W. Johnson, S. Myers, R. Schepis, J. Crew, R. Keltner

Zero Leakage Tornado Effect Pumps in Carrier Sewage Systems

Abstract

Current centrifugal pumps installed in CVN 68 class Collection, Holding, and Transfer (CHT) systems experience a high rate of seal failures within a short time period of seal installation. A CHT pump seal failure requires a significant maintenance effort to place the affected CHT pump out of service to replace the failed seal. In addition, seal leakage is not a normal condition. Seal leakage releases sewage material into the CHT pump room in violation of Navy health regulations.

Efforts to develop improved pump seals have met with less than satisfactory results. An effort was undertaken to identify an alternate pump design that met the zero external leakage criteria during operation and eliminated seal failure contamination of the pump room. This effort identified a new type design, the Eddy Pump, which is gaining increased recognition for its performance in civilian applications.

This new pump departs from conventional centrifugal pump design principles in that the pump rotor is able to create and harness a dynamic fluid eddy effect within the pump housing and inlet. This effect is similar to that which occurs in a tornado. A very strong synchronized central column flow develops from the pump rotor to the pump inlet and creates a low-pressure reverse eddy flow from the pump inlet to the pump discharge. This action results in a region of negative pressure in the vicinity of the pump seal. The seal design allows one way passage of externally supplied low-pressure air through the shaft seal toward the pump casing (volute) to oppose the ingress of sewage.

This paper is a condensation and combination of two reports that deal with the use of the alternate pump design in aircraft carrier CHT systems. These reports are:

Myers, S. and Keltner, R., "CHT Pump Trade-Off Studies - CVN 76," prepared by Newport News Shipbuilding under contract N00024-91-C-2108, Task 2318T-74, January 1993.

Schepis, R. and Crew, J., "Eddy Pump Performance," Commander, Naval Surface Warfare Center, Carderock Division, Philadelphia Site, Serial 631/110, May 1995.

Introduction

The Collection, Holding, and Transfer (CHT) system pumps on CVN 68 class aircraft carriers have not shown any significant problems with their ability to pump, however, there is a major problem with mechanical seal failure. Once seal failure occurs, sewage can leak into the CHT pump room and pose a health hazard to the crew. Navy Bureau of Medicine (BUMED) Notice 6240, BUMED-5532, dated 5 February 1978, prohibits any leakage of sewage into the CHT pump room because of the health hazard. The BUMED instruction thus places a zero external leakage constraint on CHT pumps.

Ship personnel have reported mechanical seal failure as early as thirty days after installation. When a failure occurs, pump downtime can last two to three weeks depending on ship operations and parts availability. The replacement process is labor intensive. The CHT pump in the affected pump room must be taken off line and isolated with danger tags. Then at least two mechanics are required to dismantle the pump while taking health precautions for working in a septic environment. The entire process can take six to twelve hours. A diagram of the CHT pump system is shown in Figure 1.

Background

Current CHT pumps employ an opposing double mechanical shaft seal (Figure 2). The sealing faces on each side are constructed of tungsten carbide against carbon. In the cavity or reservoir between the mechanical seals, oil is used as a lubricating substance. In order to keep the mechanical seal faces together, a diaphragm located behind the pump impeller pressurizes the dead-ended reservoir. A coil spring also helps to keep the faces together.

Investigation has shown that the mechanical seal on the pump end is the first to fail, thereby inducing sewage contamination into the pump reservoir. It is believed that when a CHT pump is started, the initial discharge pressure surge may momentarily separate the seal faces on the pump end. The existing carbon seal is too brittle too withstand the repeated chattering of the seal faces or the abrasiveness of the resulting sewage between the seal faces. As the pump reservoir builds up with sewage material, the motor end mechanical seal eventually fails due to contaminated lubrication and allows sewage to leak out into the pump room.

Alternate Pump Seals

Personnel in the carrier machinery reliability and maintenance group at Newport News Shipbuilding contacted various seal manufacturers for possible solutions to the seal problem. Two seal changes were investigated. The first change suggested the installation of a pressurized seal using a pressure tank outside the pump reservoir. A high-pressure hose using nitrogen as the pressure agent connects the pump reservoir and external tank. The external tank is charged to maintain 20 psi over CHT pump discharge pressure to prevent sewage leakage past the seal. The second change suggested is the use of a tungsten carbide to tungsten carbide seal on the pump end of the reservoir. The faces of this seal are substantially harder and tolerate sewage better than the carbon seals. This change does not require external pressurization.

Both of these seal changes were installed on CVN 73 for evaluation. The pressurized seal system was installed as suggested with the exception that the barrier fluid in the pump reservoir was changed from oil to water. This change was made for environmental considerations and to provide a cleaner workplace for personnel. If the pressurized seal were to fail, water from the external pressurizing tank would be lost instead of losing oil into the sewage side of the pump. The external pressure tank was provided with a sight glass to allow system operators to detect any seal failure that may occur, thus allowing for system repair before raw sewage can enter the barrier fluid reservoir. The second seal change of tungsten carbide to tungsten carbide seals was installed on three CHT pumps. Only the primary seals were changed on the impeller end of the assembly. Tungsten carbide to carbon seals were maintained as the secondary seals between the pump seal reservoir and atmosphere.

The tungsten carbide to carbon seal originally installed on the pressurized unit failed. Of the three non-pressurized test units, two of the seals failed, with one of these failures attributed to operator error. The seal modifications did show a significant improvement in seal life before failure, however, they still did not exhibit zero leakage. Small amounts of reservoir fluid were discovered collecting in a pocket just below the point where the pump shaft exits the seal reservoir on the motor side. This reservoir fluid can either be clean or sewage contaminated depending on the integrity of the reservoir.

Alternate Pump Design

An alternate design investigated by Newport News Shipbuilding personnel is the Eddy Pump (shown in Figure 3). This pump design appears to have the capability for zero external leakage. Due to the nature of the pump, there is a negative pressure inside the pump casing at the shaft seal. Low pressure air (Item 11 of Figure 3), from an external source, is continuously fed through the shaft seal (Item 5 of Figure 3) into the pump casing. Instead of a metal mechanical seal, the shaft seal is a non-metallic lip seal. This seal acts as a check valve in that it allows air to flow through it in one direction only. Even in the event of internal seal leak-by or seal failure, sewage should not leak into the pump room since the seal reservoir is totally enclosed.

The Eddy Pump has several attributes. Because it operates on a different hydrodynamic principle than conventional centrifugal pumps, the Eddy Pump, in a properly designed application system, creates less cavitation and vibration. The pump has comparatively fewer moving parts and the pump design has been kept relatively simple which should reduce downtime for maintenance and repair. The pump has been specifically designed such that there are no critical mechanical tolerances within the pump that would cause loss of pump efficiency due to pump wear. Due to the large clearances the pump rotor can easily pass debris that would foul other pumps.

