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[54] **FACE SEAL WITH INCREASED TORQUE TRANSFER CAPACITY**

[75] Inventors: **Steve L. Arianoutsos**, Peoria Heights; **Alan P. Dremann**, Chillicothe; **Jerry A. Metz**, Morton; **Harry B. Newman**, Washington, all of Ill.

[73] Assignee: **Caterpillar Inc.**, Peoria, Ill.

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[51] **Int. Cl.**^{5} **F16J 15/34**

[52] **U.S. Cl.** .... **277/92**; 277/81 R

[58] **Field of Search** 277/92, 84, 37, 136, 277/165; 285/917, 918

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Primary Examiner—Bernarr E. Gregory

Attorney, Agent, or Firm—O. Gordon Pence

[57] **ABSTRACT**

A shear loaded face seal having means for increasing the torque transfer capacity of the shear loaded face seal between its support ring and its load ring along their mating interface is disclosed. In such a shear loaded face seal, the load ring is of a deformable rubber and the support ring is rigid. A varying torque is transmitted through the shear loaded face seal along the interface between the load ring and the seal ring. The torque transfer means comprises a surface texture on the mating surfaces of the support ring that has a configuration sufficient to generate a projected area that is at least five per cent of the surface area of a non-textured surface of equivalent dimensions. The surface texture is provided by a plurality of uniformly spaced micro splines that are oriented normal to the direction of the torque.

13 Claims, 2 Drawing Sheets

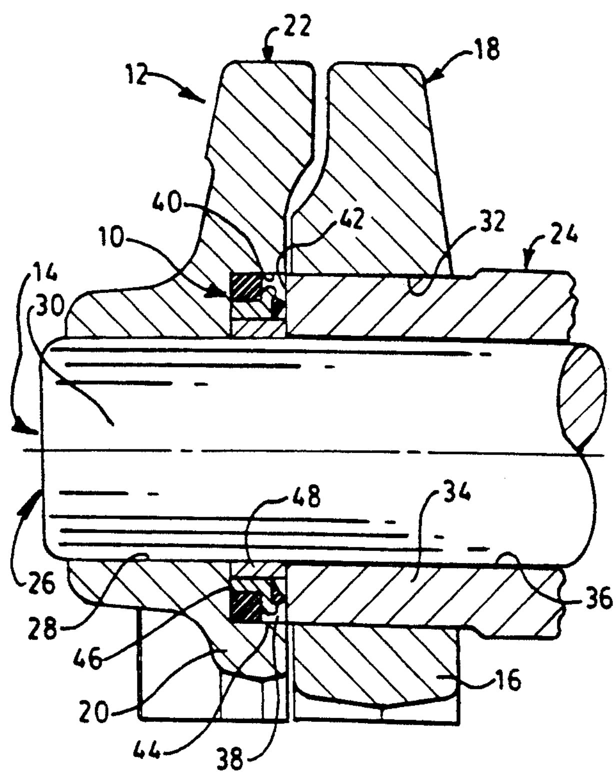

Fig. 1.

