

PUMP RERATES – WHEN AND HOW

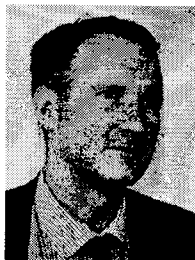
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ABSTRACT

During the life of a plant, there often arises a need to change a pump's rating to match a new plant operating condition or raise the mean time between repair (MTBR) of an unreliable pump. In these cases, rerating the pump instead of buying and installing a new one, if that is feasible, offers the plant owner worthwhile capital savings. Often the capital savings are sufficient to lower the payback period and raise the return on investment (ROI) to the point where the project becomes financially viable. The tutorial first addresses the circumstances when a rerate is warranted, then how to determine the new rating, and finally, by way of three examples, the general techniques for carrying out rerates.

INTRODUCTION

Most refinery owners seek to maximize refinery throughput in order to realize a higher ROI. They accomplish this by progressive debottlenecking of the various units that make up the refinery. Debottlenecking frequently involves increasing the mass flowrate through a particular unit, raising the pressure at which it operates, or some combination of the two. This, in turn, means the pump or pumps that charge feed to the unit need to deliver a higher flow, higher head, or both. With ingenuity it is often possible to achieve the new pumping conditions by rerating the existing pumps. Doing so lowers the investment needed for the debottlenecking thereby improving its financial viability. At the same time, particularly in older units, the pump rerate can include basic machine design improvements to correct reliability problems and thereby raise MTBR. Such improvements also help raise unit ROI by lowering maintenance expenditure and raising unit availability.

In some instances, it is necessary to reduce the rate of a particular refinery unit to balance refinery operation. This means

the pump or pumps charging feed to the unit have to operate at a lower flow. Depending on the degree of flow reduction and how it is achieved, operating units at lower rate can lead to unnecessarily high energy consumption or lower MTBR. When that is the case, rerating the pump so its hydraulics better match the actual operating flow is financially viable if the payback period on the investment to do it is short enough.

This tutorial provides guidance on pump rerates by first discussing the circumstances when a rerate may be appropriate. Next it deals with determining the new pump rating taking advantage of data from the operating unit. The last piece, how rerates are done, is a more complicated topic because each rerate is different in detail. Three examples, two for higher unit flow rate, the third for lower, are reviewed in broad detail to cover this aspect.

WHEN TO CONSIDER A RERATE

The following three circumstances warrant considering a rerate:

- *Increase in unit flow rate*—Pump ratings and the number of pumps installed generally have quite a degree of conservatism built into them (though this is falling with today's emphasis on project capital cost). It is therefore generally possible to achieve a worthwhile increase in unit flow rate just by using as much of that conservatism as the plant operators are comfortable with. If the existing equipment is operating at its practical limit before the desired unit flow rate is reached, rerating should be investigated to see whether that offers an economical means of achieving the desired flow rate.
- *Decrease in unit flow rate*—Occasionally it is necessary to reduce the flow rate in a particular unit to balance plant operation. In some of these instances, this can be achieved by simply throttling the pump, or if it is variable speed, lowering its speed. In other instances, the necessary reduction in flow, as a fraction of design, is so great that it risks lowering the pump's MTBR (or wasting energy if a permanent bypass is used). When that is the case, rerating the pump frequently offers the same MTBR while providing energy savings that help pay for the rerate.
- *Troublesome pump*—The root cause of short MTBR in many pumps is hydraulic in origin. Generally the pump's operating capacity is far from design and its energy level per stage is high enough for this to shorten seal, bearing, and running clearance life, and sometimes to cause premature erosion of the hydraulic parts, particularly the impeller. Plant management usually highlights such pumps as having high maintenance expense and reliability so low that it jeopardizes unit availability. Rerating is one means of restoring the pump MTBR and reliability to acceptable levels. That, in turn, lowers maintenance expense and can raise unit production.

DETERMINING THE PUMP'S NEW RATING

In general terms, determining the pump's new rating for a unit revamp requires the same basic discipline as for a new unit (McGuire, 1996), but with one important difference. The difference is that there is now a full-scale model available for examination, which if examined carefully can lead to an optimum

new pump rating in terms of energy consumption and MTBR. How to carry out the careful examination is best addressed by reviewing some fundamentals of pump and system hydraulics.