Tornado Effect Principle of the Alternate Pump Design

The hydrodynamic principle used by the Eddy Pump is similar to that of the aerodynamic principle of the tornado. The tornado generates a high velocity rotating air mass with a central column of increasingly higher pressure reaching toward the ground surrounded by peripheral low pressure eddies. As the lower tip of the tornado touches down on the ground, objects are drawn into the low-pressure peripheral eddy. This effect enables the tornado to pulverize and pick up and carry objects into the rotating air masses. The design of the Eddy Pump rotor and casing hydrodynamically creates a similar effect using fluid. Dr. Harry Weinrib, the inventor of the Eddy Pump, describes the operation of the pump as follows:

> "The Eddy Pump (Figure 4) consists of an energy generating **ROTOR (1)** attached to the end of a **DRIVE SHAFT (2)** and placed within a **VOLUTE (3)**. As the **ROTOR** begins to spin, it sets into motion the ambient fluid present within the **VOLUTE** and adjoining **INTAKE CHAMBER (4)**. At normal operating speed, this spinning fluid is forced down, into the hollow center of the **INTAKE CHAMBER** where it creates a high speed, swirling **SYNCHRONIZED COLUMN OF FLUID (5)** which agitates the **MATERIAL (6)** to be pumped (sludge, sand, clay, etc.). This swirling column of fluid creates a peripheral **"EDDY" EFFECT (7)**, which causes the agitated material to travel by reverse flow, **UP**, along the sides of the **INTAKE CHAMBER**, into the **VOLUTE**. Here the material, under pressure from below, is forced into the **DISCHARGE PIPE (8)**." [1]

Dr. Weinrib further explains the differences in engineering design and hardware between conventional centrifugal/vortex pumps and the tornado effect Eddy Pump as follows:

> "Most conventional centrifugal type impellers incorporate front and back shroud enclosed vanes, a center suction "eye" configuration, and discharge the solids by centrifugal force. In order to prevent fluid recirculation, critical tolerances are necessary between the wetted parts of the pumps. As wear occurs, these critical tolerances increase and the pumps progressively lose their hydrodynamic efficiency. The only way to keep these pumps functioning at an acceptable level of performance is to replace the various components through a regular maintenance program.
>
> Vortex type pumps are conventional centrifugal pumps with the suction end of the impeller shroud removed to expose the vanes. This type of impeller configuration enables the pump to create a unidirectional swirling column of liquid (vortex) in addition to the centrifugal force. The turbulence created within the suction opening draws the material into the low-pressure area within the pump casing and discharges it by centrifugal force.
>
> The Eddy Pump rotor is an open cruciform configuration (Figure 5), without any impeller shrouds. In motion, it contains and recirculates an axially solid, progressively concentrated nucleus of energy. This energy is released in the

> form of a synchronized swirling column transmitted to the center of the pump intake opening by creating a negative pressure or "eddy" effect in the intake throat area. This "eddy" current carries the material along the low pressure periphery of the swirling column of fluid to the discharge."[2]

Installation of Alternate Pump Design

Newport News Shipbuilding personnel worked with representatives from PERA(CV), Bremerton, Washington, to install Eddy pumps in carrier CHT systems for evaluation. One vertical pump was installed on USS Nimitz (CVN 68) in January 1994 and two horizontal pumps were installed on USS George Washington (CVN 73) in March and May of 1994. (CHT systems on CVN 68 through CVN 71 have six vertical pumps and CVN 72 through CVN 75 use four horizontal pumps in their CHT systems.) Figure 6 shows the vertical Eddy Pump installed on the CVN 68.

Performance Assessment of the Alternate Pump Design

BACKGROUND OF TESTS CONDUCTED

Personnel from Naval Surface Warfare Center, Carderock Division (NSWCCD-SSES), Philadelphia Site, conducted baseline pump performance and vibration assessments on one of the CVN 73 Eddy pumps in February of 1995 and continued with follow-up testing of the same pump in April 1995. The manufacturer's pump curve is shown in Figure 7. The manufacturer's rated pump capacity is 400 gallons per minute (gpm) at 135 feet of head pressure. The Eddy Pump installed as pump No. 4 on USS George Washington included a 40-horsepower 1,800-rpm motor. (Eddy pumps installed by Newport News Shipbuilding and PERA(CV) have either 40- or 50- horsepower motors.) In February 1995, NSWCCD-SSES representatives gathered baseline data on pump No. 4 on USS George Washington. They operated the pump at various flow rates by incrementally shifting the discharge valve from fully open to fully shut while recording suction and discharge pressures at the pump. Vibration and motor speed measurements were taken concurrently during this test.

Eddy Pump No. 4 was tested again in April 1995. Two test runs were conducted over a two-day period. Tests performed in February 1995 were repeated, but with additional pressure measurements taken along the pump discharge line to the pier connection. Figure 8 shows a schematic of the pumping system and the gage locations. From the recorded data the total pump head can be calculated, a pump curve can be drawn, and system friction losses can be determined.

Pressure and approximate flow measurements taken in February gave NSWCCD-SSES personnel a basic familiarization with the Eddy Pump's performance and its response to head pressure changes. The data collected in April 1995, was not intended for comparison to February 1995 data, but to the manufacturer' pump curve. Some uncertainties exist between the data collected in February and April that would not allow a fully legitimate comparison. In February the ship's in-line flow meter was used to record pump flow rates, although its accuracy was unknown at that time. During the April visit, the meter's calibration was determined as being close to that of an accurate ultrasonic meter. The February data was compared to the April data, keeping in mind that the comparison was not fully valid. The data appeared to be similar, indicating that the Eddy Pump had undergone no significant performance changes between February (880 operating hours) and April (1,100 hours).

PUMP DISASSEMBLY FOR INSPECTION

After completion of the first test run, PERA(CV) and Eddy Pump Corporation representatives disassembled the pump. Components that are normally exposed to sewage had acquired a thin black patina. This included the pump casing interior, the rotor, and the interior of the discharge piping. This is a normal reaction when Monel contacts seawater. The thickness of this black coating was not discernible to the naked eye. It was microscopically thin.

None of the pump components appeared to be corroded, eroded, or worn in any manner. The coupling (Item 13 of Figure 3 and shown assembled in Figure 9 and disassembled in Figure 10) was of particular interest since the rubber portion is considered a replaceable wear item. However, all coupling components appeared to be in nearly pristine condition.

