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Jordan

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(54) **VARIABLE MARINE JET PROPULSION**

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B63H 11/00 (2006.01)

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(58) **Field of Classification Search** 440/38, 440/46, 47, 67; 416/43, 131; 114/151
See application file for complete search history.

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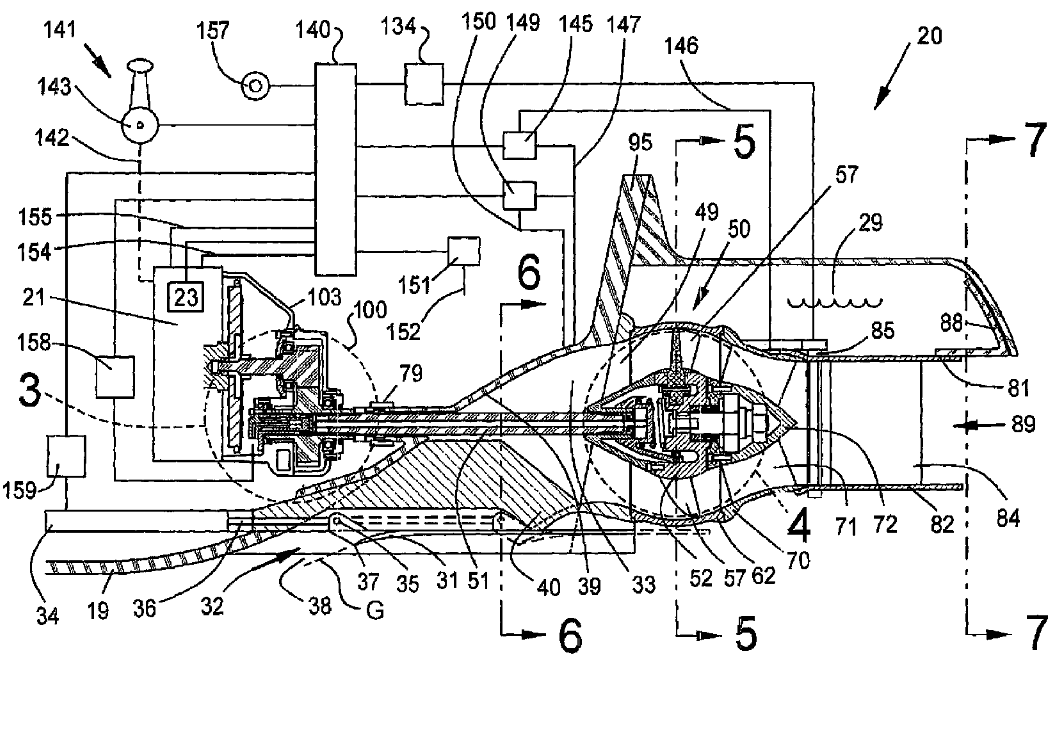
(57) **ABSTRACT**

A variable marine jet propulsion system incorporates a motor, a variable-pitch propeller pump in a spherical housing, a variable housing and a variable inlet duct, and a microcontroller. The pump, the nozzle and the inlet are controlled by the microcontroller, which is programmed to control the pump as a continuously variable power transmission for maintaining efficient motor operation, the nozzle for maintaining efficient pump operation, and the inlet for maintaining efficient recovery of the total dynamic head of the incoming water. The spherical pump housing maintains close fits to the propeller vane tips for more efficient operation at all pitches, including zero and reverse pitches. Zero pitch results in no effective pumping action, effectively a true neutral in fluid power transmission. Reverse pitch in combination with the large variable nozzle provides reverse flow and consequently reverse thrust, which eliminates the need for the "backing bucket".

21 Claims, 11 Drawing Sheets



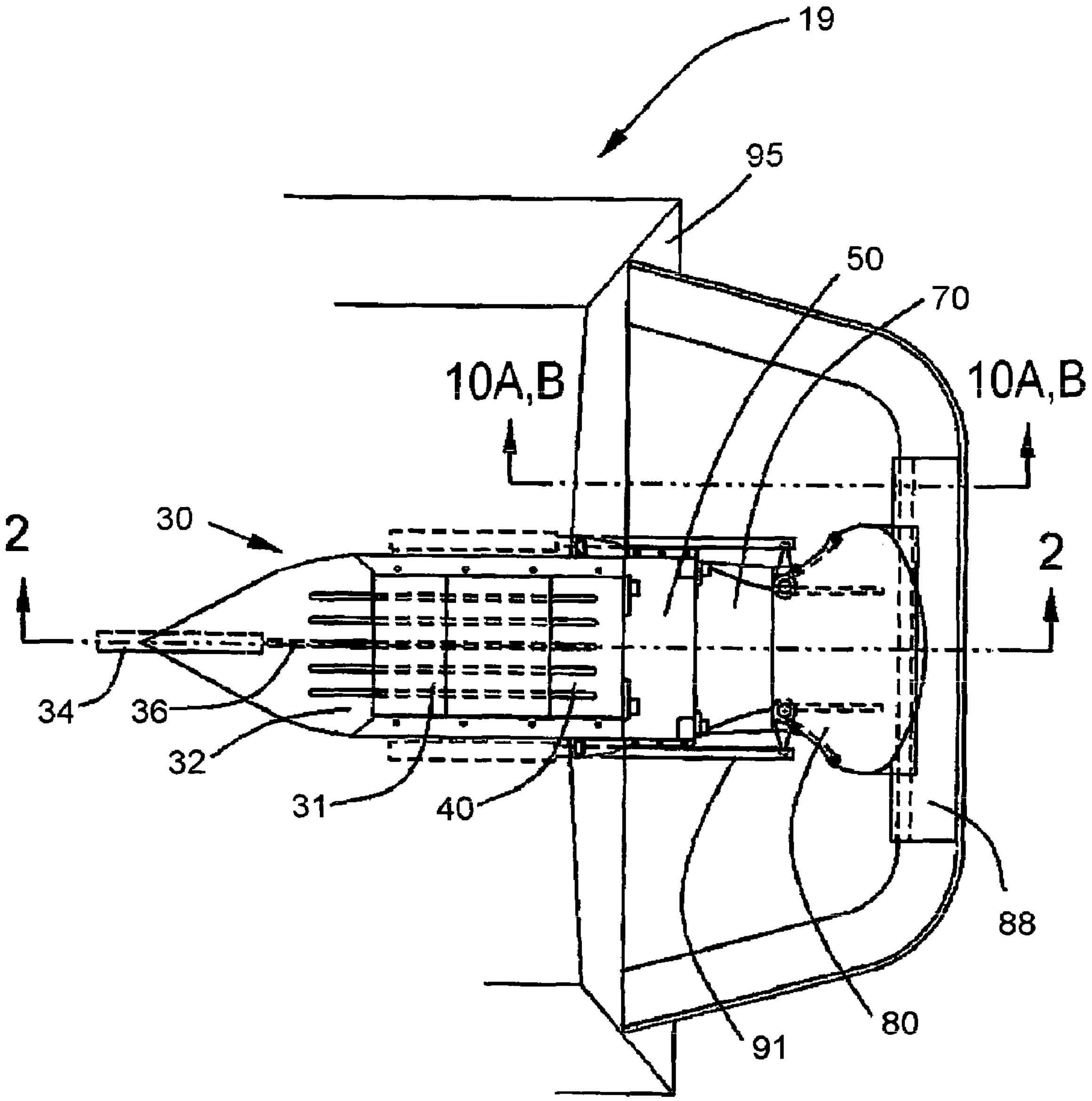


Fig. 1

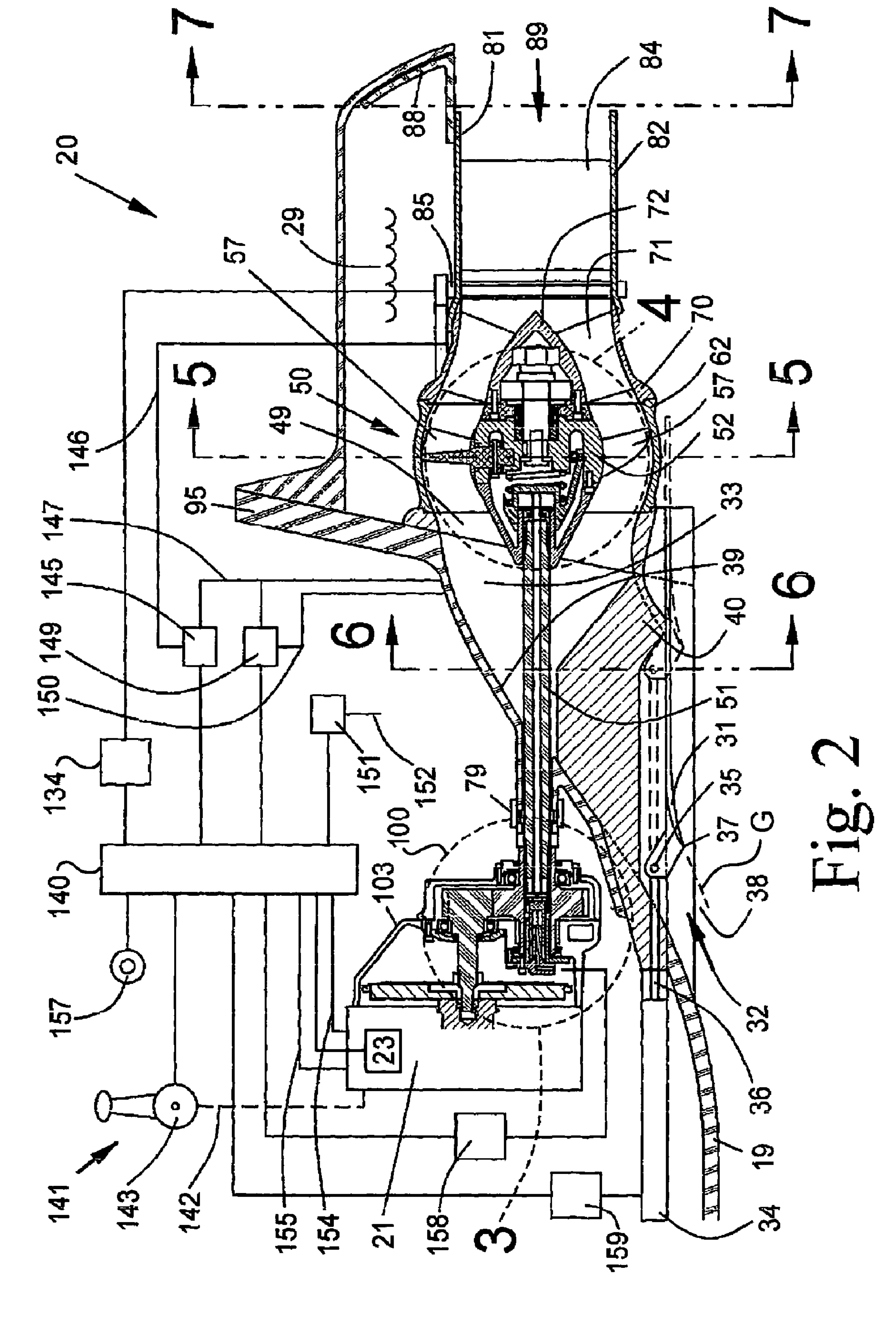

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Fig. 2

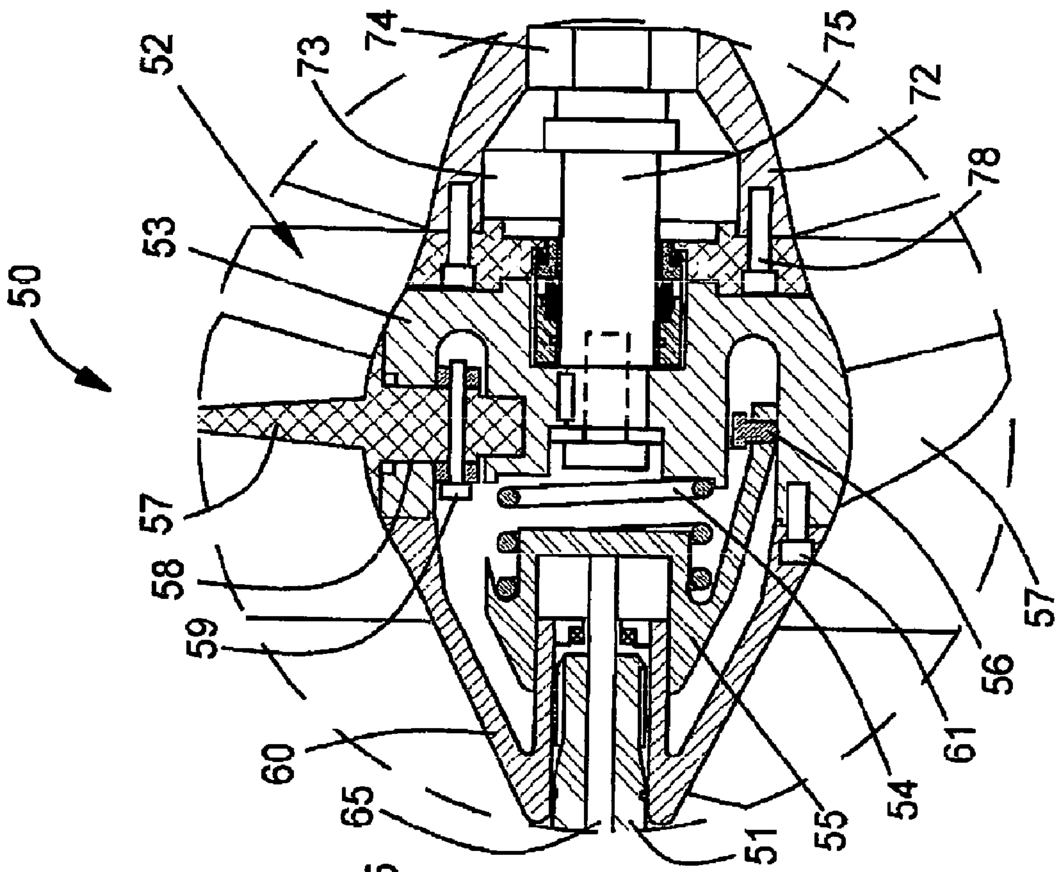
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Fig. 4

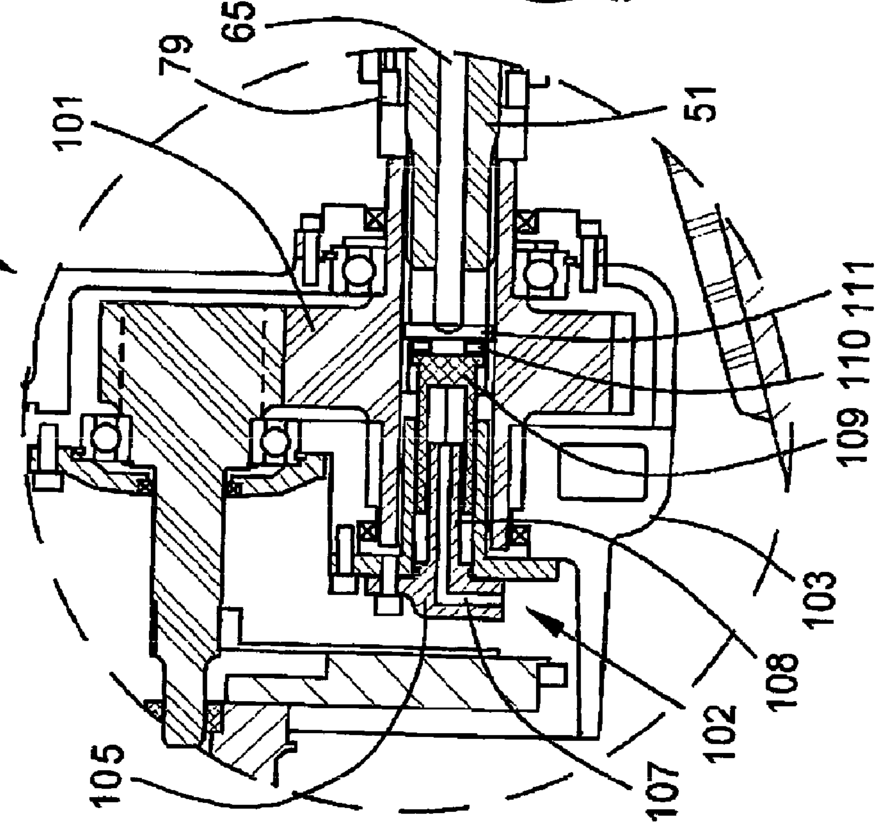
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Fig. 3