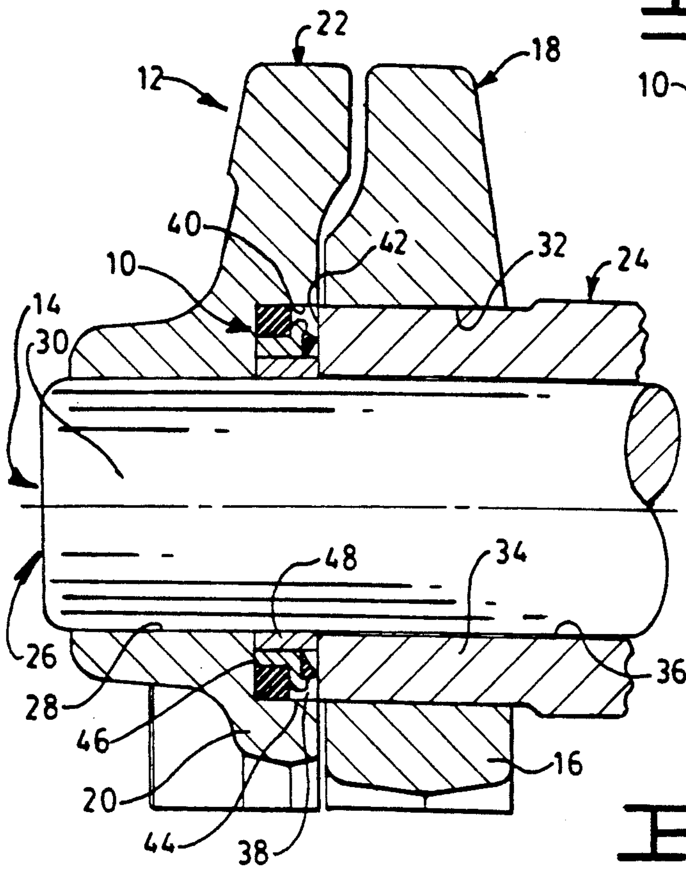

Fig. 2.

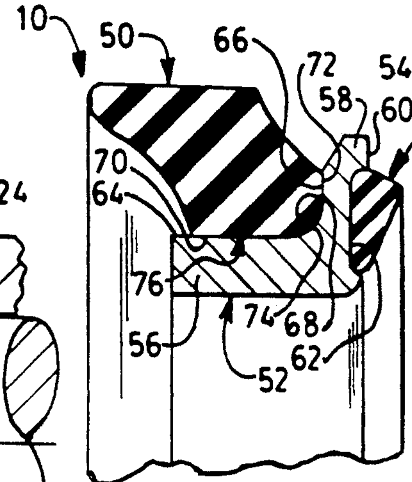

Fig. 3.

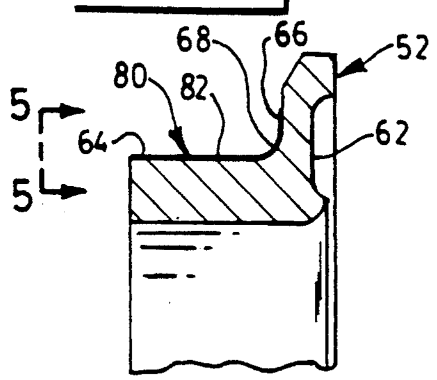

Fig. 4.