Pump System Interaction

A centrifugal pump operates at the capacity given by the intersection of its head capacity curve and the system's head capacity curve (Figure 1). At this point the energy being added by the pump equals the energy required by the system. Note in Figure 1 that the energy required by the system is often increased, by throttling across a control valve, to allow variation of the pump's capacity. Note, too, that this means of flow control is feasible only with centrifugal and other kinetic pumps, those that add energy by raising the liquid's velocity.

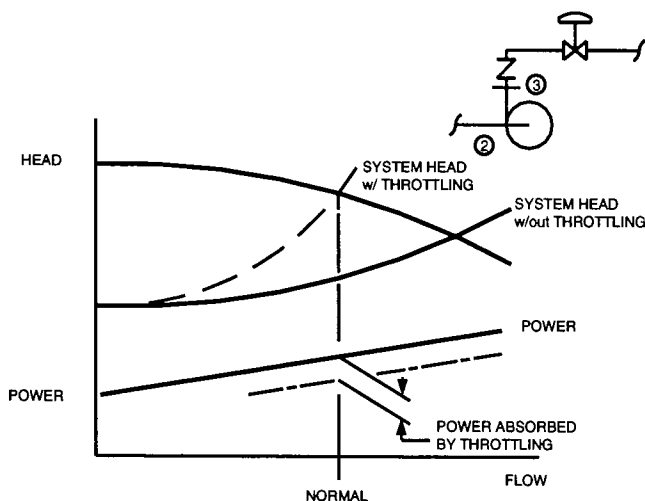


Figure 1. Centrifugal Pump Versus System with Control Valve.

Displacement pumps (Figure 2) deliver essentially a fixed capacity at a given speed, and consequently add as much energy as needed to move that capacity through the system. Care is therefore needed to ensure this energy can never be above the mechanical capability of the pump. In simple terms, this means displacement pumps must always be installed with a full capacity relief valve upstream of the first valve in the discharge system. And the relief valve must have an accumulation pressure (rise above cracking pressure to achieve full flow) that keeps the pump's discharge pressure and the corresponding power below the maximum allowable.

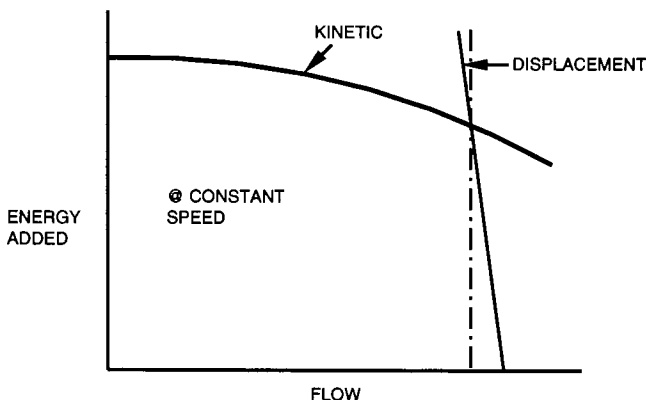


Figure 2. Flow Regulation, Kinetic Versus Displacement Pumps.

Learning from the Existing Installation

The system energy requirements for new units are estimated using various assumptions and margins. When centrifugal pumps are used, the system designer usually relies on a control valve to

balance system and pump energy at the desired flow rate. In engineering a unit revamp, it is possible to determine the actual system characteristic with accuracy and thereby avoid the energy lost to conservative assumptions and margins. This loss can be on the order of 25 percent of pump power at rated capacity, far greater than that caused by the differences in efficiency between various pump selections for the same duty.

Determining the actual system head requires accurate measurement of:

- The pump's flow rate at one condition,
- The pressure at the pump's suction and discharge and at the suction and discharge vessels,
- The liquid levels, relative to some common reference, in the suction and discharge vessels, and
- The pressure drop across the control valve, if used, taking care to measure the downstream pressure some 10 diameters from the valve to avoid the influence of any flow distortion.

To make use of the pressure measurements, it is necessary to also determine the pumped liquid's specific gravity (SG) at each measuring point. This can be determined from liquid temperature provided the liquid being pumped is known with certainty. Using Figure 3 as a reference, the system head is:

$$H_{system} = \frac{(P_4 - P_1)2.31}{SG} + (Hz_4 - Hz_1) + HL_{1-4} \quad (1)$$

where H_{z1} and H_{z4} are the static liquid levels, referred to the datum level, in the suction and discharge reservoirs respectively, and HL_{1-4} is the total friction loss in the suction and discharge piping.

