DEVELOPMENT AND TESTING OF THE *GLENN EDWARDS* DREDGE PUMPS

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ABSTRACT

Manson Construction Co. performs marine construction and dredging throughout the coastal United States and overseas. In January of 2006, Manson launched the largest trailing suction hopper dredge in the United States, the *Glenn Edwards*, having a 9,175 m^3 (12,000 $yard^3$) capacity. This dredge is fitted with two newly designed LHD-38x38-58(60.4) C/4HE GIW dredge pumps having 1.53 m (60.4 inch) diameter impellers with 1 m (39.37 inch) suctions, each being driven by a 1721 kW (2308 hp) diesel engine. These pumps were developed specially for this dredge and tested within the GIW Hydraulic Test Lab up to flows exceeding 6800 l/s (107,794 gal/min) corresponding to 307 rpm operation.

The writers in this paper describe the design considerations, CFD modeling and development of the hydraulics for this new pump, some of the mechanical details of its construction and how its performance was validated under full-scale test conditions.

Illustrative examples of pump operation for both sand-type dredging and water filling applications have been provided. In the case of the sand dredging scenario, no significant dredge limitation i.e. engine speed limit, engine power limit, pump/system suction limit (NPSH) would be reached. In the water filling illustration, the new design pump achieved ~21% greater flow-rate resulting in a 21% reduction in hopper fill time. At the maximum flow-rate, the pump consumed the full engine capacity and has reached its suction limit (NPSHR) based on the system NPSH availability indicating the design did meet the desired goals for the field application.

Keywords: Hydraulic design, trailing suction hopper dredge

INTRODUCTION

The January 2006 issue of *World Dredging – Mining & Construction* included a four page write-up on the *Glenn Edwards*. Since 1978, the United States Congress has encouraged the development of a private hopper dredge industry to replace US Army Corps of Engineers (USACE) dredges. In the recent Corps of Engineers report to Congress, the addition of the *Glenn Edwards* increases the US hopper dredge capacity by 18%. Due to the addition of this large dredge, the Corps will be retiring their dredge, the *McFarland*. With the addition of the *Glenn Edwards* to its fleet, Manson Construction Co. now operates four hopper (TH) dredges. This comprises the largest and most modern fleet in the U.S. industry as well as surpassing Great Lakes Dredge and Dock for number one in overall Trailing Hopper (TH) dredge volume.

This 9,175 m^3 capacity dredge is fitted with two newly designed LHD-38x38-58(60.4) GIW ladder dredge pumps that were developed specifically for this dredge and tested within the GIW Hydraulic Test Lab.

What makes this hydraulic design significant is that these pumps enable the dredge to utilize all of the available engine power under maximum speed condition without being suction limited during water filling operation to flows of 7000 l/s (111,000 gal/min). When considering that the reason for constructing and operating a dredge is to move solids, it is also imperative to state that these pumps are designed to also operate effectively during demanding loading operations as will be illustrated in an example later within this paper.

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There are many factors that influence the proper selection and design of a centrifugal slurry pump for a specific application. One of the best references on the subject matter would be *Slurry Transport Using Centrifugal Pumps, 3rd Ed.* (Wilson et al 2006). Some of the basic parameters that have to be considered include the slurry properties, flow range, head requirements, efficiency, suction performance, sphere passage, pressure rating, resulting component wear, capital costs, etc.. In most situations, an existing pump is selected that will perform acceptably well and meet the application requirements. When desirable, a tailored hydraulic design can be created to provide optimum performance for a specific application.

For the purposes of this paper, primary focus will be given to the hydraulic performance aspects of a new design with minimal details provided for what will here be considered less interesting parameters as associated with the mechanical end selection.

DESIGN CONSIDERATIONS

This process began with Hagler Systems Inc. working with Manson Construction to select an existing GIW pump that could meet the planned application. The more than 3,100 currently available hydraulic designs were narrowed down to the best selection being the LHD-38x38-58 C/3HE pump which has been applied to numerous slurry transport applications over the years.

Figure 1 represents operation of the existing 3-vane LHD-38x38-58 at 307 rpm on the water filling application. It should here be stated that a centrifugal pump always operates at the intersection point of the pump and pipeline system head curves. When operating this pump at a fixed speed, the flow-rate increases as the ship is filled due to the reduction in system static level. As seen here the flow-rate would initially be 5300 l/s (84,016 gal/min) into an empty hopper and progress to a maximum flow-rate achievable of 5900 l/s (93,527 gal/min) for Final Ship Draft condition. Here the pump would only be consuming 1333 kW (1788 hp) of the available 1721 kW (2308 hp). Also, the pump has not achieved its suction limit (Net Positive Suction Head Requirement, NPSHR) based on the system NPSH availability. In essence, installing this standard pump to the dredge would not maximize the usage of the available engine power nor consume the available NPSH, thus it would not maximize the water filling performance.

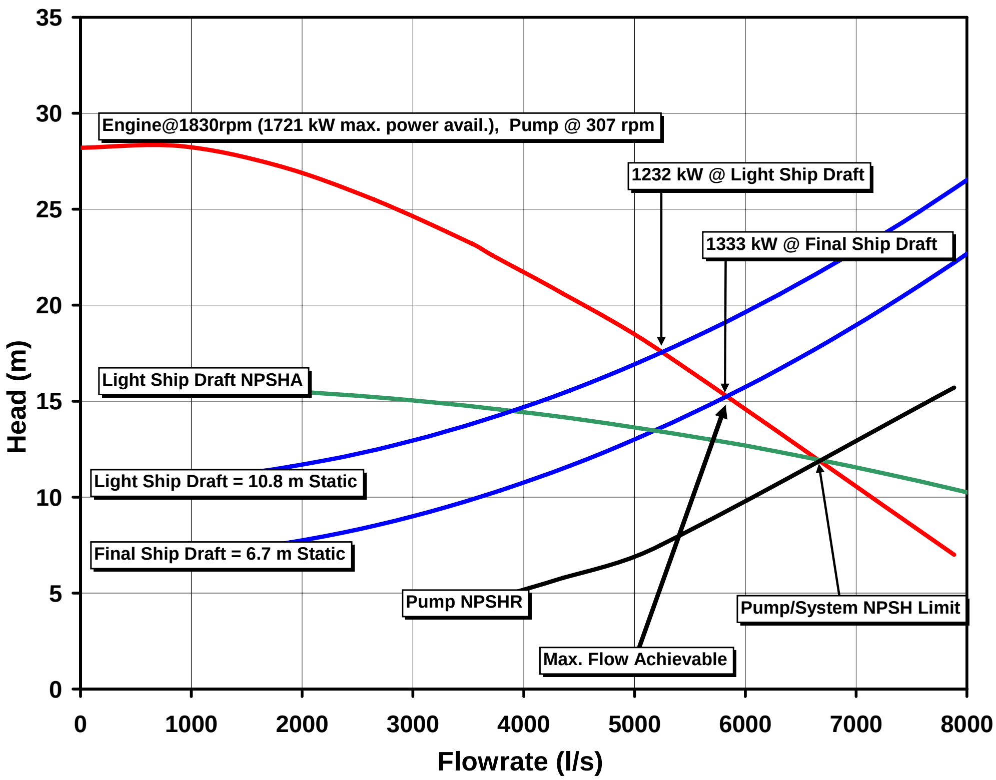

Figure 1. Existing 3-vane LHD58 pump @ 307 rpm filling hopper with water at an average flow-rate of 5600 l/s (88,772 gal/min).

One means of increasing the flow and head of the pump is to increase the shaft rotational speed. In Figure 2, the speed of the pump has been increased (to 333 rpm) until the flow-rate reached forces the engine to operate at its maximum capacity of ~1721 kW. At this condition the resulting maximum flow-rate attainable would be 6500 l/s (103,038 gal/min). It can also be seen that at higher speeds, cavitation will result in excessive head and efficiency losses once operating at or above the pump/system NPSH limit.