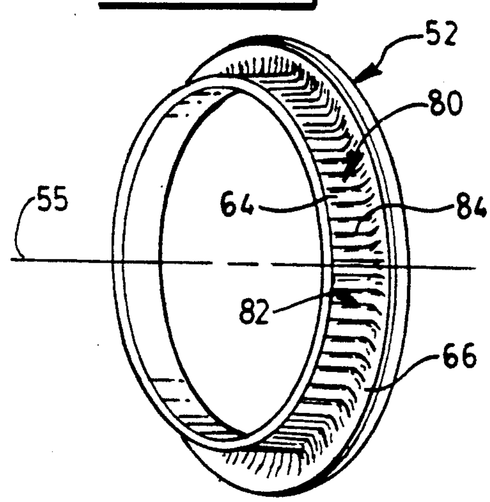

Fig. 5

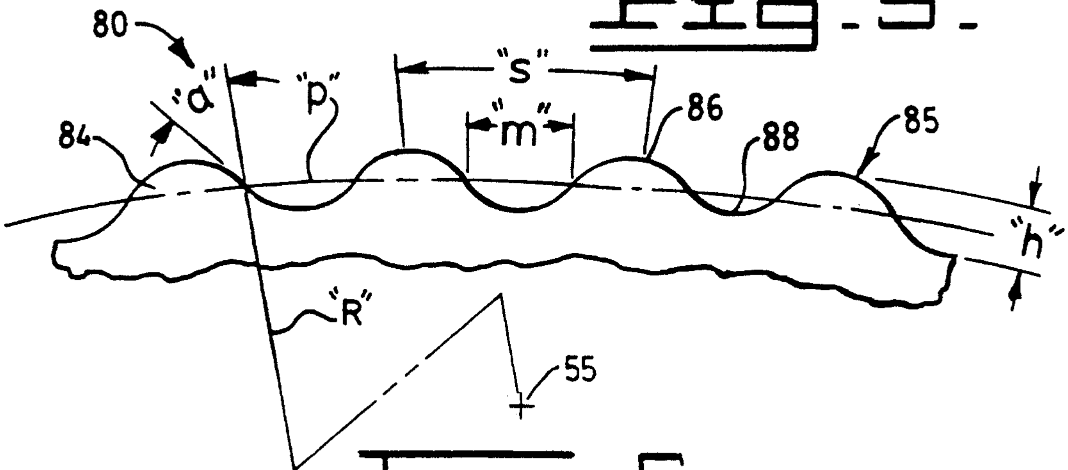

Fig. 6

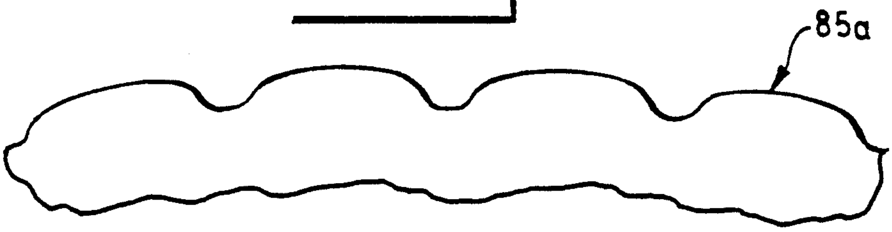

Fig. 7

Fig. 8

FACE SEAL WITH INCREASED TORQUE TRANSFER CAPACITY

DESCRIPTION

1. Technical Field

This invention relates generally to a shear loaded face seal for use in severe applications such as in the hinge joints of earthmoving vehicles and the like and, more particularly, to an improved seal with greater torque transfer capacity.

2. Background Art

Shear loaded face seals used in the hinge joints of earthmoving vehicles are exposed to the extreme conditions imposed by the elements. In hinge joint applications such as the joints of endless track chain used to propel track-type tractors or the loader linkage joints of loader-type vehicles, shear loaded face seals must be capable of sealing lubricant in the joint and sealing moisture, dirt and other abrasive matter out of the joint. Shear loaded face seals presently used in track are of the type disclosed in U.S. Pat. No. 4,195,852, issued Apr. 1, 1980 to Robert D. Roley et al., or in U.S. Pat. No. 4,262,914, issued Apr. 21, 1981 to Robert D. Roley, both of which are assigned to the assignee hereof. In a typical track joint application, the shear loaded face seal is located in a seal cavity disposed about the track pin. The cavity is typically provided by a counterbore in an outboard end collar of one of the links of the track chain and the adjacent end of a track bushing secured to an adjoining link of such chain. During track articulation, the outboard end collar rotates relative to the track bushing. The typical shear loaded face seal is a three part seal comprising a rubber load ring, a rigid support ring and a plastic seal lip. In practice, the seal lip is permanently affixed to the support ring so that these two components form a unitary member. Such unitary member is referred to herein as the "seal ring". The seal ring is placed against the radial end face of the bushing where dynamic sealing is designed to occur. The rubber load ring provides an axial force for urging a seal ring into sealing engagement against such end face. While rotational movement between the seal ring and the bushing end face is a design requirement, no such movement is intended, nor desired, between either the seal (as a unit) and link end collar or between the load ring and the seal ring. However, things do not always work as intended. In fact, slippage between the load ring and seal ring has become a problem in prior track seals. This slippage or spinning can cause an extreme amount of wear on the load ring in particular, which has resulted in early seal failure.

DISCLOSURE OF THE INVENTION

In accordance with one aspect to the present invention, means are provided for increasing the torque transfer capacity between the load ring and support ring of a shear loaded face seal along a mating interface through which a varying torque is transmitted. In such a shear loaded face seal, the load ring is of a deformable rubber and the support ring is rigid. The torque transfer means comprises a surface texture on the mating surfaces of the support ring that has a configuration sufficient to generate a projected area that is at least five per cent of the surface area of a non-textured surface of equivalent dimensions.

In another aspect of the present invention, the surface texture is provided by a plurality of uniformly spaced

micro splines that are oriented normal to the direction of the torque.