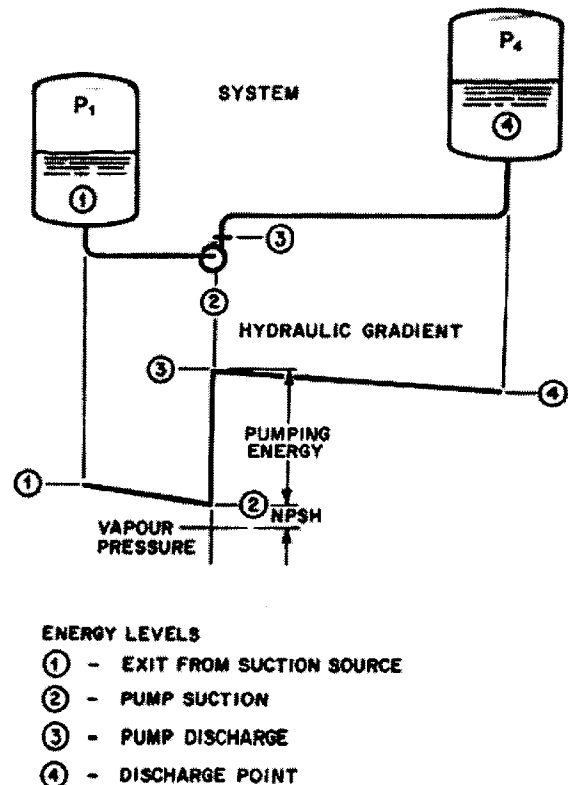


Figure 3. Hydraulic Gradient.

The pump's total head is:

$$H_{pump} = \frac{(P_3 - P_2)2.31}{SG} + H_z + HL_{2-3} + \frac{\Delta V^2}{2g} \quad (2)$$

where H_z is the correction for gauge elevation, if any, $HL_{2,3}$ the friction loss between the suction and discharge pressure gauges, and $\Delta V^2/2g$ the difference in velocity head at the points of suction and discharge pressure measurement. The friction loss is significant when there are elbows, valves, or reducers between the gauge and the pump. The difference in velocity head usually only matters when the pump head is low and there is a difference of more than one pipe size at the points of pressure measurement.

Subtracting the static head components from the pump head (Figure 4), yields the system friction head, HL_f . When a control valve is used in the system, the head being lost to throttling across the control valve is calculated from the measured valve pressure drop, then subtracted from the total system friction to give the head lost to friction in the piping, including entrance and exit losses.

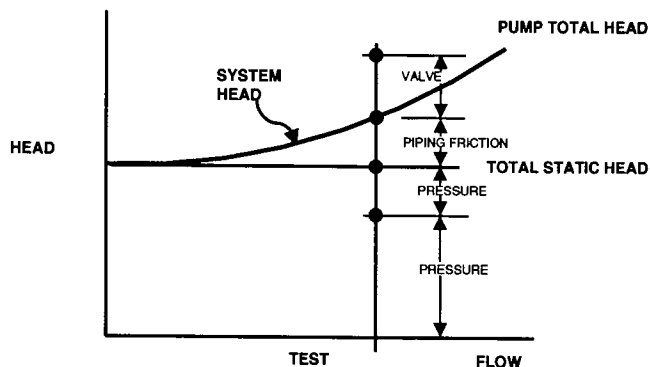


Figure 4. System Head from Measurements.

Recognizing that the head lost to friction varies as the square of the flow rate, the equivalent system friction at several other capacities can now be calculated and the system head characteristic plotted (Figure 4). If the static head varies with time, as it often does in a transfer process, then the range of system heads can be plotted after allowing for maximum and minimum liquid levels in the suction and discharge vessels.