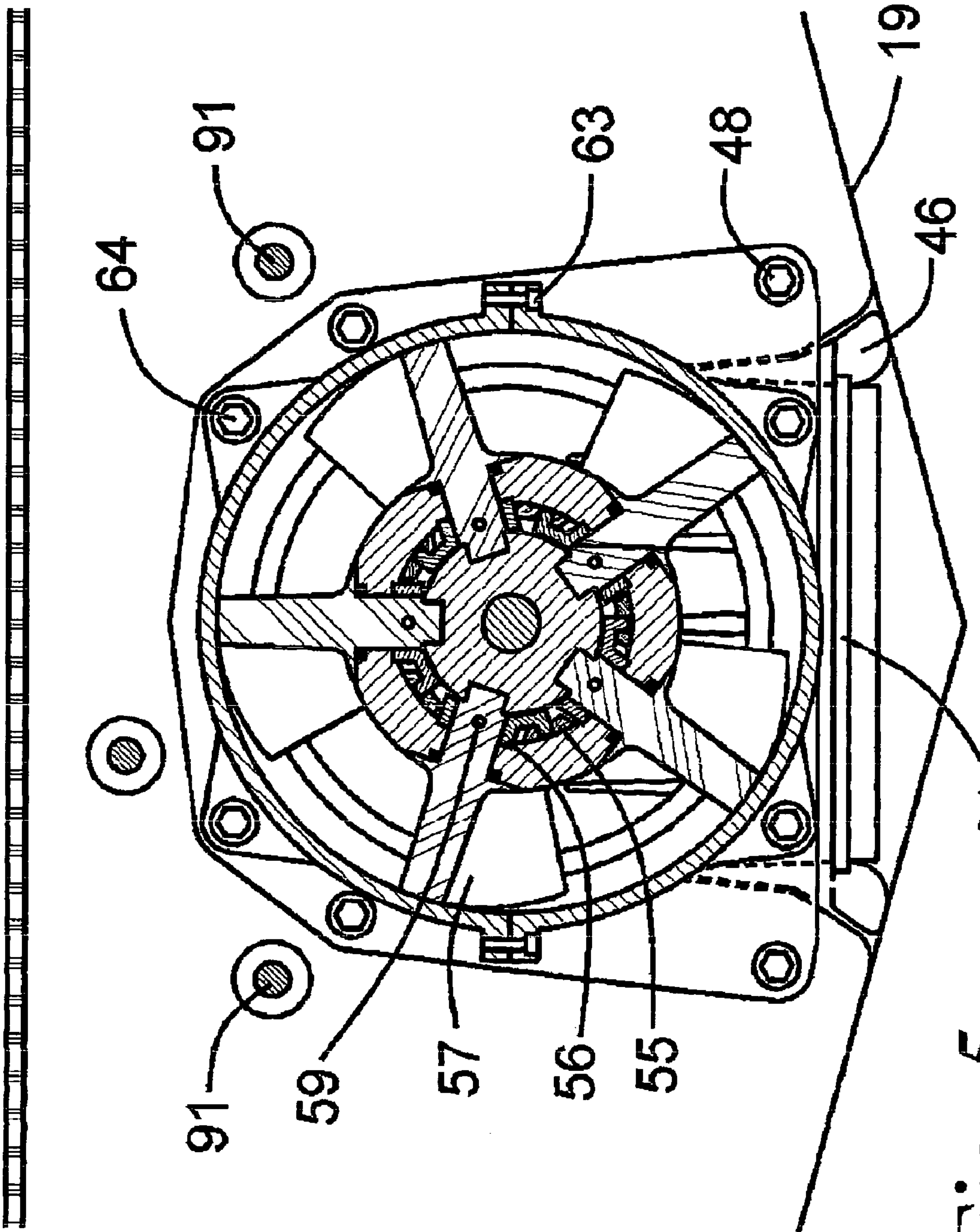

Fig. 5

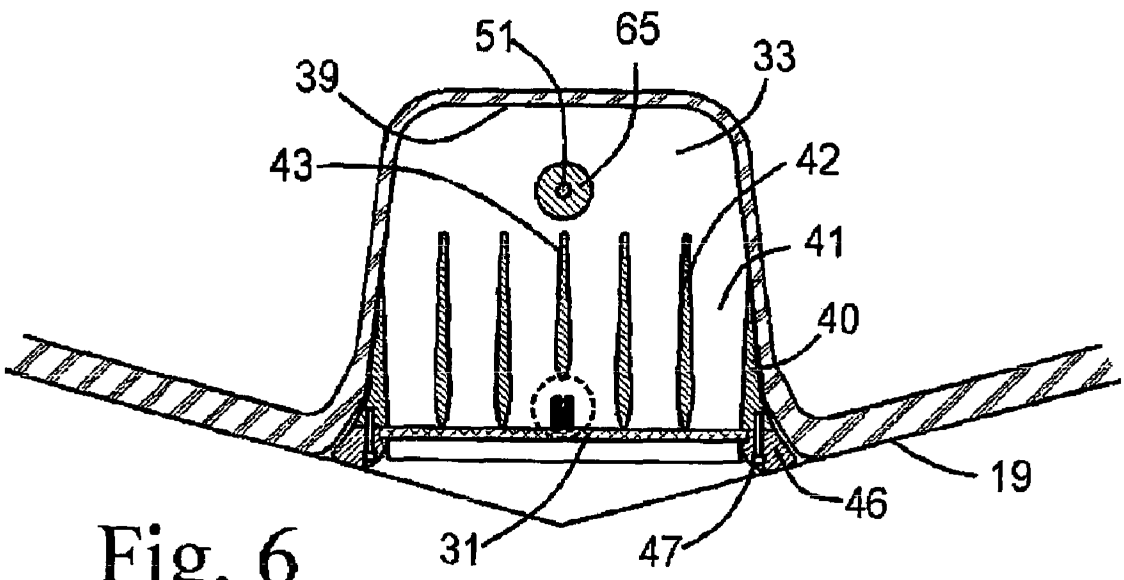
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Fig. 6

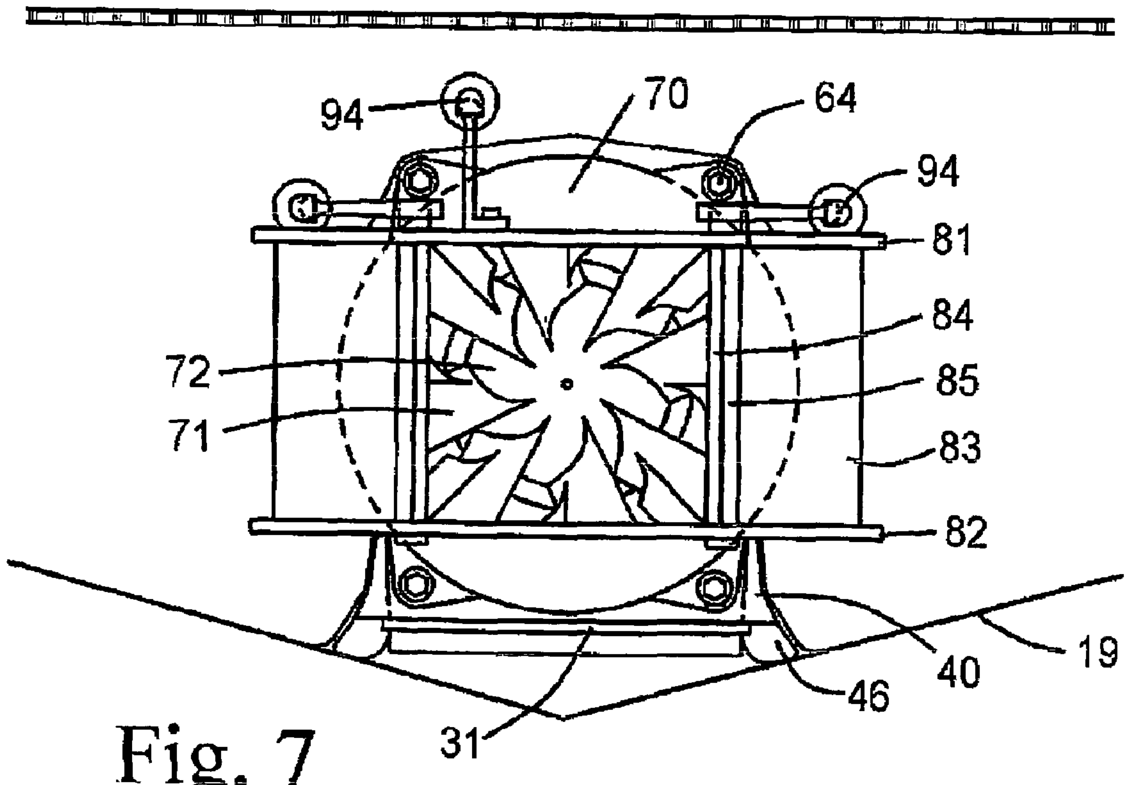
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Fig. 7

