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(54) **SIDELOAD VANES FOR FLUID PUMP**

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(22) Filed: **Jul. 13, 2006**

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(51) **Int. Cl.**
F04D 17/12 (2006.01)

(52) **U.S. Cl.** **415/195**; 415/198.1

(58) **Field of Classification Search** 416/203; 415/172.1, 224, 198.1, 195, 104
See application file for complete search history.

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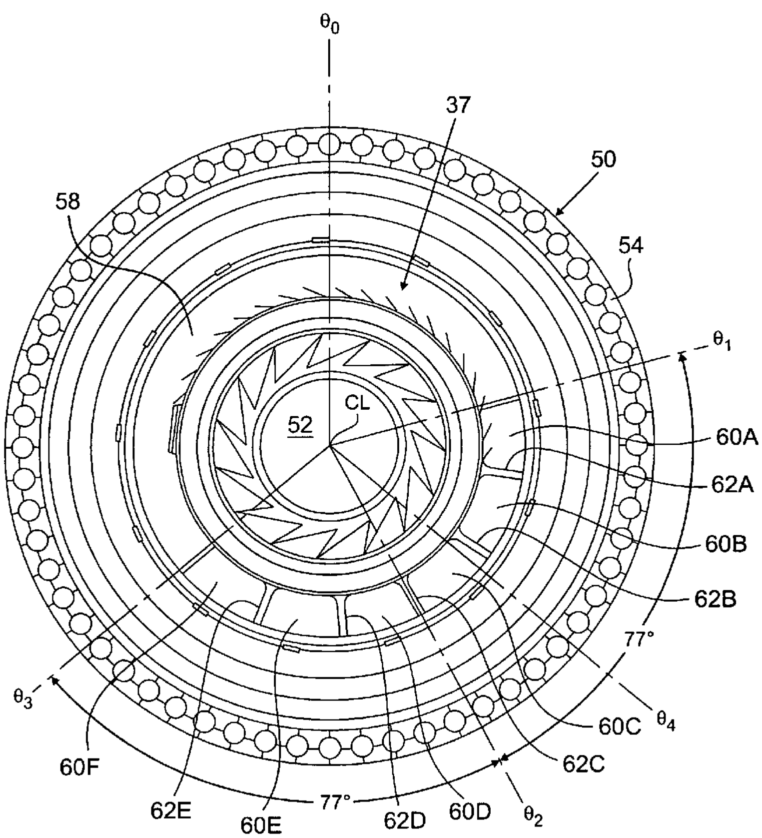
(57) **ABSTRACT**

A fluid pump assembly includes a rotatable component that can be rotated about an axis and a static vane assembly located adjacent to the rotatable component. The static vane assembly includes a circumferential surface axially spaced from the rotatable component, and one or more vanes extending from the circumferential surface toward the rotatable component. The one or more vanes are configured to produce a radial load on the rotatable component when the rotatable component is rotating about the axis and a fluid is present between the static vane assembly and the rotatable component.