There was some evidence of leakage found in the housing behind the seal plate (Figure 11). A green deposit (in the original color photo) was evident up to the seal assembly, and a smaller deposit inside the seal assembly cavity. The green color implies a copper-based oxidation from water leakage. Overall, however, the leak-by was minimal and the housing had not corroded. Most components in this pump are fabricated from Monel. It is unknown if this minor leak-by occurred during normal pump operation, or from unusual circumstance. The critical factor, in any case, is the water did not migrate fully past the seal and into the pump room. Despite some internal leak-by, the pump maintained zero external leakage integrity overall during the 1,100-hour evaluation period. It is unknown if the pump seal would have suffered a complete compromise over a longer period, and if so, how long that would have taken.

The rotor was removed and taken to the ship's machine shop. There it was modified, at the manufacturer's request, to include a shaft locking bolt. Item 10 of Figure 3 and Figure 12 show the modification. The rotor is screwed onto the threaded shaft (also shown in Figure 11). Once installed, the rotor's friction with these threads is sufficient to secure it. In fact, the fit is so tight that a special rotor removal tool and hammer impacts were needed to loosen it. The locking bolt was added only as an additional insurance measure to preclude loosening when pump direction is reversed as a result of back flow through the discharge CHT piping.

The pump was reassembled with a new shaft/bearing assembly (Figure 13). The shaft is inserted from the rear of the housing (Figure 14). A new seal assembly was then installed (Figure 15). This step revealed a machining problem with the bearing housing. The cone shaped seal assembly is inserted into the front of the bearing housing. The seal assembly has four threaded holes, which align exactly with corresponding non-threaded holes in the housing. These holes are for the four air and grease fittings (Figure 16 and Items 11 and 12 of Figure 3). Upon seal installation, the holes were found to be misaligned by about one-half degree, which was enough to prevent the fittings installations. Another seal assembly was tried, but the same problem resulted. The conclusion drawn was that the bearing housing was slightly inaccurate. The housing's holes were reamed and the fittings were successfully installed. The manufacturer stated that he would investigate this problem and fabricate a machining jig to ensure exact hole alignments on future pumps.

The second test run was performed in the same manner as the first test. One noteworthy parameter had been intentionally changed since the first test run. The seal assembly air pressure had been increased to about 17 to 18 psig and the airflow rate reduced to zero. (Air pressure is normally about 7 to 8-psig and airflow about 0.5 SCFM.)The air settings had been changed to allow the grease around the new seal assembly to seat. The original settings were to be restored after several hours of pump operation.

PUMP PERFORMANCE CURVES

Pump performance curves were constructed from the data collected during the two test runs (Figures 17 and 18). The pump performance curves are shown with comparisons to the manufacturer's curve. The curves have been smoothed to give a best fit to the data. To approximate the manufacturer's curve, it is shown as a straight line ranging from 136 feet of head pressure at zero gpm to 135 feet of head pressure at 500 gpm. The derived curves actually span an area to account for inherent gage errors and the fact that the pump discharge gage needle oscillated during the tests. The span of discharge pressures readings resulted in a range of total pump head calculations for each flow rate. The shaded areas in each curve reflect these ranges. Vertical lines in the shaded areas indicate flow rates at which measurements were taken. Dots show the mean head pressures calculated.

The pump's ability to maintain head pressure at various flow rates is less than expected. The pump's performance matches the manufacturer's curve at low flows (zero to 150 gpm) but it produces about 10 feet less head than expected at 400 to 500 gpm. Despite the lower head pressures at high flow rates, the pump is well suited for this system's configuration. It produced 6 to 7 psig discharge pressure at the deck connection, which is typical of other CHT pumps. The pump performance curves indicate that at-sea testing of the pump should be performed to establish flow rates when the CHT system is shifted to an at-sea lineup, lowering the height of overboard discharge. The lowered height of discharge means lower pump head pressure and increased flow rates. The increase in flow rates may lead to increased piping system erosion.

SYSTEM FRICTION LOSS CURVES

System friction losses from the pump to deck connection are shown in Figures 19 and 20. Only the average pump discharge gage values were used (rather than the range of readings) to generate these curves. Overall, losses decreased with increasing flow rate. This is because higher flows allow more of the pump's energy to be delivered to the deck connection. At lower flows, more energy (in the form of discharge pressure) builds up at the pump while downstream pressure eventually decreases to zero. The main source of losses at low flows is, of course, the discharge valve as it is progressively closed. The losses determined at full flow are acceptable for this CHT system configuration and Eddy Pump combination since deck connection pressures were adequate. Considering aircraft carrier CHT pump discharge piping configurations as a typical worst case scenario, the pump should provide adequate pressure at the deck connections for most, if not all, currently installed CHT systems.

VIBRATION ANALYSIS

The vibration analysis of Eddy Pump No. 4 showed that the pump's vibration after overhaul and reassembly was higher than before it was disassembled. The pre-overhaul vibration levels were in compliance with MIL-STD-167-1, Mechanical Vibrations of Shipboard Equipment, but the post-overhaul levels were not. However, the test conditions were not identical. The CHT tank levels were different, as were the seal assembly air pressures. It is uncertain whether these variations can account for the differences found in vibration. Compliance with MIL-STD-167-1 should be a requirement for the use of these pumps in Navy CHT systems. Not only should every pump meet this requirement, but also individual replacement components such as the rotor and shaft must be properly balanced to maintain the pump's vibration integrity. Any overhauls involving parts replacement must not cause the pump to depart from MIL-STD-167-1 vibration limits. Moreover, vibration trending data should be established for all approved pumps to determine the need for parts replacement. Maintenance of these pumps should be condition-based versus time-based.

COMMINUTORS

Comminutors are macerators used to break up potential clogs in the CHT system prior to reaching the CHT pumps to preclude blockage of the pump. Ship's personnel on CVN 73 reported that they had not used their comminutors during their last deployment. The Eddy Pump experienced no clogging problems during this period and no corrective maintenance actions were required.

Conclusion

The original problem of CHT pump leakage due to mechanical seal failure has been addressed by the cooperative efforts of industry and Navy representatives. Concurrent with the resolution of the primary problem, the BUMED requirement for zero leakage appears to also have been met. The Eddy Pump design offers several improvements for CHT system operation and maintenance.

Ship's force feedback has been very positive about the Eddy Pumps installed on their ships. Personnel from other aircraft carriers have requested the backfit installation of Eddy Pumps. A ship alteration package for Eddy Pumps has been issued. It is anticipated that CVN 76 will have Eddy Pumps installed as original fit CHT pumps. Other fleet communities have inquired about the use of Eddy Pumps in their CHT systems.