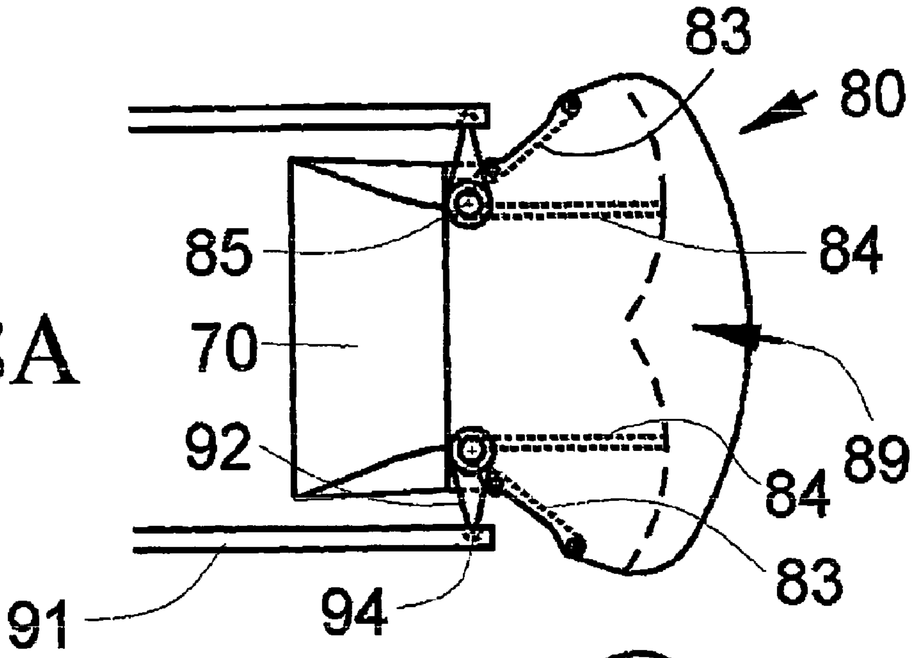

Fig. 8A

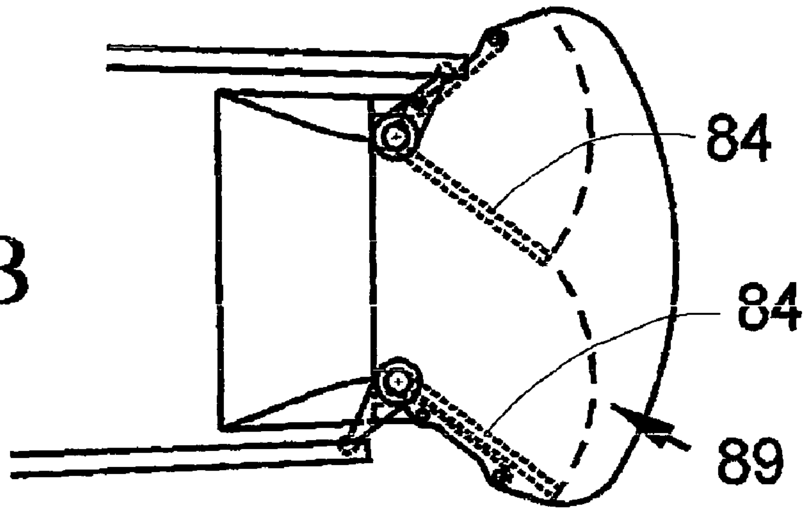

Fig. 8B

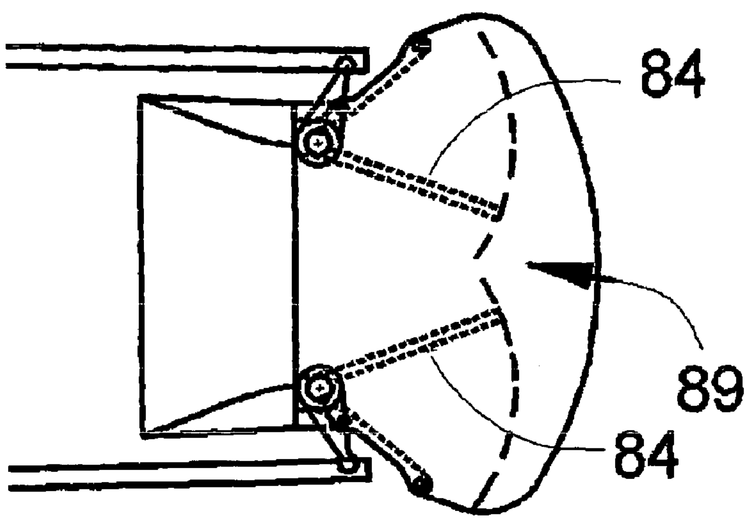

Fig. 8C

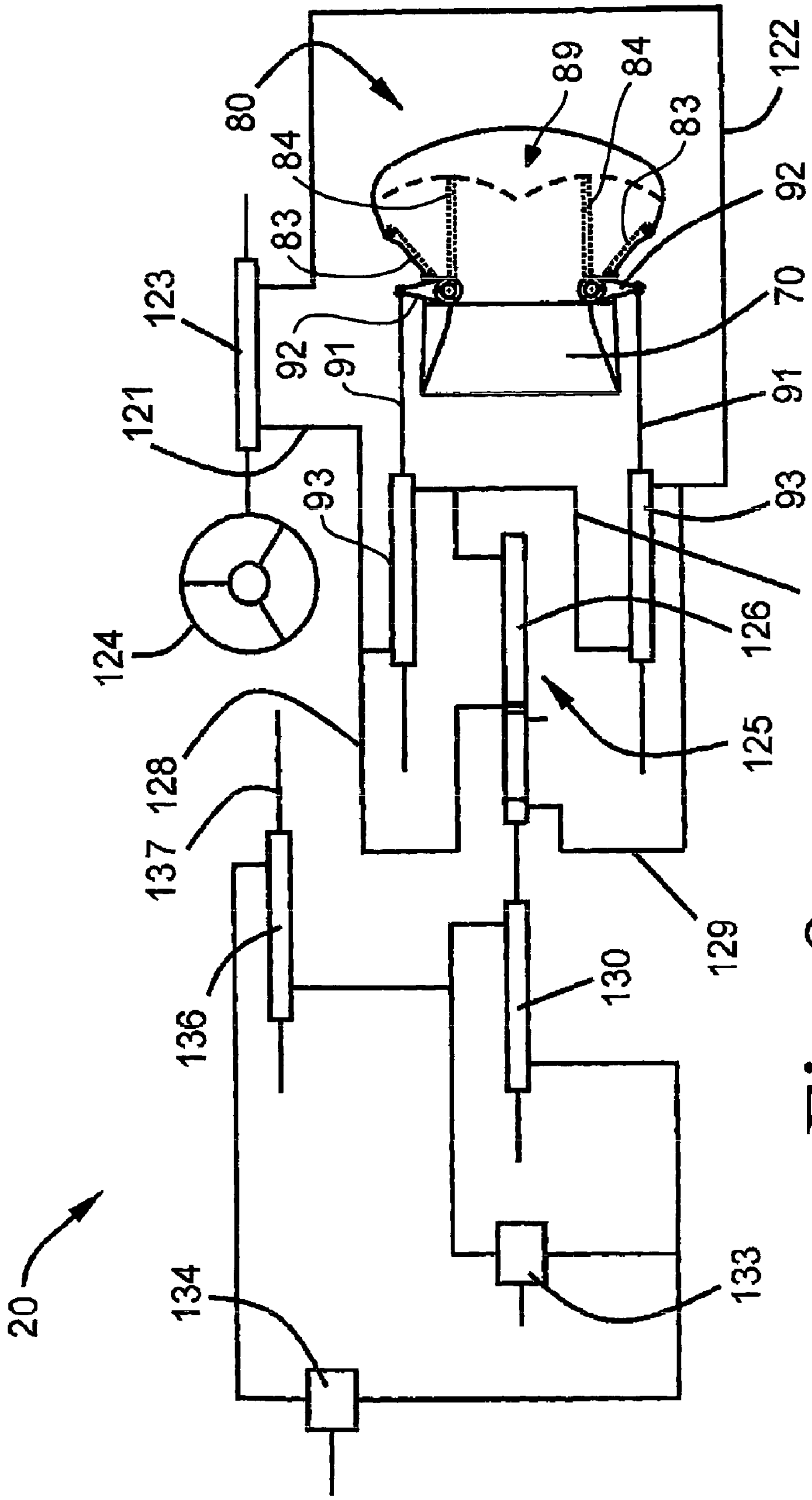
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Fig. 9

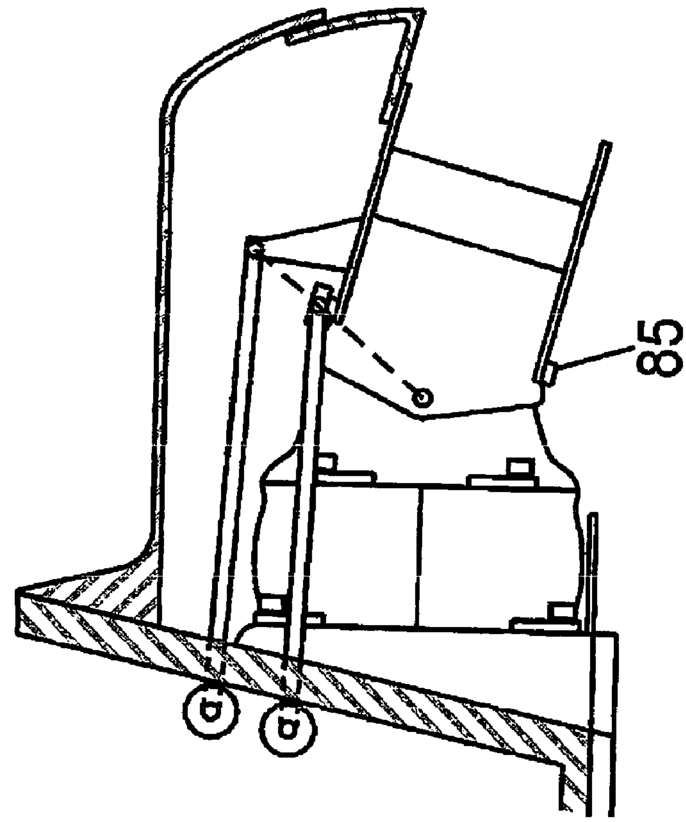


Fig. 10B

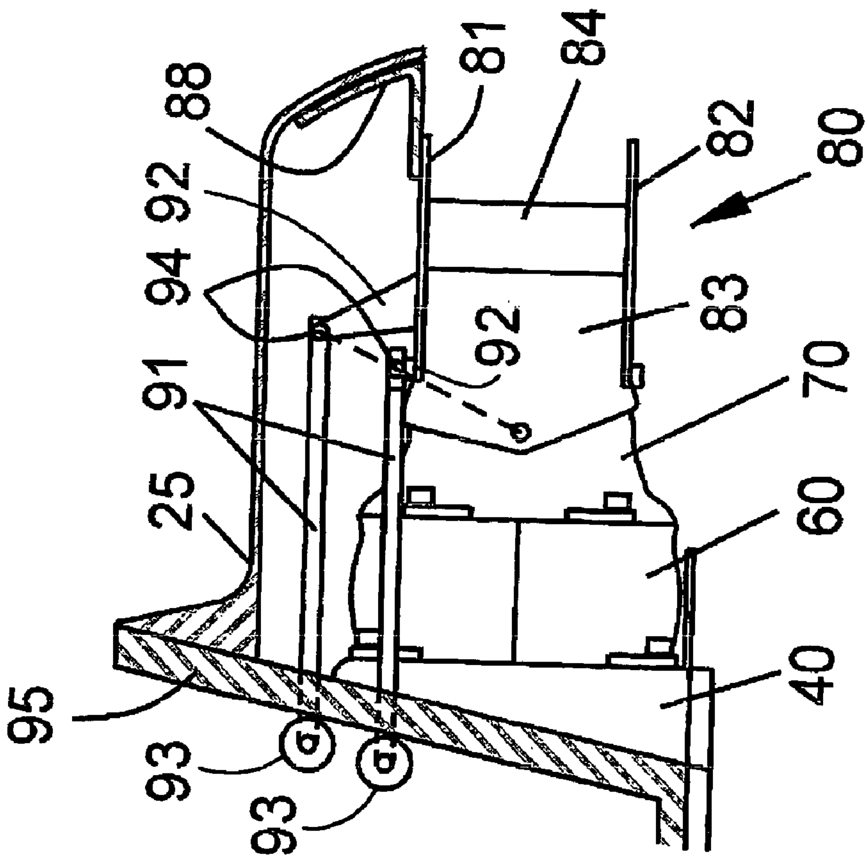
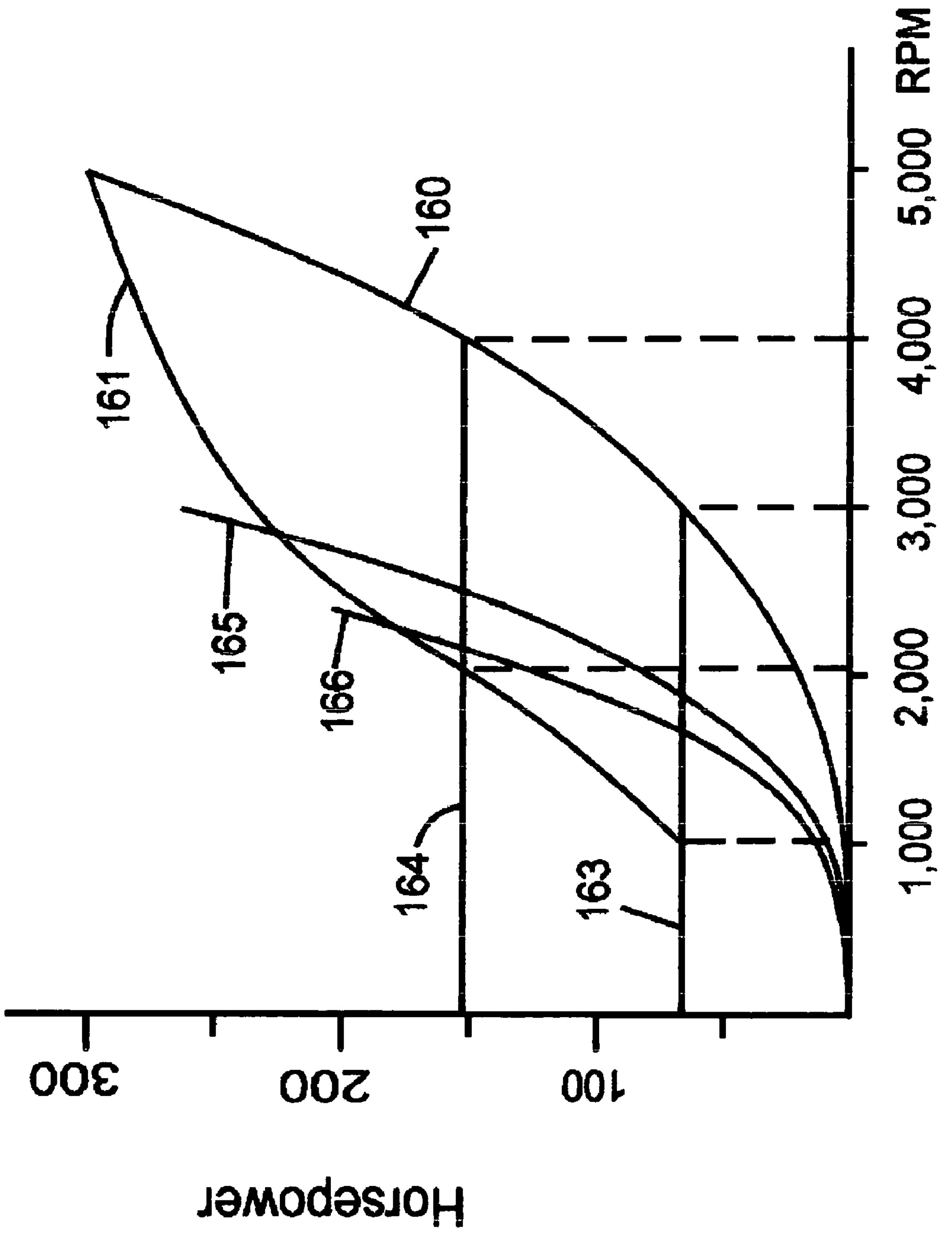


Fig. 10A


Horsepower
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200
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RPM
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3,000
4,000
5,000
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Fig. 11

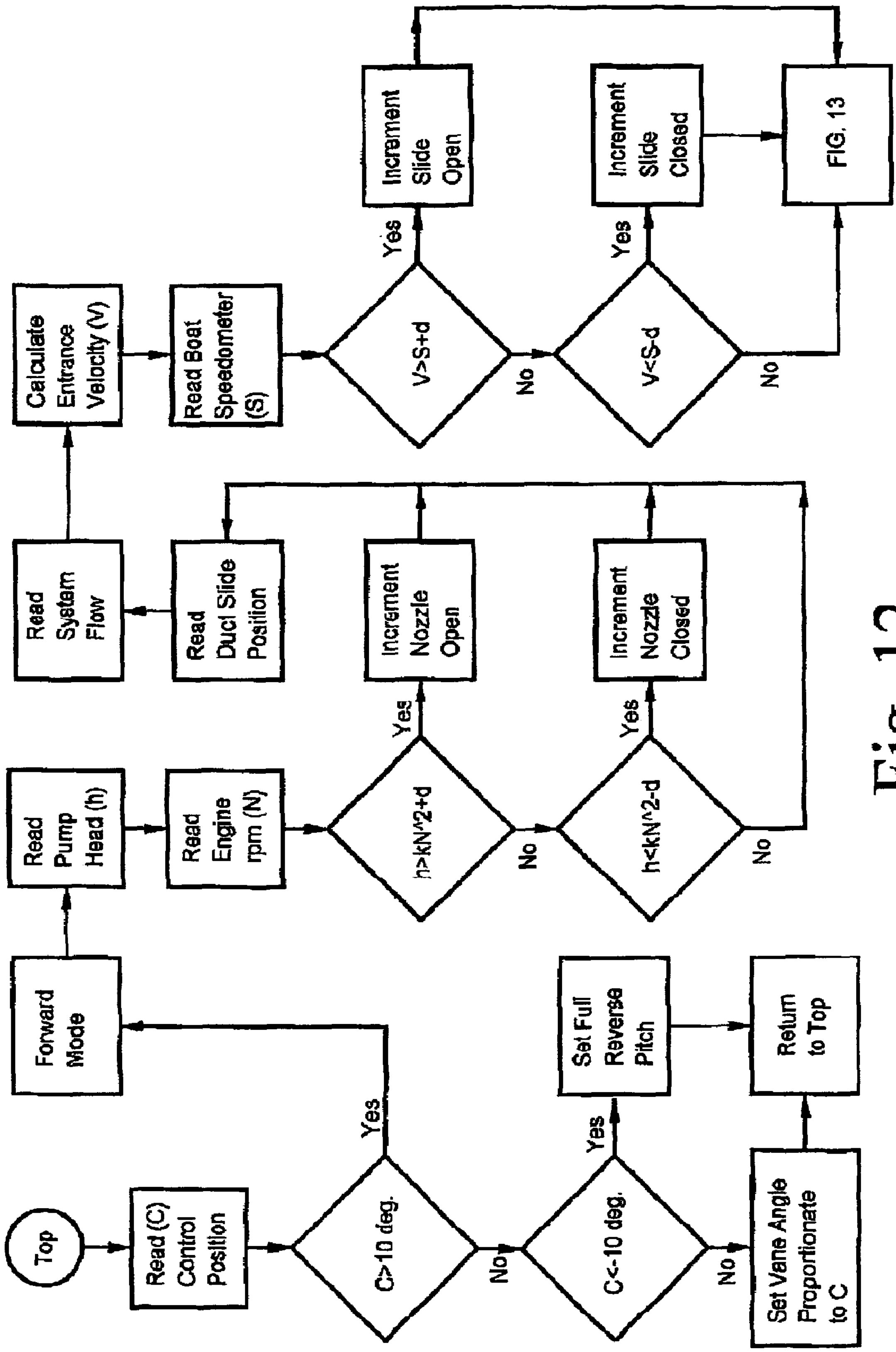
Top
Read (C) Control Position
C>10 deg.
Yes
No
Forward Mode
C<-10 deg.
Yes
No
Set Full Reverse Pitch
Set Vane Angle Proportionate to C
Return to Top
Read Pump Head (h)
Read Engine rpm (N)
h>kN^2+d
Yes
No
Increment Nozzle Open
h<kN^2-d
Yes
No
Increment Nozzle Closed
Read Duct Slide Position
Read System Flow
Calculate Entrance Velocity (V)
Read Boat Speedometer (S)
V>S+d
Yes
No
Increment Slide Open
V<S-d
Yes
No
Increment Slide Closed
FIG. 13