19 Claims, 5 Drawing Sheets



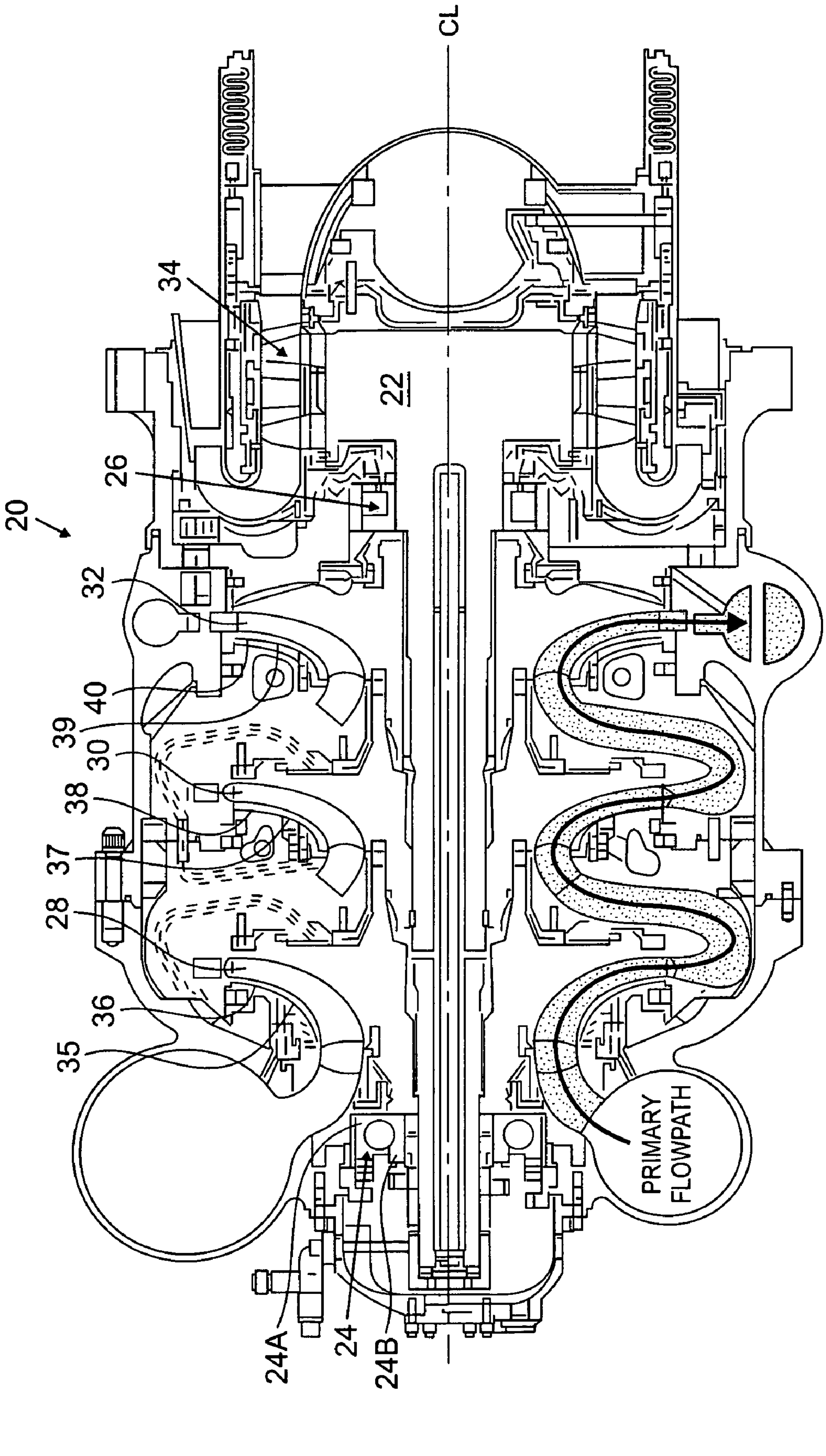
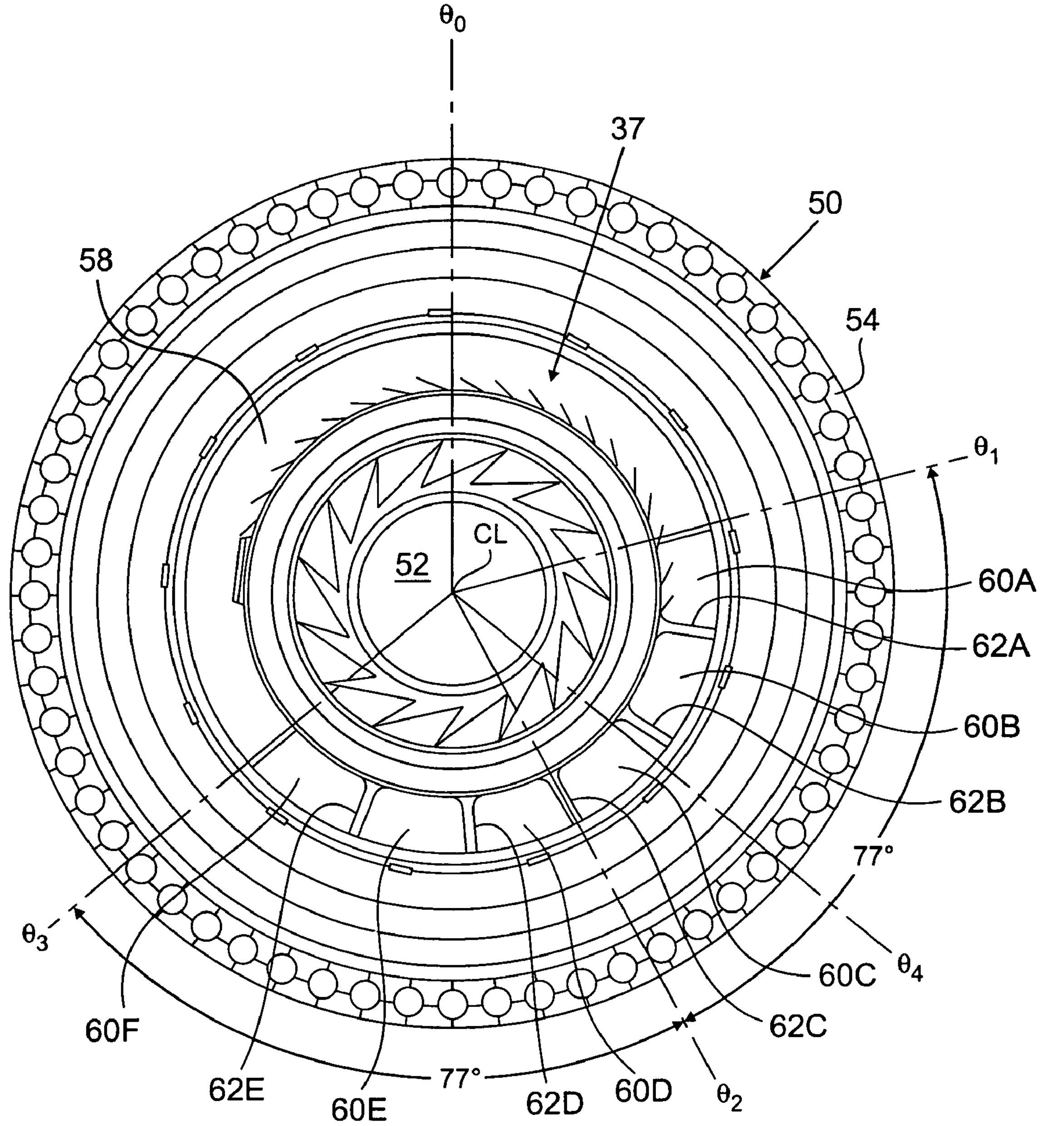

CL
34
22
26
20
32
40
39
30
38
37
28
36
35
PRIMARY FLOWPATH
24A
24
24B

FIG. 1



θ_0
37
50
58
54
θ_1
CL
52
60A
62A
60B
62B
77°
θ_3
θ_4
60C
62C
60F
θ_2
62E
77°
60E
62D
60D

