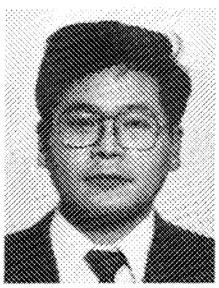
LIQUEFIED NATURAL GAS PUMP WITH HYDROSTATIC JOURNAL BEARINGS

by
Tatsuo Katayama
Senior Engineer
Plant Construction Department
Tokyo Gas Company, Limited
Tokyo, Japan
and
Akihisa Okada
Senior Engineer
Pump System Design Department, Tsuchiura Works
Hitachi, Limited
Ibaraki, Japan



Mr. Tatsuo Katayama is a Senior Engineer in the Plant Construction Department at Tokyo Gas Company, Limited, in Tokyo, Japan.

Since he joined Tokyo Gas, he has been involving in construction, improvement and maintenance of equipment in LNG terminals, such as LNG storage tanks, vaporizers, pipelines, etc.

He graduated from a technical high school in Japan in 1972.

Mr. Akihisa Okada is a Senior Engineer in the Pump System Design Department, for Tsuchiura Works, of Hitachi, Limited, in Ibaraki, Japan.

Since he joined Hitachi, Ltd., he has been involved in the development of turbo pumps, including an LNG pump project. He specializes in tribology such as journal bearings in low viscosity liquid, mechanical seal equipment.

He received a Master's degree in engineering from Tokyo Institute of Technology (1979).

ABSTRACT

In view of the inflammability of LNG (liquefied natural gas), submerged type pumps are normally used to protect against leaks. The overhaul interval of an LNG pump, therefore, is subject to the life of its ball bearings, because LNG is an extremely low temperature, low viscosity and low boiling point liquid. The use of LNG to fulfill the extremely demanding functions of lubrication and cooling of ball bearings results in substantial reduction in the bearings' maximum load tolerance.

The limits inherent in the ball bearings, consequently, will make it difficult to extend the life of the bearings beyond that currently achieved, if the bearing system itself depends on the use of conventional ball bearings. Since the service life of a ball bearing is not necessarily constant, there is a possibility of sudden breakdown. Thus, new LNG pumps were developed using a form of bearing for which service lifetime is not of primary importance, though it nevertheless has a long service life and high reliability, and is small and light. Details are given later. Two types of the static bearings were developed: a hydrostatic slide bearing and a hydrostatic guide bearing, incorporating a hydrostatic journal bearing surrounding the ball bearing.

The hydrostatic slide bearing supports the shaft by virtue of the film pressure of the LNG discharged from the pump itself. Therefore, the shaft and the bearing do not come into direct contact while the pump is in operation. Although the hydrostatic guide bearing supports the shaft with ball bearings in the usual way, the buffer effect of LNG in the hydrostatic guide section substantially reduces the radial load of the ball bearings.

A hydrostatic journal bearing does not bear any of the thrust load. A structure has, therefore, been developed in which a disk-piston-type thrust balancing system can be used to enable self balancing of the shaft thrust load, thereby reducing the thrust load to zero. The downstream side of the disk piston is designed to allow a variable clearance whereby, when it is at its maximum, the total flow area upward thrust force exceeds the downward thrust force. In this way, the two thrust forces always balance each other automatically.

A high lift impeller was designed, the number of operational stages was reduced, and, radial diffusers were used to enable the use of shorter shafts. For the stage housing, a piston ring type sealing is used, a single layer cylinder casing method was developed that enables a multistep seal provided by the piston rings at each stage. This allowed the pump diameter to be reduced. The use of hydrostatic slide bearings (hydrostatic guide bearings), a thrust balancing mechanism, and the reduction of shaft length has enabled substantial extensions to be made to the overhaul interval of the pump.

Pumps with hydrostatic slide bearings	:	100,000 hours or more
Pumps with hydrostatic guide bearings	:	50,000 hours or more

The use of hydrostatic slide bearings results in some reduction in the shaft bearing performance, due to wear of the face of the bearing. Measurement of the amount of wear, however, enables the remaining life of the bearing to be predicted, and this in turn facilitates cost reductions in terms of the failure diagnosis system employed. Furthermore, since there is little likelihood of sudden failure, maintenance schedules can be organized in a systematic way.

The use of a high lift impeller and a single wall cylindrical radial diffuser has enabled the production of a compact, lighter pump unit, which is easy to disassemble and reassemble. This technology is applicable to pumps for other liquefied gas, for example, LPG, LEG (liquefied ethylene gas), etc.

INTRODUCTION

Background of the Development

Natural gas is transformed into liquefied natural gas (LNG) through liquefaction at an extremely low temperature (−162°C). In this liquefied form, its volume is reduced to one-600th of its gaseous volume, therefore, it can be more efficiently transported by tanker. LNG is used as a material by city gas companies and as fuel by electric power companies after vaporization warming with sea water. Submersible pumps are used to transport LNG. As a result, the bearings must be lubricated and cooled by the LNG. LNG is marked by a low viscosity (about one-tenth that of water), low boiling point (if its temperature rises around 10°C, LNG vaporizes, causing a dry operation), and a low specific heat. Because of these properties, it has been regarded as unsuitable to lubricate slide bearings. Conventional LNG pumps have all employed ball bearings made of martensite stainless steel and lubricated with LNG. Unlike the case of those lubricated with oil, in ball bearings lubricated with LNG, the ball and the race roll in a state of solid contact, and abrasion is, therefore, unavoidable. Usually, the bearing life is determined by the abrasion on the rolling surface. Attempts have consequently been made to curtail abrasion by using phenol resin impregnated with a solid lubricant on the ball bearing retainer or by using a Teflon® retainer. Tokyo Gas has been working to improve the reliability of its LNG pumps since 1969, when LNG was first used as a raw material for city gas in Japan. Except for startups, most LNG pump failures have involved their ball bearings, the primary cause being imbalanced thrust. In response, the authors launched efforts to lengthen the service life of ball bearings by improving the thrust balance and the ball bearing design. These efforts dramatically increased the average time between overhaul inspections, from approximately 1,500 hr in 1969 to 15,000 hr in 1990. A bearing failure diagnosis system was developed that focuses on the changes in specific frequencies generated by the ball bearings. This system has drastically reduced the number of unscheduled shutdowns.

However, ball bearings have some inherent problems. When minute defects arise on the rolling surface, they can spread in a very short time and cause damage to the balls or other components. As such, early detection of defects calls for diagnosis at a high frequency. In addition, it is difficult to estimate the remaining service life of ball bearings. In view of these circumstances, it was decided to embark upon development of new types of LNG pumps that would feature an improved reliability and reduced maintenance costs.

Course of Development

Initially, LNG pumps were plagued with a rash of problems involving bearings and other components. This situation motivated a series of studies aimed at improving pump reliability. In 1979, Tokyo Gas initiated the actual work by arranging a contract for joint development with Hitachi, Limited.

The first phase consisted of the development of pumps with hydrostatic guide bearings to reduce the radial load. A prototype pump was constructed and tested with liquid nitrogen and LNG. This phase of the project began in 1979 and was completed in 1984.

The second phase consisted of the development of hydrostatic slide bearing for the prototype pump to achieving a longer operating time than the hydrostatic guide bearings offered. It included elementary tests, using a prototype pump in liquid nitrogen and LNG. This phase began in 1986 and was completed in 1990.

LNG PUMPS

The flow of major facilities at LNG receiving terminals is depicted in Figure 1. The LNG pumps are used to pump LNG from storage tanks to vaporizers, transport LNG between storage tanks, and circulate LNG within suction gas coolers. Therefore, they play an extremely important role in terminal operation.

There are many LNG pumps installed at LNG terminals; as of the end of 1990, two Tokyo Gas terminals had a combined total of 100 pumps. Since the pumps are so numerous, pump failures account for many facility downtimes and maintenance costs. Also, these pumps are the only type of rotating equipment that handles LNG, a flammable liquid. Overhaul inspections, consequently, involve dangerous work such as LNG purging. The basic requirements for LNG pumps are, therefore, a low incidence of sudden failures, maintainability, and long intervals between overhaul inspections. These requirements would improve reliability, reduce repair costs, and enhance safety.

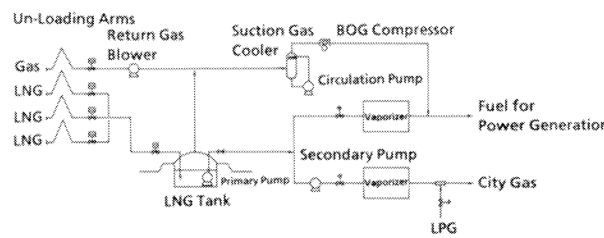


Figure 1. Flow Chart of Major Facilities at LNG Receiving Terminal.