Fig. 12

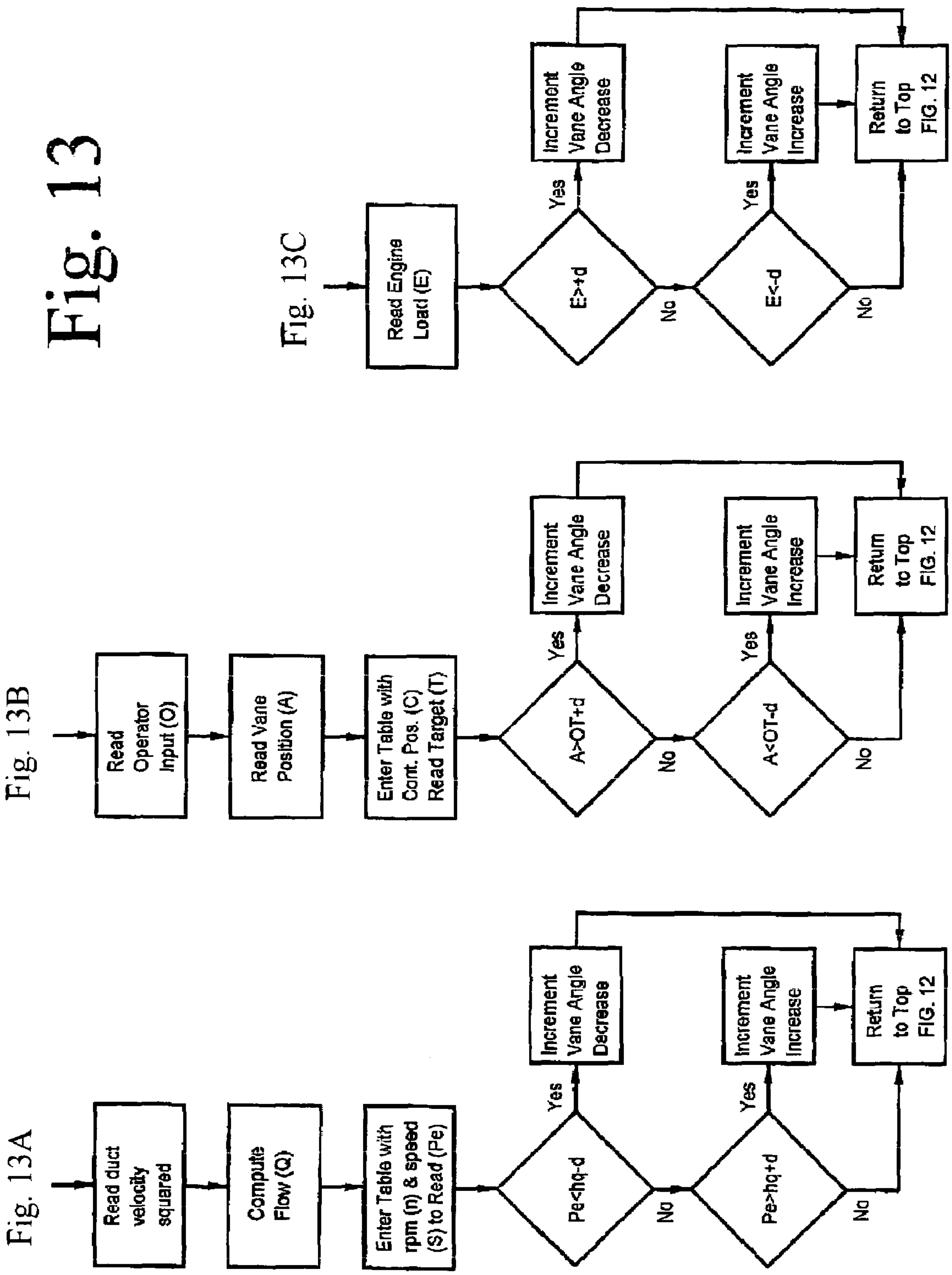
Fig. 13A
Read duct velocity squared
Compute Flow (Q)
Enter Table with rpm (n) & speed (S) to Read (Pe)
Pe<hq-d
Yes
No
Increment Vane Angle Decrease
Pe>hq+d
Yes
No
Increment Vane Angle Increase
Return to Top FIG. 12
Fig. 13B
Read Operator Input (O)
Read Vane Position (A)
Enter Table with Cont. Pos. (C) Read Target (T)
A>OT+d
Yes
No
Increment Vane Angle Decrease
A<OT-d
Yes
No
Increment Vane Angle Increase
Return to Top FIG. 12
Fig. 13C
Read Engine Load (E)
E>+d
Yes
No
Increment Vane Angle Decrease
E<-d
Yes
No
Increment Vane Angle Increase
Return to Top FIG. 12

Fig. 13

VARIABLE MARINE JET PROPULSION

The present invention claims priority benefit of International Patent Application Number PCT/US2003/039296 filed in the name of Jeff Jordan, the same inventor as the inventor of the present application, on Dec. 10, 2003, which claims priority benefit of Provisional Patent Application Ser. No. 60/432,281 filed also in the name of Jeff P. Jordan on Dec. 10, 2002, the complete disclosures of which are incorporated herein by reference.

TECHNICAL FIELD

This invention relates to Marine Jet Propulsion Systems, and more particularly to such systems of an improved design, which are more efficient over a range of vessel speeds and loads.

BACKGROUND ART

A marine jet propulsion system includes an inlet duct, a pumping means and a nozzle. The inlet duct delivers water from under the hull to the pumping means, which is driven by an engine. The pumping means delivers the water through the nozzle, which produces a water jet, thereby propelling the watercraft through the body of water in which the watercraft moves. In the prior art, a reversing bucket redirects the jet flow back under the boat fully for reverse thrust and partially for neutral thrust.

My U.S. Pat. Nos. 5,658,306, 5,679,035 and 5,683,276, which are incorporated by reference, disclose systems and methods for simultaneously optimizing the hydraulic efficiency of the inlet duct and the pumping means. Such increased hydraulic efficiency has allowed a substantial increase in the design system flow rate, which is well understood in the propulsion field of art to improve propulsion efficiency at low watercraft speeds. The increased hydraulic efficiency of the system and the methods preserves propulsion efficiency at higher watercraft speeds, so that the systems operate more efficiently over a wide range of boat speeds and accelerations.

From disclosures in my US Patents and through common knowledge in the propulsion field of art, it is known that larger mass flow rates and concomitantly lower nozzle velocities are more efficient at lower watercraft speeds, whereas lower mass flow rates and concomitantly higher nozzle velocities are more efficient at higher watercraft speeds. To achieve these ends, it is well understood in the art that a larger nozzle area is useful at low watercraft speeds, whereas a smaller nozzle area is most useful at higher watercraft speeds. Such reduction of nozzle size with watercraft speed was a natural consequence of the operation of the systems and the methods disclosed in my US Patents. However, a greater reduction of nozzle size with watercraft speed would be desirable for increased propulsion efficiency over a range of watercraft speeds.

When the watercraft is operating in a planing mode, the water jet obliquely strikes the water surface behind the watercraft, which results in turbulence on the water surface. Such turbulence is dependent on the velocity of the water jet relative to the water surface. When the velocity of the water jet relative to the water surface is high, as is common in the prior art, the water jet interacts with the water surface to produce a high turbulent spray of water behind the boat, which is commonly called a "rooster tail." The rooster tail is commonly considered objectionable for water skiing and wakeboarding behind the watercraft. Reducing the velocity

of the water jet relative to the water surface eliminates the rooster tail, but still leaves a turbulent trail of surface water in the wake of the watercraft, which is still objectionable to wake boarders, who like to use short ropes. A further reduction of the velocity of the water jet relative to the water surface would be desirable for the further reduction of the turbulent trail of surface water in the wake of the watercraft.

Another shortcoming of the prior art is the fact that the engine commonly operates at substantially higher rpm than would be most efficient, which results in greater fuel consumption, greater engine wear, and more noise than would result from operation at the engine's most efficient rpm. The operation of such systems in the prior art has been made more efficient by incorporating a two-speed transmission, but at higher cost, weight and axial length.

Many marine jet propulsion systems of the prior art feature a direct connection between the pump and the engine to eliminate the cost and axial length of a transmission or clutch. In these designs of the prior art, the neutral position that could be provided by the transmission or clutch is approximated by partially reversing the flow from the jet. The operator cannot easily maintain the balance of this partial reversing, especially given the sudden surge when starting the engine, so that the watercraft moves unpredictably. A true neutral control position would be desirable to enhance the operator's control of the watercraft.

Trash management is another shortcoming of the marine propulsion systems of the prior art. Many types of floating debris can become lodged on the grate that covers the inlet of the system, which restricts the flow of water into the pump and reduces propulsion efficiency. There are three types of such debris: solid objects, like rocks; fibrous material, like rope, fishing line, grass, reeds, and the stems of aquatic plants; and sheet material that can blind large sections of the grate, like large kelp leaves and plastic bags. The fibrous material is also well known to lodge on the leading edges of pump and stator vanes, reducing pump efficiency. The rope is particularly difficult to disentangle, when it becomes wrapped around the impeller and the drive shaft. Some jet boats carry hand rakes with right angle bends in the handle to remove debris from the inlet grate, and some integrate moveable grate sections to remove such debris, but these methods are awkward and only partially effective. Some commercial water jet propulsion systems are equipped with a reversing transmission, which is used to back flush both the pump vanes and the grate. As a last resort, commercial systems and river boats are commonly equipped with a clean-out hatch, which can be removed to allow the operator to remove debris from the pump inlet by hand. It would be desirable to reduce or eliminate the need for the trash handling mechanisms and methods by providing trash handling and back flushing methods integral to the design of the marine jet propulsion system.

In the marine jet propulsion systems of the prior art, reverse thrust is achieved by redirecting the water jet back under the boat along hydraulic reaction surfaces. Such reaction surfaces are commonly carried on a structure known as a "bucket", which is mechanically moved into the jet stream by the operator to get reverse thrust. Buckets for large jets take up considerable space and add weight and cost to the system. It would be desirable to eliminate the need for the bucket by incorporating a method of producing reverse thrust in the pump design.

DISCLOSURE OF THE INVENTION

Accordingly, it is an object of the invention to provide an improved marine jet propulsion system, which combines a variable pitch pump impeller and a variable nozzle under microcontroller controls to create a continuously variable power transmission, so that the engine is always operating close to its most efficient rpm.

It is a further object of the invention to use full pitch on the variable pitch pump impeller and maximum nozzle area on the variable nozzle at low speeds, which both increases propulsion efficiency and reduces the turbulent trail of surface water in the wake of the watercraft.

It is a further object of the invention to reduce the variable-pitch impeller pitch and the variable nozzle area with increasing watercraft speeds, so that both the impeller pitch and the nozzle area are minimum at the top boat speed, which is well understood in the art to increase propulsion efficiency.

It is a further object of the invention to maintain the variable pitch impeller pump close to its most efficient operating conditions over both a wide range of shaft rpm and a wide range of watercraft speeds, while simultaneously achieving the objects and advantages stated above.

It is a further object of the invention to achieve these objects and advantages in combination with a variable inlet duct, that efficiently converts excess velocity at the duct entrance into pressure at the pump inlet, as described in my U.S. Pat. No. 5,683,276.

It is a further object of the invention to incorporate a novel pump design, which allows the variable pitch to be reduced to near zero, which results in no effective pumping action, which is effectively a true neutral power transmission.

It is a further object of the invention to provide a method of further varying the pitch of the variable pitch impeller pump to create a reverse pumping action, which provides grate and vane cleaning by back flushing the system.

It is a further object of the invention to provide a vane design method, which results in close tolerances between the leading edges and the trailing edges of the vanes as they rotate through zero pitch, which results in a scissoring action between the leading edges and the trailing edges of the vanes, which effectively cleans the leading edges of the vanes. The scissoring action will also be seen to be effective in cutting rope and fishing line that may be sucked into the system, and it can be used to effectively chop up larger pieces of debris in the pump inlet into smaller pieces, which can escape through the grating or the nozzle.

It is a further object of the invention to provide for the further variation of the variable pitch vanes to produce a reverse pumping action through the system, which becomes an effective reverse thrust when controlled in concert with the variable inlet and the variable nozzle, thereby eliminating the need for the reversing bucket.

It is a further object of the invention to utilize the same nozzle vanes for reverse steering as are used for forward steering and nozzle flow regulation.

SUMMARY

These and other objects are met by providing an improved marine jet propulsion system, which combines a novel variable pitch spherical pump impeller and a variable steering nozzle to create a continuously variable power transmission, so that the engine is always operating close to its most efficient rpm. Reducing the pitch on the variable pitch spherical pump to near zero provides a neutral power transmission. Further reducing the pitch results in a scissoring action between the pump vanes, which cleans debris off the leading edges of the vanes. Further reducing the pitch results in reverse pitch and in reversing the pump flow, which back flushes the system for trash removal. Further reducing the pitch results in a reverse pumping action, which is an effective reverse thrust, particularly when used in concert with the variable steering nozzle and in concert with the variable inlet duct, which can act as a reverse nozzle. The swim platform and power trim function, which are both common on recreational boats of the prior art, can be used to reduce vortex formation and cavitation in the reverse thrust mode.

The variable pitch spherical pump incorporates concentric spherical surfaces on the impeller hub and on the pump housing. The axes of rotation of the variable pitch impeller vanes are radii of the concentric spherical surfaces, and the inner and outer edges of the variable pitch impeller vanes are also spherical surfaces, which fit closely to the spherical surfaces of the impeller hub and of the pump housing, respectively. This geometry allows the variable pitch impeller vanes to rotate about the axes of rotation, while constantly maintaining close fits between the inner and outer edges of the vanes and the impeller hub and the pump housing, respectively. The close fits are well known in the pump design field of art to contribute to efficient pump operation. In particular, this geometry allows the vanes to rotate to near zero pitch required for effectively neutral power transmission, while providing close fits at the full pitch required in any application. It also allows the vanes to rotate fully into reverse pitch, while maintaining the close fits, which is well understood to result in a reverse pumping action, which is useful for back flushing trash and for providing reverse thrust.

In the forward thrust mode of operation, the variable nozzle is controlled to maintain the most efficient head on the variable pitch impeller pump for the current shaft rpm, as is described in my U.S. Pat. No. 5,679,035. It is well understood in the art that the most efficient head on the variable pitch impeller pump is largely dependent on the square of the shaft rpm. It is also well understood in the art that the most efficient head on the variable pitch impeller pump is only very slightly dependent on impeller pitch. Hence, the pump will always be operating close to peak efficiency, when the variable nozzle is controlled to maintain pump head as a function of square of the shaft rpm.

It is well understood in the art that efficiency is nearly constant over a broad range of impeller pitch. The resulting flow through the pump is well understood to be a function of the impeller pitch. The shaft power demand of the pump is well understood to be directly dependent on the product of pump head and flow, when efficiency is constant. From this, it is clear that varying the impeller pitch varies the pump shaft power demand. It is further clear that this variation of power demand occurs without significant loss of efficiency, when most efficient pump head is simultaneously maintained by varying nozzle area. It will also be clear to those schooled in the art that knowledge of instantaneous pump head and shaft rpm can be used to compute the system flow by means of the pump affinity constants, and hence the shaft power demand of the pump. It will also be clear to those schooled in the art that knowledge of actual system flow can be compared to the flow indicated by pump head and shaft rpm to monitor the efficiency of the pump operation, which can be used to alert the operator of pump inefficiency, which is probably due to debris on the inlet grate or on the pump vanes.