BRIEF DESCRIPTION OF THE DRAWINGS

FIG. **1** is fragmentary cross-sectional view of a track joint illustrating a shear loaded face seal embodying the principles of the present invention.

FIG. **2** is an enlarged fragmentary cross-sectional view of the shear loaded face seal shown in FIG. **1** by itself and in a free or unloaded state.

FIG. **3** is a view similar to FIG. **2**, but showing the support ring of the seal of FIG. **2** by itself.

FIG. **4** is an isometric view of the entire support ring shown in FIG. **3**. and showing the torque increasing splines of the present invention illustrated thereon.

FIG. **5** is a greatly magnified end view of the support ring taken generally along line **5**—**5** of FIG. **3** to diagrammatically illustrate a preferred embodiment of the splines of the present invention.

FIG. **6** is a diagrammatic representation similar to FIG. **5**, but of an actual trace of the splines constructed in accordance with the present invention.

FIGS. **7** and **8** are greatly enlarged profiles of spline geometries, similar to FIG. **5**, diagrammatically illustrating acceptable limits that the splines of the present invention may have.

BEST MODE FOR CARRYING OUT THE INVENTION

A shear loaded face seal embodying the principles of the present invention is generally depicted by reference numeral **10**. In FIG. **1**, shear loaded face seal **10** is shown, for illustrative purposes only, in an endless track chain **12** for a track type vehicle (not shown). Such track chain **12** includes a plurality of track joints, one of which is partially depicted at **14** in FIG. **1**. In the partial depiction of FIG. **1**, the track joint **14** includes an inboard end collar **16** of a first link **18**, an outboard end collar **20** of a second link **22**, a bushing **24**, and a pin **26**. The outboard end collar **20** has a pin bore **28** into which an end portion **30** of the pin **26** is nonrotatably secured by means of a heavy press fit. The inboard end collar **16** has a bushing bore **32** into which an end portion **34** of the bushing **24** is nonrotatably secured by a similar heavy press fit. The bushing **24** has a pin bore **36** of a size to freely rotatably mount the pin **26** therewithin.

The seal **10** of the present invention is disposed about the pin **26** within a seal cavity **38** defined by a counterbore **40** and a radial end face **42** of the bushing **24**. The counterbore **40** is located at the inboard end of the pin bore **28** and has a cylindrical surface **44** and a radial surface **46**. The joint **14** further includes a thrust ring **48** which is disposed radially inboard of the seal **10** to set the minimum axial length of the seal cavity **38** between the radial surface **46** of the counterbore **40** and the bushing end face **42** and to prevent the seal **10** from being crushed during assembly of the joint **14** or during operation of the track chain **12**. It should be noted that the portion of the track joint **14** depicted in FIG. **1** is, in essence, only the left hand half of the track joint, it being understood that such joint the track joint, it being understood that such joint **14** also includes a right hand half which is not shown, but is the mirror image of the left hand half. It should also be understood that the construction of the track joint **14** shown and described herein is merely by way of illustration and that the present invention is not intended to be limited thereby, as the seal **10** hereinafter described may be equally employed in any oscillating hinge joint found on earthmoving vehicles.

As shown in FIG. **2**, the seal **10** includes three components of different materials. The first component is a load ring **50** which is made of a soft, deformable rubber having a hardness within a range of from 50 to 70 Shore A durometer, with a 60 Shore A durometer being preferred. Also, the load ring **50** is preferably constructed of an oil resistant synthetic rubber, such as a nitrile rubber or a rubber made of epichlorohydrin.

The second component is a support or stiffener ring **52**, which may be made of any rigid material having a minimum flexural modulus of one million p.s.i. Preferably, a hard reinforced thermoplastic, such as glass filled polycarbonate, is used to make the support ring **52**.

The third component is a seal lip **54** which is preferably of a polyurethane material.