As shown in Table 1, there are two basic types of LNG pump installations: on ground and in tank. In the case of the latter, the pump is installed in the outer casing called the pump barrel. Here, the pump can be extracted by means of a foot valve attached to the bottom end of the barrel and a hoist installed on the tank roof. The configuration is shown in Figure 2 of the "in tank" pump in an LNG storage tank.

Table 1. Number of Liquefied Gas Pumps Installed at LNG Terminals.

Terminal	Application	Type of Installation	Type of Bearing	Number of Pumps
Negishi	Primary Pump	On-ground	Ball Bearing	13
		In-tank		9
	Transport Pump			1
	Secondary Pump	On-ground		7
	Pump for Cryogenic Power Generation		Hydrostatic Guide Bearing	2
			Total	32
Sodegaura	Primary Pump	On-ground	Ball Bearing	9
		In-tank		35
	Transport Pump			4
	Secondary Pump	On-ground		9
			Hydrostatic Guide Bearing	1
			Hydrostatic Slide Bearing	1
	Drain Pump		Ball Bearing	4
			Hydrostatic Guide Bearing	2
			Total	68
			Grand Total	100

The location and general design of the hydrostatic journal bearing that resulted from the LNG pump development program is shown in Figures 3 and 4. The notable design features of these pumps are as follows.

• The pump and motor driver are completely submerged in the LNG to eliminate leakage from seals to the atmosphere. LNG is

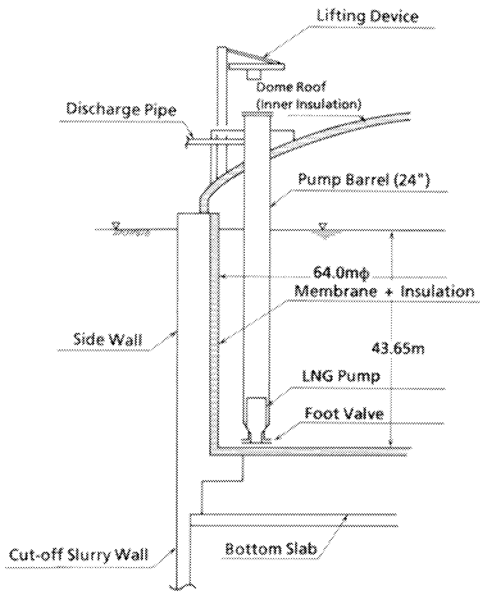


Figure 2. Configuration of LNG Inground Storage Tank and Its Pump Barrel System.

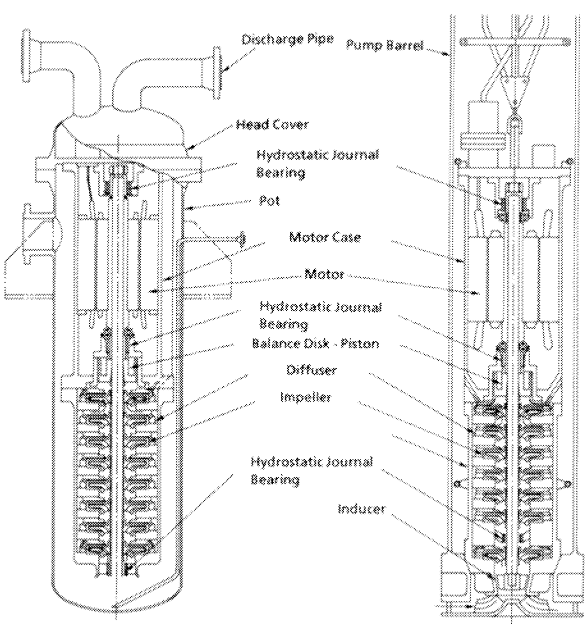


Figure 3. Hydrostatic Journal Bearing LNG Pumps.

perfectly suited for this type of motor, since it is not corrosive and is an insulating liquid. Insulation needs to be applied only between phases, and the conductor copper may be in contact with the LNG.

• The major components are a centrifugal pump with a vertical multistage diffuser and a three-phase squirrel cage motor. The LNG is drawn in from below and pumped out from the discharge nozzle on the top of the pump barrel cover.

• The bearings supporting the rotor are the most crucial determinants of pump service life and reliability. The pump in question employs two types of highly reliable hydrostatic journal bearings. Hydrostatic journal bearings (guide and slide types) are compared in Figure 4. In both case, the pumped fluid is supplied to the bearing pocket, providing the liquid film that supports the shaft. More specifically, in the guide type bearings, the hydrostatic bearings are installed on the outer periphery of conventional ball bearings and support their flotation. Slide type bearings are not installed with ball bearings and the shaft flotation is directly supported by the liquid film. It should be noted that pumps with hydrostatic slide bearings must be equipped with ball type thrust bearings below the motor. The flow of fluid is shown in Figure 5 for bearing lubrication, bearing cooling and motor cooling.

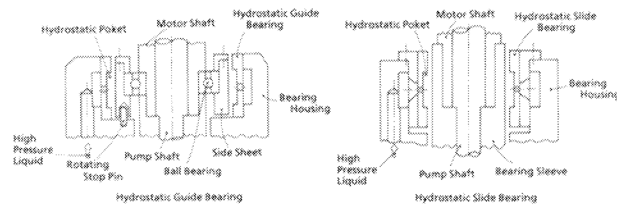


Figure 4. Hydrostatic Guide and Hydrostatic Slide Bearing.

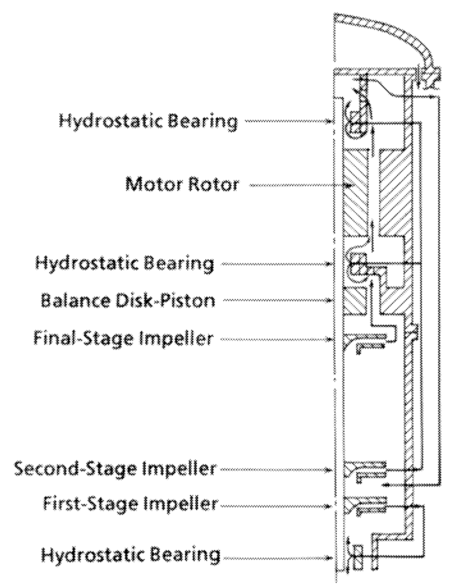


Figure 5. Flow of Bearing Lubrication, Bearing Cooling and Motor Cooling.

• Throughout the pump curve, downward thrust induced by the impellers is counter balanced by the upward thrust force generated with a balance disk-piston device.

• The diffuser, which guides the high speed flow from the impeller and transforms it into static pressure, is of the radial type. This is different from conventional vertical turbine pumps, which employed diffusers of the axial type.

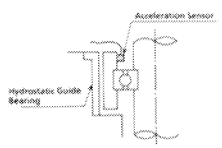
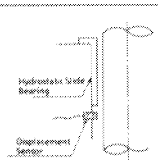
• Each pump stage has unique stainless steel sealing rings.

• The major material for impellers, diffusers, and certain other components is aluminum alloy. Stainless steel was selected as the material for hydrostatic guide bearings, since they are installed on the outer periphery of the ball bearings, and are not in contact with the shaft. For the hydrostatic slide bearings, however, a special type of carbon with a low friction coefficient and high resistance to abrasion was selected, since contact with the shaft is conceivable.

• The pumps are bipolar with a speed of 3000 (50 Hz) or 3600 rpm (60 Hz), and are designed to be safe at speeds up to 20 percent faster than these standard speeds.

• The pump diagnostic system varies depending on the type of pump. As mentioned above, LNG is not a corrosive liquid, and there is, consequently, little danger of a deterioration of the pump or motor materials. As such, the main determinant of bearing life is the interval of pump overhaul inspection. The system of diagnosis for pumps with hydrostatic guide bearings and hydrostatic slide bearings is reflected in Table 2. Pumps with hydrostatic guide bearings are diagnosed like the conventional ball bearing types. An accelerometer is installed on the bearing housing and sends ball bearing vibration signals to the control center room, monitoring equipment. This system is shown in Figure 6. The diagnosis procedure is (a) calculation of the general acceleration value and comparison to a prescribed vibration level setpoint, and (b) analysis of acceleration frequencies to determination if the acceleration (at the specified frequency) exceeds the prescribed frequency setpoint.