A microcontroller incorporates inputs from differential pressure transducers to determine the head on the pump and the flow through the system. The microcontroller gets an rpm input from an engine tachometer. The control program in the microcontroller incorporates a look-up table of the pump efficiency as a function of shaft rpm. From these inputs the control program determines the shaft power demand of the pump. The control program also incorporates a look-up table, which allows interpolation of the most efficient power supplied at each shaft speed by the engine, as is well understood in the art of industrial controller programming. The control program compares the calculated pump power demand to the power most efficiently supplied by the engine at the input rpm, and adjusts the pitch on the variable pitch impeller to adjust pump shaft power demand to approximate the most efficient power supply of the motor at the input rpm. Simultaneously, the variable steering nozzle is adjusted to maintain the pump at its most efficient operating head for the shaft rpm.

In an alternate embodiment, the pitch on the variable pitch impeller is controlled by reference only to the throttle position on the engine. The efficient power supplied by the motor is largely dependent on the throttle position, and the pump power demand is largely dependent on impeller pitch, so linking the impeller pitch to the throttle position approximately maintains efficient engine operation. Simultaneously, the variable steering nozzle is adjusted to maintain the pump at its most efficient operating head for the shaft rpm.

In another alternate embodiment, the pitch on the variable pitch impeller is adjusted based on an engine loading output from a combustion microcontroller on the engine. It is well understood that such combustion microcontrollers commonly use a variety of sensors on the engine to control fuel injection, ignition timing and electric servo valve timing. Such combustion microcontrollers also commonly output engine-loading signals to automobile transmission microcontrollers, which incorporate engine conditions into their shift point control calculations. By these means, variations in elevation, humidity, fuel quality, and other engine operating parameters are incorporated in the most efficient shift point control decisions, so that the engine operates most efficiently. Similarly, this alternate embodiment adjusts the impeller pitch to operate the engine most efficiently. Simultaneously, the variable steering nozzle is adjusted to maintain the pump at its most efficient operating head for the shaft rpm. It will be clear from the following disclosure that several fortunate consequences result from this pump and nozzle design and from these control methods.

When the watercraft is at the dock, the operator can manually control the pitch on the variable pitch impeller to be effectively zero, so that no pumping action results from the rotation of the variable pitch impeller. This is a true neutral position for starting the engine and for sitting at rest in the water. The operator can also reverse the pitch to clean the vanes and to back flush the system. By increasing the pitch, the operator increases the flow through the jet in a controllable way, either in forward or reverse, eliminating any starting jerks or uncontrollable movement of the watercraft. The same steering wheel or other steering control method is effective in steering the boat in either forward or reverse. When the operator has set the impeller at full forward pitch and increases the engine rpm, the microcontroller maintains efficient operation, as described above.

At low speeds, the power demanded to propel the boat at constant speed is low. To match the power demanded by the pump to the most efficient rpm of the engine, the microcontroller sets the pump impeller pitch near maximum. To maintain the pump close to its most efficient operating conditions, the microcontroller opens the variable steering nozzle to maximum. In addition to maintaining engine efficiency, this control strategy has the fortunate consequence of providing maximum flow at low speeds for maximum propulsion efficiency. The flow through the maximum nozzle opening also occurs at the lowest possible velocity. Thus, motor efficiency, pump efficiency, and flow rate efficiency are all close to optimum, and wake turbulence is minimized.

When the system is under full acceleration, as in pulling up a water skier, the control system will reduce the pump impeller pitch to match the pump's shaft power demand to the engine's most efficient power supply at the instantaneous shaft rpm. The control system will also reduce the nozzle area to maintain the most efficient head on the pump for its current rpm.

When the boat reaches steady wakeboarding speed in the approximate range of 15 to 20 mph, the impeller is close to full pitch to reduce the engine rpm to the most efficient operating point. The variable nozzle is close to being fully open to maintain the most efficient pump head at the relatively low shaft rpm. A further advantage is that the variable inlet duct opening is near maximum due to the high flow, which results in no losses from the conversion of inlet entrance velocity to pressure at the pump inlet. This again has the fortunate consequence of providing close to maximum flow at this relatively low boat speed for maximum propulsion efficiency, which also results in minimum nozzle velocity through the large nozzle area and consequently in minimum wake turbulence. The system rpm is further reduced relative to systems of the prior art by this higher propulsion efficiency, which requires less shaft power and consequently lower shaft rpm to maintain the boat speed. Thus, motor efficiency, pump efficiency, and flow rate efficiency are all close to optimum, and wake turbulence is minimized.

When the boat reaches steady water skiing speed at approximately 30 mph, the recovery of pressure in the inlet duct has increased, which will cause a slight reduction in nozzle area to maintain the most efficient system flow and head on the pump. The power required to maintain this higher boat speed is also higher, so the engine must operate at a higher rpm to supply the necessary power. The most efficient pump head rises as the square of the shaft rpm. Higher engine rpm causes the control system to reduce the impeller pitch, which reduces the most efficient pump flow. The nozzle area control function implicitly accounts for higher inlet head at this boat speed, higher pump head at the higher shaft rpm, and the reduced flow resulting from reduced impeller pitch. As a result of all these factors, the nozzle area is reduced and the nozzle velocity relative to the boat is increased. However, the nozzle velocity relative to the water surface is reduced by the increased boat speed, so that the velocity of the jet relative to the water surface has only slightly increased. Wake turbulence is thereby only slightly increased, and the use of longer towropes at this higher boat speed makes wake turbulence less critical, since it has more time to dissipate before the skier reaches it.

Further increases in boat speed demand increased engine power, which the engine can only supply at higher rpm. The control system reduces impeller pitch to allow the engine higher rpm. Reduced impeller pitch requires a commensurate reduction in nozzle area. Pump head is rising as the square of the rpm. Inlet head is rising as the square of the boat speed. The increasing pump rpm, the reducing pitch, and the higher inlet pressure are all factors, which will result in the control system's reducing the nozzle area to maintain peak pump efficiency. Hence, nozzle area is reduced with increasing rapidity as boat speed increases as a natural consequence of the system operation, until minimum nozzle area is reached at the top design speed of the system. The minimum nozzle area at top speed is also ideal for reducing the system flow rate, hence improving propulsion efficiency at the higher speed.

BRIEF DESCRIPTION OF THE DRAWINGS

FIG. **1** is a plan view of the bottom of a boat, which incorporates an Improved Marine Jet Propulsion System, showing the hull, inlet duct, pump housing, variable nozzle, and the swim platform.

FIG. **2** is a midline vertical section view indicated on FIG. **1**, showing the internal details of the improved marine jet propulsion system and the control system schematic.

FIG. **3** is an enlarged view of the area indicated on FIG. **2**, which shows the details of the hydraulic control piston for the vane pitch.

FIG. **4** is an enlarged view of the area indicated on FIG. **2**, which shows the details of the impeller hub and vane pitch operating mechanism.

FIG. **5** is a section view indicated on FIG. **2** showing the vanes in the inlet duct and the sliding gate beneath the vanes.

FIG. **6** is the section view indicated on FIG. **2** showing the variable vane operating mechanism of the pump.

FIG. **7** is a rear section view of the boat indicated on FIG. **2** showing the variable rectangular nozzle under the swim platform.

FIGS. **8**A, **8**B and **8**C are schematic overhead views of the variable steering nozzle showing the various vane positions that result from the actions of the hydraulic nozzle controls.

FIG. **9** is a schematic representation of the nozzle hydraulic system, which shows the integration of the steering function, the nozzle area reduction function, and the nozzle pitch function.

FIGS. **10**A and **10**B are section views indicated on FIG. **1** showing the power trim adjustment of the propulsion system and the maximum declination, which is used in reverse mode.

FIG. **11** is a graph on which shaft power is plotted against shaft rpm, showing the relationships between pump power demand and efficient engine power supply.

FIG. **12** is a flow chart for the microcontroller program used to control the variable pump vane pitch, the variable nozzle area and the variable inlet entrance area in all embodiments of the invention.

FIGS. **13**A, **13**B and **13**C are flow charts for three alternative microcontroller programs used to control the variable impeller vane angle for efficient engine operation in the forward mode of operation.

BEST MODE FOR CARRYING OUT THE INVENTION

In the accompanying FIGS. **1-13**, there is shown an improved marine jet propulsion system, generally referred to as **20**, designed to achieve higher propulsion efficiency, greater maneuverability, and better injury prevention features than currently available marine propulsion systems.

The system **20** includes a variable water inlet duct **30** for admitting water into the system **20**, a variable-pitch spherical pump **50** capable of receiving and pumping a relatively large amount of incoming water, and an adjustable, large, variable rectangular discharge steering nozzle **80** capable of forcibly exiting the water pumped by the spherical pump **50** to propel the watercraft **19** through the body of water **29**. A microcontroller **140** controls the variable inlet duct **30**, the variable pitch spherical pump **50** and the variable discharge steering nozzle **80**. By simultaneously controlling the variable inlet duct **30**, the variable-pitch spherical pump **50**, the large variable rectangular discharge steering nozzle **80**, the propulsion efficiency of the system **20** is greatly improved over marine jet propulsion systems of the prior art.

The inlet duct **30** is designed so that hydraulic efficiency of the system **20** is optimally maintained at all watercraft **19** velocities, as described in my US Patents. In this embodiment, the entrance area of the inlet entrance opening **32** is varied by the action of the inlet hydraulic slide cylinder **34** on an adjustable inlet slide **31** to match the velocity of the water in the inlet entrance opening **32** to the velocity of the water passing under the watercraft **19**.

As shown in FIGS. **1-6**, the inlet duct **30** includes adjustable inlet slide **31** located over the inlet entrance opening **32** of the hydraulically efficient, elongated inlet tunnel **33** formed or attached to the bottom of the watercraft **19**. The inlet hydraulic slide cylinder **34** moves the adjustable inlet slide **31** to vary the effective area of the inlet entrance opening **32**. The inlet tunnel **33** is longitudinally aligned on the watercraft **19** with a front inlet entrance opening **32** and a rear exit opening **49**. The inlet tunnel **33** gently curves upward inside the watercraft **19** and has a larger cross-sectional area at its exit opening **49** than at its inlet entrance opening **32**, when the adjustable inlet slide **31** is in its forward position as shown in FIG. **2**. The surrounding surface of the inlet entrance opening **32** of the inlet tunnel **33** is gently curved from tangent to bottom of the watereraft **19** so that turbulence is minimal at the inlet entrance opening **32** of the inlet duct **30**.

A grate structure **40** fits in the elongated inlet tunnel **33** and attaches to the watercraft **19** with grate structure fasteners **48**, so that the conversion of excess entrance velocity at the inlet entrance opening **32** into pressure at the rear exit opening **49** takes place largely in the rectangular passages or flow channels **41** between grate vanes **42**. It is well understood in the art of hydraulic design that dividing the flow into such rectangular flow channels **41** reduces turbulence losses in the water system flow (indicated in FIG. **2** by arrow **32**), which are larger than the frictional losses against the vane surfaces. The system **20** moves the adjustable inlet slide **31** by means of the inlet hydraulic slide cylinder **34** to adjust the size of the inlet entrance opening **32** so that the velocity of the incoming water therethrough matches the velocity water under the watercraft **19** in the body of water **29** in which the watercraft **19** moves. By controlling the relative velocity of the incoming water through the inlet entrance opening **32** and by using a hydraulically efficient inlet tunnel **33**, which gradually increases in cross-sectional area from its inlet entrance opening **32** to its exit opening **49**, the dynamic head of the incoming water may be efficiently recovered at the spherical pump **50**.

As shown in FIGS. **1**, **2** and **6**, the grate structure **4O** includes a plurality of spaced apart, longitudinally aligned elongated grate vanes **42**. The middle grate vane **43** is vertically truncated to allow passage for the inlet slide cylinder shaft **36** of the inlet hydraulic slide cylinder **34**, which passes through the watercraft **19**. The inlet slide cylinder shaft **36** is attached to the adjustable inlet slide **31** with clevis pin **35**, so that the action of the inlet hydraulic slide cylinder **34** moves the adjustable inlet slide **31** in response to the microcontroller **140**. The adjustable inlet slide **31** is held in place by the slide rails **46**, which are attached to the grate structure **40** with the slide rail fasteners **47**. The leading edge **37** of the adjustable inlet slide **31** bends downward so that the effective entrance angle of the leading edge **37** is approximately parallel to the upper inlet surface **39** of the inlet tunnel **33**, so that the velocity of inlet entrance flow (indicated in FIG. **2** by arrow **32**), which is parallel to the upper surface **39** will approximately match the entrance angle of the leading edge **37**, which is well understood in the art of hydraulic design to provide efficient separation of the inlet flow from the water under the boat.

When the watercraft **19** is stationary or at low speed, water enters the inlet entrance opening **32** via the suction created by the spherical pump **50**. During this stage, the adjustable inlet slide **31** is in its rearmost position as shown by the ghost line position G in FIG. **27**, so that the inlet entrance opening **32** is wide open and the grate vanes **42** act as diffusers to reduce entrance swirl. As the watercraft's speed increases, water enters the inlet entrance opening **32** by the forward movement of the watercraft **19** through the body of water **29** and by the suction of the spherical pump **50**. The microcontroller control system **140** adjusts the position of the adjustable inlet slide **31** through the inlet hydraulic slide cylinder **34** and inlet slide cylinder shaft **36** so that the velocity of the water entering the inlet entrance opening **32** matches the velocity of the water under the watercraft **19**. At the top design speed of the system **20**, the adjustable inlet slide **31** is in the forward position shown in FIG. **2**.

As the velocity of the incoming water at the inlet entrance opening **32** relative to the velocity of the incoming water at the exit opening **49** in the inlet tunnel **33** increases, the microcontroller **140** progressively moves the adjustable inlet slide **31** forward. It can be seen that this has two effects—first, it reduces the effective area of the inlet entrance opening **32** of the inlet tunnel **33**; and second, it increases the effective length of the inlet duct **30**. It can be seen that the changes both in cross-sectional area and change in flow direction within the inlet tunnel **33** are always gradual, which are design requirements well known in the art for the efficient recovery of pressure head in the turbines and venturi flow meters. It can also be seen that the increasing effective length of the inlet tunnel **33** with decreasing effective area of the inlet entrance opening **32** maintains a nearly constant rate of change in area over the inlet tunnel's range of operation. The total dynamic head of the incoming water can then be efficiently recovered at the spherical pump **50**.