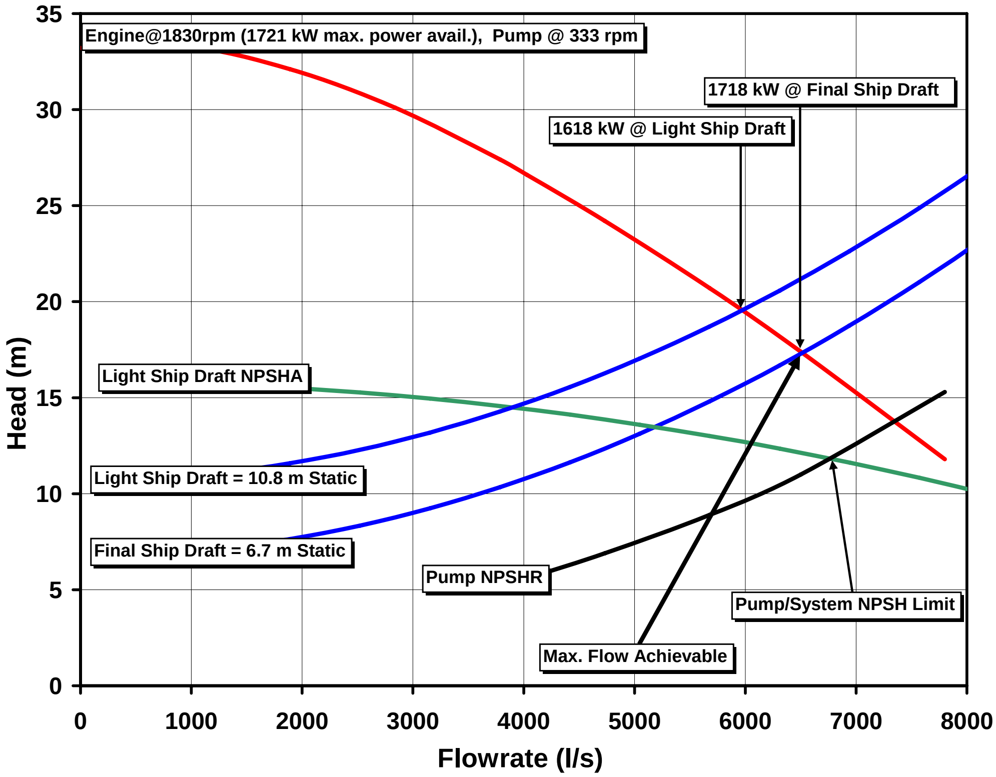

Figure 2. Existing 3-vane LHD58 pump @ 333 rpm filling hopper with water at an average flow-rate of 6225 l/s (98,680 gal/min).

It should here be stated that pump wear, due to operation on slurry, approximately increases in proportion to the speed change raised to the 3rd power. As a result, the pump wetted parts lives would be reduced if the gear reducer ratio was changed to reflect 333 rpm operation. In addition, operating at lower speeds is also beneficial from an NPSHR view point.

It was established that this existing design though possible for the application would be more desirable if it were tailored to better utilize the project capital investment. Subsequently, to accomplish this task, Manson partnered with GIW Industries Inc. who is well known for custom designing large dredge pumps when supporting the international dredge industry.

The goal of the hydraulic design effort was to maximize the production of the pumping unit within the constraints of the chosen pump/motor combination with the available motor horsepower being 1721 kW (2308 hp) at rated speed of 307 rpm. In order to achieve this goal, it was necessary for the pump to absorb all of the available motor power and to convert as much as possible into energy in the form of flow and pressure. This would require a high head impeller with optimum pump efficiencies over a wide range of flows.

One of the design challenges was that traditional methods used to increase the output energy of a pump, such as increasing pump speed or number of vanes, would generally have an adverse impact on the NPSHR performance and could result in the pump entering into cavitation before absorbing full power.

It was also agreed that it would be advantageous to cast the test impeller of a softer material thus ensuring that modifications would be possible during the laboratory testing for refining the design and resulting performance. The test impeller was therefore cast of ductile iron, rather than the 650 Brinnell hardness high chrome white iron which production parts would be.

The design goals could be summarized as follows:

1. To maximize the ratio of head produced relative to NPSH required across the range of flow, but especially at higher flows where cavitation was most likely. Ideally, no cavitation would occur before 100% of the available power was consumed.

2. To maximize pump efficiency across the range of flow in order to maximize the available output energy, especially at higher flows where the power requirements of the pump would be greatest. Ideally, at least 80% overall pump efficiency would be achieved from 3,790 to 6,310 l/s (60,000 to 100,000 gpm).

CFD MODELING AND DEVELOPMENT

Cavitation in a centrifugal pump impeller almost always begins at the leading edges of the vanes where the system pressure is lowest and the acceleration of the fluid around the vane inlet creates a region of higher velocity and correspondingly lower pressure (see Figure 3). This effect can be minimized by proper vane inlet design, but it can never be fully eliminated. Of special impact is the attack angle of the flow relative to the leading edge of the vane.

To assist in achieving the design goals, GIW engineers employed specialized Computational Fluid Dynamics (CFD) tools. These are computer programs which solve for the flow and pressure fields within the impeller by dividing the flow passages into thousands of small bricks (or “elements”) for which generalized equations of flow based on the well known Navier-Stokes equations can be solved. Taking advantage of the computational power and speed of the computer, these programs solve the flow equations for thousands of elements simultaneously, ensuring that the results match at each adjacent boundary. The value of these tools to the designer is that they provide a complete picture of the flow and pressure fields within the pump within an accuracy of a few percent, provided that enough elements are used to properly resolve the small scale flow patterns. In the present case, particular attention was paid to the following items:

1. The total developed pressure of the impeller and the corresponding input power for the purpose of insuring that all of the available power would be utilized in the most efficient manner.

2. The existence of recirculation or “stalled” areas within the impeller flow field which generally lead to pressure and efficiency losses. These are common in slurry pump impellers due to the need for large sphere clearance, but can be minimized.

3. The small scale pressure distribution around the leading edges of the vanes for the purpose of predicting the required NPSH and the onset of cavitation.

Before using the CFD tools, which provide only feedback and not direction, the designers must use their knowledge of slurry pump dynamics to generate design alternatives which would have the most likely chance of success. The CFD tools are then used to evaluate each option and confirm/disprove (as the case may be) the designers’ approach.

In the present case, it was chosen to increase the energy output of the pump without increasing pump speed (which generally has an adverse effect on the NPSHR performance) by first maximizing the impeller diameter within the existing shell from 1473 to 1534mm (58 to 60.4 inches). Next, the number of vanes was increased from 3 to 4. While this usually has an adverse effect on NPSHR by increasing the velocity around the inlet edges of the vanes, the effect can be minimized by careful profiling of the leading edge shape and by moving the edge to a slightly larger radius from the axis of the suction inlet.

To optimize the NPSHR performance over the range of flows, the vane leading edge angle was adjusted for best angle of incidence resulting in the smallest pressure drop. Cavitation in the centrifugal pump impeller almost always begins at the leading edges of the vanes where the system pressure is lowest and the acceleration of the fluid around the vane inlet creates a region of higher velocity and correspondingly lower pressure. Although this effect cannot be eliminated, it can be minimized by optimizing the angle of attack and reducing the local pressure drop. The vane angles must vary along the leading edge of the vane since different points along the edge sit at different radii from the center and therefore move at different circumferential speeds. This variation in angles results in a twisted vane shape. A first approximation for the optimum inlet angles may be derived geometrically based on the known rotational speed of the impeller and inlet velocity of the flow.

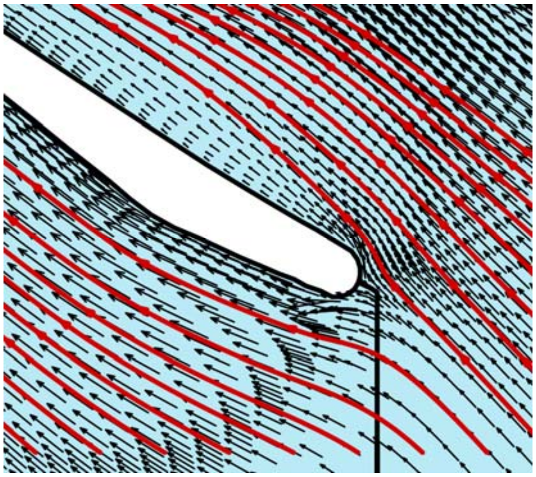

Figure 3. Optimization of the inlet edge angle resulting in twisted vane inlet geometry.

In the present case, design variants with inlet angles corresponding to flow-rates from 4,000 to 5,500 l/s were investigated. Although higher flow-rates were to be approached during operation, the angles eventually become too steep leading to flow instabilities at the lower flow-rates.

Finally, different vane angles at the outlet of the impeller were investigated to find the best balance between high angle (which would increase power) and lower angles which would improve vane shape and efficiency.

In total, 11 different designs were evaluated, including the original tested 3 vane impeller. Each design was evaluated in eight flow increments from 3,150 to 7,570 l/s (50,000 to 120,000 gpm) for a total of 88 solutions. Each solution utilized approximately 35,000 elements and required 2-4 hours of computer runtime to solve.

In Figures 4 and 5, we can see some CFD outputs for the original 3 vane and optimized 4 vane impellers at a flow-rate of 6,310 l/s (100,000 gpm). These plots show two important aspects of the designs. On the left side are the velocity vectors and streamlines of flow at a point on the leading edge midway between the shrouds. On the right is a contour plot of the developed energy (expressed in terms of angular momentum) as seen on the meridional section mid-way between two vanes.

In the vector plots, we see an improved angle of attack with the new design. Similar plots were made for other operating conditions and for other locations on the vane leading edge to insure the best overall result. In the meridional plots, we see the new design producing more energy (head), as well as a more regular development of head along the path from inlet to outlet, a result that tends to indicate best vane efficiency. By comparison, the original design produces more energy near the inlet and less towards the outlet.