Table 2. System of Diagnosis for Pumps with Hydrostatic Guide Bearings and Hydrostatic Slide Bearings.

Type of Bearing	Hydrostatic Guide Bearing	Hydrostatic Slide Bearing
Type of Sensor	Piezo-Electric Accelerometer	Eddy Current Displacement-meter
Position of Sensor		
Purpose of Measurement	To measure vibration of ball bearings to catch the condition of their surface and/or their failures.	To measure change of shaft clearance to catch the degree of wear of the Slide Bearig.
Measurement	Acceleration (~ 10 kHz)	Displacement of the shaft (~ 500 Hz)

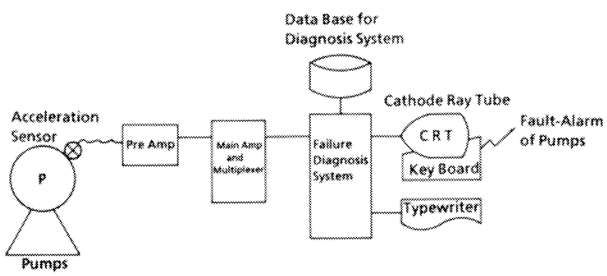


Figure 6. Diagnostic System for Hydrostatic Guide Bearing.

The vibration level judged to constitute failure is determined based on experiment and experience. It's typical value is, e.g., 1160 in/s^2.

On the other hand, pumps with hydrostatic slide bearings are diagnosed by means of noncontact eddy current probes installed at the bearing. These sensors send vibration amplitude signals to a displacement meter installed outside the pump. The meter displays these signals on a readout and/or records them in chart form. The configuration of this system is shown in Figure 7. The hydrostatic slide bearing clearance is measured when the pump is stopped. This is because, when the pump is stopped, the discharge pressure is lost, causing the bearings to lose their supporting force. As a result, the shaft rotates near the inner surface of the bearing, and the bearing wear can be calculated by measuring the bearing clearance. An example of measurement is shown in Figure 8. In contrast to hydrostatic guide bearings, for which the wear is automatically measured once a day, the frequency of wear measurement for hydrostatic slide bearings is approximately once every three months. This low frequency reflects the slow wear rate in the slide bearings, which come into contact with the shaft, only during startup and

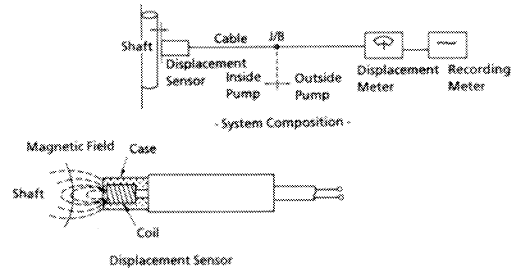


Figure 7. Noncontact Eddy Current Displacement System.

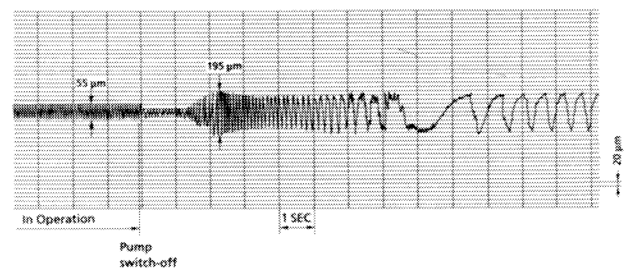


Figure 8. Measurement Example of Noncontact Eddy Current Displacement.

shutdown, and eliminates the possibility of sudden failure experienced with ball bearing damage.

DESIGN OF HYDROSTATIC JOURNAL BEARINGS

Hydrostatic journal bearings provide shaft support based on supply of external fluid under high pressure to the bearing periphery. They are the most suitable type of bearing for submersion in liquids with a low viscosity, such as LNG. In hydrostatic journal bearings, the rigidity of the lubricating film (K) has no direct connection with the viscosity of the lubricant or rotational speed, and can be derived by the following equation.

$$K \propto (D\,L\,P_s / h) \times f(P_o / P_s) \qquad (1)$$

Here:
- D : inner diameter of bearing
- L : bearing length
- h : bearing clearance
- P_s : supply pressure
- P_o : pocket pressure
- f : orifice ratio (P_o / P_s) function

By contrast, the corresponding rigidity (K) for dynamic pressure bearings is obtained by the following equation.

$$K \propto (D\,L\,N\,\eta / h) \times (D / h)^2 \times f(\varepsilon) \qquad (2)$$

Here:
- N : rotational speed
- η : coefficient of viscosity
- f : eccentricity (ε) function

As shown in Equation (2), the film rigidity for hydrodynamic pressure bearings depends on the viscosity of the lubricant, and such bearings are consequently unsuitable for use with LNG, which has a low viscosity. An adequate lubricating film could not be formed, and there would be great danger of bearing failure due to shaft contact and frictional heat. However, as shown in Equation (1), the film rigidity of hydrostatic journal bearings does not depend on viscosity, but on the fluid pressure supplied to the bearings. As a result, they are well suited for use with LNG and other low viscosity fluids.

The configuration of hydrostatic journal bearings is shown in Figure 9. The inner surface of the bearings is equipped with pockets into which the high pressure lubrication fluid is introduced. As shown in Figure 10, when the shaft shifts from one side to the other, the differential pressure arising in the two opposing pockets restores the shaft to its proper orientation. This is the basic principle of the operation of hydrostatic journal bearings.

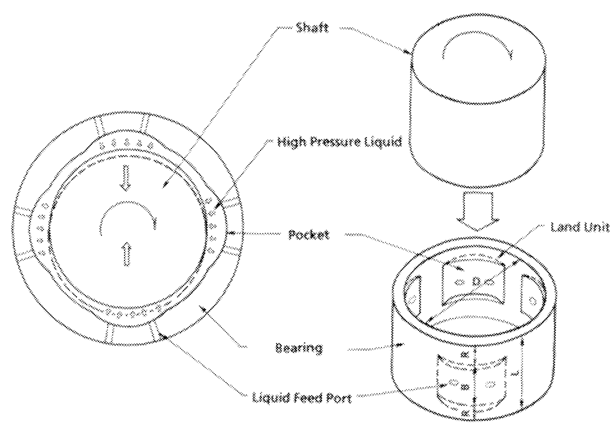


Figure 9. Configuration of Hydrostatic Slide Bearing.

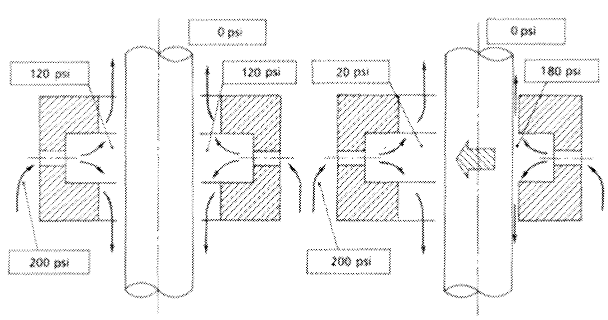


Figure 10. Principle of Hydrostatic Slide Bearing.

In Figure 10, the pocket pressure is shown to be half as great as the lubrication fluid pressure. If, however, the pocket pressure is higher and approaches the lubrication supply pressure, the amount of increase in the pocket pressure would be smaller. In such a case, it would be difficult to derive the increased pocket pressure needed to restore the shaft orientation from the pocket on the close clearance side. Conversely, if the pocket pressure is lower, it would be difficult to derive the lower pocket pressure needed to restore the shaft from the pocket on the larger clearance side. For example, to derive the force needed to center the shaft by varying the two opposing pocket pressures. As implied in Equation (1), inducing a great repulsion force for hydrostatic journal bearings calls for the selection of a proper orifice ratio, that is, a ratio that will provide a large $f(P_0/P_s)$. The rigidity of these bearings is determined by this ratio, a parameter that does not apply in hydrodynamic pressure bearings. The relationship is shown in Figure 11 between bearing load and shaft eccentricity when the lubrication fluid pressure is constant for a certain hydrostatic journal bearing. The selection of an improper orifice ratio can greatly reduce the rigidity. If correctly designed, hydrostatic journal bearings will provide shaft support based on the supply of lubricant fluid under high pressure and prevent shaft contact.