The other critical aspect of the system to be verified using the measurements already made is the NPSH available at the pump. For measurements at the pump suction, again referring to Figure 3, the equation is:

$$NPSHA = \frac{(P_2 + P_a - P_{vap})2.31}{SG} + H_z + \frac{V_s^2}{2g} \quad (3)$$

where P_a is atmospheric pressure at site, P_{vap} is the vapor pressure of the pumped liquid at the pumping temperature, H_z is any correction for gauge elevation to the pump's reference level, and V_s is the velocity at the point of suction pressure measurement. The pump's reference level is the shaft centerline for horizontal machines, and the centerline of the suction nozzle for vertical machines.

Since NPSH available is also equal to:

$$NPSHA = \frac{(P_1 + P_a)2.31}{SG} + H_{z1} - HL_{1-2} \quad (4)$$

It is possible with the measurements already made to calculate the friction loss in the suction side of the system, and following the same procedure as used for the system head, develop the system NPSHA characteristic (Figure 5). If the liquid level in the suction vessel can vary with time, the range of NPSHA can be plotted in the same manner as the range of system heads.

With the net system head now known accurately, the pump head necessary to move the required flow and allow flow control can be kept to a minimum. At the same time, an accurate NPSHA characteristic eliminates hidden margins, which means an NPSH margin appropriate to the application can be set, thereby allowing

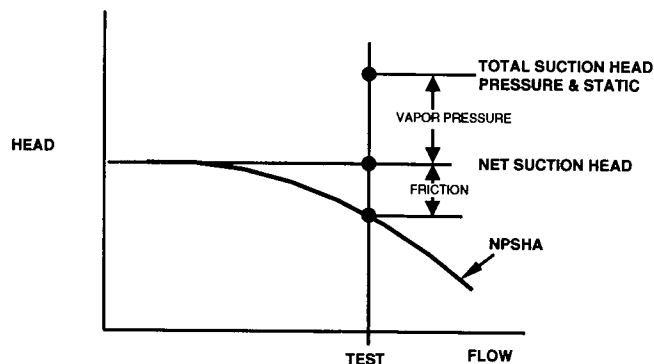


Figure 5. NPSH Characteristic from Measurements.

the selection of an optimum hydraulic design. Keeping the pump head to the minimum necessary lowers energy consumption, and having an optimum hydraulic selection can contribute to both lower energy consumption and longer MTBR.

Before hastening off to prepare the pump specification, there are two more items to be addressed while at the site. The first is to study the suction piping. Many pump problems are caused by poor suction piping, so a unit revamp is an opportunity to correct that. And in the case of a revamp for higher unit rate, poorly laid out suction piping may need to be corrected to allow the pump to operate at the higher flow. The important features of the suction piping layout are the orientation of reducers, the proximity of elbows to each other when in different planes, the orientation of the elbow immediately upstream of double suction pumps, suction piping slope, and submergence over the vessel outlet. If it is thought there are problems in the suction piping, consult Karassik and Krutzsch (1986) or Karassik and McGuire (1998) for guidance on correction.

The second item is to ask the maintenance department about the pump's service history. What needs to be looked for in this review is evidence of problems with the pump's application, its materials of construction, or its mechanical design.

Poor application is usually evident from frequent shaft seal and bearing failures, rapid wear at the running clearances, frequent shaft failure, noise and vibration, or premature impeller erosion. All these are symptoms of prolonged operation at low flows. Whether this is the case can be determined by comparing known flow rates with the pump's performance curve to see where it has been operating relative to the pump's best efficiency point (BEP).

Better materials of construction are warranted if the pump has a history of components failing from general corrosion, corrosion-erosion, erosion, fatigue, or corrosion-fatigue. It is often hard to differentiate between causes of component failure, so it may be necessary to consult a metallurgist. In some cases, changing materials may not be enough; it may be necessary to either correct a problem in the process, for example lowering the concentration of abrasive solids or bringing the pH closer to neutral, or change to a more suitable type of pump.

Mechanical design becomes suspect only when the influences of application and the pumped liquid have been eliminated. (This is probably the reverse of common practice, but is the sequence to be followed in troubleshooting modern pumps.) A major source of mechanical problems is strain caused by piping loads. If the pump has had a high incidence of seal, bearing, coupling, or shaft failures, the cause might be piping loads. The question then is whether the piping loads are too high or the pump not stiff enough. A computer analysis of the as-built piping is the first step in resolving this question. If the piping loads are reasonable or high but cannot be changed, it is necessary to change to a pump with higher piping load capacity.