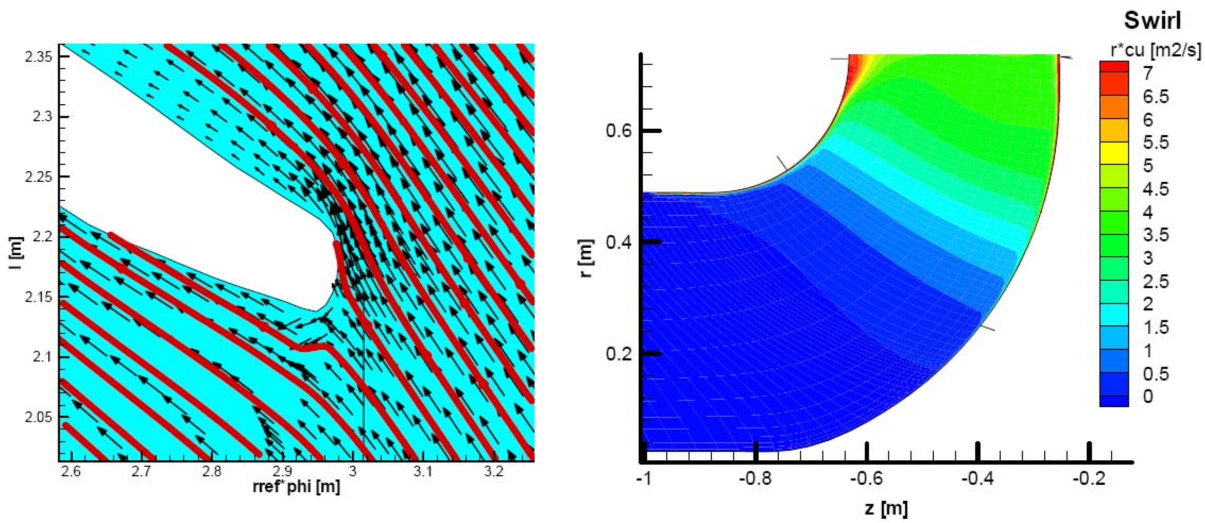

Figure 4. CFD results of original 3 vane design.

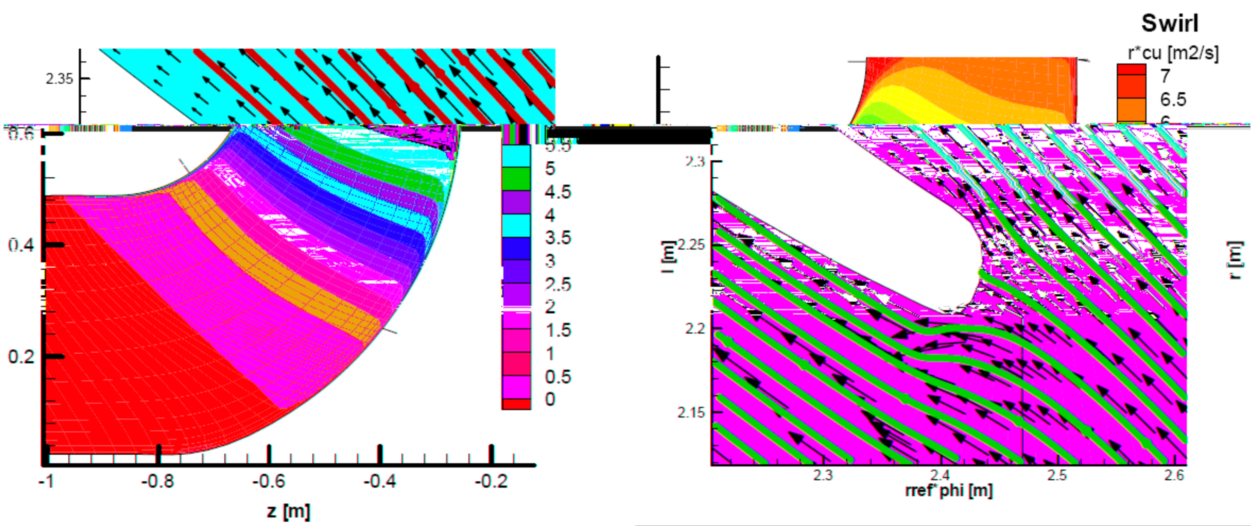

Figure 5. CFD results of optimized 4 vane design.

To give an overview of the entire analysis, a summary plot was created (see Figure 6) showing the CFD head, efficiency and incipient NPSHR for both designs together with the tested performance for the original 3 vane and estimated performance for the new 4 vane designs. To generate the CFD head curves, the calculated pressures and velocities over the entire inlet and outlet areas were summed up for each design at each operating point. The CFD efficiency curves were determined in a similar way, but by instead considering the power absorbed based on the pressure differential over the leading and trailing faces of the blades (converted to torque). Note that these CFD efficiency curves represent only the impeller efficiency, with casing and mechanical end losses not included. The CFD NPSH curves represent incipient NPSHR, that is, the point at which visible cavitation bubbles begin to form. This is generally well in advance of the NPSHR3%, but the curves are still useful for comparative purposes. The measured performance curves for the original 3 vane impeller come from full scale, laboratory testing and the estimated 4 vane performance was determined by using the relative difference between the 3 and 4 vane CFD results, as well as cross checking against standard pump design scaling rules.

Of special note, the new design is seen to produce more head, require less NPSH and maintain good efficiency over the range of interest, all goals of the design effort.

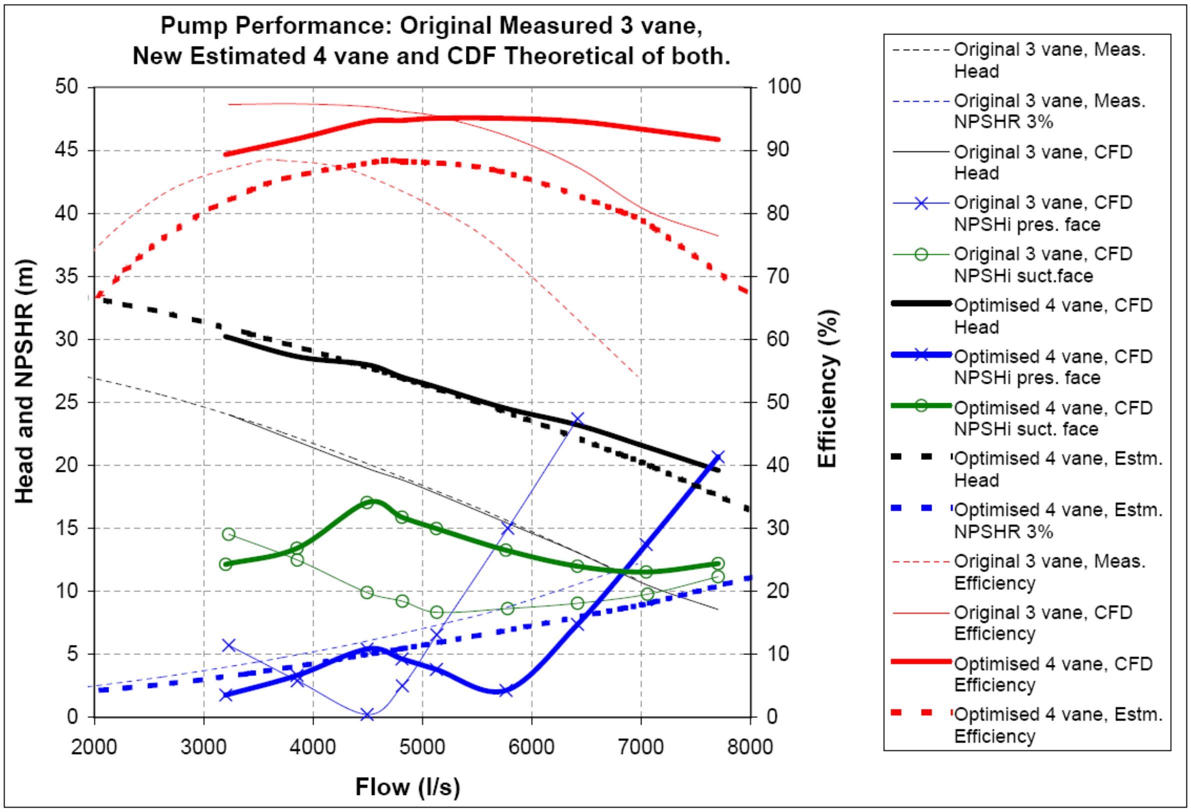

Figure 6. Summary of CFD analyses for the original and final designs at 307 rpm.

The estimated performance of the new 4-vane design in the planned application can be seen in Figure 7 along with the 3-vane performance. As seen in this figure, the combination of more head, good efficiency and improved NPSHR performance would result in the full absorption of available motor power and a more than 15% increase in the maximum flow-rate condition relative to the original.