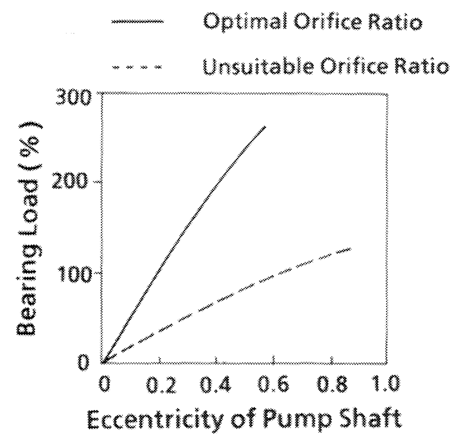


Figure 11. Relationship between Bearing Load and Eccentricity of Pump Shaft.

The optimum lubrication fluid orifice ratio must be found through experiments. The results of experiments provide the design basis for the bearing orifice ratio used with LNG pump hydrostatic journal bearings. It should be noted, however, that the pocket pressure (P_0) should be set a little on the high side for manufacture, so that the orifice ratio (f) reaches its maximum after a slight amount of initial wear. Therefore, care must be taken to make the orifice a little bigger than the optimal size.

In a radial bearing, the bearing clearance (h) is basically determined by the rotor design. As such, it may be desired to increase the land unit length to curtail the bearing flow quantity (Q), but such a step would decrease the pocket area (A) and lower the load weight (W). In bearings for LNG pumps, it may be desirable to lengthen the land unit to curtail change in the orifice ratio, even in the event of slight wear on the land unit arising from minute particles (ranging from five to several tens of microns in diameter) of iron powder and certain other substances contained in the LNG. However, if the land unit is made too large, the pocket area (A) will be lessened and the load weight (W) greatly reduced as a result.

The optimal values for such parameters as pocket depth, width, and number of fluid supply apertures also must be found empirically through experiments.

TESTING OF HYDROSTATIC JOURNAL BEARINGS

As described previously, the rigidity of hydrostatic journal bearings is largely determined by such factors as the pocket pressure, pocket array and pocket depth, width, and number of fluid supply apertures. To find the optimal values for these parameters, tests were conducted using bearing test stand equipment. More specifically, an external pump was used to feed water under high pressure to a submerged bearing, which was also subjected to a load by a hydraulic piston. This made it possible to determine the relationship between the load weight applied to the shaft and the shaft's eccentricity. The displacement of the shaft was measured by a noncontact sensor. The findings provided the basis for the completion of a hydrostatic journal bearing of the type shown in Figure 9. The features of this bearing are:

- The bearing is of the double orifice type, there being two fluid supply orifices for each pocket.
- The pockets are equipped with tapered grooves.

• The pocket length is shorter than usual.

Maintaining the pressure within the bearing on levels that are as uniform and high as possible enables the bearing to offer a high rigidity and excellent vibration control characteristics. Unlike hydrostatic journal bearings used for machine tools, these hydrostatic journal bearings deliver an adequate performance, even if the bearing clearance is wide or a certain amount of wear occurs during use.

The hydrostatic bearings used in the pump have inner diameters of 50 to 200 μm, and bearing clearances of 150-200 μm, which means that they are not easily affected by impurities in the LNG. Safety is further increased in the bearing shown in Figure 9 by an impurity extraction groove parallel with the shaft. In tank pumps cannot have filters installed at the inlet. However, on ground pumps are fitted with 100 μm mesh or similar filters as a safety feature. In the extremely unusual occurrence of a large piece of foreign matter, it is prevented from entering the pump.

From experience, the authors have found that bearings like this with wide clearances need a supply pressure of at least 50 psi, if they are to function as hydrostatic bearings. Consequently, the design assumes supply pressure of at least 50 psi and ensures that shaft vibration is kept to within approximately 1/3 of the size of the clearance.

For the bearing rolling element, a special carbon material was selected. It is impregnated with resin and contains such additives as molybdenum disulfide and certain other solid lubricants. The addition of solid lubricants curtails friction and wear even in liquids in which there is no supply of oxygen and an oxide film can therefore not be formed.

BENEFITS OF USE OF HYDROSTATIC JOURNAL BEARINGS

The newly developed pumps adopted a triple bearing structure in which the aforementioned hydrostatic journal bearings were installed above the motor, below the motor, and at the lower end of the pump. The use of hydrostatic journal bearings for LNG pumps can be expected to yield the following benefits.

Bearing Unit

In conventional LNG pumps, the entire radial load generated by the shaft vibration is imposed on the ball bearings. By contrast, in the newly developed hydrostatic guide bearings, the radial load on the ball bearings is greatly reduced by the buffer action of the fluid, since flotational support for the outer ring of the ball bearing is provided hydraulically, i.e., by fluid pressure. In the hydrostatic slide bearings, radial load is not even a factor, since flotational support for the shaft is provided by the discharge liquid film of the pump itself, there being no ball bearings. In addition, there is absolutely no abrasion during pump operation, since there is no contact between the shaft and the bearing. The influence of minute dust and other particles is also thought to be negligible. As a result, there is no risk of sudden failure. Furthermore, the contact between the shaft and bearing during startup and shutdown is restricted to an extremely short duration and small contact energy. The contact, consequently, is thought to entail almost no abrasion.

These improvements can enable an extension of the bearing life to several times that of the conventional ball bearing life.

Wearing Parts Other Than Bearings

With conventional LNG pumps, cylindrical bushes are attached to each stage (Figure 12) to curtail shaft vibration at the centrifugal pump unit. However, due to LNG's low viscosity, the shaft and bush rotate in a state of solid contact, and bush abrasion is, therefore, unavoidable. The bush abrasion, in turn, causes abrasion of the impeller sealing (e.g., wear rings). This abrasion is apt to progress in proportion to the amount of shaft vibration and to the shaft length. In some cases, it can make the bush and sealing life shorter than the bearing life. For the new pumps, hydrostatic slide bearings, which offer an excellent control of vibration, are attached to the lowest end of the pump. Radial diffusers are employed to enable a shortening of the required shaft length. These steps reduce the amount of shaft vibration at the centrifugal pump unit, affording prospects for an extension of the life of wearing parts to several times the conventional figure. Because there is little shaft vibration, there is also little radial load generated at the bearing below the motor, which means an additional contribution to extended bearing life.

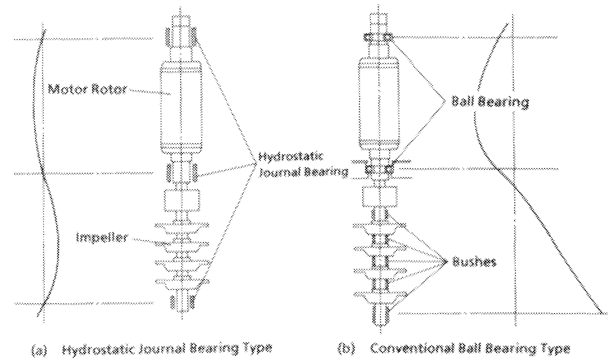


Figure 12. Vibration Mode of the Shaft.

SHAFT THRUST BALANCE EQUIPMENT

The shaft thrust balance equipment employs both a balance disk and balance piston to accurately balance the downward thrust induced by the impellers. When the pump is shut down, the rotor sags under its own weight, and the structure is designed so that an end rolling element gap automatically opens. When the pump is started up, the ball bearing moves upward along with the rotor, due to the discharge pressure, and continues to operate with a balanced gap. Even if the downward thrust of the impellers changes along with the flow quantity, the balance gap automatically changes as well, and maintains the balance by changing the upward thrust of the balance disk piston. As a result, there is very little thrust load imposed on the ball bearings. In addition, on the outer periphery of the ball bearing, there is a hydrostatic guide bearing that does not rotate, but moves freely in an axial direction. This bearing ensures a smooth upward and downward movement, alleviates the radial load, and extends the service life.

The following is an account of the principle behind the operation of the balance disk piston shown in to Figure 13. The status is shown in Figure 13(b) during normal pump operation. The downward impeller force (A) is balanced by the upward disk piston force (B). If, however, the rotor moves downward for some reason, the balance disk piston's rear balance gap enlarges, increasing the inflow of fluid discharge, as shown in Figure 13(a). As a result, the back pressure (B2) decreases, and the rotor automatically moves upward, returning to the status shown in Figure 13(b). Similarly, if the rotor moves upward for some reason, the balance disk piston's back balance gap contracts, decreasing the inflow of fluid discharge, as shown in Figure 13(c). As a result, the back pressure (B2) increases, and the rotor automatically moves downward, returning to the status shown in Figure 13(b). The structure is therefore designed so that the balance gap automatically changes to balance both thrusts.

In designing the balance disk piston, it is particularly vital to ascertain:

· The pressure distribution on the balance disk piston back surface.