Disposed adjacent to the exit opening **49** of the inlet tunnel **33** is the spherical pump **50**, which is coupled via a drive shaft **51** and transmission **100** to an engine **21**. In the embodiment shown, the spherical pump **50** is contained in a split spherical pump housing **62**, which is attached to the grate structure **40** with the fasteners **64**. The spherical pump **50** is axially aligned with the inlet duct exit opening **49**, so that the splined drive shaft **51** extends forward there from and connects to the transmission **100**. In the embodiment shown, the spherical pump **50** includes a spherical impeller **52**, which rotates to forcibly deliver the incoming water from the exit opening **49** to the discharge steering nozzle **80** located on the opposite side of the spherical pump **50**. In the preferred embodiment, the spherical pump **50** is designed to be used with a 300 horsepower engine so that the mass flow equals approximately 2200 lbs/sec and the pump head is approximately 70 feet at full power with 18-degree discharge angle on the variable pump vanes. The spherical pump **50** uses a 16-inch spherical impeller **52**, which matches the size of the diffuser **70**, which is disposed over the aft position of the spherical pump **50** to recover the vortex velocity produced by the spherical pump **50** as useful propulsive momentum, as is common in the art of pump design. The stator vanes **71** of the diffuser **70** support the diffuser hub **72**, which contains the tapered roller bearings **73** and **74**.

Referring to FIGS. **2**, **3**, **4** and **5**: In the assembly of the spherical pump **50**, the bearings **73** and **74** are first mounted on the pump shaft **75**, which is inserted into the diffuser hub **72**. The bearing collar **75** is bolted to the hub **72** to carry the thrust of the bearing **73** and to provide mounting surfaces for the mechanical seal. The spherical impeller hub **53** is bolted to the pump shaft **75**. The return spring **54** and the spider **55**, engaging the operating arms **56**, are held in place by a press, while the vanes are inserted radially through the bearing holes **58** and the operating arms **56**. Each vane is rotated to align with the pin holes in the operating arms **56** and the locking pins **59** are inserted to lock the assembly together. When the press is removed, the spider **55** is held in place by the operating arms **56**, which restrains the return spring **54**. The hub cone **60** is fitted over the spider **55** and drawn up against the impeller hub **53** by the progressive sequential tightening of the impeller hub cone bolts **61**. This process compresses the return spring **54** and results in the pump impeller vanes **57** being held in the full pitch position by the return spring **54** against the hub cone **60**.

The split spherical pump housing **62** is assembled around the impeller **52** and pinned together circumferentially with the fasteners **63**. The diffuser **70** is attached to the pump housing **62** with the fasteners **78**. The splined drive shaft **51** is assembled into the internally splined driven gear **101** trapping the water seal **79**. Matching the internal spline in the pump hub cone **60** to the splined drive shaft **51**, the assembled spherical pump **50** and diffuser **70** slide onto the splined drive shaft **51** and are attached to the grate structure **40** with the fasteners **64**. Internal to the splined drive shaft **51** is the pushrod **65**, which acts on the spider **55**.

A vane adjustment means is connected to the pump impeller **52** for controlling pitch of the pump impeller vanes **57**, and, hence, the most efficient flow rate of the spherical pump **50**. As shown in FIGS. **2**, **3**, **4** and **5**, the vane adjustment means includes the hydraulic cylinder assembly **102** internal to the splined driven gear **101**. The hydraulic cylinder assembly **102** is solidly mounted to the bell housing **103** using the fasteners. The end piece **105** of the hydraulic cylinder assembly **102** incorporates a hydraulic fluid passage **107** and a square post **108**, which fits a square hole in the vane actuator piston **109** to prevent the rotation of the vane actuator piston **109**. The vane actuator piston **109** acts on the roller thrust bearing **110**. On the other side of the roller thrust bearing **110**, the bearing plate **111** engages the internal spline in the splined driven gear **101**, so that the bearing plate **111** rotates with the splined driven gear **101** and the splined drive shaft **51**.

It can be seen that when hydraulic fluid is forcibly introduced through the hydraulic fluid passage **107**, the vane actuator piston **109** is driven against the bearing **110**, which acts on the bearing plate **111** and the push rod **65**, which is driven through the rotating drive shaft **51** to act on the spider **55** to compress the spring **54** in the impeller hub **53** and move the operating arms **56**, which reduce the pitch of the pump impeller vanes **57**.

Located aft position of the pump's diffuser **70** is the variable rectangular discharge steering nozzle **80**. The steering nozzle **80** is formed between a top plate **81** and a bottom plate **82**, which are held parallel by their attachment to the two wing walls **83**. The nozzle steering vanes **84** have integral nozzle vane shafts **85**. The nozzle vane shafts **85** are born by bearing holes in the top plate **81** and bottom plate **82**. The nozzle steering vanes **84** are formed so that their top and bottom edges fit closely to the top plate **81** and bottom plate **82**, respectively. The axes of the nozzle vane shafts **85** are held perpendicular to the plates **81** and **82**, so that the rotation of the nozzle vane shafts **85** results in the movement of the nozzle steering vanes **84** between the plates **81** and **82**, while maintaining close fits between the edges of the nozzle steering vanes **84** and the plates **81** and **82**. As a result of this geometry, there is formed a rectangular nozzle discharge opening **89** which is bounded by the plates **81** and **82** and the nozzle steering vanes **84**.

FIGS. **8**A, **8**b and **8**C show top views of the steering nozzle **80**, which show how the angle of the nozzle steering vanes **84** can be controlled both to provide steering control and to reduce nozzle area. Each of the nozzle steering vanes **84** is positioned by a hydraulic steering ram **91**, which operates on the respective vane operating arm **92**. The hydraulic nozzle steering cylinders **93** are mounted inside the transom **95**, so that only the steering rams **91** penetrate the transom **95**. FIG. **8**A shows the nozzle steering vanes **84** in the wide-open straight position. FIG. **8**B shows the nozzle steering vanes **84** in the full low speed turn position. FIG. **8**C shows the nozzle steering vanes **84** in the high-speed flow reduction position.

FIG. **9** is a schematic of a hydraulic system for controlling the nozzle steering vanes **84** for steering, flow reduction, and nozzle azimuth simultaneously. The azimuth movement is commonly used in planing watercraft as a power trim to adjust the planing angle of the boat, as is well understood in the art. As will be seen in the discussion of FIG. **10** below, the adjustment of the nozzle discharge angle in the vertical plane is also useful for reducing vortexing in the reverse mode.

The double acting hydraulic nozzle steering cylinders **93** penetrate the transom **95** with ball-ended fittings or rubber grommets, as is common in the art, and are connected to the vane operating arms **92** with ball-ended couplings **94**, as is common in the art. The hydraulic steering lines **121** and **122** are connected to a hydraulic helm **123**, which is driven by the steering wheel **124**, as is common in the art. The hydraulic nozzle steering cylinders **93** are series connected for reverse action, so that the hydraulic nozzle steering cylinders **93** move equal distances in opposite directions in response to fluid delivered from the hydraulic helm. This steering action can be seen to result in the common rotation of the nozzle vane shafts **85**, until the nozzle steering vanes **84** reach the position shown in FIG. **8**B.

The balancing cylinder **125** of FIG. **9** is composed of three hydraulic cylinders in tandem. The driven cylinder **126** is single ended, and is so constructed that the area of the piston of driven cylinder **126** is twice the area of the shaft. The nozzle closing circuit **127** is connected on the closed end of the driven cylinder **126**, so that the fluid displacement is proportionate to the area piston of driven cylinder **126**. The hydraulic balancing circuit **128** is connected on the shaft side of the piston of driven cylinder **126**, so that the fluid displacement in the circuit **128** is equal to half of the displacement of the piston of driven cylinder **126** in the nozzle closing circuit **127** and opposite in direction. The hydraulic balancing circuit connection **129** is made to tandem balancing cylinder **125** in the end connected to driving cylinder **130**, so that the displacement in the hydraulic balancing circuit **129** is also equal to half of the displacement of the piston of driven cylinder **126** in the nozzle closing circuit **127** and opposite in direction. It can be seen that the result of this arrangement is that the hydraulic nozzle steering cylinders **93** move in the same direction and by the same amount in response to the movement of the common shaft of the tandem cylinders **126** and **130**. It can also be seen that no net fluid displacement occurs in the hydraulic steering lines **121** and **122** from the hydraulic steering helm **123** to the hydraulic nozzle steering cylinders **93**. Hence, the displacement of the common shaft of the tandem cylinders **126** and **130** has the effect of increasing and decreasing the angle between the nozzle steering vanes **84**, and this movement may cause the nozzle steering vanes **84** to reach the positions shown in FIG. **8**C. It should also be noted that the action of the tandem cylinders **126** and **130** is independent of the action of fluid flows from the steering helm **123** at hydraulic steering lines **121** and **122** and may occur simultaneously, so that the system allows simultaneous nozzle area control and steering.

Referring further to FIG. **9**, the flow control module **133** controls the displacement of the tandem cylinders **126** and **130**. The driving cylinder **130** moves in response to two hydraulic power sources. The flow control valve **134** responds to commands from the microcontroller **140**. It is a 4-way valve that controls the motion of the driving cylinder **130**, as is common in the art. Through this means the microcontroller **140** acts to adjust the effective nozzle area of the rectangular nozzle discharge opening **89** in order to maintain the efficient operation of the spherical pump **50**, as is detailed below. The flow control valve **134**, is controlled by the operator to adjust the azimuth of the steering nozzle **80** for power trim, as shown in FIGS. **10**A and **10**B. The hydraulic circuit from the trim control valve **134** is connected in series with the trim cylinder **136**. It can be seen that the effect of this circuit is to displace the trim cylinder **136** and the driving hydraulic cylinder **130** in the same direction. As a result of the motion of the driving cylinder **130**, the hydraulic nozzle steering cylinders **93** are also displaced in the same direction. This common motion can be seen to reduce the effect of power trim adjustment on the position of the nozzle steering vanes **84**.

FIGS. **10**A and **10**B are side elevation section views indicated on FIG. **1**, showing the action of the optional trim cylinder **136** on the steering nozzle **80**. The piston area of the trim cylinder **136** is so chosen relative to the piston area of the hydraulic nozzle steering cylinders **93** and the driving cylinder **130** that the angular displacement of the connection points **94** is approximately equal. As a result, the action of the trim control valve **134** of FIG. **9** is to extend both the trim cylinder ram **137** and the steering cylinder rams **91** by a proportion that minimizes the effect of the trim movement on the steering and nozzle area control functions. The extreme down position of the trim range shown in FIG. **10**B is useful in increasing submergence of the steering nozzle **80**, which acts as a water inlet in the reverse mode. The nozzle guard **88** serves both to prevent vortex cavitation and to prevent human limbs and other objects from approaching possible pinch points in the steering nozzle **80** mechanisms.

Propulsion system efficiency is the product of four efficiency components: inlet duct **32**, pump **50**, steering nozzle **80**, and engine **21**. The steering nozzle **80** has relatively small losses, which can be ignored without significant loss of system efficiency. The recovery efficiency of inlet duct **32** is maximized independently by maintaining the duct entrance velocity to approximate the velocity of the water under the boat **19**, as detailed in my US Patents. Pump **50** efficiency is maximized independently by adjusting the nozzle area **89** to maintain the most efficient head, h, on the pump **50** for the current shaft rpm, as detailed in my US

Patents. In this disclosure engine **21** efficiency is maintained by incorporating a variable pitch spherical pump **50** in the propulsion system **20** design, which provides continuously variable power demand to track the most efficient power supply of the engine **21**. The head nozzle control method and the inlet duct control methods from my US Patents work well in concert with the variable pitch pump **50**. The propulsion system **20** simultaneously maximizes all of the four efficiency components: inlet duct **32**, pump **50**, steering nozzle **80**, and engine **21**, over a wide range of boat speeds and accelerations. As a result, the design flow of the system **20** can be increased with a smaller efficiency penalty, which allows the use of a higher mass flow rate for better propulsion efficiency, as is well understood in the art. The relevant principles and their interrelation are discussed in more detail below.

FIG. **2** shows the schematic diagram connections of the microcontroller **140**. The operator uses the single handle control **141** to control both propulsion direction and the throttle **142** for the engine **21**, as is common in recreational boats. The single handle control **141** incorporates a throttle dead band, so that the throttle is set at idle from about 10 degrees forward of the straight-up or neutral position to about 10 degrees back of the neutral position. In the prior art these forward 10 degree and reverse 10 degree travels operate a gear, which shifts the transmission into forward and reverse, respectively, and further travel of the handle out of the throttle dead band increases the engine throttle **142**, as is well known in the art. In the present embodiment, the single handle control **141** has the same appearance and function to the operator, but the integral gear shifting mechanism is omitted and replaced with a shaft position encoder **143**, which provides the angular position of the single handle control **141** to the microcontroller **140**. As is more fully explained in the discussion of FIGS. **12** and **13** below, the microcontroller **140** is programmed to position the vane actuator piston **109** (shown in FIG. **3**) through the hydraulic control module **133**, so that the vane angle follows the position of the single handle control **141** over the throttle dead band.

Another input to the microcontroller **140** shown in FIG. **2** is the head differential pressure transducer **145**, which provides the difference between the pitot tube pressure **146** after the spherical pump **50** and the pitot tube pressure **147** at the inlet of the spherical pump **50**. This difference is well understood in the hydraulic art to be the commonly accepted measure of the head, h, on the spherical pump **50**.

Another input to the microcontroller **140** shown in FIG. **2** is the flow differential pressure transducer **149**, which provides the difference between the inlet pitot tube pressure **147** and the pump inlet static pressure **150**. For purposes of the calculations discussed below it is well known that the differential pressure on the transducer **149** is equal to the flow velocity, V, squared divided by twice the acceleration of gravity, g, or: V^2/2 g It is also well understood that the volume flow rate, Q, is the product of the velocity, V, and the cross section flow area according to: Q=VA, and that mass flow rate, q, is the product of volume flow rate and the density of the fluid, w, so that; q=Qw.

Another input to the microcontroller **140** shown in FIG. **2** is the speed pressure transducer **151**, which provides the speedometer pitot tube pressure **152** from the boat speedometer pitot tube. For purposes of the calculations discussed below it is well known that this pressure is approximate to the speed of the water craft **19** divided by twice the acceleration of gravity, so the discussion in the previous paragraph also applies here.

Another input to the microcontroller **140** shown in FIG. **2** is the engine tachometer **154**. This tachometer input **154** is commonly a pulse train that is read with a timed counter integral to the microcontroller, as is well known in the art.

Another input to the microcontroller **140** shown in FIG. **2** is the engine load signal **155**, which is output by the engine combustion microcontroller. This interface is well known in the automotive art.

Another input to the microcontroller **140** shown in FIG. **2** is the operator preference input **157**, which is a variable resistance or optical encoder to indicate the operator preference for performance or economy operation.

The microcontroller **140** has several control outputs, through which it controls the movement of the nozzle steering vanes **84**, the pump impeller vanes **57**, and the adjustable inlet adjustable inlet slide **31**. The operation of the flow control valve **134** and flow control module **133** has been discussed in relation to FIG. **9** above. The balancing cylinder **125** of FIG. **9** has internal positional feedback to the microcontroller **140**, as is well known in the art. The inlet control module **159**, shown in FIG. **2**, uses hydraulic power and incorporates positional feedback. The impeller vane hydraulic control module **158** also uses hydraulic power and incorporates positional feedback.

In one embodiment, the program for microcontroller **140** is a PICmicro® Microcontroller, which is available from Microchip Technology. Programs for these devices are developed using the Microchip's C programming environment. This development system is capable of incorporating a wide range of mathematical functions in the control program. The following paragraphs provide background on the functions to be incorporated in the control program.

The Basis of the Control Relationships

The relationships for controlling the inlet duct and nozzle are developed in detail in my said US Patents, and reviewed in the discussion of FIG. **12** below. The technical basis for controlling the pump impeller vane **57** pitch to maintain the engine **21** at its most efficient operation follows.