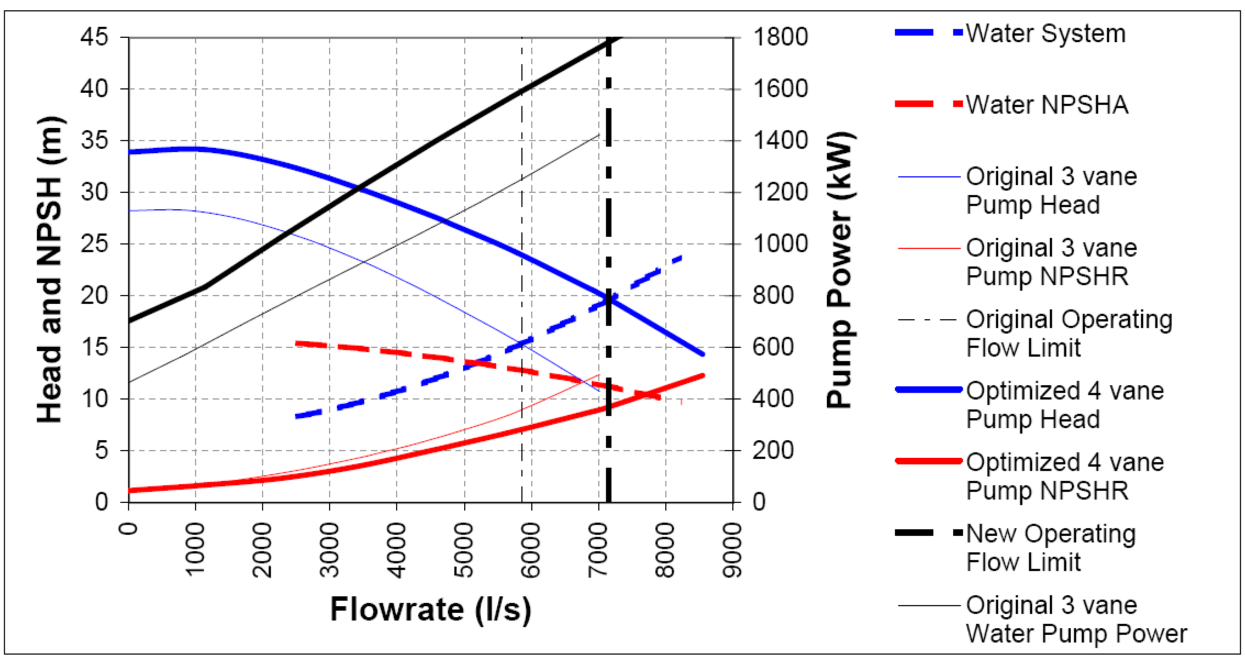

Figure 7. Estimated maximum operating condition for optimized 4 vane design at 307 rpm.

MECHANICAL DETAILS & CONSTRUCTION

A centrifugal slurry pump is more than just a wet-end or hydraulic; it must be equipped with the necessary mechanical assembly capable of handling the inherent static loads of supporting in this case a 3728 kg (8221 lb.) rotating assembly and the hydraulically induced dynamic loads. This aspect of the pump selection will not be dwelled upon herein, with focus simply on describing the selected mechanical end.

For more comprehensive understanding, the readers are directed towards the new ANSI/HI standard on Rotodynamic (Centrifugal) Slurry Pumps, ANSI/HI (2005), as produced by the Hydraulic Institute. That standard provides examples of all the different types of slurry pumps available and is beneficial when considering the various aspects associated with selection. Wetted materials of construction are also described with recommendations made for those along with the arrangement details. Recommended bearing housing and rotating shaft seals are described in some detail with a special section on mechanical seals and where they may be used. The standard also lays out shaft alignment limits and recommends minimum bearing lives.

Figure 8 provides an illustrative view of the LHD-38x38-58(60.4) C/4HE pump assembly. The hydraulic is supported by a fabricated steel pedestal that cradles the stiff 10-1/4" cartridge type bearing assembly. This assembly is fitted with DUO-Cone seals designed to meet underwater sealing requirements as necessary in a ladder pump operation such as this. The wet-end to shaft interface is sealed by a standard packed stuffing box. The entire bare-shaft pump assembly weighs approximately 28,576 kg (63,000 lbs.)

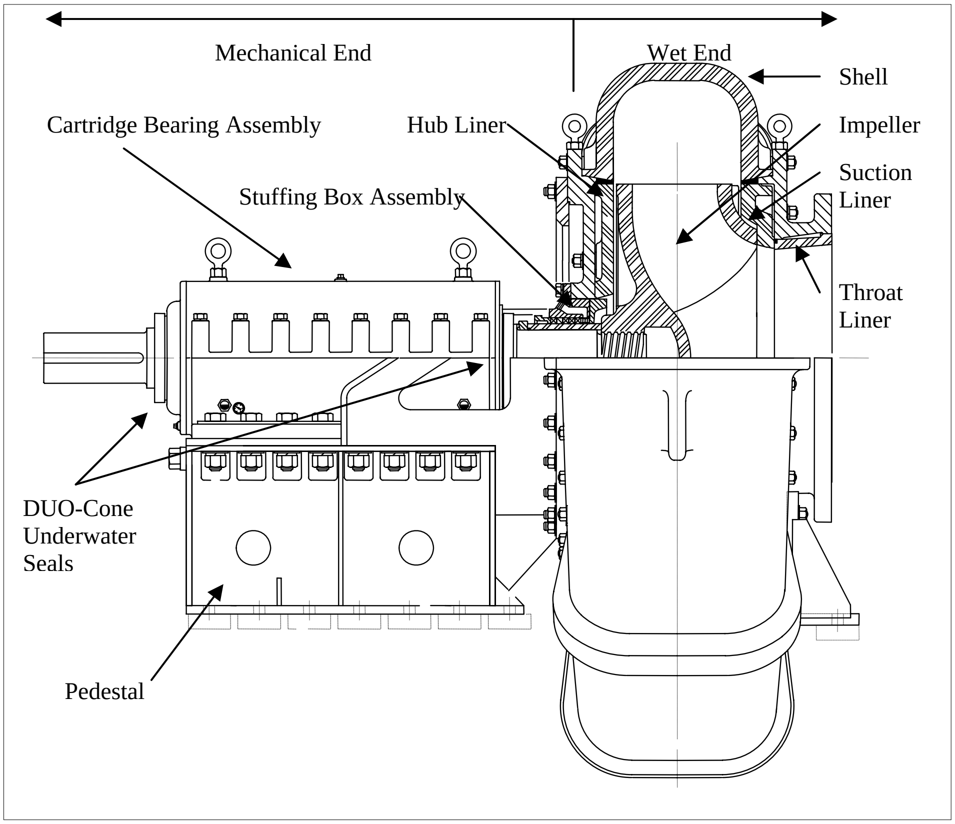

Figure 8. Bare-shaft LHD-38x38-58(60.4) pump assembly.

FULL-SCALE PERFORMANCE VALIDATION

Test Overview

In August of 2005, a series of tests were conducted on the LHD-38x38-58(60.4) C/4HE pump in the 1.22 m (48 in) diameter closed loop system at the GIW Hydraulic Test Lab. This testing was performed to determine the pump performance characteristics of the newly designed 4 vane impeller. Testing was to acquire data for establishing the pump's flow-rate versus total dynamic head (TDH), shaft power, suction performance (Net Positive Suction Head Require) and efficiency characteristics. Primary focus of this program was to collect data out to the highest possible flow-rate in order to provide Manson confidence in the improved performance characteristics of this design as it would relate to their application. The testing complied with Hydraulic Institute of Standards (ANSI/HI 1.6-2000) guidelines.

The Test System

A picture of the interior pump and pipeline setup can be found in Figure 9, with an overall schematic of the test system in Figure 10. As shown, the system ran from a 214,000 liter (56,400 gal) sealed tank to the pump and then back to the tank. This tank served as a reservoir to trap and remove entrained air from the system. It was mounted with a water inlet, drain valve and pressure/vacuum relief valves to protect against excessive loading. The suction pressure was controlled by changing the liquid level in the tank with the system sealed. A 1.22 m (48 inch) automated butterfly valve located before the tank on the return line, provided system flow control.

Pump head pressure taps were located two diameters away from the suction and discharge flange connections. Standard wall 1 m (39.37 inch) suction and 0.93 m (36.5 inch) discharge piping was used.

The system drive train was powered by a 4160 volt, 1830 kW (2450 hp), 450 rpm AC motor. The output of this motor was connected to a variable speed fluid drive using a jackshaft v-belt drive arrangement. The output of the fluid drive was then run through two gear reducers providing the final test speed of 270 rpm to the pump. Results were scaled to the design speed of 307 rpm via the affinity laws for evaluation.

The test impeller was cast of ductile iron rather than the harder high chrome white iron so that after initial testing, minor modifications to the vane profiles could easily be made for investigation of possible improvements.