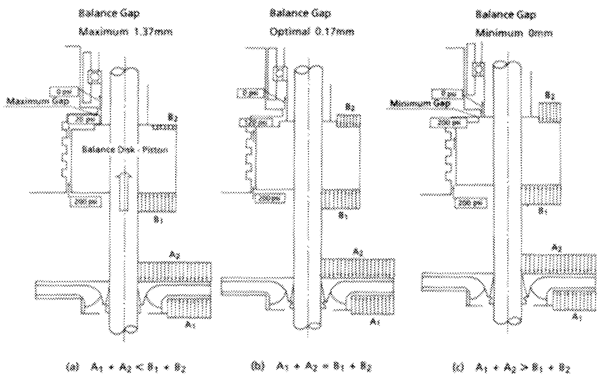


Figure 13. Principle of Balance Disk Piston.

· The disk piston flow quantity.

In Figure 13, the pressure around the balance disk piston is shown to be uniform in a radial direction. In reality, however, it exhibits the distribution shown in Figure 14. This is because the rotation of the disk piston induces a forced eddy current in the same direction in the fluid. Therefore, the pressure rises as one proceeds further on the periphery. As can be understood from Figure 14, due to the eddy flow, the pressure on the undersurface of the balance disk piston is lower, and that on its upper surface, higher, thus greatly reducing the balance disk piston's upward force. This poses a risk of imbalance with the downward force of the impellers. Generally, this pressure distribution can be calculated by the following equation, assuming the fluid between the balance disk piston and seat rotates at half the peripheral speed of the balance disk piston.

$$\begin{aligned}\Delta H &= (\Delta u^2 / 2\,g) + (\Delta u^2 / 2\,g) = \Delta u^2 / g = \omega^2\,\Delta R^2 / g \\ &= (\omega^2 / g) \times (R_o^{\,2} - R_i^{\,2}) \\ &= (1 / g) \times (\Omega / 2)^2 \times (R_o^{\,2} - R_i^{\,2}) \qquad (3)\end{aligned}$$

Here, ΔH : difference between pressure on the outer surface and that on the inner surface of the balance disk-piston

μ : rotational speed of the fluid

ω : rotational angular speed of the fluid

R_o : outer diameter of the balance disk-piston

R_i : inner diameter of the balance disk-piston

Ω : rotational angular speed of the balance disk-piston

g : gravitational acceleration

$(\Delta u^2 / 2\,g)$: pressure rise due to centrifugal force

$(\Delta u^2 / 2\,g)$: dynamic pressure rise due to rotation

However, it would be extremely dangerous to depend entirely on such calculated values and not ascertain the correct pressure distribution, since this could lead to inappropriate design. The quantity of flow on the outer periphery of the balance disk piston is influenced by such factors as this pressure distribution and the labyrinth form. The flow quantity also changes depending on the extent of the balance up. Failure to determine this flow quantity accurately could result in insufficient cooling of the motor on the downstream side and, by extension, vaporization (boiling) of the fluid within the motor.

Consequently, there is a need for the establishment of a design procedure that is based on determination of the items above through elementary testing. In addition, this procedure should provide for accurate balancing of the thrust load at any flow quantity and provide adequate the motor cooling.

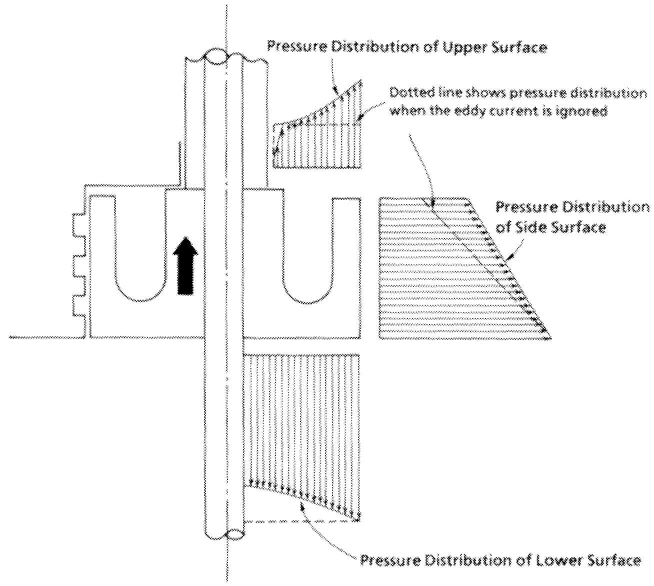


Figure 14. Pressure Distribution Around Balance Disk Piston.

To determine the pressure distribution and flow quantity around the balance disk piston accurately, a test unit was built with a full scale rotor. The thrust force on the impeller side was varied and measurements were taken for pressure distribution around the disk piston, flow quantity, and amount of balance gap at the different thrust force values.

The test revealed that, due to the strong eddy flow, the rear pressure was much higher than the anticipated design figure. Consequently, the disk piston's upward force was much lower than anticipated. Because the rear pressure was high, the flow quantity was lower than anticipated. The disk piston diameter had to be enlarged to increase the upward force. An improved balance disk piston was built according to these findings, and yielded favorable results in retesting. In short, the test findings provided the design procedures for very good balance at all flow rates during pump operation.

BENEFITS OF USE OF THE BALANCE DISK PISTON

Use of the aforementioned balance disk piston in LNG pumps can eliminate the imposition of a thrust load on the ball bearings in the normal range of pump operation (20 to 150 percent in terms of flow quantity).

As a result, ball bearing life in hydrostatic guide bearing pumps is determined entirely by the size of the radial load. In these pumps, this radial load is low, roughly less than 100 lb.

These pumps offer the elimination of the thrust load up to several hundred pounds produced by imperfect thrust mitigation mechanisms, and therefore, can dramatically lengthen ball bearing life. It is said that the abrasion based life of ball bearings lubricated with LNG basically corresponds with the fifth power of the load, and the life with reference to fatigue destruction of the rolling surface, which is a cause of sudden failure, to the cube of the load. As such, the benefits of reducing the thrust load to zero are very considerable indeed.

In hydrostatic slide bearing pumps, there is absolutely no thrust load during pump operation, and a thrust load is imposed on the auxiliary ball bearings for only a short time during startup. As a result, these auxiliary ball bearings are subject to virtually no abrasion. The bearing life is, therefore, determined entirely by the life of the hydrostatic slide bearings. The life of these bearings is thought to be considerably long, since they too undergo almost no abrasion, due to their lack of contact with the shaft during rotation.

PUMP DESIGN

As noted above, a radial diffuser was adopted for the LNG pumps. As compared to axial diffusers, this radial diffuser occupies less axial space, allowing the shaft length to be significantly reduced. Since the structure has an axial symmetry, it is more unlikely for a radial load to emerge in any part of the operational range. This short shaft configuration enhances the vibrational stability, reduces the bearing load, and lengthens the service life.

To reduce the pump diameter, a single wall, cylindrical casing was constructed. This casing is constructed of aluminum alloy stage case and each stage is sealed with stainless steel piston ring type sealing rings with a low thermal expansion. As shown in Figure 15, in conventional pumps, the diffuser has guide vanes for each stage with an outer casing. Several stages are stacked and the assembly is secured with more than 10 tie bolts. This construction, which is generally used for multistage centrifugal pumps, is difficult to maintain and results in increased weight.

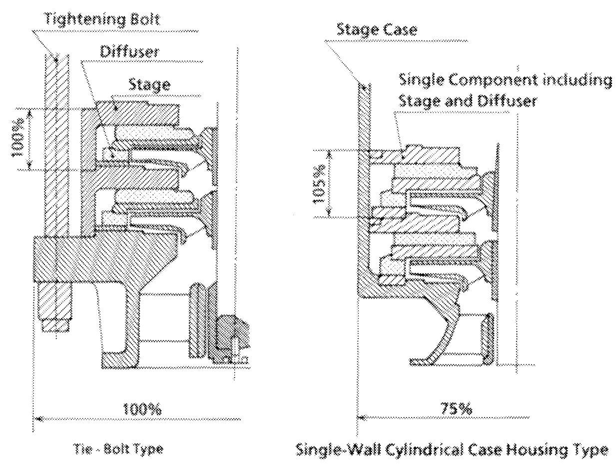


Figure 15. Configuration of Hydraulic Flow Unit.