REFERENCES

[1]. Weinrib, Dr. Harry, Eddy Pump Sales Brochure, Eddy Pump Corporation.
[2]. Weinrib, Dr. Harry, Eddy Pump Information White Paper, Eddy Pump Corporation.

ACKNOWLEDGMENTS

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Stephen Myers is a Senior Engineer at Newport News Shipbuilding in the Carrier Construction Engineering Project, Machinery Group. He is coordinating the design of the CVN 76 aircraft arresting gear. In his previous assignment with the Aircraft Carrier Piping Design Group he worked on the Aircraft Carrier Engineering Services (ACES) study of CHT pumps and the use of the Eddy Pump for that application.

Robert Schepis is a Special Projects Engineer in the Environmental Quality Systems In-Service Engineering Branch of Naval Surface Warfare Center Carderock Division (NSWCCD-SSES), Philadelphia Site. He completed sewage handling performance testing of the Eddy Pump installed on the USS George Washington (CVN 73) and documented the tear-down inspection of the pump.

Joseph Crew (Ret.) was a Mechanical Engineering Technician in the Environmental Quality Systems In-Service Engineering Branch of Naval Surface Warfare Center Carderock Division (NSWCCD-SSES), Philadelphia Site. He has extensive background with shipboard sewage pumps and piping systems, and assisted with the USS George Washington (CVN 73) Eddy Pump evaluation.

Ronald Keltner is a Senior Technical Analyst at Newport News Shipbuilding working in the Aircraft Carrier Machinery Reliability and Maintenance group. Employed at Newport New Shipbuilding since retiring as a Senior Chief Machinist Mate in 1985, Ronald has worked in many areas of aircraft carrier engineering support. He worked with Stephen Myers on the Aircraft Carrier Engineering Services (ACES) study of CHT pumps and now is coordinating the installation of the Eddy Pumps on aircraft carriers.

The statements and opinions expressed in this paper are solely those of the contributing authors and not necessarily those of John J. McMullen Associates, Naval Sea Systems Command, Naval Surface Warfare Center Carderock Division or Newport News Shipbuilding.

Collection, Holding, and Transfer (CHT) System

A - Overboard discharge
B - Shore connection (P+S)
C - Pressure relief
D - Supply to washdown
E - CHT tank washdown
F - Washdown hose connection
G - Riser washdown
H - CHT pumps

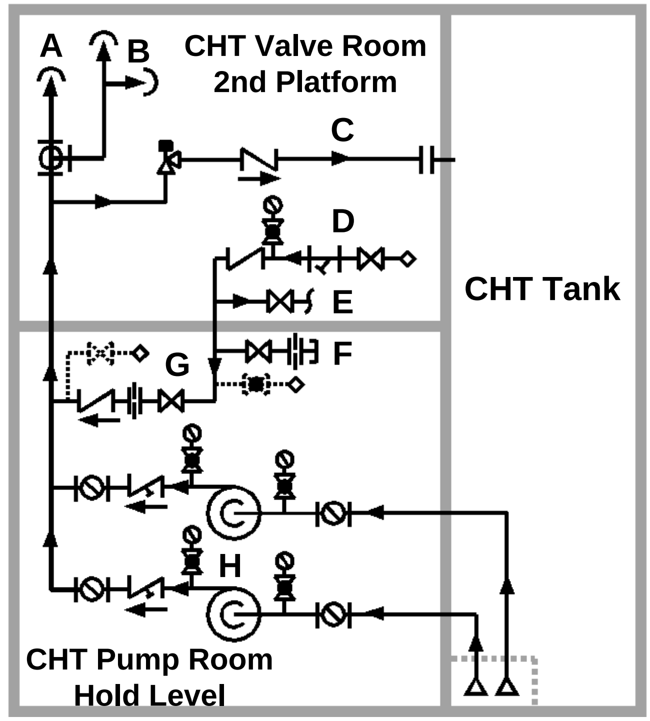


Figure 1. Typical CVN CHT system.

- **Seal faces constructed of tungsten carbide against carbon**
- **Seals are oil lubricated**
- **Dead-ended reservoir is pressurized to maintain seal contact**

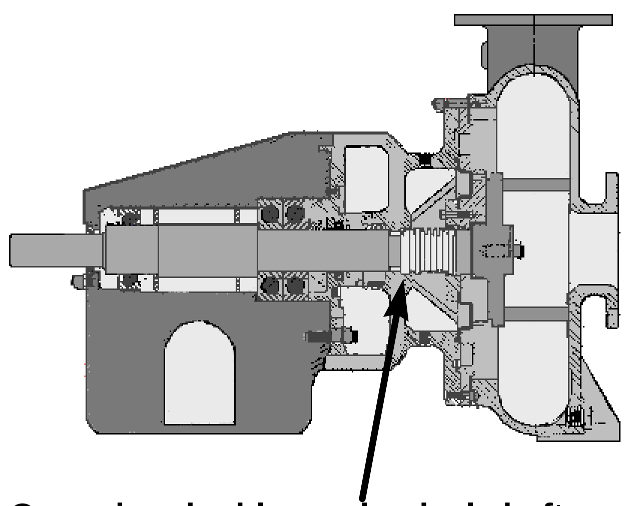


Figure 2. Current CHT pump.

1. Rotor
2. Pump casing (volute)
3. Suction intake
4. Discharge outlet
5. Non-metallic lip shaft seal
6. Thrust bearing
7. Radial bearing
8. Shaft
9. Bearing housing
10. Locking bolt
11. Air fitting
12. Grease fitting
13. Motor-pump coupling

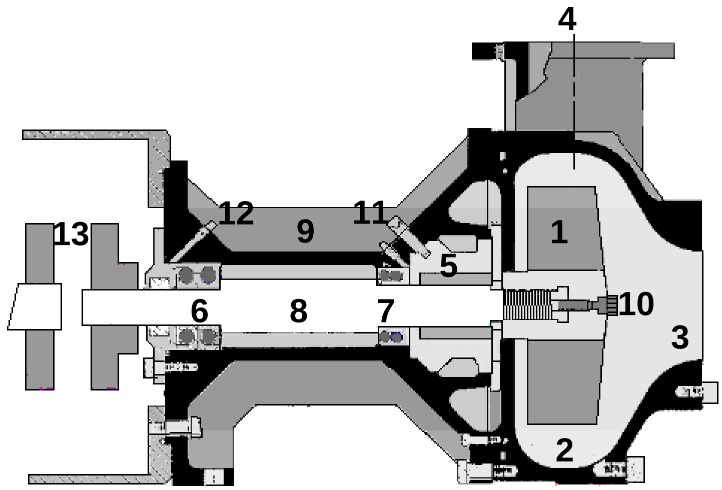


Figure 3. Eddy Pump NF-4000 section view.