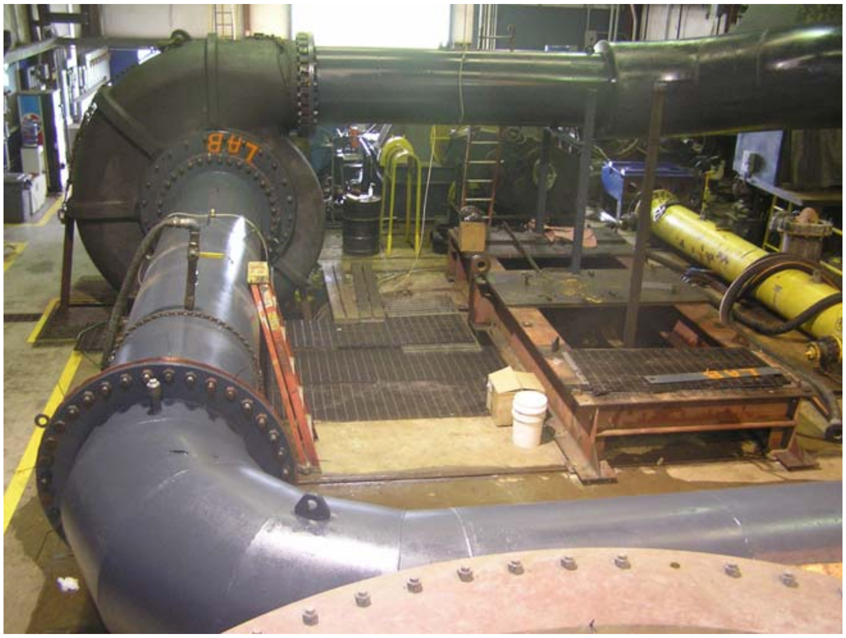

Figure 9. LHD-38x38-58(60.4) C/4HE pump on GIW test stand.

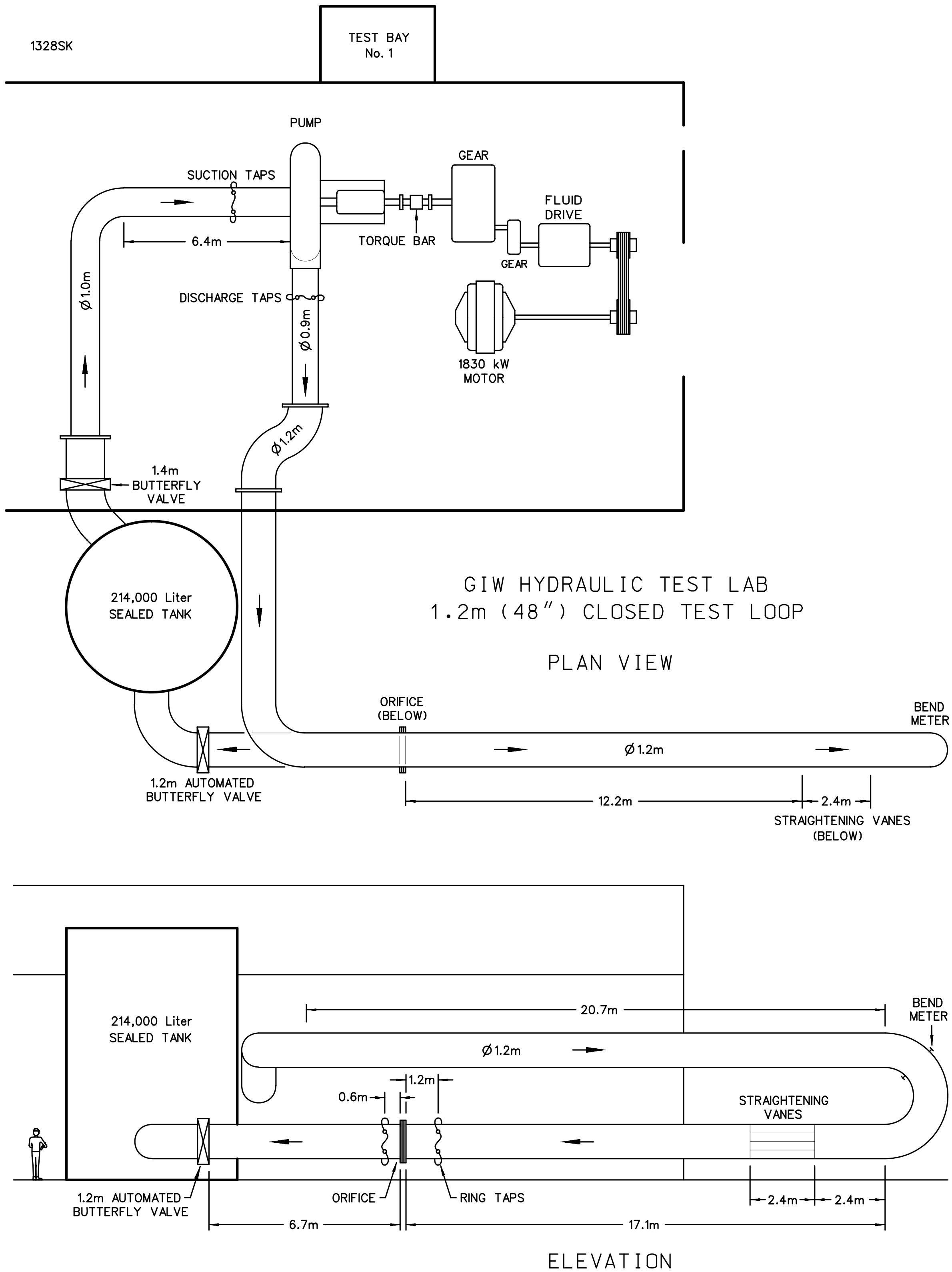

Figure 10. GIW 1.2 m (48 inch) diameter closed loop test system.

Test Instrumentation

The GIW Hydraulic Test Lab instrumentation was used for the testing. Instrumentation selection, accuracy, calibration intervals and installation was in accordance with ANSI/HI 1.6-2000 standards requirements. During testing, measurements were taken with both primary and secondary instruments.

The primary flow meter for the 1.2 m (48 inch) system was a machined square edged stainless steel orifice plate. This plate had an opening of 0.8382 m (33.0 inch) and was installed in a 1.2 m (48 inch) pipe. As shown in Figure 10, this plate was located downstream from the pump before the control valve. Pressure taps for the plate were located one diameter upstream and a half a diameter downstream. Multiple differential pressure transducers were assigned to this orifice plate of various ranges to ensure accurate measurements. The flow coefficient for the orifice plate was calculated according to the American Society of Mechanical Engineers, Fluid Meters book, 6th Edition.

The secondary flow meter in the system was a 1.2 m (48 inch) bend flow meter. This meter calculates flow rate from the measured pressure difference between the inner and outer curvature of the bend. As shown in Figure 10, this elbow meter was located downstream of the pump. Due to the best accuracy limit of this meter being +/- 5% it was used as a backup indication. (Throughout all of this test work the bend meter read 4 to 7% higher flow rates than the orifice plate when using the calculated flow coefficient.)

All pressure measurements used for the pump suction, discharge, orifice and bend meters were measured during the tests with differential pressure transducers. These transducers are certified twice a year using a certified dead weight tester, inclined mercury manometer and a 6.1 m (20 ft) water column. Transducers that had converted readings that varied more than 0.25% of full scale were re-calibrated.

To account for variation in the density and vapor pressure of water with changes in temperature, an RTD type 100-ohm platinum temperature transducer was located in the tank. A second RTD was used to measure the lab ambient temperature.

The pump input power was measured using a 500,000 in-lb torque sensor mounted directly between the pump and marine gear box. A magnetic pickup in conjunction with a 60 tooth gear provided the shaft rotational speed reading. Pump shaft input power was determined from the torque and speed. The resulting pump efficiency was calculated from dividing the power put into the water (in terms of flow and head) by the shaft input power.

All instrumentation signals were fed through an analog to digital (A-D) conversion unit. Measurement sampling occurred simultaneously and automatically at a rate of 100 readings per second over any previously selected 32 channels. Monitoring and averaging display times could be varied at the console during a test. Storage of data was arranged by depressing a key at the operators' console. Display during a test included readings and calculated results and was according to test type and the software selected.

Test Results

Testing of the as-delivered pump was begun on August 1st 2005. Once the pump was installed and all settings made according to the GIW Maintenance Manual, initial performance testing was conducted. Operational data for pump flow rate versus head, power, % efficiency and suction performance (NPSHR) was collected. Though the results indicated the design closely agreed with that estimated, it was believed that improvements were possible. As a result the impeller was removed with vane inlets modified as instructed by the GIW design team. Upon completion and reinstallation, 2nd phase testing was performed up to flows exceeding 6800 l/s (107,794 gal/min) corresponding to 307 rpm operation. Figures 11, 12, 13 and 14 reflect performance comparisons of the initial and modified impeller designs to that estimated, along with the original 3 vane performance. By observing these Figures it can be seen that the performance was relatively close to that estimated.