FIG. 2

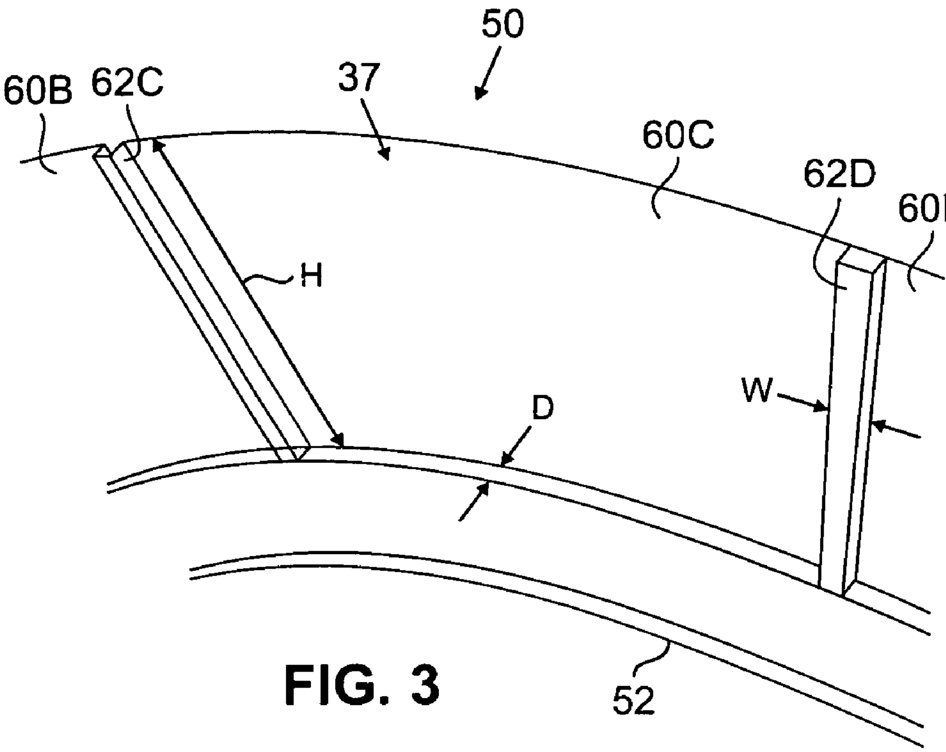

50
60B
62C
37
60C
62D
60D
H
W
D
52

FIG. 3

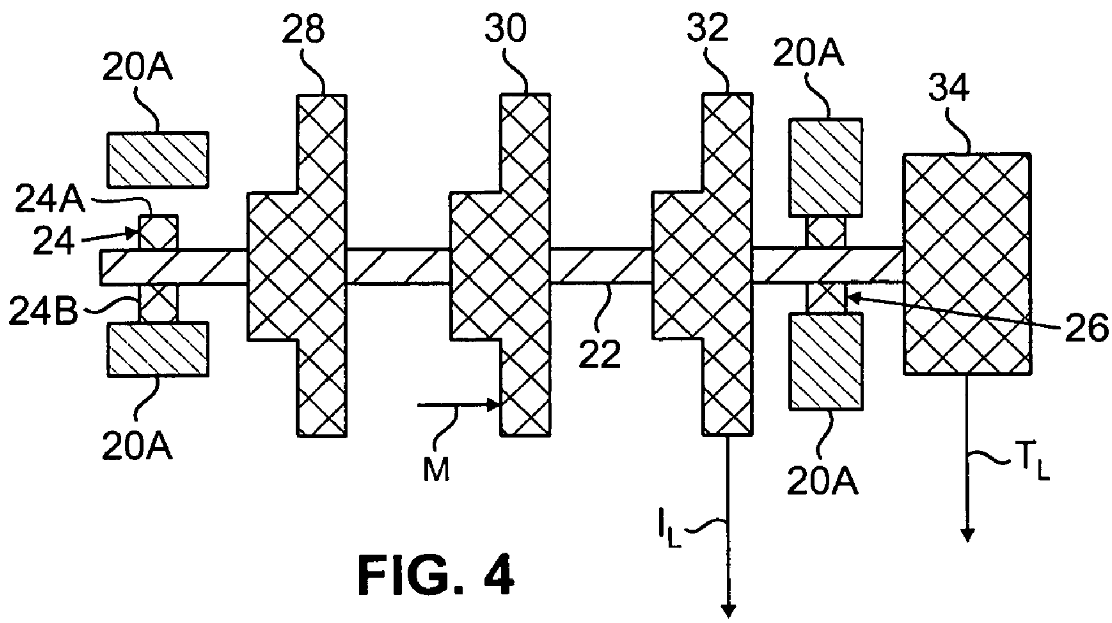

28
30
32
20A
34
20A
24A
24
24B
20A
22
26
M
20A
T_L
I_L