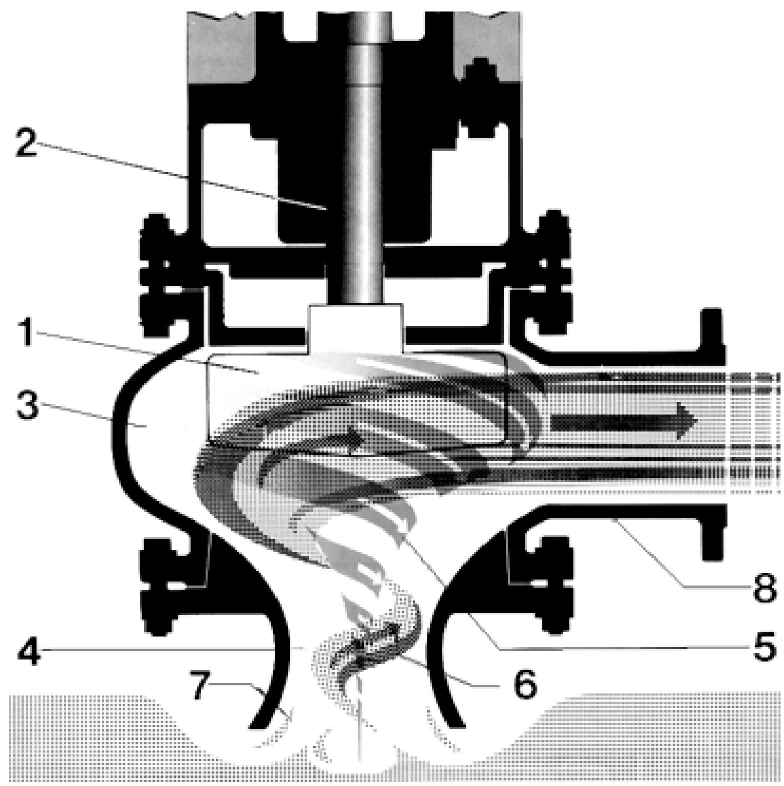


Figure 4. Illustration of Eddy Pump operation.

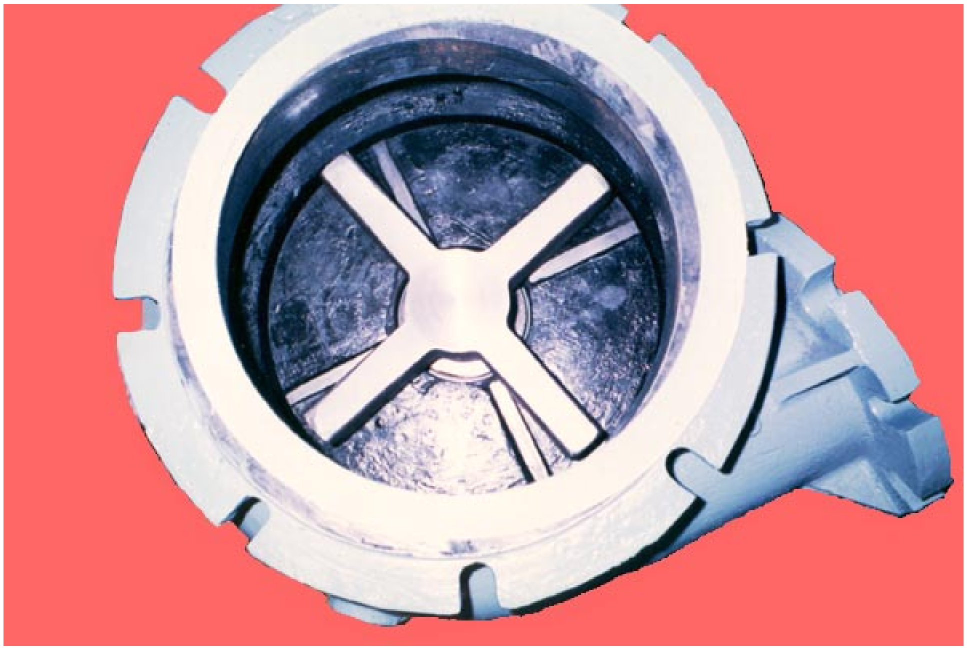


Figure 5. End view of Eddy Pump rotor in casing.

Figure 6. Eddy Pump installation in CVN 68.