FIG. **11** is a graph of shaft power versus shaft rpm, showing the relationship between pump shaft power demand and a typical engine's most efficient power supply. In a typical water jet propulsion system design, the gearing between the pump and engine is chosen so that pump power demand curve **160** intersects the engine power supply curve **161** at the highest allowable engine rpm, which is taken to be 5000 rpm in FIG. **11**. When the pump is maintained at its most efficient head and flow, the pump power demand curve **160** is approximately a cubic curve as shown in FIG. **11**, as is well known in the art of pump design, and particularly in the area of pump affinity relationships. The difference between the most efficient power supply curve **161** and the pump power demand curve **160** is unfortunately greatest in the most frequent operating range, which falls between the horizontal lines **163** and **164**. Hence, the engine **21** is operating furthest from its most efficient operating rpm most of the time. When the pump power demand curve **160** represents the variable pitch spherical pump **50** with the pump impeller vanes **57** at about 18 degrees beta-2, as vane pitch is commonly designated in the pump art, the full-pitch pump power demand curve **165** represents the power demand curve of the same spherical pump **50** with the pump impeller vanes **57** set at full pitch of about 40 degrees beta-2. It is well understood in the art of pump design that this range of efficient operation is common to variable pitch propeller pumps. The spherical pump **50** has the additional efficiency advantage of having close fits between the tips of the pump impeller vanes **57** and the housing **62**, even at pitches greater than 40 degrees beta-2. It is clear that the full-pitch pump power demand curve **165** much more closely matches the engine's most efficient power supply curve **161** in the most frequent operating range between the horizontal lines **163** and **164**. Such reduction of engine rpm is widely used in the automotive power transmission art to increase fuel efficiency and engine life. At the bottom end of operating range on the horizontal line **163**, the engine rpm is reduced from about 3,000 to about 2000. At the top end of the operating range along the horizontal line **164**, the engine rpm is reduced from about 4,000 to about 2,600 as indicated by full-pitch pump power demand curve **165**. There is a continuous range of efficient spherical pump **50** power demand curves between the curves **165** and **160**, which result from the continuous variation in pump impeller vane **57** pitch possible in the spherical pump **50**. One of these intermediate curves can be seen to be the most efficient for each possible engine rpm between 3,000 and 5,000 in FIG. **11**.

The pump power demand curves **160** and **165** and the range of efficient curves in between are based on the assumption that the pump is maintained at its most efficient head and flow for every shaft rpm and for every vane pitch. Following my said US Patents, this function is approximated by a control function based on the pump affinity relationship: head, h, equals an affinity constant multiplied by the square of the pump rpm, N, or h=kN^2. This nozzle control function and method are detailed in my U.S. Pat. No. 5,679,035, which is incorporated here by reference. It is well understood in the art of pump design that the affinity relationship between pump head, h, and shaft rpm holds true for variable-vane pumps over a wide range of vane settings. It is also well understood in the art that the pump affinity constant is only approximate, because the pump efficiency is reduced at higher shaft rpm, N. This efficiency deviation from the affinity relationship generally does not cause significant losses in employment of the nozzle control function, because the pump efficiency does not drop significantly so long as the operating head, h, and flow, Q, are close to the most efficient operating point. However, factoring in an efficiency correction factor based on shaft rpm, N, can increase the accuracy of the head affinity control relationship. In practice, the efficiency reduction in the pump **50** with higher shaft rpm can be largely captured in the head affinity constant, so that the control relationship is still: head, h, equals a constant, k, (corrected for efficiency reduction with increasing rpm) multiplied by the square of the pump shaft rpm, N: h =kN^2. This efficiency correction is also useful in the pump shaft power demand calculation, which is discussed below.

The curve **166** in FIG. **11** represents the power demand curve that can be achieved at somewhat reduced pump efficiency by either further increasing vane pitch or by reducing the nozzle area below that required to maintain the pump at its most efficient operating head, h, and flow, Q. In the preferred embodiment this occurs at low watercraft speeds and low engine rpm, N. For example, the steering nozzle **80** of the preferred embodiment is designed for a rectangular nozzle discharge opening area **89** of 10 inch by 10 inch, which is sufficient to maintain the pump **50** at its most efficient operating point on the full-pitch pump power demand power curve **165** at a watercraft speed of 20 mph, where the inlet duct **30** is recovering about 12 feet of total dynamic head at the pump inlet. However, at zero watercraft speed, and in the absence of the 12 feet of recovered head at the nozzle in addition to the pump head, the maximum nozzle area restricts full-pitch pump flow. Hence, pump head and shaft power demand are increased as is well understood in the pump art. This results in a zero-watercraft-speed, full-pitch power demand curve **166**. The curve **166** can be shifted down and to the right by reducing the impeller pitch, which reduces the efficient flow and the corresponding efficient nozzle area to approach the maximum effective nozzle area of the variable rectangular steering nozzle **80**. Hence reducing the pitch of the pump impeller vanes **57** reduces the pump shaft power demand in this range, just as it does between the curves **166** and **165**. The control area between the curves **166** and **165** is used to get more thrust at low engine rpm at low boat speeds.

FIGS. **12**, **13**A, **13**B, and **13**C are flow diagrams for the microcontroller **140** program. The "d" values in FIGS. **12** and **13** are control dead band factors to prevent hunting, as is common in the art. These are discussed in the Operation of the Invention below.

Operation

The operation of the invention is controlled by the microcontroller **140** using the control program diagrammed in FIGS. **12** and **13**. The physical components are shown in FIG. **2**. The control loop of FIG. **12** begins with reading the control position C of the shaft encoder **143** on the single handle control **141**. If C is in the throttle dead band range, the microcontroller **140** increments the vane impeller vane hydraulic control module **158** to set the pitch of the pump impeller vanes **57** is set to follow C. This has the effect of giving the operator direct control over the forward or reverse flow through the spherical pump **50**. The concentric spherical surfaces of the split spherical pump housing **62**, the pump impeller vanes **57** and the spherical surface of impeller hub **53** allow the pump impeller vanes **57** to rotate through 90 degrees or more, while maintaining close fits between the pump impeller vanes **57** and both housing **62** and the hub **53**. In the preferred embodiment, the impeller vanes rotate to about plus 40 degrees beta-2 for full forward pitch and through zero to minus 20 degrees beta-2 for full reverse pitch. The impeller vane hydraulic control module **158** positions the vane actuator piston **109** (FIG. **3**) by controlling the flow of fluid through the hydraulic fluid passage **107**. The vane actuator piston **109** acts through the roller thrust bearing **110**, the bearing plate **111**, the pushrod **65**, and the spider **55** to move the operating arms **56**, which rotate the pump impeller vanes **57**. If C is out of the dead band in reverse, the microcontroller **140** holds full reverse pitch on the pump impeller vanes **57**.

If C is greater than idle in the Forward Mode, the program of FIG. **12** branches to adjust the nozzle according to the pump head affinity relationship. The program of FIG. **12** then adjusts the adjustable inlet slide **31** to match entrance velocity to boat speed, and passes control to **13**A, **13**B, or **13**C for setting the pitch of the pump impeller vanes **57**.

When the operator moves the single handle control **141** out of the dead band range in the forward direction, the microcontroller program branches to the "Forward Mode" as shown in FIG. **12**. In accordance with the discussion of FIG. **12** above, the control program adjusts the nozzle steering vanes **84** between the positions shown in FIGS. **8**A and **8**C to maintain the most efficient head on the spherical pump **50**, according to the pump affinity relationship h=kN^2. This action maintains the spherical pump **50** at its most efficient head for the current shaft rpm. As also shown in FIG. **12**, the control program sets the adjustable inlet slide **31** to match the velocity of the inlet entrance **32** flow, Q, to the velocity of the water under the boat. This maintains the most efficient possible recovery of total dynamic head at the inlet of the spherical pump **50**.

Control is then passed to one of three methods to match the shaft power demand of the spherical pump **50** to the most efficient power supplied by the engine **21**. FIG. **13**A sets the pitch of the pump impeller vanes **57** based on pump shaft power demand calculated from measured head and flow on the spherical pump **50**. Alternatively, FIG. **13**B sets the pitch of the pump impeller vanes **57** based on throttle position as measured by the shaft position encoder **143** on the single handle control **141**. Alternatively, FIG. **13**C sets the pitch of the pump impeller vanes **57** based on feedback from the combustion control computer **23** on the engine **21**. It will be appreciated by those skilled in the art that each of these alternative methods accomplishes the same function: they all adjust the pitch on the pump impeller vanes **57**, so that the shaft power demand of the spherical pump **50** approximates the most efficient power supplied by the engine **21** at its current rpm.

This program of FIG. **12** has the following consequences. When the handle **141** is at the forward end of the dead band range, the pump impeller vanes **57** are at full forward pitch, which provides maximum forward thrust. When the handle **141** is in the middle of the dead band range, the pump impeller vane **57** pitch is about zero, which provides no pumping action and therefore a true neutral. When the handle **141** is at the back end of the dead band range, the vane pitch is in the maximum negative position, which provides reverse thrust and back flushing of trash. As the handle moves out of the dead band range in either direction, it increases the engine throttle **142**, which increases thrust, as is common with single handle controls on recreational boats. From this it is clear that the action of the propulsion system **20** in response to the position of the control handle **141** is identical to the action of propulsion systems of the prior art.

In the "Forward Mode" of FIG. **12**, and referring to FIGS. **2**, **9** and **10**A, the microcontroller **140** reads the head differential pressure transducer **145** to input the pump head, h, and the engine tachometer input **154** to input engine rpm N. If the measured head is higher than the pump affinity value kNˆ2 plus a small dead band factor, d, to prevent hunting, the microcontroller **140** uses the flow control module **133**, which positions the driving cylinder **130** and consequently the driven cylinder **126**, which forces fluid into the hydraulic balancing circuits **128** and **129**, while removing an equal amount of fluid from the hydraulic nozzle closing circuit **127**. Following the explanation of FIG. **9** above, this results in a balanced fluid flow to the hydraulic nozzle steering cylinders **93**, so that the steering rams **91** are equally retracted, acting through ball-ended couplings **94** on the nozzle vane operating arms **92** to increase the distance between the nozzle steering vanes **84**, thus increasing the effective nozzle discharge opening area **89**. This has the effect of increasing the water flow **44** and consequently reducing the head h on the spherical pump **50**. If the measured head, h, is lower than the pump affinity value kNˆ2 minus a small dead band factor, d, to prevent hunting, the microcontroller **140** uses the flow control module **133**, which positions the driving cylinder **130** and consequently the driven cylinder **126**, which removes fluid from the hydraulic balancing circuits **128** and **129**, while it forces an equal amount of fluid from the hydraulic nozzle closing circuit **127**. Following the explanation of FIG. **9** above, this results in a balanced fluid flow to the hydraulic nozzle steering cylinders **93**, so that the steering rams **91** are equally extended, acting through ball-ended couplings **94** on the nozzle vane operating arms **92** to reduce the distance between the nozzle steering vanes **84**, thus reducing the effective nozzle discharge opening area **89**. This has the effect of reducing the water system flow **44** and consequently increasing the head, h, on the spherical pump **50**. If the head, h, is within the dead band range, no nozzle control action is taken.

The next sequence in the control loop of FIG. **12** is setting the adjustable inlet slide **31**. Referring to FIG. **2**, the microcontroller **140** reads the position of the inlet hydraulic slide cylinder **34** from the inlet control module **159** and computes the effective inlet entrance area. It reads the flow differential pressure transducer **149** and computes the water system flow Q (indicated in FIG. **2** by arrow **32**) as in the description of the flow differential pressure input on the transducer **149** above. The microcontroller **140** then computes the entrance velocity through the inlet entrance opening from V=water system flow, Q, divided by the effective inlet entrance area, A. The microcontroller **140** reads the boat speed pressure transducer **151** and compares watercraft speed S. If V>S+d, the microcontroller **140** outputs to the inlet control module **159**, which actuates the inlet hydraulic slide cylinder **34** to move the adjustable inlet slide **31** back, which increases the effective entrance area and reduces the entrance velocity V. If V<S−d, the microcontroller **140** outputs to the inlet control module **159**, which actuates the inlet hydraulic slide cylinder **34** to move the adjustable inlet slide **31** forward, which reduces the effective entrance area and increases the entrance velocity V. This control function meets the requirement of my said US Patents that inlet duct efficiency requires that the flow velocity through the inlet entrance approximate the velocity of the water under the hull, which is indicated by the boat speedometer **152**. From the discussion above, it is clear that the microcontroller **140** can also be programmed to calculate the water system flow, Q, from positional feedback from the impeller vane control module **158** on the angle of the pump impeller vanes **57**, which would allow the elimination of the flow differential pressure transducer **149** input in the control loop.

At the end of FIG. **12** the microcontroller program control passes to FIG. **13**A, **13**B, or **13**C to match the spherical pump **50** power demand to the power most efficiently supplied by the engine **21** by varying the pitch of the pump impeller vanes **57**.

The control scheme in FIG. **13**A first computes the hydraulic power produced by the spherical pump **50**, which is the product of pump head, h, and system mass flow rate, q. The pump shaft power demand is the hydraulic power divided by the hydraulic efficiency, e, so that the control equation for efficient engine operation may be written: P=hq/e or Pe=hq. The latter formulation is convenient, because both the efficient engine power, P, and the hydraulic efficiency, e, are dependent on shaft turn rate in rpm, N, Hence a table of control equation Pe values, which is entered with the shaft turn rate, rpm N, and the boat speed, S, can be highly accurate. In the preferred embodiment, the boat speed factor, S, is only useful for boat speeds of less than 20 mph, where the power demand curve falls between the curves **166** and **165** of FIG. **11**. The pump head, h, is constantly available from the nozzle control loop of FIG. **12**. The system mass flow rate, q, calculation is discussed above in the description of the flow differential pressure transducer **149** input. It will be apparent to one skilled in the control systems art that other methods of inputting the mass flow rate could be used, including mechanical, acoustic and optical flow sensing devices. The microcontroller **140** sets the pump impeller vane **57** pitch by outputting to the impeller vane control module **158**. Control passes back to the top of FIG. **12**.

Alternatively, in FIG. **13**B the control loop uses a table of vane pitch targets, T, which is entered with control position C. The vane pitch targets values, T, include adjustments for pump efficiency variations and other factors based on test results. The control position, C, is a measure of the engine throttle **142** setting, which has an associated most efficient operating shaft turn rate in rpm, N. This shaft turn rate in rpm is implicitly included in the table of values for vane pitch targets, T, which is entered with control position, C. It will be obvious to those skilled in the art that the watercraft speed, S, may be incorporated in the table of values for vane pitch targets, T, to improve performance, as discussed above. The microcontroller program then sets the vane pitch to the vane pitch target, T. This method presumes that the engine is operating at peak performance. The operator preference input **157** may be used to reduce the shaft power demand when the engine is out of tune or laboring. It may also be used to choose between low-speed performance and fuel economy, as is common in automotive power transmission. This preference factor is "0" in the control equations of FIG. **13**B. This operation may be similar to trimming the propeller pitch in an airplane. After the pump impeller vane **57** pitch is adjusted, control passes back to the top of FIG. **12**. This method requires no flow input to and no power calculations by the microcontroller **140**. It is particularly useful for legacy diesel engines at sea level. The microcontroller **140** sets the pump impeller vane **57** pitch by outputting to the impeller vane control module **158**, as described above. Control passes back to the top of FIG. **12**.