For the LNG pumps, the newly developed hydraulic, or liquid end, casing was selected in place of this tie bolt configuration. The new design integrates the return blade and diffuser into a single component. These individual stage casings are then stacked and housed within a cylindrical case. Since there is no outer cylinder for each stage, the high pressure fluid from the latter stages would leak into the former stages through the clearance between the stage case and the stage gap, if nothing were done to prevent this. As shown in Figure 16, this design problem was solved by incorporating martensite stainless piston ring type steel sealing rings capable of providing a sealing action at low temperatures, because their coefficient of thermal expansion is twice that of aluminum alloy. Since there is a larger clearance between the sealing ring and casing during assembly at normal temperatures, the assembly does not entail the use of great force or special tools as required with tie bolts; assembly can be done by merely slipping the outer cylindrical casing over the stages and securing it with bolts. As a result, the time required for assembly was greatly reduced.

Conventionally, radial diffuser pumps have been characterized by a larger hydraulic casing diameter and a bigger pump outer casing than axial diffuser pumps. The development of this new hydraulic casing with cylindrical outer housing has reduced the

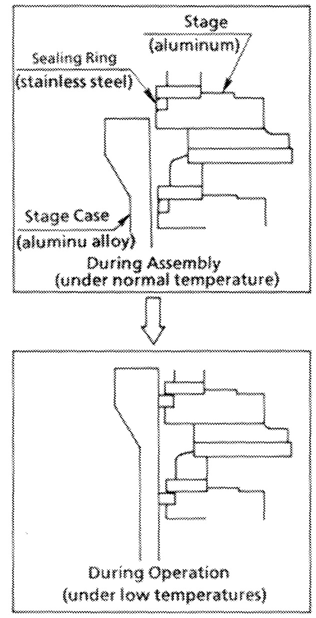


Figure 16. Sealing Structure of Single-Wall Cylindrical Case Housing.

diameter by about 25 percent, bringing it to a par with the diameter of axial diffusers. Since the shaft is short, the pump installation space and weight requirements have been reduced to values approaching their lower limits. While this structure therefore has some excellent features, it must be noted that the compressive load generated by the pump's operational pressure imposes a burden on the diffuser blade and the return blade. In large high pressure pumps, this force can exceed 100 tons. The following test was conducted to obtain empirical design data for a blade structure that could withstand such force. A stage of actual size was manufactured and equipped with numerous strain gauges. The stage was compressed to the point of destruction using a compression tester, and measurements of the stress were taken in the process. The strength of this maximal stress part was reinforced by increasing the wall thickness there.

BENEFITS OF USE OF THE SINGLE-WALL CYLINDRICAL CASE HOUSING TYPE HYDRAULIC FLOW UNIT

The newly developed pumps employ a low relative speed model for the impellers. Use of this model enables an increase in the impeller diameter and a decrease in the required number of stages. The new pumps use impellers with a diameter that is about 20 percent larger than that of conventional impellers, but have only about half the conventional number of stages. This relationship can be explained with the following equation. Since the head generated by the impellers is basically proportionate to the cube of the diameter,

$$n / n' = (D' / D)^3 = (D' / 1.2\ D')^3 = 0.58 - 0.5 \qquad (4)$$

Here, n : number of impeller stages for the newly developed pumps

D : impeller diameter stages for the newly developed pumps

n' : number of impeller stages for conventional pumps

D' : impeller diameter stages for conventional pumps

In short, part of the space in a diametrical orientation freed by adoption of the single-wall cylindrical case housing type hydraulic flow unit was used to lengthen the impeller diameter. This enabled a reduction of the required number of impeller stages and of the required shaft length. As noted under BENEFITS OF USE OF HYDROSTATIC JOURNAL BEARINGS, a shorter shaft length reduces the amount of shaft vibration, and therefore, extends the life of bearings and other wearing components. Because shaft flexure is proportionate to the fourth power of the shaft length, the halving of the number of stages represents an extremely great benefit.

TESTING OF PROTOTYPE PUMPS

As noted previously, two types of hydrostatic journal bearing pumps (guide and slide) were developed. For each type, a prototype pump was manufactured and subjected to various tests.

Hydrostatic Guide Bearing Pump Prototype.

Pump specifications

Discharge	410 gpm	(45 t/h)
Total head	2477 ft-LNG	(755 m-LNG)
Temperature	–260°F	(–160°C)
Specific gravity	0.465	
Motor	150 kW x 3,000 V	

Test items

- Tests using liquid nitrogen test equipment
  - Spin test
  - Performance test
  - Continuous operation test
- Field tests at LNG terminal
  - Insulation test
  - Spin test
  - Performance test
  - Continuous operation test

Test flow

The flow of tests using the liquid nitrogen test equipment is shown in Figure 17. The flow of the field tests at an LNG receiving terminal is shown in Figure 18. The former tests were conducted at Hitachi's Tsuchiura Works, and the latter, at Tokyo Gas's Negishi Receiving Terminal.

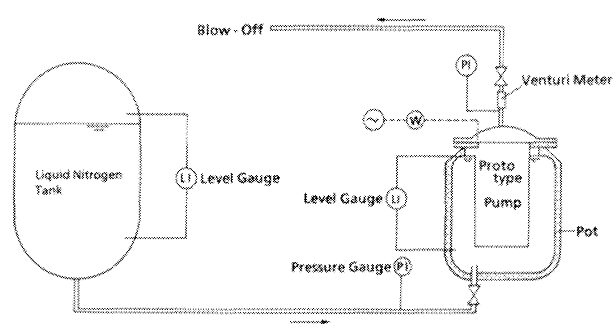


Figure 17. Flow of Liquid Nitrogen Test Equipment.

Test procedures

The various tests were implemented in 1980. The continuous operation test on LNG was completed in 1984, when the cumulative hours of test operation reached 24,000.

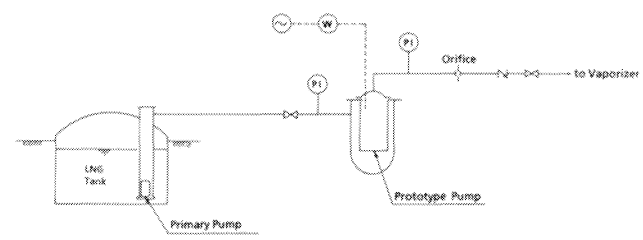


Figure 18. Flow of Field Tests at LNG Receiving Terminal.

Test results

Favorable results were obtained from all tests. The results are shown in Figure 19. The results were quite satisfactory to meet such specification requirements as the stable performance curve. Observation of the movement of the axial thrust balancing device was done by a detective sensor mounted on the motor casing that measured axial displacement of the shaft. The result, shown in Figure 20, indicates the axial displacement of the shaft. Moving upward to reduce the balancing clearance at startup, the disk keeps a relatively stable clearance during operation, and at switchoff it moves downward to increase the clearance. The behavior of the balancing device corresponds to the expected behavior, which verifies the good performance of the device. These results are outlined in Table 3. The results of measurements of the bearing clearance during the testing are represented in Table 4. Although it increased gradually, the ball bearing radial clearance in the hydrostatic guide bearing remained well within the specified tolerances.

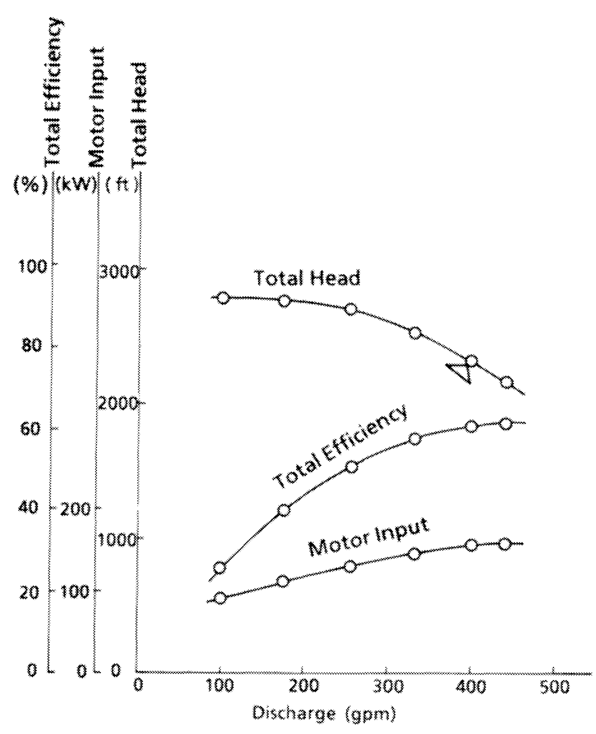


Figure 19. LNG Pump Performance Test Results.

Hydrostatic Slide Bearing LNG Pump

The testing was based on the hydrostatic guide bearing pump, except that alterations were made in the following respects. The same tests as performed for the guide bearing pump were done for the slide bearing pump (the test items and flow were also the same).