Short MTBR caused solely by the mechanical design of the pump is rare in modern designs, but not uncommon in many older

designs (30 years or more). The usual difficulties are rotor stiffness, rotor construction, bearing capacity, bearing cooling, bearing housing stiffness, and casing and baseplate stiffness. These typically manifest themselves as frequent seal, bearing, and shaft failures, and rapid running clearance wear. Most of these are also symptoms of poor application, so care is needed in sorting out the true cause of the problem.

Pump Options for the Unit Revamp

Armed now with accurate data on the system head and NPSH available, and knowing whether the suction piping or pump need correction as part of the revamp, it is time to look at what has to be done and how best to do it.

First, data developed by the process designer need to be checked against the actual system head and NPSH available characteristics, and corrected where necessary. As already discussed, the decisions at this phase are how much pressure drop does the control valve need to control reliably or is it more economical to change to a variable speed pump, and what NPSH margin is necessary to ensure rated pump performance and expected life. The former is a question for the valve designer. A starting point for the latter can be taken from Table 1.

Table 1. Typical NPSH Margins.

Application	NPSH Margin %NPSHR ₃
Water, cold	10 – 35 ⁽¹⁾⁽²⁾
Hydrocarbon	10 ⁽²⁾
Boiler feed, small ⁽³⁾	50
High energy ⁽⁴⁾	100-200

Notes:

1. Depends on size; higher margin for larger pumps
2. Minimum 3 ft
3. Up to 2500 hp at 3600 rpm
4. U₁ greater than 100 ft/s

To meet the new conditions of service required for the unit revamp, there are three options: rerate the existing pump or pumps, buy an additional pump or pumps of the same design, or buy pumps of a new design. These choices may, in turn, be influenced by the operating history of the existing pumps. Table 2 summarizes the needs developed from investigation of the existing pumps and the usual options for satisfying them.

Table 2. Usual Options for Pump Changes in a Unit Revamp.

Need	Options				
	Rerate	Add	Replace	Mat'ls	Const'n
Lower flow	Δ				
Higher flow - small	Δ				
Higher flow - large		Δ	Δ		
Corrosion resistance			Δ	Δ	
Erosion resistance			Δ	Δ	Δ
Better mech'l design			Δ		Δ

Rerating the existing pump is the simplest course and the only one dealt with in this tutorial. To be successful, the rerate must be designed to meet the new conditions of service and at the same time overcome any deficiencies in the original application, such as being oversized for the normal flow or being of too high a suction specific speed.

Turning now to the mechanical aspects, a hydraulic rerate of older design pumps would typically be combined with a mechanical upgrade to raise MTBR. Most manufacturers now have standard upgrades available for pumps ranging from single-stage overhung to multistage.

If the existing pumps have suffered corrosion or erosion abnormal for the service and the class of pump, changing the materials should be considered. Conversely, if the corrosion or erosion appears more related to the type of pump than to the service, it may be better for the long term to change the pump.

In rare instances, it will be clear that the existing pump is the wrong configuration for the service, a circumstance that likely would be aggravated by rerating the pump to a yet higher energy level. A high-energy overhung pump on a severe service is a typical example. This will be obvious from the service history of the pump. Replacing it as part of the revamp is really mandatory because the success of the project will be jeopardized if an unreliable pump is retained in the unit.

Once the pump's new rating has been determined and its operating history established, these data need to be reviewed with the original equipment manufacturer (OEM) or a competent manufacturer of equivalent equipment to establish whether a rerate is feasible. This should be done even when the owner or his engineer think the pump should be replaced, because sometimes a rerate and radical upgrade might be feasible and more cost effective.

Whether the choice is a rerate or a new pump, the next step is to prepare the specification. The essential rule for a good specification is to keep it simple. Many a good solution has turned into a purchasing nightmare, to the detriment of the revamp project, because those preparing the specification for the pumps forgot this simple rule. The goal should be:

- A one or two page data sheet.
- Scope of supply summary, supplemented with a terminal point diagram if necessary.
- A schedule of events to complete the rerate.
- Agreed terms and conditions.

For a more detailed discussion of this critical phase and of the two means of purchasing the equipment, refer to McGuire (1996).