FIG. 4

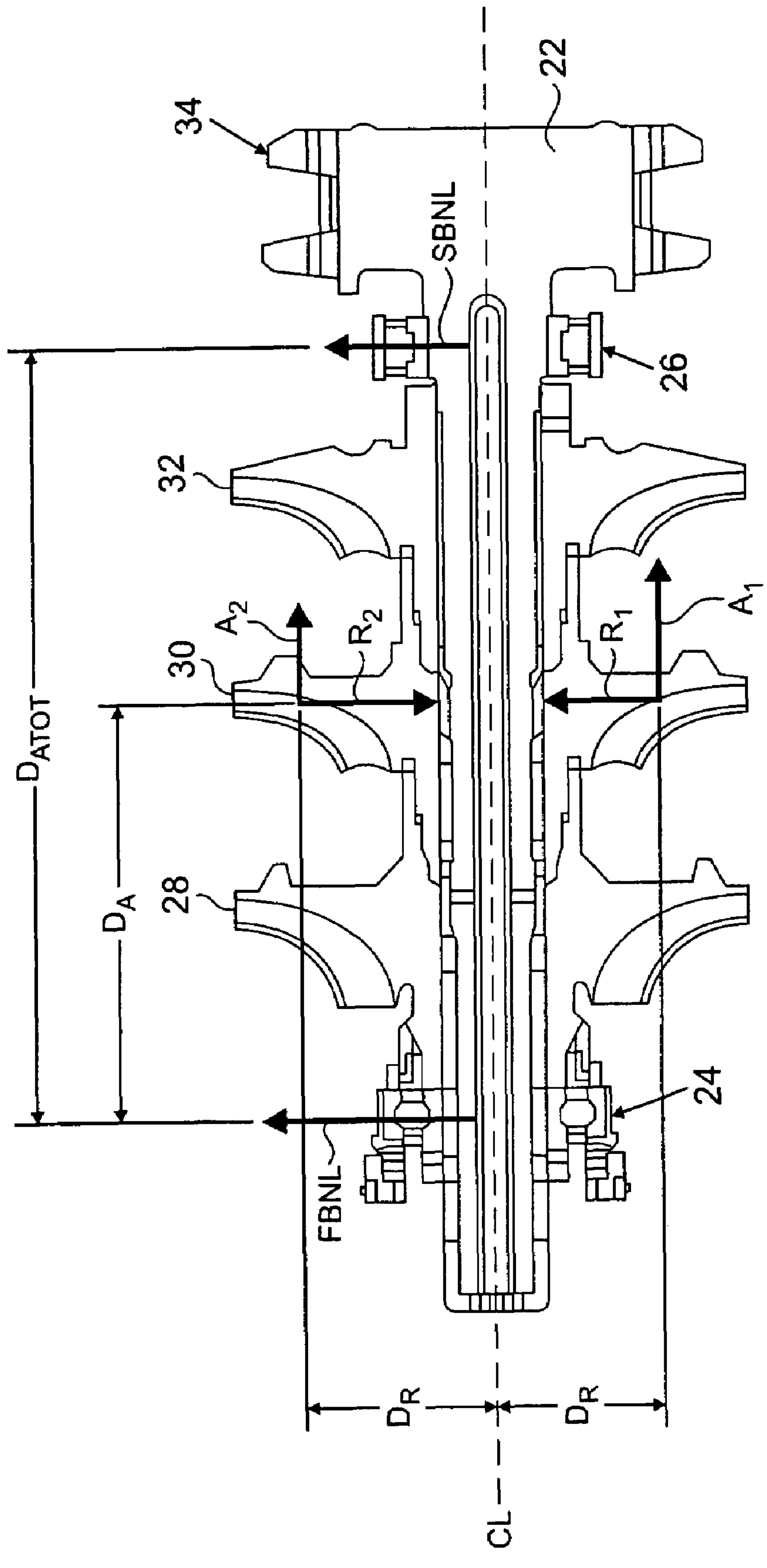


FIG. 5

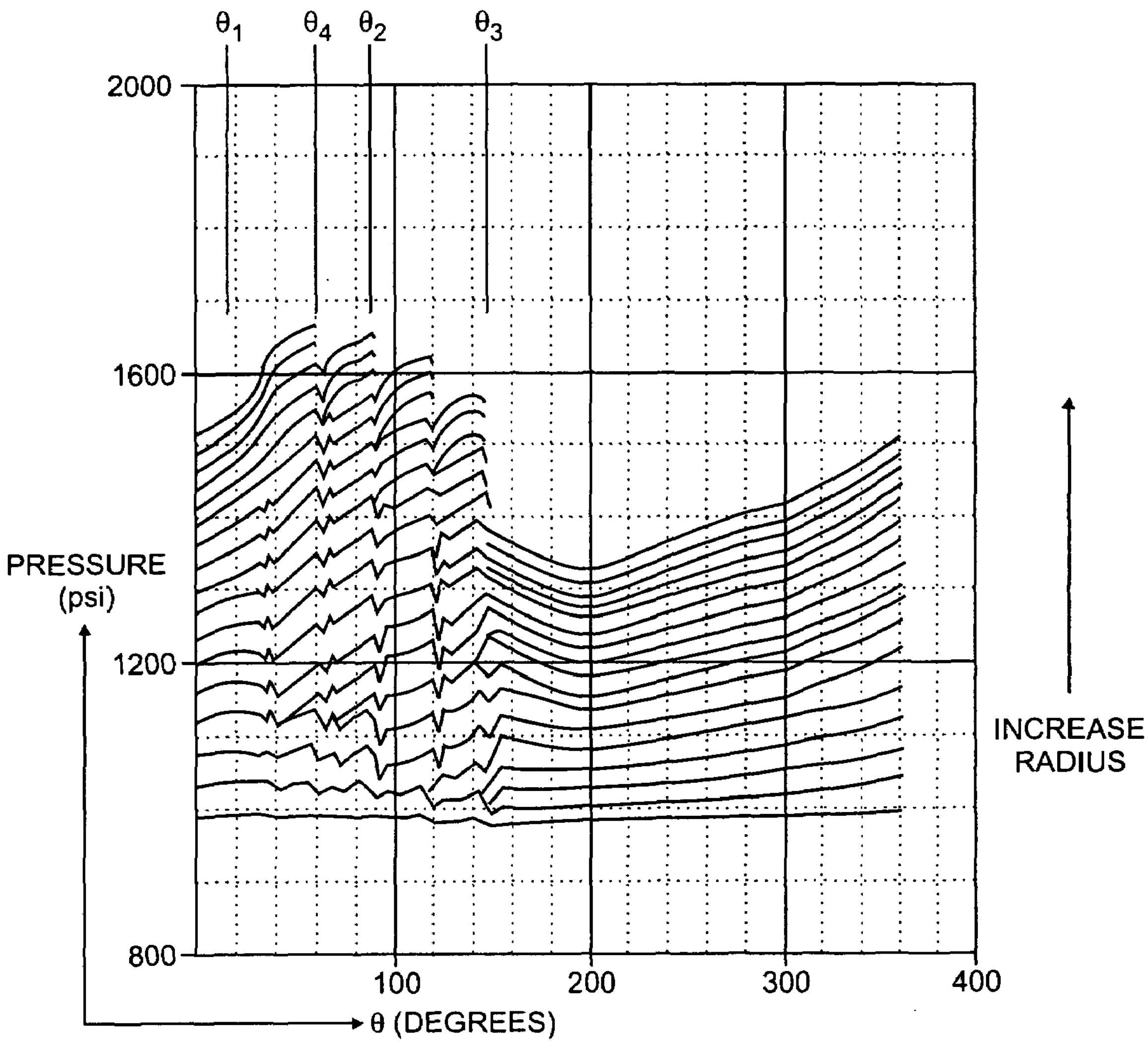

θ1
θ4
θ2
θ3
2000
1600
1200
800
PRESSURE
(psi)
100
200
300
400
θ (DEGREES)
INCREASE
RADIUS