The support ring **52** is disposed about a central axis **55** (FIG. **4**) and has a cylindrical portion **56** and a radial flange portion **58** extending outwardly from one end of the cylindrical portion **56**, providing the support ring **52** with a generally L-shaped cross sectional configuration. The flange portion **58** has an outer surface **60** which is preferably provided with a groove **62** for receiving the seal lip **54**. In practice, the seal lip **54** is preferably bonded to the flange portion **58** so that the support ring **52** and seal lip **54** become a unitary member, i.e., the seal ring referred to earlier. The support ring **52** also includes an outer cylindrical surface **64** on the cylindrical portion **56** and an inner radial surface **66** on the flange portion **58**. The outer cylindrical surface **64** and the radial surface **66** are joined by a radiused corner surface **68**.

The load ring **50** has an inner cylindrical surface **70** and a radial surface **72** which are joined by a rounded corner **74**. The inner cylindrical surface **70**, the radial surface **72**, and the rounded corner **74** of the load ring **50** mate with and contact the corresponding outer cylindrical surface **64**, the radial surface **66**, and the radiused corner surface **68**, respectively, of the support ring **52** along an interface **76** therebetween when the load ring is mounted upon the support ring **52**.

Of principle importance to the present invention is means **80** for increasing the torque transfer capacity between the support ring **52** and the load ring **50** along their interface **76**. Means **80** comprises a surface texture **82** which is applied to the mating surfaces, i.e., outer surface **64**, radial surface **66**, and radiused corner surface **68**, of the support ring **52**. The textured surface **82** is of a configuration sufficient to generate a projected area that is at least 5% of the surface area of a non-textured or substantially smooth surface of equivalent dimensions. Preferably, such projected area is about 12% of the surface area of a smooth surface. As best shown in FIGS. **4** and **5**, the surface texture **82** preferably comprises a plurality of uniformly spaced, micro splines **84**. As used herein, the term "projected area" refers to a summation of the areas of each individual surface texture element (i.e., spline **84**) that each such element would project onto a longitudinal plane (a plane in which the central axis **55** lies) which extends through such element. In the case of splines, the projected area would be an area defined by the length of such spline multiplied by the height thereof. Splines **84** are oriented axially on the cylindrical outer surface **64** and continue around the radius corner surface **68** onto the radial surface **66**, where the splines are oriented radially. To optimize the desired torque transfer capacity of the splines **84**, it has been found that the splines must have a predetermined profile **85** that falls within certain dimensional and geometric parameters. As best shown in FIG. **5**, the profile **85** undulates between peaks **86** and valleys **88**. The profile **85** is provided with a peak to peak spacing "s" of less than 2 mm and a peak to valley height "h" of less than 0.5 mm. Preferably, the profile of the splines have a peak to peak spacing of about 0.5 mm and a peak to valley height within a range of from 0.048 to 0.084 mm. The profile of the splines **84** also undulates about a median pitch circle "p" located midway between the peaks and valleys. The profile has a circumferential spacing "m" between adjacent splines at the median pitch circle of between 2.5 to 5.5 times the peak to valley height "h" of such splines. In addition, no portion of the profile is oriented at an angle "a" of less than 25° from a radial line "r" extending through such portion of the profile.

Industrial Applicability

The shear loaded face seal **10** constructed in accordance with the teachings of the present invention advantageously provides the seal with sufficient torque transfer capacity to prevent slippage between the support ring **52** and the load ring **50**. In order to prevent such slippage, the torque capacity, or "driving torque", between the load ring **50** and the support ring **52** must always equal or exceed the torque generated between the seal lip **54** and the bushing end face **42** during sealing, referred to herein as the "sealing torque".

In the past, prior seals have relied on friction alone between their support rings having a generally smooth surface and their load rings to generate the driving torque necessary to handle the amount of sealing torque generated. It has been found, however, that the amount of sealing torque varies significantly in use and is dependent on several factors. Such factors include: the amount of face load exerted on the seal lip by the load ring; static and kinetic friction between the seal lip and the end face of the bushing; static adhesion between the seal lip and bushing end face; and inertia. The face load on the seal lip of shear loaded face seals is purposely relatively high, but can vary in use within a normal force range from a minimum unit load of 40 pounds per inch (7 kN/m) of seal lip circumference to a maximum unit load of 80 pounds/inch (14 kN/m). The minimum force is needed to ensure good sealing of the seal lip against the seal face. The amount of face load influences, to a large extent, the other factors noted above.