Upon completion of the test program, multi-speed performance curve E19B-04 (Figure 15) was then generated to reflect the tested performance; the pump was dismantled with the impeller patterns updated to match final test geometry leading to high chrome white iron impellers being cast for installation by Manson.

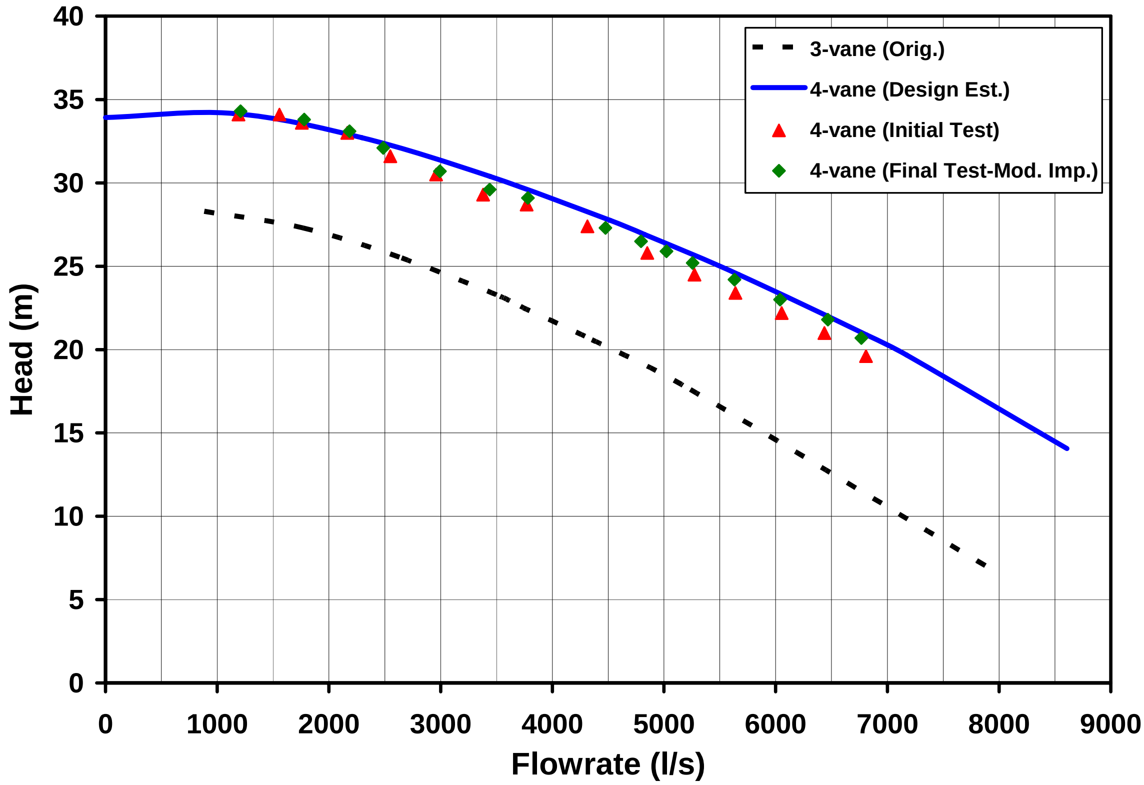

Figure 11. Test data comparison to design estimate for flow-rate versus head at 307 rpm operation.

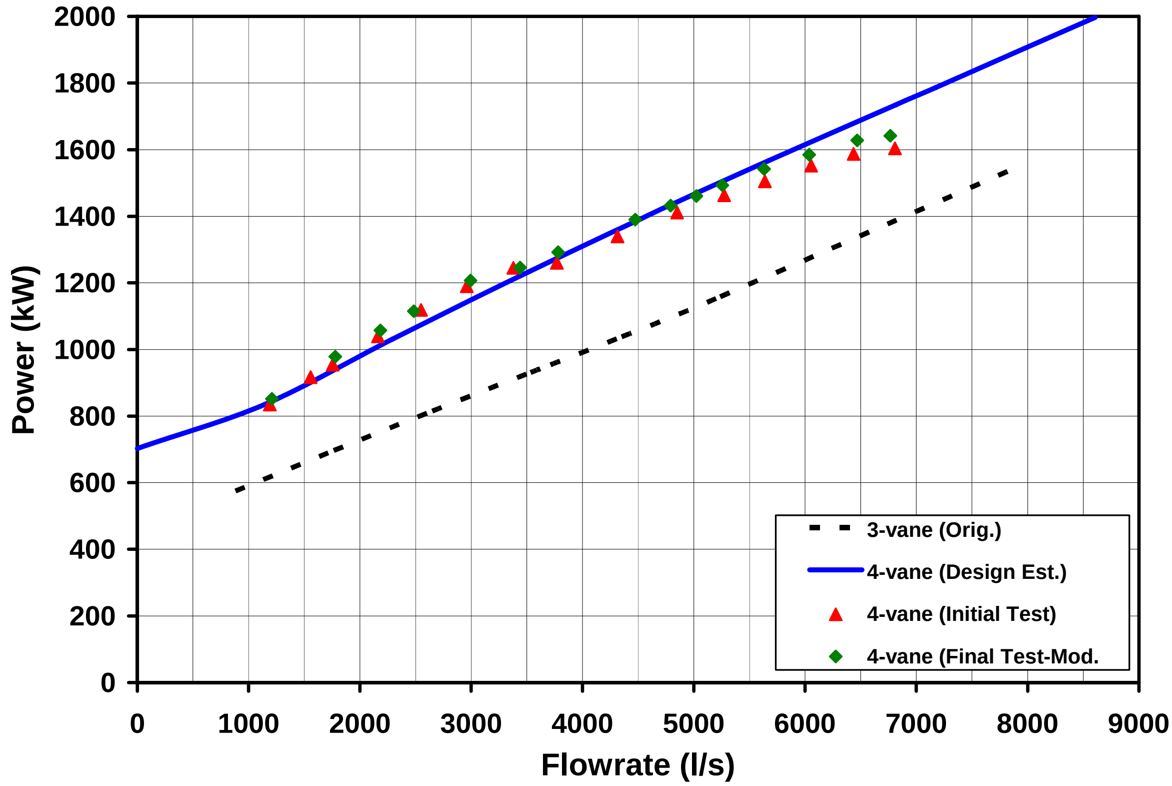

Figure 12. Test data comparison to design estimate for flow-rate versus power at 307 rpm operation.

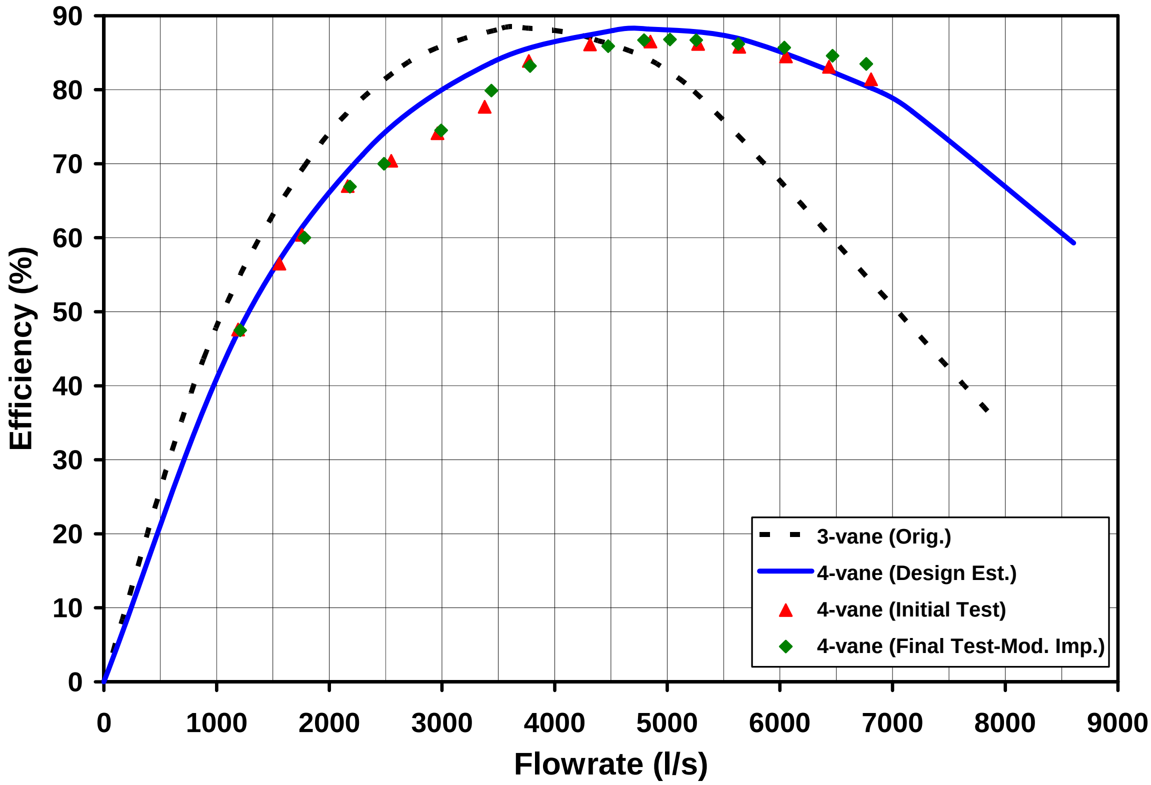

Figure 13. Test data comparison to design estimate for flow-rate versus efficiency at 307 rpm operation.