FIG. 6

SIDELOAD VANES FOR FLUID PUMP

STATEMENT OF GOVERNMENT INTEREST

The present invention was made, in part, with government funding under NASA Contract No. NAS8-36801. The U.S. Government has certain rights in this invention.

BACKGROUND OF THE INVENTION

The present invention relates to vane assemblies suitable for use in fluid pumps, and more particularly to static vane assemblies for producing radial loads on turbopump components.

Rocket engines can utilize turbopumps to deliver propellants to an injector assembly in the combustion chamber. Such turbopumps have rotors that rotate as the turbopump operates, and impellers that rotate as part of the rotor to increase the pressure of propellants or propellant mixtures. It is desired to obtain a low, steady synchronous vibration response during turbopump operation. However, for a variety of reasons, a particular turbopump may produce an undesired sub-synchronous response. Sub-synchronous vibration responses can be caused, at least in part, by insufficient radial loading on a given bearing set of the turbopump.

Undesired asynchronous vibration response issues could be addressed in a number of ways. However, many potential solutions are overly complex, insufficiently robust, or are otherwise undesirable, for instance, resulting in an unsatisfactory turbopump performance loss. As one example, the rotor bearings could be redesigned, but redesigns of rotor bearings are difficult and complex. Moreover, flow inlets and outlets create load vectors that could be optimized relative to undesired vibrations, but optimal inlet and outlet flow paths may undesirably increase engine size and/or mass and may provide optimal design "windows" (i.e., tolerances on desired vibration characteristics) that are too small to be practical.

BRIEF SUMMARY OF THE INVENTION

A turbopump assembly according to the present invention includes a rotatable component that can be rotated about an axis and a static vane assembly located adjacent to the rotatable component. The static vane assembly includes a circumferential surface axially spaced from the rotatable component, and one or more vanes extending from the circumferential surface toward the rotatable component. The one or more vanes are configured to produce a radial load on the rotatable component when the rotatable component is rotating about the axis and a fluid is present between the static vane assembly and the rotatable component.

BRIEF DESCRIPTION OF THE DRAWINGS

FIG. **1** is a schematic cross-sectional view of a turbopump.

FIG. **2** is front view of a sideload vane assembly according to the present invention.

FIG. **3** is a perspective view of a portion of the sideload vane assembly of FIG. **2**.

FIG. **4** is a simplified schematic cross-sectional view of a portion of the turbopump of FIG. **1** that is radially loaded.

FIG. **5** is a schematic cross-sectional view of a portion of the turbopump of FIG. **1**.

FIG. **6** is a graph of fluid pressure versus angular location calculated for a number of radial locations in a secondary flowpath of a turbopump.

DETAILED DESCRIPTION

The present invention provides an apparatus and method for reducing undesired vibration of components of a fluid pump. In particular, the present invention provides advantages in producing radial loading on bearing supports for pump rotors, which otherwise permit undesired vibrations in an unloaded condition. The present invention utilizes sideload vanes positioned adjacent to rotating members that work upon the fluid in the pump. The sideload vanes produce a non-uniform circumferential pressure field in a fluid in the pump, as fluid moves in a flowpath adjacent to the vanes. The non-uniform circumferential pressure field in turn, imparts radial loading to rotor bearings that otherwise would be substantially unloaded and prone to undesirable vibration issues.

FIG. **1** is a schematic cross-sectional view of a turbopump **20** that includes a rotor shaft **22** located at a centerline CL, a first bearing set **24**, a second bearing set **26**, and three impellers **28**, **30**, **32** (referred to as the first through third stage impellers, respectively). A turbine assembly **34** is mechanically connected to the rotor shaft **22**. The first bearing set **24** is a ball bearing set that includes an outer race **24**A and an inner race **24**B. The inner race **24**B rotates with the rotor shaft **22**, while the outer race **24**A is static. As used herein, the term "static" refers to being stationary relative to a pump mounting location, and applies even where the entire pump or turbopump has a mounting location in a moving vehicle (or on another movable object). The second bearing set **26** is a roller bearing set. The first and second bearing sets **24** and **26** support the rotating components of the turbopump **20** (see FIG. **5**) relative to the static components of the turbopump **20**. The impellers **28**, **30**, **32** and the rotor shaft **22** are rotating components when the turbopump **20** is operational. The impellers **28**, **30**, **32** all rotate together with the rotor shaft **22**, which is driven by rotation of the turbine assembly **34**. In operation, a fluid is pumped sequentially through the impellers **28**, **30**, **32**, which move and pressurize the fluid. The impellers **28**, **30**, **32** generally move the fluid through the turbopump **20** along a primary flowpath, a portion of which is indicated schematically in FIG. **1**. Those skilled in the art will recognize that the primary flowpath has a complex shape defined by the rotating impellers **28**, **30**, **32** and connecting passageways.