Independent external factors can also have a significant affect on the amount of sealing torque generated. These external factors may include, but are not intended to be limited to, mud packing around the seal lip between the support ring and the bushing end face and/or freezing of mud or liquid slurry therebetween.

The combination of the factors noted herein generate a substantial and varying amount of sealing torque on the seal lip. This sealing torque is transmitted from the support ring into the load ring through the interface along the mating contacting surfaces therebetween. In order to accomplish this torque without slippage, the driving torque along the interface must equal or exceed the sealing torque, as earlier noted.

The amount of driving torque is also influenced by the amount of face load on the seal, but not by the other factors noted above that affect sealing torque. Additionally, a change in face load has a greater affect on driving torque than on sealing torque, principally due to larger area of contact that the load ring has with the support ring than the seal lip has with the bushing end face. In other words, the amount of friction between the load ring and support ring decreases more rapidly than the amount of friction between the seal lip and the bushing end face per unit of decrease in face load. Another factor that can adversely affect driving torque is the presence of lubricant along the interface between the load ring and the support ring, which reduces the amount of friction between these two components. Under appropriate conditions then, the sealing torque of prior seals can exceed driving torque, resulting in slippage between the load ring and support ring.

The construction of the present invention advantageously overcomes this slippage problem by the incorporation of means **80** for increasing the torque transfer capacity of the seal **10** between the load ring **50** and the support ring **52**. Means **80** comprises providing a predetermined surface texture **82** on the mating surfaces **64,66,68** of the support ring **52** substantially along the interface **76** with the load ring **50**. The surface texture **82** is of a configuration sufficient to generate a projected area that is at least five per cent, and preferably twelve percent, of the surface area of an non-textured surface, i.e., a substantially smooth surface of the type found on prior support rings, of equivalent dimensions. The increased amount of surface area is effective in increasing the amount of friction between the load ring **50** and the support ring **52**.

Preferably, the surface texture **82** is created by means of large number of uniformly spaced micro splines **84** that are oriented normal to the direction of the driving torque. As used herein, the phrase "oriented normal to the direction of the driving torque" means an orientation that is in or generally in the direction of a longitudinal plane in which the central axis **55** of the support ring **52** lies. In other words, the splines are oriented axially if on an axially extending portion of the mating surface, as is the cylindrical surface **64**, and oriented radially if on a radially disposed portion of the mating surface, as is the radial surface **66**. As described earlier, the splines **84** have a predetermined profile **85** selected to optimize their torque transmission capabilities. The splines **84** not only generate greater friction through an increase in the surface area, as noted above, they also increase torque transfer capacity mechanically by providing a physical locking mechanism. This is accomplished by causing the load ring **50** to deform or work in order for it to slip on the support ring **52**. For slippage to occur, not only must friction be overcome, but energy must be expended to deform the load ring **50** over the splines **84**, thereby increasing the driving torque capacity. In effect, the load ring **50** develops mating notches to "lock" onto the splines **84**. As a result, the driving torque exerts a shearing force on the load ring **50** along the interface **76**, again in a direction normal to the driving torque. To accomplish this, the load ring **50** must be sufficiently soft and the profile **85** of the splines **84** must be such to allow the load ring **50** of such hardness to substantially conform to the undulations of the profile **85**.

Variations of size, number, shape and spacing of the splines **84** have been investigated to arrive at the parameters described herein that are needed to optimize the torque transfer capacity of the present invention. For instance, it was found that a large number of very small splines **84** performed substantially better than a small number of large splines. While torque capacity is

deemed to be proportional to the height of the splines **84**, the taller the splines become, the further apart they must be to ensure that the valleys are filled by the load ring **50**. In addition, large splines are not desirable because such large splines can unduly stress the rubber of the load ring **50**, which may lead to failure of the load ring under the high loads exerted upon it. Likewise, it has been found that the slope of the sides of the spline profile should not be less than 25 degrees from a radial line "r" extending through the slope as depicted in FIG. **5**. This side slope greatly facilitates the ability of the load ring **50** to conform to and substantially fully contact the entire profile **85** of the splines **84**.