FIG. **13**C uses the output from the combustion microcontroller on the engine **21** as the best measure of the power most efficiently supplied by the engine at the current shaft turn rate in rpm, N. This control method is well understood in the automotive field of art, as it is widely used to determine shift points in automatic transmission controllers and to control continuously variable transmissions. In effect, the microcontroller **140** is programmed to act as a slave to the engine **21** combustion microcontroller, which dynamically determines the most efficient power demand for the motor based on a complex set of environmental and combustion variables, as is well understood in the art. In each control cycle the microcontroller **140** incrementally increases, decreases, or leaves unchanged the impeller pitch, based solely on the input from the combustion control computer **23**. This method requires no pump head input, no rpm input, no system flow input and no positional feedback for controlling the impeller vane pitch to match shaft power demand to the most efficient power supplied by the motor. After the pump impeller vane **57** pitch is adjusted, control passes back to the top of FIG. **12**.

Note that the method of FIG. **13**C requires neither flow measurement nor vane positional feedback, because it incorporates control feedback from the engine combustion control computer. When the direct flow measurement means is not required in the "Increment Vane Angle" control sequence, as in FIG. **13**C or when flow is estimated by pump impeller vane **57** pitch, it can be compared with the positional feedback from the pump impeller vane **57** angle to monitor the operating efficiency of the marine jet propulsion system **20**. If the calculated flow is lower than that indicated by the pump impeller vane **57** position, the likely cause is debris on the inlet grate vanes **42**, pump impeller vanes **57**, and/or stator vanes **71**. The microcontroller **140** may be programmed to alert the operator by an alarm means, such as a light or a horn.

With any of these combinations, the microcontroller **140** is programmed to adjust the pitch of the pump impeller vanes **57** through the hydraulic impeller vane control module **158**, so that the spherical pump **50** shaft power demand is made to approximate the most efficient power supplied by the engine **21** at the current shaft turn rate in rpm, N.

The functional advantages of this program of operation are described more fully below.

When the operator switches the ignition on, the microcontroller **140** outputs to the vane control module **158** to set the pump impeller vane **57** pitch to the position indicated by the shaft position encoder **143** on the single handle control **141**, which is generally zero pitch for neutral pump flow. The operator then starts the engine **21**, which idles at for example about 1,000 rpm. In response to the movement of the single handle control **141** in an approximately +/−10-degree dead band range, the microcontroller **140** adjusts the pump impeller vane **57** angle to continuously vary the forward, neutral, and reverse thrust of the marine jet propulsion system **20**, as detailed in FIG. **12** and the associated discussion above. Moving the handle through the straight up position results in a scissoring action between the pump impeller vanes **57**, which cleans debris off the leading edges of the vanes. Moving the handle further back results in reverse pitch and in reversing the pump flow, which back flushes the system for trash removal. Such reverse pump flow also produces an effective reverse thrust. Moving the handle **141** back further increases engine rpm and consequently the magnitude of the reverse thrust, just as is common in propeller driven boats.

This operation provides smooth, quick shifting from forward to reverse thrust for low speed maneuvering, because there is no change of shaft direction in transitioning from forward to reverse or from reverse to forward. The swim platform and power trim function, which are both popular on recreational boats of the prior art, may be used to reduce vortex formation and cavitation in the reverse thrust mode, as shown in FIGS. **10**A and **10**B. The operator independently controls power trim, just as in stem-drive and outboard propulsion systems of the prior art.

The steering wheel controls the action of the nozzle steering vanes **84** through the range of motion shown in FIGS. **8**A and **8**B through the hydraulic helm and hydraulic circuits shown in FIG. **9** and described above. Turning the wheel **124** of FIG. **9** to the right results in the left position of the nozzle steering vanes **84** in FIG. **8**B. The resulting directional change of momentum of the system flow creates a reaction steering force to the right along the transom **95** (shown in FIG. **1**), so that when the wheel is turned to the left as in FIG. **8**B, the transom **95** is driven to the right. The reaction force resulting from reverse system flow is in the same direction as with forward flow, so that the transom **95** always moves to the right when the wheel **124** is turned to the left. Such reaction forces are well understood in the hydraulic art.

In the forward thrust mode of operation, the variable steering nozzle **80** is controlled to maintain the most efficient head on the variable pitch spherical impeller pump **50** for the current shaft rpm, as is described in my the U.S. Pat. No. 5,679,035 and as further described above. It is well understood in the art that efficiency is fairly constant over a broad range of impeller pitch. The resulting flow through the pump **50** well understood to be a function of the impeller pitch. At low cruising speeds, the power demanded to propel the boat

19 at constant speed is low. To match the power demanded by the pump 50 to the most efficient rpm of the engine 21, the microcontroller 140 sets the pump impeller pitch near maximum, which is the state in which it is passed from the low speed maneuvering mode to the forward mode. To maintain the pump 50 close to its most efficient operating conditions, the microcontroller 140 opens the variable steering nozzle 80 to maximum. In addition to maintaining engine efficiency, this control strategy has the fortunate consequence of providing maximum flow at low speeds for maximum propulsion efficiency. The flow through the maximum nozzle opening also occurs at the lowest possible velocity. Thus, motor efficiency, pump efficiency, and flow rate efficiency are all close to optimum, and wake turbulence is minimized.

When the system 20 is under full acceleration, as in pulling up a water skier, the microcontroller 140 will reduce the pitch on the pump impeller vanes 57 to match the pump's shaft power demand to the engine's 21 most efficient power supply at the instantaneous shaft rpm. The control system will also reduce the nozzle discharge opening area 89 to maintain the most efficient head on the pump 50 for its current rpm.

When the boat reaches steady wakeboarding speed in the approximate range of 15 to 20 mph, the pump impeller vanes 57 are close to full pitch to reduce the engine rpm to the most efficient operating point along the line 163 of FIG. 11. The variable steering nozzle 80 is close to being fully open to maintain the most efficient pump head at the relatively low shaft rpm. A further advantage is that the variable inlet duct opening is near maximum due to the high flow, which results in no losses from the conversion of inlet entrance velocity to pressure at the pump inlet. This also again has the fortunate consequence of providing close to maximum water system flow, Q, at this relatively low boat speed for maximum propulsion efficiency, which also results in minimum nozzle velocity through the large nozzle effective nozzle discharge opening area 89 and consequently in minimum wake turbulence. The system 20 rpm is further reduced relative to systems of the prior art by this higher propulsion efficiency, which requires less shaft power and consequently lower shaft rpm to maintain the boat speed. Thus, engine 21 efficiency, inlet duct 30 efficiency, spherical pump 50 efficiency, and inlet entrance opening area 32 flow rate efficiency are all close to optimum, and wake turbulence is minimized.

When the boat reaches steady water skiing speed at approximately 30 mph, the recovery of pressure in the inlet duct 30 has increased, so the microcontroller 140 has made a slight reduction in effective nozzle discharge opening area 89 to maintain the most efficient system flow rate, Q, through inlet entrance opening 32 and head, h, on the spherical pump 50. The power required to maintain this higher boat speed is also higher, so the engine 21 must operate at a higher rpm to supply the necessary power. The most efficient spherical pump 50 head rises as the square of the shaft turn rate in rpm, N. Higher engine 21 rpm causes the microcontroller 140 to reduce the pitch on the pump impeller vanes 57, which reduces the most efficient system flow rate, Q, through inlet entrance opening 32. The nozzle discharge opening area 89 head-affinity control function implicitly accounts for higher inlet head at this boat speed, higher spherical pump 50 head at the higher shaft rpm, and the reduced system flow rate, Q, through inlet entrance opening 32 resulting from reduced pitch on the pump impeller vanes 57. As a result of all these factors, the nozzle discharge opening area 89 is reduced and the nozzle velocity relative to the boat velocity is increased. However, the nozzle velocity relative to the water 29 surface is reduced by the increased boat speed, so that the velocity of the jet relative to the water surface has only slightly increased. Wake turbulence is thereby only slightly increased, and the use of longer towropes at this higher boat speed makes wake turbulence less critical, since it has more time to dissipate before the skier reaches it.

Further increases in boat speed demand increased engine 21 power, which the engine 21 can only supply at higher rpm. The microcontroller 140 reduces the pitch on the pump impeller vanes 57 to maintain efficient engine 21 operation at the higher rpm. Reduced pitch on the pump impeller vanes 57 requires a commensurate reduction in nozzle discharge opening area 89. Spherical pump 50 head is rising as the square of the engine 21 rpm. Inlet 30 head is rising as the square of the boat speed. The increasing spherical pump 50 rpm, the reducing pump impeller vane 57 pitch, and the higher inlet 30 pressure are all factors, which will result in the microcontroller 140 reducing the nozzle discharge opening area 89 to maintain peak spherical pump 50 efficiency. Hence, nozzle discharge opening area 89 is reduced with increasing rapidity as boat speed increases as a natural consequence of the microcontroller 140 operation, until minimum nozzle discharge opening area 89 is reached at the top design speed of the system 20. The minimum nozzle discharge opening area 89 at top speed is also ideal for reducing the system flow rate 32, hence improving propulsion efficiency at the higher speed.

In compliance with the statute, the invention, described herein, has been described in language more or less specific as to structural features. It should be understood, however, the invention is not limited to the specific features shown, since the means and construction shown comprised only the preferred embodiments for putting the invention into effect. The invention is, therefore, claimed in any of its forms or modifications within the legitimate and valid scope of the amended claims, appropriately interpreted in accordance with the doctrine of equivalents.

The invention claimed is:

1. An improved water jet propulsion apparatus for a jet propelled watercraft, the water jet propulsion apparatus comprising:

a) a variable-pitch propeller water pump structured for operating as a continuously variable power transmission for a motor external to the pump, the pump comprising a plurality of vanes and a mechanism structured for adjusting a pitch on the vanes in such manner that water flow and pump power demand vary as a function of the pitch of the vanes;

b) a mechanism structured for coupling the pump to the external motor for driving the pump; and

a spherical impeller having the plurality of vanes, the spherical impeller residing within a substantially spherical housing structured to maintain close fits with tips of the vanes over a range of vane pitch.

2. The apparatus of claim 1, further comprising a spherical hub that is concentric to the spherical housing, and a substantially uniform water flow space between the spherical hub and the spherical housing.

3. The apparatus of claim 1, further comprising a vane pitch control mechanism coupled to the pump and structured for controlling the vane pitch within the range of vane pitch.

4. The apparatus of claim 3, further comprising a microcontroller coupled for controlling the vane pitch control mechanism.

5. An improved water jet propulsion apparatus for a jet propelled watercraft, the water jet propulsion apparatus comprising:

a) a variable-pitch propeller water pump structured for operating as a continuously variable power transmission for a motor external to the pump, the pump comprising a plurality of vanes and a mechanism structured for adjusting a pitch on the vanes in such manner that water flow and pump power demand vary as a function of the pitch of the vanes;

b) a mechanism structured for coupling the pump to the external motor for driving the pump; and

wherein an outlet of the pump is coupled to an inlet of a variable orifice discharge nozzle, and an orifice control mechanism is coupled for controlling an orifice of the variable orifice discharge nozzle.

6. The apparatus of claim **5** wherein the orifice of the nozzle is further controlled to maintain a head affinity relationship of the pump.

7. An improved water jet propulsion system for a jet propelled watercraft, the water jet propulsion apparatus comprising:

a) water pump having a variable pitch impeller and being structured for being coupled to an external driver;

b) an inlet duct structured for receiving water from an external body of water and directing received water to the water pump, the inlet duct having a variable inlet orifice and an outlet orifice structured for being coupled to an inlet orifice of the water pump; and

c) a discharge nozzle structured for receiving water from the water pump and discharging received water from a variable discharge orifice.

8. The system of claim **7**, further comprising a pitch control mechanism structured for controlling a pitch of the variable pitch impeller of the pump to provide variable forward, neutral and reverse flows of received water.

9. The system of claim **7** wherein the variable pitch impeller of the pump is further structured for operating in zero and reverse pitch.

10. The system of claim **7** wherein leading and trailing edges of vanes of the variable pitch impeller are further structured to provide a cutting action therebetween at zero pitch.

11. The system of claim **7** wherein the variable pitch impeller further comprises a spherical impeller, and wherein the water pump further comprises a substantially spherical housing structured to maintain close fits with the spherical impeller over a range of vane pitch.

12. The system of claim **7** wherein the variable inlet orifice of the inlet duct further comprises a slide that is structured for reducing the variable inlet orifice and is further structured to elongate an enclosed tunnel between the variable inlet orifice and the outlet orifice thereof.

13. The system of claim **7**, further comprising a watercraft having the water pump, inlet duct and discharge nozzle installed therein, the variable inlet orifice of the inlet duct being coupled to a surface of the watercraft in a position for receiving water from an external body of water with the outlet orifice of the inlet duct being coupled to the inlet orifice of the water pump, and the discharge nozzle being coupled for receiving water from the water pump with the variable discharge orifice being coupled to a surface of the watercraft in a position for discharging therefrom water received from the water pump;

and an external driver coupled to drive the water pump.

14. A method of water jet propulsion for propelling a watercraft, the method comprising:

a) in a water pump, providing a variable pitch impeller;

b) coupling an inlet duct to the water pump for receiving water into an inlet orifice thereof and directing received water to an inlet orifice of the water pump;

c) coupling a discharge nozzle to the water pump for receiving water from the water pump and discharging the received water from a discharge orifice thereof; and

d) varying at least one of the inlet orifice of the inlet duct and the discharge orifice of the discharge nozzle.

15. The method claim **14** wherein the method further comprises varying a pitch of the variable pitch impeller between different forward, zero and reverse pitches.

16. The method claim **14** wherein the method further comprises controlling a pitch of the variable pitch impeller in combination with controlling one of a variable inlet orifice of the inlet duct, and a variable discharge orifice of the discharge nozzle.

17. The method claim **14** wherein the method further comprises:

coupling the inlet orifice of the inlet duct to a surface of a watercraft in a position for receiving water from an external body of water;

coupling the discharge orifice of the discharge nozzle to a surface of the watercraft in a position for discharging therefrom received water; and

coupling to the water pump an external power supply for driving the water pump.

18. An improved water jet propulsion system for a jet propelled watercraft, the water jet propulsion apparatus comprising:

a) water pump being structured for being coupled to an external driver;

b) a inlet duct structured for receiving water from an external body of water and directing received water to the water pump, the inlet duct having an elongated inlet tunnel between a variable inlet orifice and an outlet orifice structured for being coupled to an inlet orifice of the water pump;

c) a grate structure internal of the inlet duct and positioned within the elongated inlet tunnel; and

d) a discharge nozzle structured for receiving water from the water pump and discharging received water from a variable discharge orifice.

19. A jet propelled watercraft, comprising:

a watercraft;

a water pump having a variable pitch impeller;

a power supply coupled for driving the water pump;

an inlet duct having a variable inlet orifice coupled to a first surface of the watercraft in a position for receiving water and directing received water to an inlet orifice of the water pump;

a discharge nozzle coupled to the water pump for receiving water from the water pump, and coupled to a second surface of the watercraft in a position for discharging the received water from a discharge orifice thereof.

20. The watercraft of claim **19** wherein the variable pitch impeller further comprises a spherical impeller, and wherein the water pump further comprises a substantially spherical housing structured to maintain close fits with the spherical impeller over a range of vane pitch.

21. The watercraft of claim **19** wherein the discharge orifice further comprises a variable discharge orifice.

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