A sideload portion **35** of a first diffuser **36** is located adjacent to the first impeller **28**, a sideload portion **37** of a second diffuser **38** (also called the 1-2 diffuser) is located adjacent to the second impeller **30**, and a sideload portion **39** of a third diffuser **40** (also called the 2-3 diffuser) is located adjacent to the third impeller **32**. The diffusers **36**, **38**, **40** are static components located at a forward or upstream side of the respective adjacent impellers **28**, **30**, **32** (to the left of the impellers **28**, **30**, **32** as shown in FIG. **1**). A portion of a secondary flowpath is defined in a gap between the sideload portions of the diffusers and the adjacent impellers, for instance, between the sideload portion **37** of the second diffuser **38** and the second impeller **30**. The secondary flowpath corresponds to a fluid flow that is generally outside the primary flowpath that carries the majority of fluid through the turbopump **20**. In a conventional prior art turbopump, the secondary flowpaths between the sideload portions **35**, **37**, **39** of each of the diffusers **36**, **38**, **40** and the impellers **28**, **30**, **32** would be circumferentially uniform, a condition which would produce no net radial load on the rotor shaft **22** or first bearing set **24**.

The turbopump **20** includes numerous other components not specifically identified herein. Those skilled in the art will understand the basic operation of turbopumps. Therefore, further explanation here is unnecessary.

FIGS. **2** and **3** illustrate one embodiment of a sideload vane assembly **50**, which is positioned at the second diffuser **38** (see FIG. **1**). It should be recognized that the sideload vane assembly **50** could be positioned adjacent to any of the impellers **28**, **30**, **32** of the turbopump **20** in alternative embodiments. FIG. **2** is front (axial) view of the sideload vane assembly **50** (viewed from the second impeller **30** toward the first impeller **28**), and FIG. **3** is a perspective view of a portion of the sideload vane assembly **50** shown in FIG. **2**. Reference markers for angles Θ_0-Θ_3 are shown in FIG. **2** in order to better explain angular positioning of various features about the centerline CL.

The sideload vane assembly **50** is a static component that includes a central opening **52** for the rotor shaft **22** and flange **54** at the perimeter of the assembly having bolt holes for mounting the assembly **50** in the turbopump **20**. The assembly **50** can be made of a metallic material, such as aluminum. A sideload wall **37** is positioned (radially) between the central opening **52** and the flange **54**. The sideload wall **37** extends circumferentially about the entire assembly **50**, that is, the sideload wall **37** has an angular sweep of 360° about the centerline CL. The sideload wall **37** is radially positioned so as to align with one of the side of the diffusers **36**, **38** or **40** adjacent to one of the corresponding impellers **28**, **30** or **32**.

The sideload wall **37** includes a substantially smooth wall portion **58** and six pockets **60**A-**60**F. The pockets **60**A-**60**F form five vanes **62**A-**62**E at the circumferentially spaced edges thereof. As shown in FIG. **2**, the vanes **62**A-**62**E are located within a first angular region, which has an angular sweep of 154° about the centerline CL between angles Θ_1 and Θ_3. The vanes **62**A-**62**E are substantially equally angularly spaced within the first angular region. The substantially smooth wall portion **58** is located within a second angular region, which has an angular sweep of 205° about the centerline CL between angles Θ_3 and Θ_1. The first and second angular regions have a combined angular sweep of 360°, that is, together they extend circumferentially about the entire assembly **50**. It should be noted that in alternative embodiments, the number and arrangement of the vanes can vary. For example, more or fewer than six pockets can be formed. Moreover, the first angular region can have a greater or lesser angular sweep, and the position of the first angular region (i.e., the "rotational" position of the reference markers with respect to a pump mounting location) can vary. Furthermore, the vanes need not be equally angularly spaced.

In FIG. **3**, a portion of the sideload wall **37** is shown, including pockets **60**B-**60**D and vanes **62**C-**62**D. A number of reference dimensions are indicated in FIG. **3**, including a vane height H, a vane width W, and a pocket depth D. The dimensions H, W, and D can be adjusted for particular applications, to provide desired performance characteristics, including desired radial loading.

Each of the vanes **62**A-**62**E has a substantially rectangular shape, and the pockets **60**A-**60**F and vanes **62**A-**62**E can be formed by milling the sideload wall **37**. Use of rectangular vanes simplifies manufacture while still providing sufficient structural integrity. In alternative embodiments, the shape of the vanes can vary as desired.

In operation, as fluid is being pumped through the turbopump **20**, the sideload vane assembly **50** interacts with the fluid in the secondary flowpath (i.e., in the gap between the sideload vane assembly **50** and the adjacent second impeller **30**). The vanes **62**A-**62**E of the assembly **50** act like asymmetric swirl brakes and generate a non-uniform circumferential pressure field in fluid in the secondary flowpath. The non-uniform circumferential pressure field imparts a moment on the adjacent second impeller **30**, and that moment produces a radial force component in the second impeller **30** that, in turn, radially loads the rotor shaft **22** and the first bearing set **24**.

FIG. **4** is a simplified schematic cross-sectional view of a portion of the turbopump **20**, showing a net moment M on the second impeller **30** due to a non-uniform circumferential pressure field generated in conjunction with an adjacent sideload vane assembly **50** (not shown). The moment M is shown in FIG. **4** with a generally axial orientation at a location radially spaced from the rotor shaft **22** (and its centerline CL). The magnitude and location of the moment M will vary according to the characteristics of particular applications. As explained below, the moment M, in turn, produces radial loading in a first direction (at an angle Θ_4, not shown) on the rotor shaft **22** and the first bearing set **24** (and/or the second bearing set **26**) as force is transmitted through the impeller **30** and the rotor shaft **22**. As shown in FIG. **4**, the second bearing set **26** acts like as a fulcrum, while the first bearing set **24** has some freedom of radial movement with respect to a pump housing or ground **20**A. This is because the configuration of the turbopump **20** provides sufficient stiffness to the second bearing set **26** to keep it engaged. It should be recognized that the particular characteristics of the bearing sets **24** and **26** will vary depending on the particular configuration of the turbopump **20** and its housing **20**A.