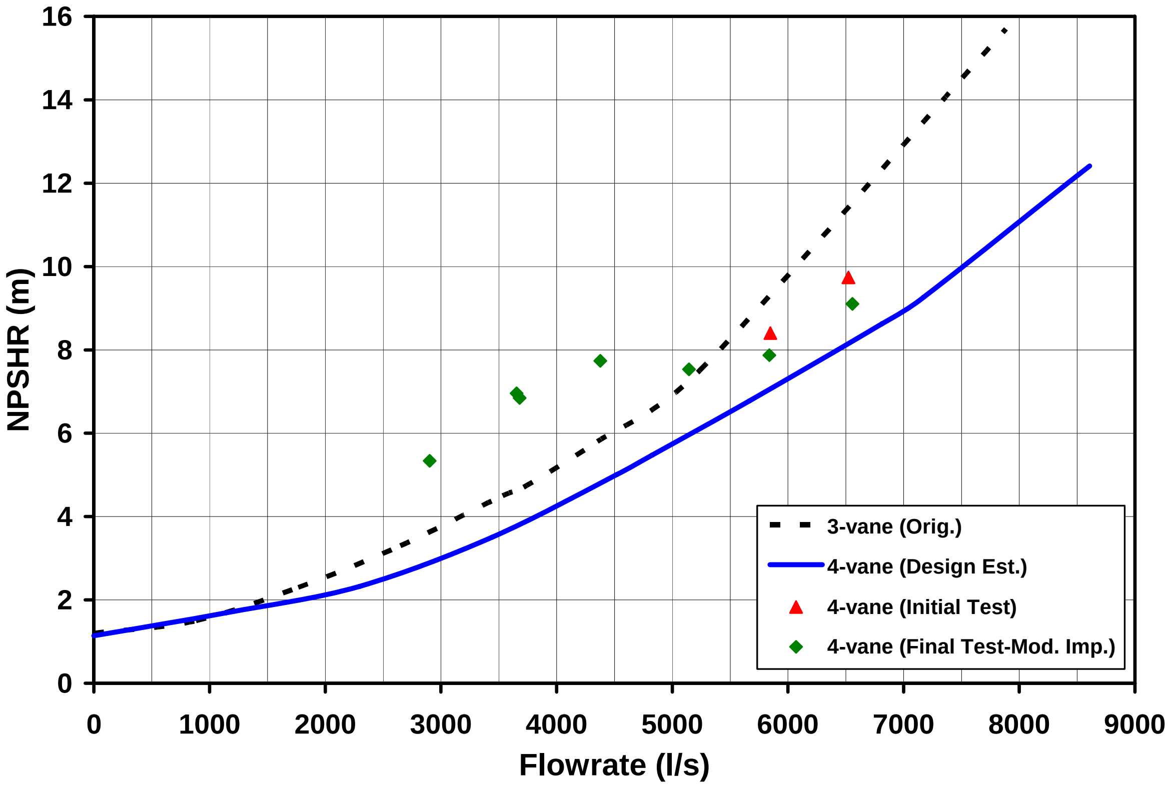

Figure 14. Test data comparison to design estimate for flow-rate versus NPSHR at 307 rpm operation.

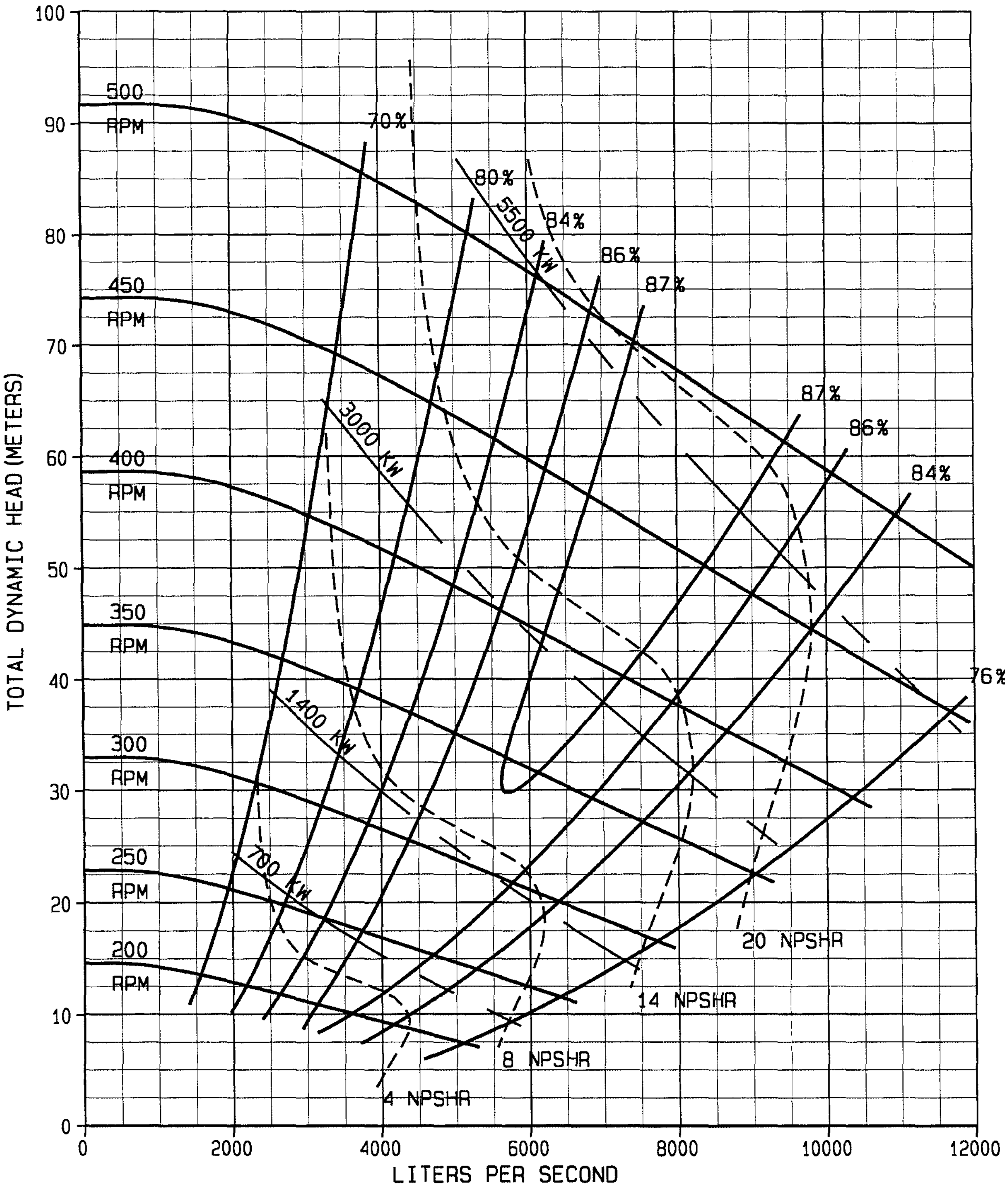

Figure 15. New 38x38LHD58(60.4) multi-speed water performance curve based upon test data.

FIELD INSTALLATION AND PERFORMANCE INDICATIONS

With laboratory testing confirming the design improvements were realized, high chrome impellers were cast to the final design dimensions. Upon completion the pumps were installed onto the dredge drag arms in preparation for the water field trials. Figure 16 illustrates one of the drag arms as fitted with the new pump.

Figure 16. New 38x38LHD58(60.4) C/4HE pump installed onto one of the *Glenn Edwards* drag arms

Figure 17 illustrates operation of the new LHD-38x38-58(60.4) C/4HE at 307 rpm during water filling of the hopper. As seen here the flow-rate would initially be 6500 l/s (103,038 gal/min) into an empty hopper and progress to a maximum flow-rate achievable of 7100 l/s (112,550 gal/min) for Final Ship Draft condition, thus resulting in an average filling rate of 6800 l/s. At the maximum flow-rate we see that the pump is consuming the full 1721 kW (2308 hp) engine capacity. Also, the pump has reached its suction limit (Net Positive Suction Head Requirement, NPSHR) based on the system NPSH availability indicating the design has met the desired goals in the field application.

This translated to 21% increase over the existing 3-vane impeller design due to the additional head developed and improved suction performance of the new version. Subsequently, this would result in a reduction in filling time from 13.7 to 11.25 minutes, i.e. 21 % for 5600 versus 6800 l/s for 2 pumps operating at the average fill rate, respectively, for the 9175 m^3 hopper.