HOW A RERATE IS DONE

The objective is to achieve the required new pump rating while retaining as much of the original pump as possible, and correcting any reliability problems evident from operating history or deemed likely from engineering analysis of the proposed rerate. Three examples are discussed in broad detail to illustrate the overall approach for rerates involving more than just changing to maximum diameter impeller(s).

Gulf Coast Refinery—FCC Feed Hydrotreater

When first put into service in 1984, the fluid catalytic cracking (FCC) hydrotreater was rated at 55,000 bpd. Feed was charged to the unit by three half-capacity 12-stage double casing (API type BB5) feed pumps. Each pump was rated 1061 gpm, 5212 ft, 1370 hp at 3580 rpm.

A subsequent rerate raised unit capacity to 85,000 bpd. At the same time, reactor pressure was raised to increase the hydrogen content of the feed to the FCC, thereby improving its yield. Charging the unit at the higher flow and pressure was accomplished by running all three charge pumps in parallel, effectively turning them into three one-third capacity pumps.

In 1993, the refinery sought to raise the unit flow rate yet again, this time to 100,000 bpd. Preliminary process design put the required charge pump rating for the higher unit rate 1315 gpm, 5684 ft at 3580 rpm based on three pumps in parallel. This performance could not be achieved with the existing hydraulic design.

The operating history of the existing pumps had been acceptable (once some early difficulties with the shaft seals had been overcome), so the only engineering issue was finding or developing a set of hydraulics able to achieve the new rating and fit in the existing casing. A search of the barrel pump hydraulics available within the author's company yielded a "diffuser" type element of 11 stages that fit in the casing and needed only 10 stages to meet the rating, thus allowing a spare stage for a future head increase of up to 10 percent.

The one difficulty with the 11-stage element was that it was designed for counterclockwise rotation (viewed on the coupling) whereas the existing pumps were clockwise rotation. Since the pumps were motor driven, this fundamental issue was overcome by changing the motors' rotation.

To be able to test the new elements without disrupting unit operation and have a complete pump as a warehouse spare, the balance of the parts to make a fourth pump (casing, head, seal housings, and line and thrust bearings) were included in the rerate proposal.

With the projected investment needed to rerate the charge pumps on the order of \$2.4 million, the refinery determined that the project was financially viable and so elected to proceed with it.

Following classical engineering practice, a detailed layout was made of the 11-stage element in the existing casing. Figure 6 shows the section of the final design. A key objective of the layout was to use as much of the existing pump as possible. This was accomplished with the following:

- Suction guide or spacer modified to match the existing casing.
- Casing head modified and a new discharge spacer provided to match the new element.
- New shaft designed to match the original bearings.

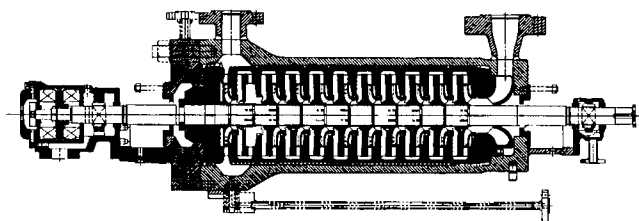


Figure 6. Section, 6 Inch Discharge 11-Stage FCC Hydrotreater Charge Pump.

Since the standard shaft for the 11-stage element was larger under the impellers and shaft seals than the original, the new shaft was about 36 percent stiffer than the original (L^4/D^2 2.4×10^6 in² versus 3.8). An analysis of the new rotor's dynamics showed that it was free from potential vibration problems with running clearances at "new" and two times "new" values.

Engineering analysis included a check of the pump's pressure boundary. The existing casing was checked using finite element analysis against ASME Section VIII, Division 2, allowable stresses and rerated for 2650 psig at 600°F. In the process of making that check, a potential resonance was identified in the casing wall, so the wall thickness of the new casing was increased to ensure this could not occur. There being no reports of such problems with the existing casings, they were left in their original condition. Analysis of the regions of the casing normally subject to suction pressure showed that their design pressure was limited to 1650 psig at 600°F, the limitation being seal housing bolting strength. The refinery's design practice required that such regions be good for maximum pump discharge pressure. Modifying the pump to achieve this would have involved a major redesign of the casing, hence three new casings and project delay, so the refinery elected to protect against accidental overpressure of the suction regions by other means.