A vector I_L represents natural radial loading of the third impeller **32**, and a vector T_L represents natural radial loading of the turbine assembly **34**. Vector I_L is oriented at about 0-50° with respect to a given angular reference point Θ_5 (not shown), and vector T_L is oriented at about 0° with respect to the reference point Θ_5. The vectors I_L and T_L arise due to the rotation of and interaction with fluids by the third impeller **32** and the turbine assembly **34**, and due to configurations of fluid inlets and outlets of the turbopump **20**. Vectors I_L and T_L establish a preferred direction of radial loading for the turbopump **20**, based on the natural characteristics of the turbopump **20**, that is, based on factors substantially independent from radial loading imparted by the sideload vane assembly **50**. The vectors I_L and T_L generally have small magnitudes that, alone, do not provide significant stiffness to the first bearing set **24**.

The sideload vane assembly **50** is configured such that the first direction of radial loading imparted by assembly **50** substantially aligns with the preferred direction of radial loading of the turbopump **20** (i.e., such that $\Theta_4 \approx \Theta_5$). Such alignment, although not strictly necessary, improves the effectiveness of the radial loading and reduces performance losses.

FIG. **5** is a free body diagram of the rotatable components of the turbopump **20**, showing the impellers **28**, **30**, **32**, the shaft **22**, and a portion of the turbine assembly **34** in a schematic cross-sectional form. A number of reference markings for vectors, distances, etc. are indicated in FIG. **5** to illustrate some of the parameters that influence loading on components of the turbopump **20** during operation. The definitions of these reference markings are given in Table 1.

TABLE 1

Reference Marking	Definition
A_1	Vector for axial load on the selected impeller associated with the first angular region (of the adjacent sideload vane assembly)

TABLE 1-continued

Reference Marking	Definition
A_2	Vector for axial load on the selected impeller associated with the second angular region (of the adjacent sideload vane assembly)
CL	Turbopump centerline axis aligned at the center of the rotor shaft
D_A	Axial distance between the selected impeller and the first bearing set (measured midpoint-to-midpoint)
D_{ATOT}	Axial distance between the first bearing set and the second bearing set (measured midpoint-to-midpoint)
D_R	Radial distance between the selected impeller and CL (measured midpoint to midpoint)
R_1	Vector for radial load on the selected impeller at the first angular region
R_2	Vector for radial load on the selected impeller at the second angular region, with vector R_2 being positioned 180° from vector R_1
FBNL	Vector for the net radial load on the first bearing set
SBNL	Vector for the net radial load on the second bearing set

It should be noted that although reference markings are shown in FIG. **5** primarily with respect to the second impeller **30**, similar parameters would exist for the other impellers **28**, **32** where sideload vane assemblies were located adjacent to the other impellers **28**, **32**.

The magnitude for the vector FBNL (i.e., the net radial load on the first bearing set **24**) is given by the following equation:

$$FBNL = \left(1 - \frac{D_A}{D_{ATOT}}\right)R_1 + \left(\frac{D_A}{D_{ATOT}} - 1\right)R_2 + (A_2 - A_1)\frac{D_R}{D_{ATOT}} \quad (1)$$

The vector FBNL for the sideload vane assembly **50** of FIG. **2** is directed at an angle designated as Θ_4, which can be determined empirically. The angle Θ_4 generally lies within the first angular region of the vane assembly **50**, and is generally offset from the symmetry line of the first angular region (i.e., angle Θ_2) in a direction opposite to the direction of rotation of the impellers **28**, **30**, **32** and rotor shaft **22**. The offset of angle Θ_4 from angle Θ_2 is due to the fluid dynamics within the pump.

The magnitude for the vector SBNL (i.e., the net radial load on the second bearing set **26**) is given by the following equation:

$$SBNL = \frac{D_R}{D_{ATOT}}(A_1 - A_2) + \frac{D_A}{D_{ATOT}}(R_1 - R_2) \quad (2)$$

The vector FBNL gives the anticipated radial loading on the first bearing set **24**, and the sideload vane assembly **50** can be configured such that the anticipated radial loading provides desired stiffness to maintain engagement of the first bearing set **24** (e.g., to maintain engagement of the first bearing set **24** with the housing **24**A). Equations (1) and (2), and the free body diagram in FIG. **5** help illustrate the relationship of the forces that produce radial loading on the first bearing set **24** due to the non-uniform circumferential pressure field created by the sideload vane assembly **50**.

EXAMPLE

A bench test experiment was performed on an embodiment of the vane assembly **50** like that described above. The turbopump **20** was run under normal operating conditions pumping water. The sideload vane assembly **50** had five vanes **62**A-**62**E and six pockets **60**A-**60**F, where the vane length L was 3.429 cm (1.35 inches), the vane width W was 0.635 cm (0.250 inches), the pocket depth D was 0.1524 cm (0.060 inches). The vanes **62**A-**62**E were equally circumferentially spaced within a first angular region having an angular sweep of about 154°.

FIG. **6** is a graph of fluid pressure versus angular location calculated for a number of radial locations in the secondary flowpath of the turbopump **20**. On the X-axis, Θ represents the angular location about the turbopump centerline CL in degrees (measured 0-360° from an arbitrarily selected reference point). On the Y-axis, pressure in pounds per square inch (psi) represents the measured static fluid pressure. A number of plots are shown on the graph of FIG. **6** each based on pressures at different radial locations from the centerline CL (with the greater radii corresponding to greater average pressures in the fluid).

