
Medium-spread pivots, plain bearing	1 1 1 1 1 1 1 0	3	0.083
Medium-spread pivots, a-f bearing	1 1 1 1 1 1 1 0	4	0.111
Medium-spread pivots, X-flexure	1 1 1 1 1 1 1 1	5	0.139
	36		

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		<u>Positive Decisions</u>	<u>Relative Emphasis Coefficient</u>
4. Slack			
Universal joint, plain bearing	0 1 0 0 0 0 0 0	1	0.028
Universal joint, a-f bearing	1 1 0 0 0 0 0 0	2	0.056
Crossed cylinder pivot	0 0 0 0 0 0 0 0	0	0.0
Widespread pivots, plain bearing	1 1 1 1 1 1 1 1	3	0.083
Widespread pivots, a-f bearing	1 1 1 1 1 1 1 1	6	0.167
Widespread pivots, X-flexure	1 1 1 1 1 1 1 1	7	0.194
Medium-spread pivots, plain bearing	1 1 1 1 1 1 1 1	4	0.111
Medium-spread pivots, a-f bearing	1 1 1 1 1 1 1 1	5	0.139
Medium-spread pivots, X-flexure	1 1 1 1 1 1 1 1	8	0.222
		<u>36</u>	
5. Resistance to Motion			
Universal joint, plain bearing	0 0 1 0 0 1 0 0	2	0.056
Universal joint, a-f bearing	1 1 1 1 0 1 1 0	6	0.167
Crossed cylinder pivot	1 0 0 0 0 0 0 0	2	0.056
Widespread pivots, plain bearing	0 0 0 0 0 0 0 0	0	0.0
Widespread pivots, a-f bearing	1 0 1 1 1 1 1 1	4	0.111
Widespread pivots, X-flexure	1 0 1 1 1 1 1 1	7	0.194
Medium-spread pivots, plain bearing	0 0 0 0 0 0 0 0	2	0.056
Medium-spread pivots, a-f bearing	1 0 0 0 0 0 0 0	5	0.139
Medium-spread pivots, X-flexure	1 0 0 0 0 0 0 0	8	0.222
		<u>36</u>	
6. Useful Life			
Universal joint, plain bearing	0 1 0 0 0 0 0 0	1	0.028
Universal joint, a-f bearing	1 1 1 0 0 1 0 0	3	0.083
Crossed cylinder pivot	0 0 0 0 0 0 0 0	0	0.0
Widespread pivots, plain bearing	1 0 1 1 1 1 1 1	3	0.083
Widespread pivots, a-f bearing	1 0 1 1 1 1 1 1	6	0.167
Widespread pivots, X-flexure	1 0 1 1 1 1 1 1	7	0.194
Medium-spread pivots, plain bearing	1 0 0 0 0 0 0 0	2	0.056
Medium-spread pivots, a-f bearing	1 0 0 0 0 0 0 0	5	0.139
Medium-spread pivots, X-flexure	1 0 0 0 0 0 0 0	8	0.222
		<u>36</u>	