Structural analysis established that the discharge end bearing bracket had to be stiffer to ensure adequate separation from

possible exciting frequencies. A new design was included in the new pump. The bearing bracket of each existing pump was modified to meet the new design when its element was changed.

As engineering and manufacture were progressing, process design was finalized and the pump rating revised to 1195 gpm, 5503 ft, 1628 hp at 3570 rpm. The hydraulics were adjusted to meet this slightly lower rating.

Each element was shop tested in the new pump casing to verify its hydraulic performance and mechanical operation, the latter including axial thrust. The test curve for one of the pumps is shown in Figure 7. Rotor vibration on the test stand was limited to 1.25 mils peak-to-peak, including mechanical and electrical runout, at rated speed and within 10 percent of rated flow. This is slightly below the limit in API 610, Eighth Edition, which for this speed equates to 1.5 mils peak-to-peak from 70 to 120 percent of BEP.

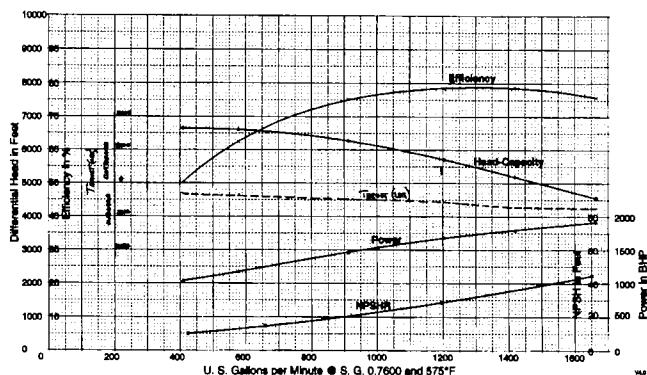


Figure 7. Test Curve of Pump in Figure 6.

The new pump was shipped directly to the refinery for installation, while the elements went to the local service center. Each of the existing pumps was then sent to the service center where it was dismantled, the casing inspected, existing parts modified where necessary, and finally reassembled with its new element.

In parallel with the charge pump rerate, the bottoms pumps for this unit were also upgraded. These radially split, single-stage, double-suction pumps, API 610 type BB2, had been a source of operating problems, often leading to low unit availability, since the unit first went online. Past experience with similar pumps had shown that the shafts were not stiff enough to ensure satisfactory shaft seal operation during adverse operating conditions. The upgrade therefore consisted of providing a stiffer shaft (larger diameter under the impeller and seals), modifying the impeller to accommodate the larger shaft, and providing new shaft seals.

The refinery's purchase orders for the rerate and upgrade were placed in July 1993. All the equipment was shipped from the manufacturing plant in January 1994, seven months from order placement. The hydrotreater went online at the higher rate three months later in March 1994.

The refinery's rotating machinery engineer for the project reports that since going back online the unit has operated as expected, in terms of both performance and availability. Speaking for future, similar projects, he emphasized the need for detailed coordination between the manufacturing plant and the local service center.

West Coast Refinery—FCC Feed Hydrotreater

After installing a "through the wall" hydrogen plant, the refinery found it had surplus hydrogen. Process design studied what could be done with this hydrogen and determined the best use was to simultaneously increase the rate of the FCC hydrotreater from 55,000 bpd to 65,000 bpd (with the possibility of running at 70,000 bpd in the future), and raise the pressure in the reactor to increase the hydrogen content of the feed to the FCC. Their estimate of the economics showed additional revenue of \$13 million per year for an investment of \$4.5 million.

The hydrotreater was served by two full-capacity charge pumps, each rated 1900 gpm, 5500 ft, 2955 hp at 3570 rpm. Preliminary process design required that the pumps be rerated to 2400 gpm, 6065 ft at 3570 rpm. Estimated power at the new rating was 3955 hp, which meant that the existing 3000 hp drivers also had to be replaced.

Operating experience with the existing charge pumps, 8 inch discharge, 11-stage barrel-type with an axially split, volute-type inner casing, had shown problems with leakage across the inner casing split joint to the lower pressure stages (typically second and third) and leakage past the radial seal between the outboard end of the element and the casing head (Figure 8). This leakage had often reduced hydrotreater flow rate and the erosion caused by it required welding and remachining to restore the inner casing. These difficulties were compounded by the inner casing being CG-8M stainless steel (equivalent of wrought type 317) to resist naphthenic acid corrosion, which meant a large "cold" clearance at the element to casing radial fits to allow for differential thermal expansion at the maximum operating temperature of 450°F. This posed a difficulty in centering the rotor since the stator centerline rose approximately 12 mils relative to the casing as the pump came up to temperature.