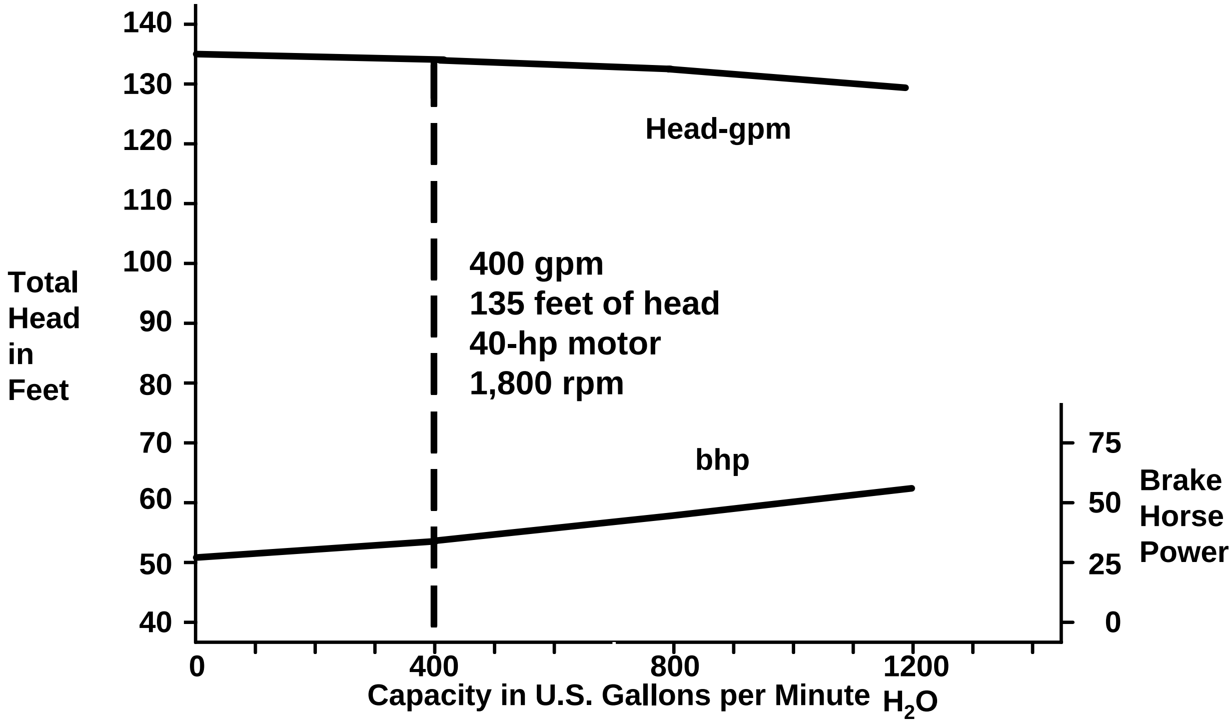
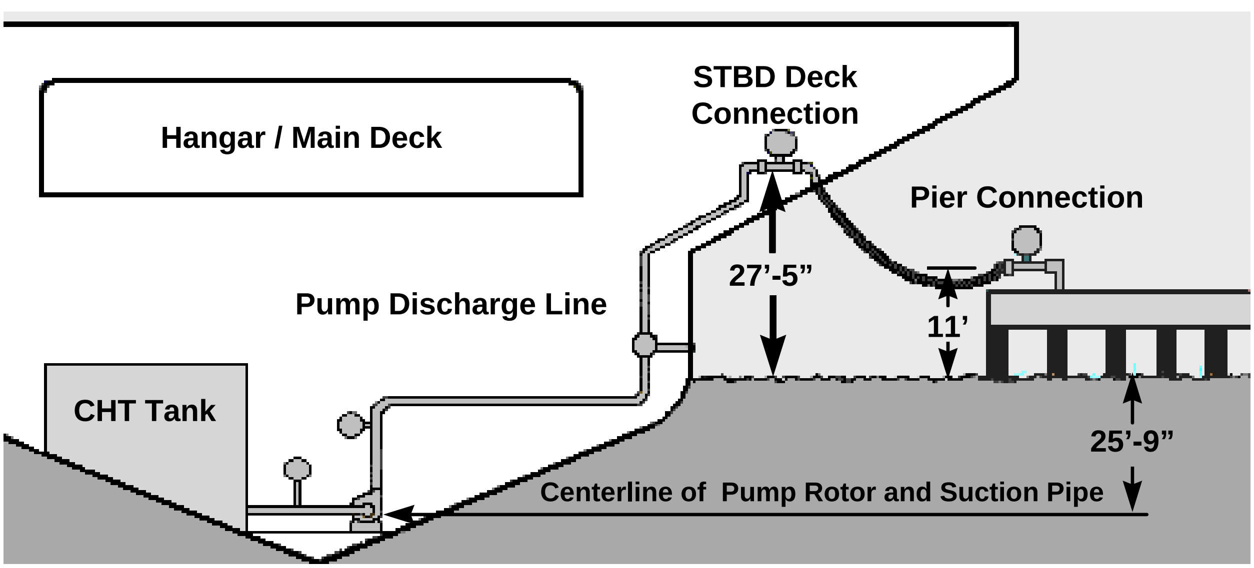


Figure 7. Manufacturer's pump curve for the Eddy Pump.



USS George Washington (CVN 73) (~ Frames 195-205, looking forward)
FEB and APR 1995 (not to scale)

Figure 8. Schematic of Eddy Pump discharge configuration.

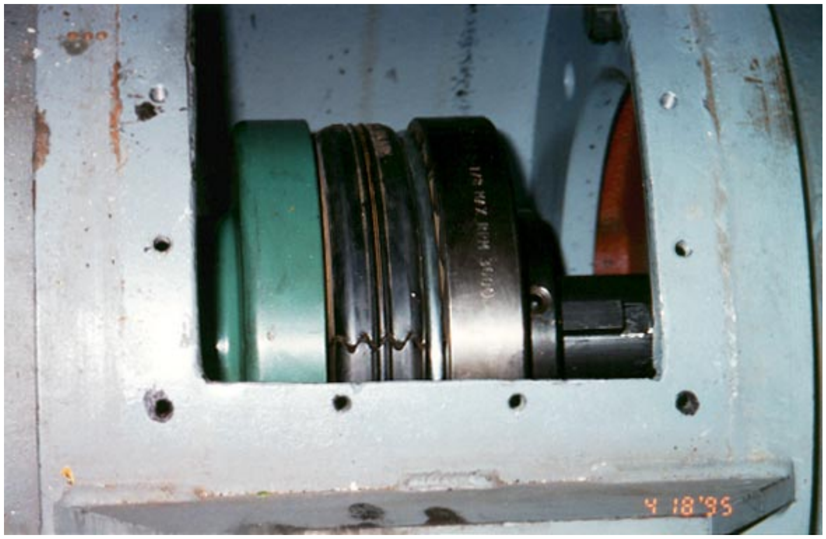


Figure 9. Eddy Pump to motor coupling.

Figure 10. Motor coupling disassembled view.

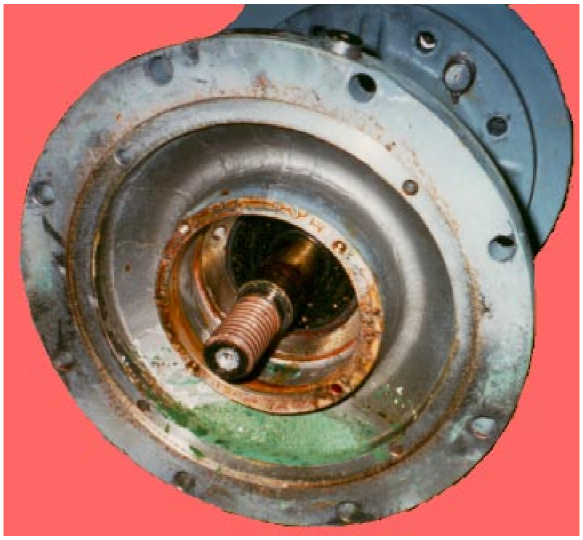


Figure 11. Indications of slight leakage residue.

Figure 12. Rotor locking device installed for additional security.

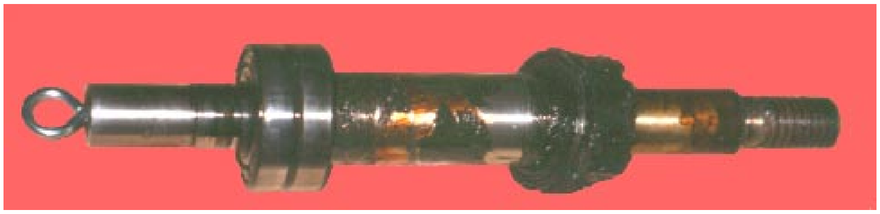


Figure 13. Replacement shaft/bearing assembly.