		<u>Positive Decisions</u>	<u>Relative Emphasis Coefficient</u>
7. Axial Load Capacity			
Universal joint, plain bearing	1 1 0 1 1 0 1 1	6	0.167
Universal joint, a-f bearing	0 1 0 0 1 0 0 1	3	0.083
Crossed cylinder pivot	0 0 0 0 0 0 0 0	0	0.0
Widespread pivots, plain bearing	1 1 1 1 0 1 1	7	0.194
Widespread pivots, a-f bearing	0 1 1 0 0 1	4	0.111
Widespread pivots, X-flexure	0 0 1 1 0 0 0	1	0.028
Medium-spread pivots, plain bearing	1 1 1 1 1 1 1	8	0.222
Medium-spread pivots, a-f bearing	0 1 1 0 1 0 1	5	0.139
Medium-spread pivots, X-flexure	0 0 1 0 0 1 0 0	2	0.056
		<u>36</u>	
8. Temperature Tolerance			
Universal joint, plain bearing	1 1 0 1 0 0 1 0	4	0.111
Universal joint, a-f bearing	0 1 0 0 0 0 0 0	1	0.028
Crossed cylinder pivot	0 0 0 0 0 0 0 0	0	0.0
Widespread pivots, plain bearing	1 1 1 1 0 0 1 0	5	0.139
Widespread pivots, a-f bearing	0 1 1 1 0 0 0 0	2	0.056
Widespread pivots, X-flexure	1 1 1 1 1 1 1 0	7	0.194
Medium-spread pivots, plain bearing	1 1 1 1 1 1 0 1 0	6	0.167
Medium-spread pivots, a-f bearing	0 1 1 1 0 1 0 0 0	3	0.083
Medium-spread pivots, X-flexure	1 1 1 1 1 1 1 1 1	8	0.222
		<u>36</u>	
9. Post-Failure Operation			
Universal joint, plain bearing	0 1 0 0 0 0 0 0	1	0.028
Universal joint, a-f bearing	1 1 1 0 0 1 0 0	4	0.111
Crossed cylinder pivot	0 0 0 0 0 0 0 0	0	0.0
Widespread pivots, plain bearing	1 0 1 1 0 0 0 0	2	0.056
Widespread pivots, a-f bearing	1 1 1 1 1 1 0 1 0 0	5	0.139
Widespread pivots, X-flexure	1 1 1 1 1 1 0 1 0	6	0.167
Medium-spread pivots, plain bearing	1 1 0 1 1 1 0 0 0	4	0.111
Medium-spread pivots, a-f bearing	1 1 1 1 1 1 0 1 0	6	0.167
Medium-spread pivots, X-flexure	1 1 1 1 1 1 1 1 1	8	0.222
		<u>36</u>	

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	Emphasis Coeff. from Page 1.	U-joint, pln brg		U-joint, Af brg		Cross Cyl Pivot		Wide Pivot, Plain brg		Wide Pivot AF Brg		Wide Pivot, X-Flex		Medium Pivot, Pln Brg		Medium Pivot, AF Brg		Medium Pivot, X-Flex	
1. Lubrication	0.111	0.139	0.0154	0.028	0.0031	0.0	0.0	0.111	0.123	0.083	0.0092	0.194	0.0215	0.167	0.0185	0.056	0.0062	0.222	0.0246
2. Longitudinal Rigidity	0.222	0.139	0.0308	0.111	0.0246	0.056	0.0124	0.028	0.0062	0.028	0.0062	0.056	0.0124	0.194	0.0430	0.167	0.0370	0.222	0.0492
3. Torsional Rigidity	0.194	0.028	0.0054	0.056	0.0108	0.0	0.0	0.167	0.0323	0.194	0.0376	0.222	0.0430	0.083	0.0161	0.111	0.0215	0.139	0.0269
4. Slack	0.028	0.028	0.0007	0.056	0.0015	0.0	0.0	0.083	0.0023	0.167	0.0046	0.194	0.0054	0.111	0.0031	0.139	0.0038	0.222	0.0062
5. Resistance to Motion	0.000	0.056	0.0	0.167	0.0	0.056	0.0	0.0	0.0	0.111	0.0	0.194	0.0	0.056	0.0	0.139	0.0	0.222	0.0
6. Useful Life	0.056	0.028	0.0015	0.083	0.0046	0.0	0.0	0.083	0.0046	0.167	0.0093	0.194	0.0108	0.056	0.0031	0.139	0.0077	0.222	0.0124
7. Axial Load Capacity	0.167	0.167	0.0278	0.083	0.0138	0.0	0.0	0.194	0.0323	0.111	0.0185	0.028	0.0046	0.222	0.0370	0.139	0.0232	0.056	0.0093
8. Temperature Tolerance	0.111	0.111	0.0123	0.028	0.0031	0.0	0.0	0.139	0.0154	0.056	0.0062	0.194	0.0215	0.167	0.0185	0.083	0.0092	0.222	0.0246
9. Post Failure Operation	0.111	0.028	<u>0.0031</u>	0.111	<u>0.0123</u>	0.0	<u>0.0</u>	0.056	<u>0.0062</u>	0.139	<u>0.0154</u>	0.167	<u>0.0185</u>	0.111	<u>0.0123</u>	0.167	<u>0.0185</u>	0.222	<u>0.0246</u>
		0.0970		0.0738		0.0124		0.1116		0.1070		0.1377		0.1316		0.1271		0.1778	