The first angular region of the sideload vane assembly **50** corresponds approximately to values of Θ between Θ_1 and Θ_3 (inclusive of Θ_2), as shown in the graph of FIG. **6**. The second angular region of the sideload vane assembly **50** corresponds roughly to values of Θ between Θ_1 and Θ_3 (exclusive of Θ_2) on the graph. The graph shows acute pressure rises that generally correspond to when the fluid passed each of the vanes **62**A-**62**E in the first angular region, with the pressure rise being greatest at locations further from the centerline CL. At the smallest radii, nearest the centerline CL, the effects of the vanes **62**A-**62**E is less pronounced. However, because of radial fluid movement during operation of the turbopump **20**, causing fluid to move away from the vanes **62**A-**62**E, the number of acute rises in fluid pressure did not strictly correspond to the number of vanes (five). The angle of radial loading Θ_4 in the present example was approximately 60° with respect to the arbitrarily selected reference point shown on FIG. **6**. Θ_4 can be calculated according to the following equation, where P is the pressure, r is a variable for the radial location and Θ is the variable in the horizontal axis of FIG. **6**:

$$\Theta_4 = \frac{\int P*\Theta*d\Theta*dr}{\int \Theta*d\Theta*dr} \quad (3)$$

The magnitude of FBNL (i.e., the net radial load on the first bearing set **24**) was 185.519 kg (409 lbs.), and that value was obtained by integrating the area under the plots of the graph of FIG. **6**.

Although the present invention has been described with reference to preferred embodiments, workers skilled in the art will recognize that changes may be made in form and detail without departing from the spirit and scope of the invention. For instance, a sideload vane assembly according to the present invention can have a variety of vane and pocket configurations. Moreover, a fluid pump can utilize one or more sideload vane assemblies according to the present invention in a variety of locations. In addition, sideload vane assemblies according to the present invention can be used to reduce the net radial loads on components (e.g., bearings) of a fluid pump, as desired, by configuring the sideload vane assemblies to produce radial loads in opposition to existing radial loads.

What is claimed is:

1. A turbopump assembly comprising:

a rotor defining an axis of rotation;

an impeller assembly supported on the rotor for rotation therewith;

a case structure adjacent to the impeller assembly and having one or more vanes extending therefrom; and

a secondary flowpath for a fluid medium, the secondary flowpath defined between the impeller assembly and the case structure, wherein rotation of the impeller assembly generates a non-uniform circumferential pressure field in the secondary flowpath that produces radial loading on the rotor.

2. The turbopump assembly of claim **1**, wherein the one or more vanes are configured to generate the non-uniform circumferential pressure field such that the radial load on the rotor is produced in a direction aligned with a preferred radial direction of movement of the rotor.

3. The turbopump assembly of claim **1**, wherein one of the vanes is substantially rectangularly shaped.

4. The turbopump assembly of claim **1**, wherein the one or more vanes are arranged within a first region of the case structure, and wherein the one or more vanes are substantially equally circumferentially spaced within the first region.

5. The turbopump assembly of claim **4**, wherein the first angular region has an angular sweep of less than 180°.

6. The turbopump assembly of claim **4**, wherein the first angular region has a total angular sweep of about 154°.

7. The turbopump assembly of claim **1** and further comprising:

a rotor bearing for supporting the rotor, wherein the radial loading on the rotor in turn causes radial loading of the rotor against the rotor bearing.

8. The turbopump assembly of claim **1**, wherein the non-uniform circumferential pressure field in the secondary flowpath produces a substantially axial load on the rotor at a location radially spaced from the axis of rotation to produce a net moment.

9. A method of modifying a turbopump assembly to reduce vibrations, the turbopump assembly including a rotor defining an axis of rotation, an impeller subassembly and a static case, the method comprising:

identifying a preferred direction of movement of the rotor;

determining a non-uniform circumferential pressure field that can be formed in a secondary flowpath between the impeller subassembly and the static case to produce a radial load in the preferred direction of movement of the rotor; and

forming vane structures that extend from the case in a pattern that facilitates generation of the non-uniform circumferential pressure field.

10. The method of claim **9**, wherein the non-uniform circumferential pressure field is determined so as to produce a substantially axial load on the rotor at a location radially spaced from the axis of rotation to produce a net moment.

11. A fluid pump assembly comprising:

a rotatable component that can be rotated about an axis; and

a static vane assembly located adjacent to the rotatable component, the static vane assembly comprising:

a circumferential surface axially spaced from the rotatable component, wherein the circumferential surface comprises a first angular region and a second angular region, the first and second angular regions defined substantially perpendicular to the axis of the rotatable component and having a combined angular sweep totaling 360°, and wherein an angular sweep of the first angular region is less than 180°; and

a plurality of circumferentially spaced vanes extending from the circumferential surface toward the rotatable component, the plurality of vanes all located within the first angular region and configured to produce a radial load on the rotatable component when the rotatable component is rotating about the axis and a fluid is present between the static vane assembly and the rotatable component.

12. The fluid pump assembly of claim **11**, wherein the rotatable component is an impeller assembly mounted on a rotor.

13. The fluid pump assembly of claim **12**, wherein the plurality of vanes are configured to produce a radial load on the rotor in a first radial direction of movement of the rotor.

14. The fluid pump assembly of claim **1**, wherein the plurality of circumferentially spaced vanes are substantially equally angularly spaced within the first angular region.

15. The fluid pump assembly of claim **1**, wherein the static vane assembly comprises five vanes substantially equally angularly spaced with the first angular region.

16. The fluid pump assembly of claim **1**, wherein one of the vanes is substantially rectangularly shaped.

17. The fluid pump assembly of claim **1**, wherein the circumferential surface defines a case wall.

18. The fluid pump assembly of claim **1**, wherein the first angular region has a total angular sweep of about 154°.

19. The fluid pump assembly of claim **1**, wherein the plurality of vanes are configured to produce a produce a substantially axial load on the rotatable component at a location radially spaced from the axis to produce a net moment when the rotatable component is rotating about the axis and a fluid is present between the static vane assembly and the rotatable component.

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