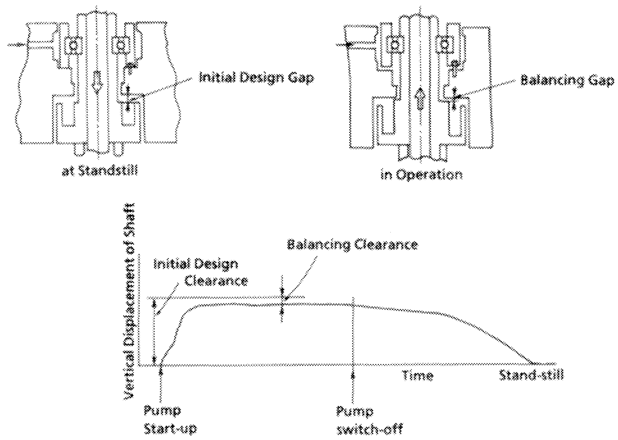


Figure 20. Test Result of Axial Thrust Balancing Device.

Table 3. Results of Tests of Prototype Hydrostatic Guide Bearing Pump.

Test Item	Test Procedure	Test Results
LN_2 Tests		
Spin Test	Ten repetitions of one minute operation formed smoothly, and it was confirmed that the current and discharge pressure indications were stabilized.	Start-up and shutdown were both per-
Performance Test	Measurement of data for pressure, current, and certain other parameters for single discharge quantity	Calculations of discharge quantity, total head, input power, and total efficiency confirmed that the design performance level was met.
Continuous Operation Test	Repetitions of operation for more than ten minutes at a time and measurement of bearing unit temperature, shaft vibration, and certain other parameters.	Operation for a cumulative total of six hours revealed no abnormalities in any component.
Tests in LNG		
Insulation Test	Measurement of coil resistance and insulation resistance	Both coil resistance and insulation resistance were confirmed to be problem-free.
Spin Test	Several repetitions of operation ranging from several seconds to several tens of seconds at a time.	Start-up and shutdown were both performed smoothly, and it was confirmed that the current and discharge pressure indications were stabilized.
Performance Test	Measurement of data for pressure, current, and certain other parameters for five different discharge quantities.	Calculations of discharge quantity, total head, input power, and total efficiency confirmed that the design performance level was met.
Continuous Operation Test (1)	Continuous operation and measurement of bearing unit temperature, shaft vibration, and certain other parameters.	Operation for a cumulative of 18 hours revealed no abnormalities in any component.
Continuous Operation Test (2)	Continuous operation with implementation of overhaul inspections and measurement of motor insulation resistance and abrasion of rolling elements when the cumulative number of hours of operation reached 600, 1,600, 9,300, and 24,000.	Each inspection found that the motor insulation resistance was normal, that the abrasion of rolling elements was negligible, and that values for all other parameters were within the tolerable limits (See Table 3)

Table 4. Record of Pump Operation and Overhaul (Hydrostatic Guide Bearing Pump).

Operational Record		Record of Overhaul (Gap between the shaft and the bearing)		
Cumulative Hours of Operation	Cumulative Number of Start-ups	Ball Bearings Above the Motor	Ball Bearings Below the Motor	Bearings at the Lower End of the Pump
0 h	0	20 μm	38 μm	0.25 mm
600	66	22	40	0.25
1,600	91	25	42	0.26
9,300	113	35	63	0.27
24,000	440	81	99	0.27
Tolerable Limit		200	200	0.45

Bearing	Hydrostatic guide	==>	Hydrostatic slide
Motor	3,000 V	==>	6,000 V
Impeller	7 - stage	==>	3 - stage *

* : Cutback of the stages was due to reduction of operational pressure in the LNG receiving terminal where the test was taken place.

Test procedures

The various tests were implemented in 1986. The continuous operation test in LNG was completed in 1989, when the cumulative hours of test operation reached 18,000.

Test results

Favorable results were obtained from all tests. The results of measurement of the shaft vibration and bearing gap during the testing are shown in Table 5. The following points should be noted about the findings shown in this table.

• The level of shaft vibration during operation in LNG was greater than during operation in liquid nitrogen, because the bearing fluid supply pressure is lower in LNG.

• Shaft vibration stayed on the same level right from the initial period of testing. In addition, analysis revealed no rolling abrasion in any of the three bearings, and no abnormality was found in any other components either. In this table, the difference between the shaft vibration during pump shutdown and the bearing gap during pump overhaul reflects such factors as shaft sag and installation eccentricity between bearings.

Table 5. Record of Pump Operation and Overhaul (Hydrostatic Slide Bearing Pump).

Operational Record		Record of Overhaul (Gap between the shaft and the bearing)		
Cumulative Hours of Operation	Cumulative Number of Start-ups	Bearings Above the Motor	Bearings Below the Motor	Bearings at the Lower End of the Pump
0 h	0	0.25 mm	0.28 mm	0.24 mm
1,826	17	0.25	0.28	0.24
17,892	929	0.25	0.28	0.24
Tolerable Limit		0.70	0.70	0.70

RESULTS

Installation Record

The record of hydrostatic journal bearing pump installation to date is reflected in Table 6. Actual and scheduled installation inside and outside Tokyo Gas amounts to 29 pumps.

Table 6. Record of Hydrostatic Journal Bearing Pump Installation to Date.

Number of pumps	29
Discharge quantity	20 - 2,500 gpm (2 - 270 t/h)
Total head	65 - 5,050 ft-LNG (20 - 1,540 m-LNG)
Rate of motor output	1.5 - 800 kW
Power source	200 - 6,600 V

Example of Application and Operational Record

Noted below are the specifications of hydrostatic journal bearing pumps installed in Tokyo Gas's Sodegaura Works, exemplifying actual application and the operational record of these pumps.

Pump specifications

Discharge quantity		1,000 gpm	(100 t/h)
Total head		4,600 ft-LNG	(1,400 m-LNG)
Suction Temperature		–260°F	(–160°C)
FluidSpecific gravity		0.44 - 0.485	
Impeller		8 - stage	
Weight	Pump motor	5,730 lb	(2,600 kg)
	Suction pot	2,200 lb	(1,000 kg)
Motor	Type: submerged squirrel cage rotor type		
	Rated output :	710 kW	
	Power source :	AC 6,600 V x 3-phase x 50 Hz	

Measurements	Pump diameter	28 in(motor)	(730 mm)
	Total pump length	102 in	(2,600 mm)
Bearing type	Hydrostatic guide bearing type	4	
	Hydrostatic slide bearing type	1	
Year of installation	Hydrostatic guide bearing type	1982	
	Hydrostatic slide bearing type	1987	

Pump appearance (see Figure 21)

Record of operation

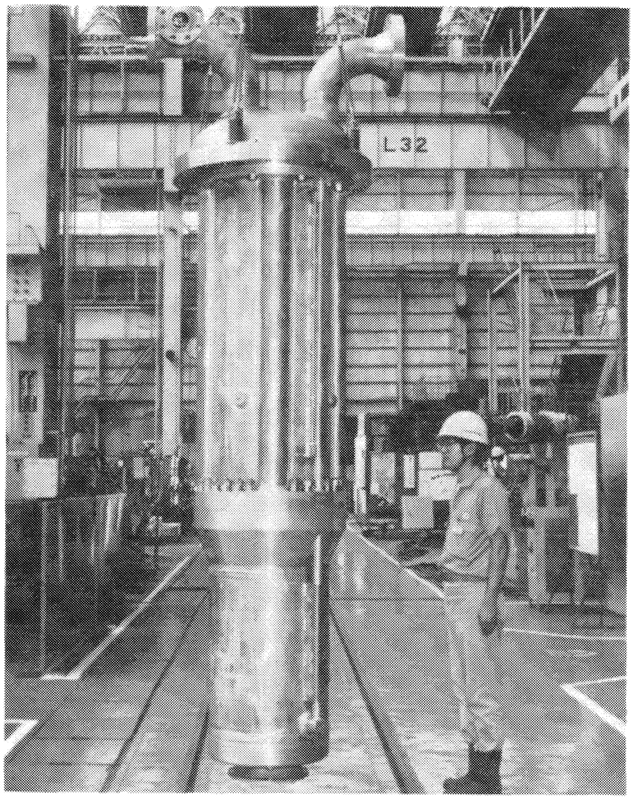


Figure 21. Pump Appearance.