Any suitable manufacturing process may be used to create the splines **84** in the support ring **52**. However as the support ring **52** is preferably made by molding, the splines **84** may be economically produced during the molding process of the support ring **52** by chemically etching the mold pattern used to produce the support ring **52**.

The particular shape of the splines **84** is not critical as any spline shape is satisfactory as long as it falls within the designated limits and the load ring **50** is capable of conforming to the peaks **86** and valleys **88** of the splines **84**. The spline profile **85** illustrated in FIG. **5**, for instance, diagrammatically illustrates a profile having a sinusoidal wave pattern that falls within the broader desired dimensional and geometric limits set forth herein. The spline profile **85*a*** illustrated in FIG. **6** diagrammatically depicts an actual trace made of the splines on an experimental support ring **52** that satisfactorily increased torque capacity by about 150 inch-pounds. FIGS. **7** and **8**, on the other hand, diagrammatically illustrate spline profiles **85'**, **85''** having the preferred maximum height and preferred minimum height limits, respectively, set forth herein that would be acceptable for splines produced during the molding process of the support ring **52**, as heretofore discussed.

Other aspects, objects and advantages of the present invention can be obtained from a study of the drawings, the disclosure and the appended claims.

We claim:

1. In a shear loaded face seal having a deformable rubber load ring and a rigid support ring, said load ring and support ring having mating surfaces contacting each other along an interface through which a varying torque is transmitted, the improvement comprising:

means for increasing the torque transfer capacity between the support ring and the load ring along said interface, said means comprising a surface texture on the mating surfaces of said support ring having a configuration sufficient to generate a projected area that is at least five percent of the surface area of a non-textured surface of equivalent dimensions.

2. The shear loaded face seal of claim **1**, wherein said load ring has a hardness substantially in a range of from 50 to 70 shore A durometer and said textured surface has a configuration sufficient to allow the load ring of such hardness to substantially conform to the shape of said textured surface.

3. The shear loaded face seal of claim **2**, wherein said textured surface comprises a plurality of uniformly spaced micro splines, said splines being oriented normal to the direction of said torque.

4. The shear loaded face seal of claim **3**, wherein said splines have a predetermined profile that undulates between peaks and valleys, said profile having a peak to peak spacing of less than 2 mm and a peak to valley height of less than 0.5 mm.

5. The shear loaded face seal of claim **4**, wherein the profile of said splines undulates about a median pitch circle between said peaks and valleys and has a circumferential spacing midway between adjacent splines of between 2.5 to 5.5 times the peak to valley height of said splines measured at the median pitch circle.

6. The shear loaded face seal of claim **5**, wherein no portion of the profile of said splines is oriented at an angle of less than about 25 degrees from a radial line extending through said portion.

7. The shear loaded face seal of claim **3**, wherein the profile of said splines has a peak to peak spacing of about 0.5 mm and a peak to valley height within a range of from 0.048 to 0.084 mm.

8. In a shear loaded face seal having a deformable rubber load ring and a rigid support ring, said load ring and support ring having mating surfaces contacting along an interface through which a varying torque is transmitted, the improvement comprising:

means for providing a surface texture on the mating surfaces of said support ring, said surface texture being of a configuration sufficient to generate a projected area that is at least five percent of the surface area of a non-textured surface.

9. The shear loaded face seal of claim **8**, wherein said means includes a plurality of uniformly spaced micro splines, said splines being oriented normal to the direction of said torque.

10. The shear loaded face seal of claim **9**, wherein said splines have a circumferential spacing of less than 2 mm and a peak to valley height of less than 0.5 mm.

11. The shear loaded face seal of claim **9**, wherein said splines have a circumferential spacing of about 0.5 mm and a peak to valley height within a range of from 0.048 to 0.084 mm.

12. The shear loaded face seal of claim **11** wherein said splines undulate about a median pitch circle between peaks and valleys and have a circumferential spacing between adjacent splines of between 2.5 to 5.5 times the peak to valley height of said splines at the median pitch circle.

13. The shear loaded face seal of claim **12** wherein no portion of said splines is oriented at an angle of less than about 25 degrees from a radial line extending through said portion.

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