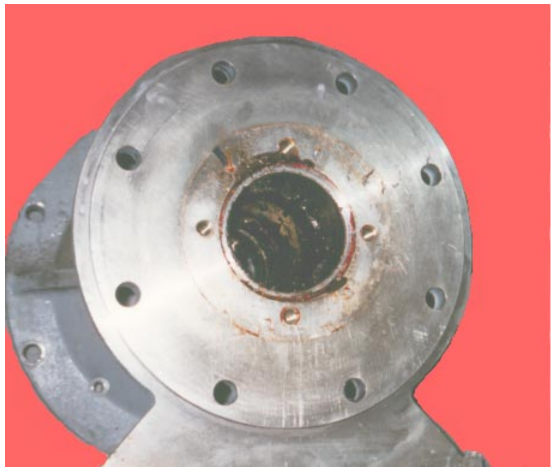


Figure 14. End housing of Eddy Pump.

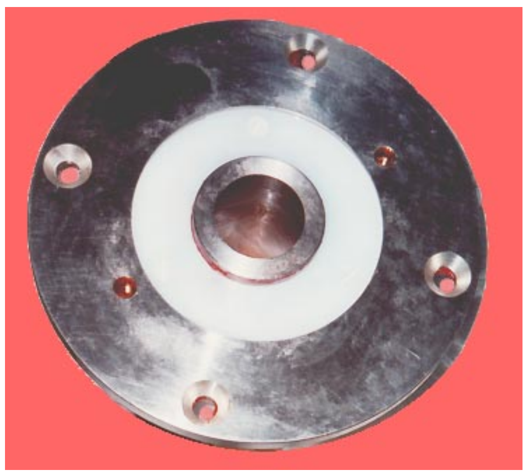


Figure 15. New seal assembly.

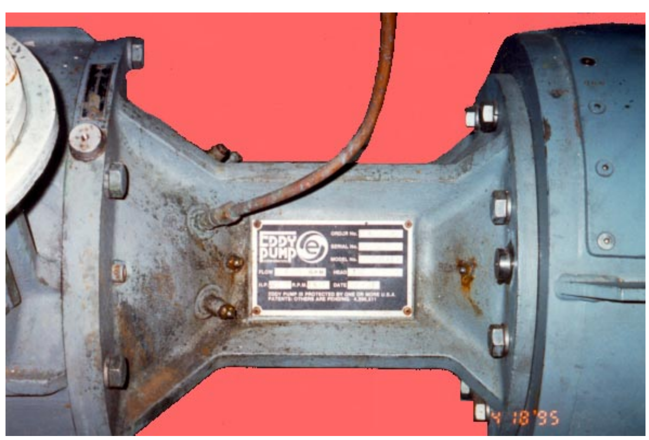


Figure 16. View of Eddy Pump showing location of fittings.

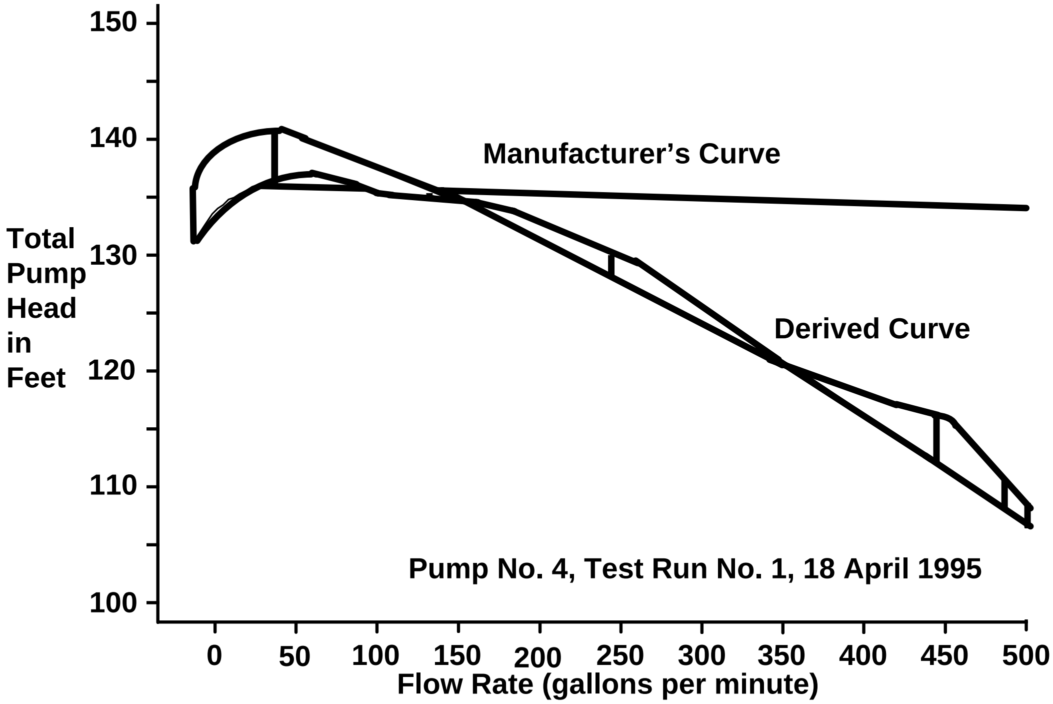


Figure 17. Pump performance curve Run No. 1.