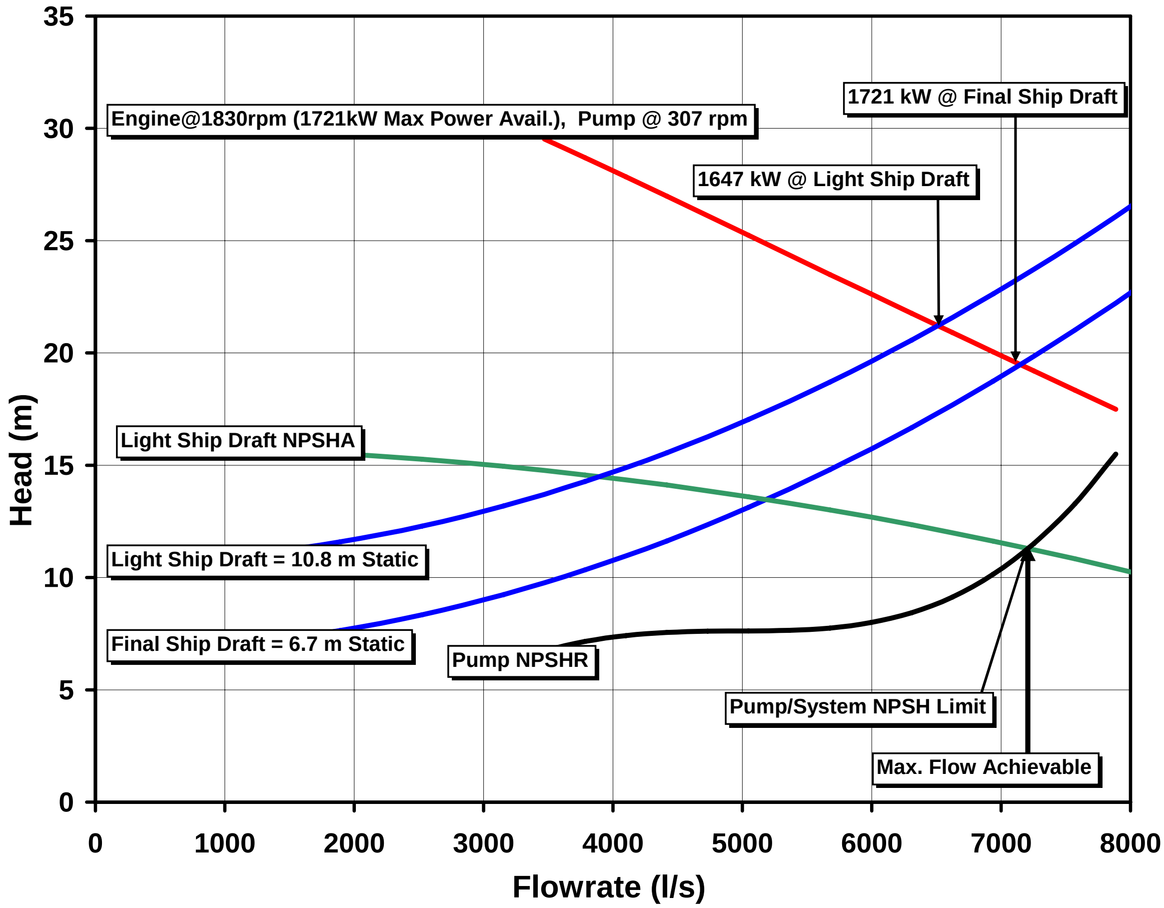

Figure 17. New LHD-38x38-58(60.4) C/4HE pump @ 307 rpm filling hopper with water at an average flow-rate of 6800 l/s (107,794 gal/min).

It would now be beneficial to evaluate and illustrate a solids loading application. This example is considered to be representative of a typical upper operating limit for hopper dredging within the United States within shallow channels at a high concentration.

Consider Figure 18 which reflects a solids loading operation on 1.4 SG sand slurry at 4750 l/s (75,300 gal/min) for a 15 m (50 ft) dredging depth. Loading at 7300 m^3/hr (9,550 yd^3/hr) of solids per pump into the 9,175 m^3 hopper, would result in a 40 minute fill time when pump speed is increased to maintain a constant flow-rate. It can be stated that no significant dredge limitation i.e. engine speed limit, engine power limit, pump/system suction limit (NPSH) would be reached.

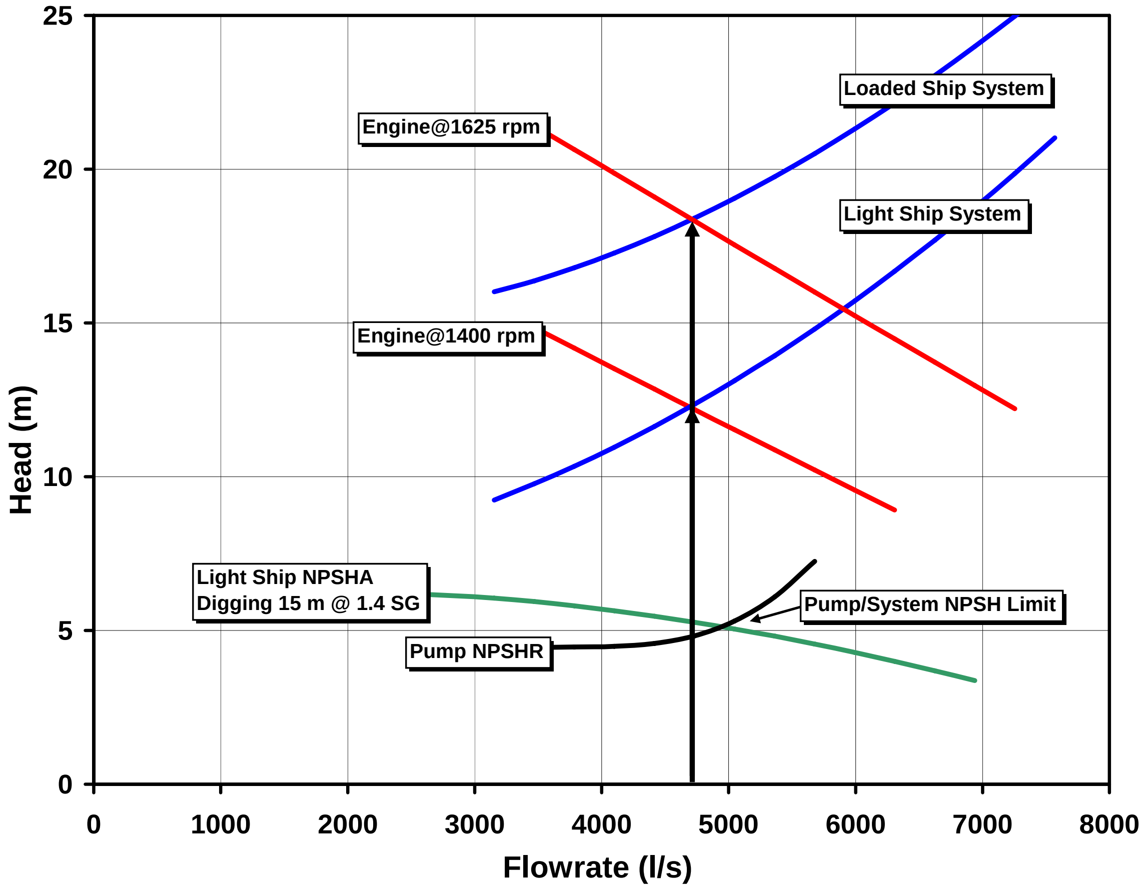

Figure 18. Illustration of hopper solids loading at 1.4 slurry SG and 4750 l/s for 15 m dredging depth on sand.

CONCLUSIONS

The writers in this paper have described the design considerations, CFD modeling and development of the hydraulics for a new LHD-38x38-58(60.4) C/4HE pump, some of the mechanical details of its construction and how its performance was validated under full-scale test conditions.

Laboratory water testing has been presented verifying the design to flow-rates of 6800 l/s (107,794 gal/min) corresponding to 307 rpm operation. Flow-rate versus head, power, efficiency and NPSHR characteristic curves were established which indicated the newly designed 4-vane impeller performed closely to the estimated design performance. Test data obtained formed the basis for generating a multi-speed sales curve to be used for applying this pump to future applications.

Illustrative examples of pump operation for both sand-type dredging and water filling applications have been provided. In the case of the sand dredging scenario, no significant dredge limitation i.e. engine speed limit, engine power limit, pump/system suction limit (NPSH) would be reached.

In the water filling illustration, the new design pump achieved ~21% greater flow-rate resulting in a 21% reduction in hopper fill time. At the maximum flow-rate, the pump consumed the full engine capacity and has reached its suction limit (Net Positive Suction Head Requirement, NPSHR) based on the system NPSH availability indicating the design has meet the desired goals for the field application.

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