The hours logged by the subject pumps in actual operation and the results of overhaul inspection of the pumps are shown in Table 7. Since there had been no previous record of hydrostatic journal bearing pump operation, the pumps were overhauled for inspection once they had been in operation either 10,000 or 20,000 hr. Such inspection turned up no problems, and it was consequently decided to shift the emphasis to preservation of the operational status through diagnostic equipment.

Table 7. Example of "Actual Record" of Operation of LNG Pumps Installed at the Terminal (as of the end of June 1991).

Pump Code	Type of Bearing	Cumulative Hours of Operation	Overhaul Record (Bearing Gap)	
			Hours of Operation	Overhaul Findings
V_C-1P_2 (Installed in 1982)	Hydrostatic Guide	33797 hours	22254 hours (As of Mar 1989)	Note 1 Note 2
V_C-2P_2 (Installed in 1982)	Hydrostatic Guide	41558	23453 (As of Nov 1987)	Note 1 Note 2
V_C-3P_2 (Installed in 1982)	Hydrostatic Guide	35800	11458 (As of Oct 1986)	Note 1 Note 2
V_C-4P_2 (Installed in 1982)	Hydrostatic Guide	35800	15857 (As of Mar 1985)	Note 1 Note 2
			25306 (As of Nov 1989)	Note 1 Note 2
V_D-1P_2 (Installed in 1987)	Hydrostatic Slide	22622		Overhaul inspection not yet implemented.

Note 1 : Overhaul inspection for the purpose of confirmation when the number of hours of operation passed 10,000 or 20,000.
Note 2 : The inspection revealed no abnormality in any component, but the ball bearings were replaced.

Effects for Lengthening Service Life

The service life of the components of the newly developed pumps may be estimated as follows, based on the results of the continuous operation tests described in the previous section.

Hydrostatic guide bearings	At least 50,000 hr
Hydrostatic slide bearings	At least 100,000 hr
Bushes and other wearing parts	At least 100,000 hr

Consequently, the pump overhaul interval is expected to be extended to several times that of the conventional ball bearing LNG pumps, at least 50,000 hr for the guide bearing type and at least 100,000 hr for the slide bearing type.

Improved Safety

Since they represent a great improvement in LNG pump dependability, hydrostatic journal bearing pumps also make a great contribution to the stable operation of LNG terminals. Furthermore, the significant reduction in the frequency of overhaul inspection also reduces in purging, valve operation, and other complex work requiring great care that accompanies this inspection.

Improved Maintenance

The reduction in the frequency of overhaul inspection affords a large reduction in maintenance costs.

In hydrostatic slide bearings, the remaining life expectancy can be estimated by measuring shaft vibration. This feature enables prevention of sudden failure and the implementation of systematic maintenance management. In addition, the shaft vibration measurement can be performed with a simple measuring device, and there is no need for expensive failure diagnosis equipment.

The pumps are also lightweight and compact, and therefore easy to handle (disassembly and assembly each require only a single day).

Range of Application

Hydrostatic journal bearing pumps come in a wide range of capacities, from large to small. They are suitable for various application with low temperature liquefied gases other than LNG as well.

CONCLUSION

The main characteristics of the pump are radial balance, due to being a hydrostatic bearing, thrust balance, due to using a balance disk piston, short shaft structure, due to the radial diffusers and high lift impellers, and a single layer cylindrical stage housing using piston ring type sealing rings. The hydrostatic bearing, balance disk piston and short shaft structure that together give both radial thrust and balance are substantially better than in previous pumps. This extends the life of slide components, and increases by a factor of several times the interval between open-up inspection. Because hydrostatic slide bearings are used, the bearing sensitivity to breakdowns and enabling more planned maintenance. Consequently, the pumps are much more reliable than previous pumps, and will make a major contribution to the safe running of LNG plants. The short shaft structure, and single layer cylindrical stage housing using piston ring type sealing rings mean that the pump is lighter and more compact, and make pump dismantling and assembly easier. The authors now have substantial operating experience with the new pumps, and can report that they function as designed.

APPENDIX

The following is an outline account of the design procedure for the orifice contraction ratio, which exerts a great influence on hydrostatic journal bearing performance.

Optimal Orifice Ratio

The orifice contraction ratio that would maximize the bearing rigidity for a disk form hydrostatic thrust bearing of the type shown in Figure 22 may be calculated as follows. The load weight (W) may be expressed as follows.

$$W = Ae\,P_o \qquad \text{(A-1)}$$

The value of the effective bearing area (Ae would be smaller than the actual bearing area (A) ($= \pi r_1^2$) and vary depending on the pocket form. The flow quantity (Q) discharged from the bearing gap may be expressed as follows.

$$Q = \pi / (6 \ln (R_1 / R_0)\,\mu) \times h^3 P_o = K_B h^3 P_o \qquad \text{(A-2)}$$

Furthermore, the flow quantity (Q) passing through the orifice may be expressed as follows.

$$Q = C_f \pi r^2 (2/\rho)^{1/2} \times (P_s - P_o)^{1/2} = K_o \times (P_s - P_o)^{1/2} \qquad \text{(A-3)}$$

As a result,

$$W = Ae\,K_o^2 / (2 K_B^2 h^6) \times (1 + ((4 K_B^2 h^6)/K_o^2) \times P_2)^{1/2} - 1 \qquad \text{(A-4)}$$

The precondition for maximizing the bearing rigidity is therefore the same as that for maximizing the load weight (W). Since $\partial W^2/\partial^2 h = 0$, then

$$P_o / P_s = 0.691 \qquad \text{(A-5)}$$

In the case of cylindrical hydrostatic radial bearing, the theoretical precondition for maximizing the bearing rigidity is known to be as follows (calculations omitted).

$$P_o / P_s = 0.586 \qquad \text{(A-6)}$$

The orifice contraction ratio function $f(P_o / P_s)$ at this time is shown in Figure 23. If this ratio becomes smaller or larger than $P_o / P_s = 0.586$, this function will become smaller. In actuality, the optimal ratio for maximizing rigidity will vary slightly depending on such factors as the pocket form and the presence or absence of grooves.

Here, W : load weight
- Ae : effective bearing area
- P_o : pocket pressure
- P_s : supply pressure
- r : radius of orifice
- R_1 : outer radius of bearing
- R_0 : inner radius of pocket
- μ : coefficient of viscosity
- h : bearing clearance
- C_f : coefficient of orifice
- ρ : density

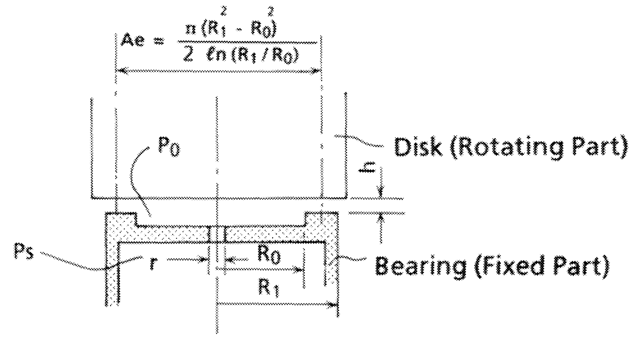


Figure 22. Disk Form Hydrostatic Bearing.

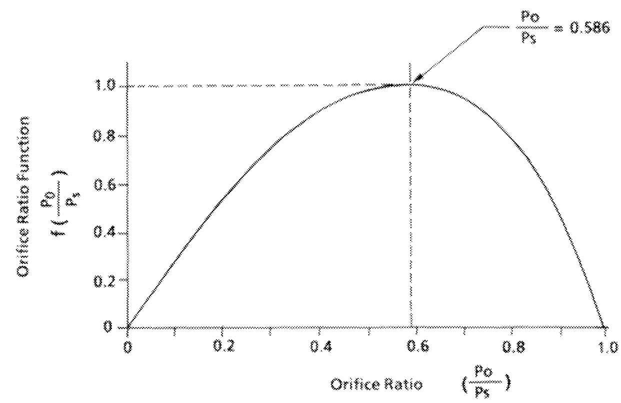


Figure 23. Orifice Ratio Function.

REFERENCES

1. Miyata, Y., Seta, A., and Kaneda, Y., "Tokyo Gas Experience in Improving LNG Pump Reliability," The 9th International LNG Conference, Paper 11 in Session 4 (1989).
2. Yoda, H., Kumata, K., and Shimizu, K., "Submerged Motor Pump for Cryogenic Plants," Hitachi Review, 32, pp. 19-22 (1983).
3. Reddecliff, J.M., Vohr, J.H., "Hydrostatic Bearings for Cryogenic Rocket Engine Turbopumps," ASME Trans., J. of Lubrication Technology, pp. 557-574 (July 1969).