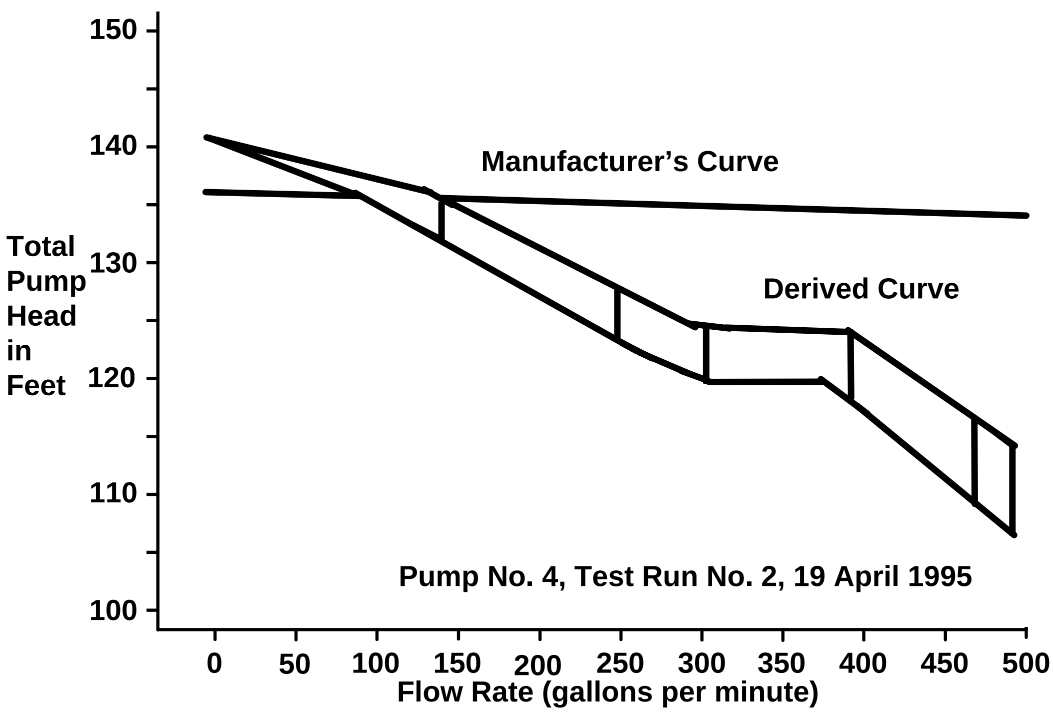


Figure 18. Pump performance curve Run No. 2.

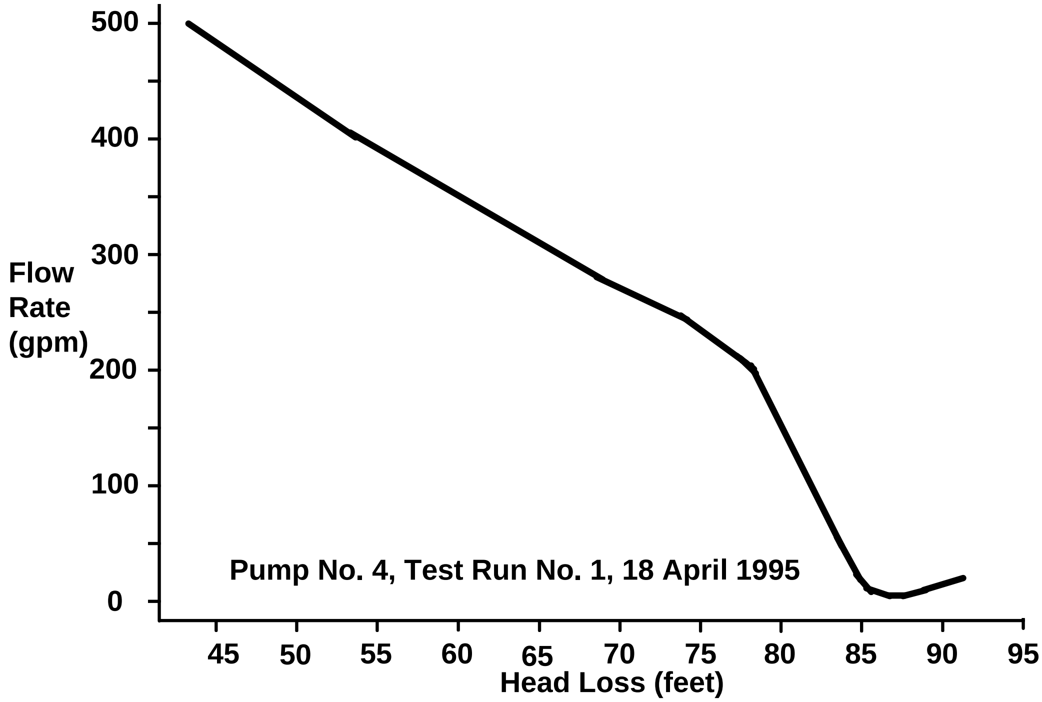


Figure 19. Pump head loss curve Run No. 1.

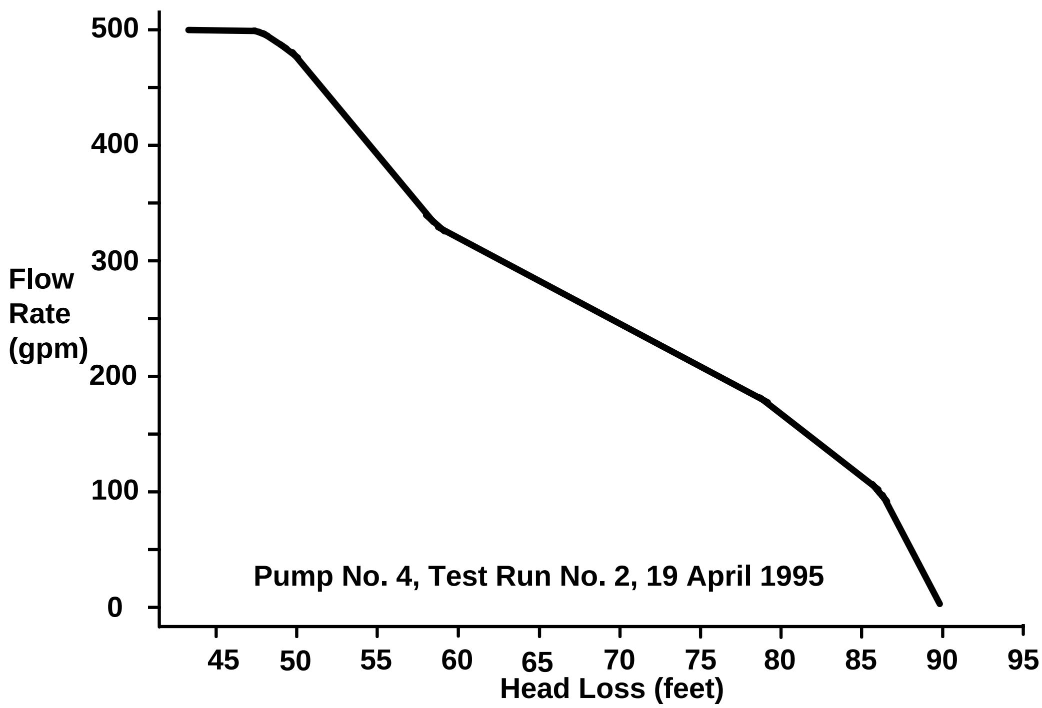


Figure 20. Pump head loss curve Run No. 2.

This paper was originally presented at the 5th Fleet Maintenance Symposium, October 1995, under the title of “Application of Zero Leakage Tornado Effect Pump in Carrier CHT Systems.” The paper was subsequently published in the March 1997 issue of the Naval Engineers Journal by the American Society of Naval Engineers under the current title.

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