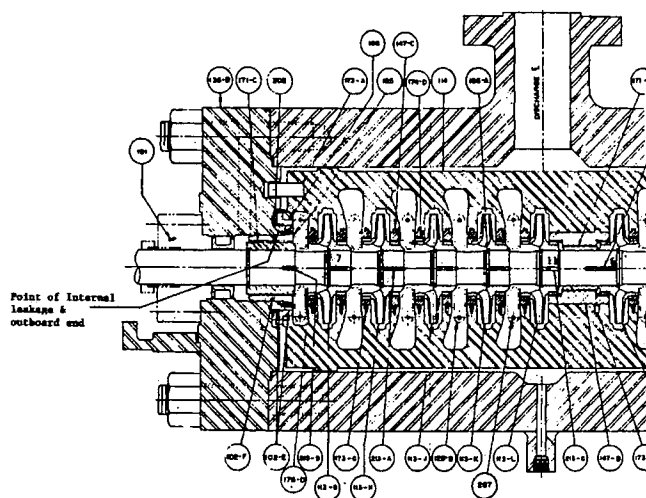


Figure 8. Point of Internal Leakage in Original Pump.

Taking account of the known internal leakage and rotor centering problems, the uncertainty of rerating the existing hydraulics (volute modification was limited), and the need for short delivery, it was decided to retrofit the existing pump with a radially split diffuser type element of existing design. Initial engineering showed that a "diffuser" type element of 10 stages would fit in the existing casing, and needed only nine stages to meet the rating, thereby leaving a "dummy" stage for a future head increase of up to 11 percent. To maintain complete interchangeability with the existing hydraulics, the new element was supplemented with new casing heads, seal housings, and complete line and thrust bearings. The final design is shown in Figure 9.

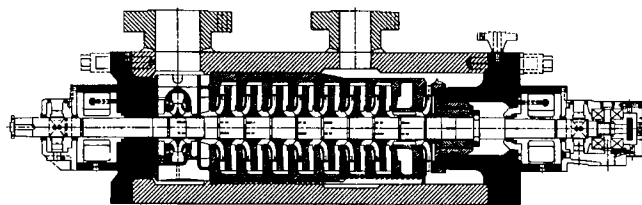


Figure 9. Section, 8 Inch Discharge 10-Stage FCC Hydrotreater Charge Pump.

Given the extent, hence cost, of the rerate and the owner's good past experience with the line of pumps from which the element was chosen, it was decided to rerate only one of the two installed pumps. This meant zero pump redundancy for operation above 55,000 bpd.

Because the existing pump had an opposed impeller rotor, its discharge was located midway along the casing. The 10-stage element has tandem (inline) impellers, so liquid discharged at the outboard end of the element has to pass back over half of the element to reach the discharge nozzle. To provide a large enough annulus for this flow, the OD of the latter stages of the element was reduced (Figure 9). Engineering analysis verified that the resultant compressive stress in the stage piece wall from radial pressure difference and axial force was still acceptable.

Being radially split with metal-to-metal face seals and shrink-fit assembled stage pieces, the new design element provided very good insurance against leakage from the discharge to the lower pressure stages. At the outboard end of the element, the pressure difference across the radial locating fit (Figure 9) is the pressure rise through the diffuser, which is negligible in terms of erosion potential.

To avoid the need for a large "cold" clearance at the element's radial locating fits, the replacement element was made from CD4MCu, a duplex stainless steel with a coefficient of thermal expansion much closer to that of the forged carbon steel casing.

Early in the engineering phase of the project, the pump selected for rerating was shutdown and its element removed so the casing could be inspected and measured. Inspection established that the critical fits and faces would need to be restored by overlaying with austenitic stainless steel and remachining. This was added to the work to be done when the casing finally came to the manufacturing plant.

Using the actual casing dimensions, the casing's pressure rating was checked by finite element analysis against ASME Section VIII, Division 2, allowable stresses. By this approach, the design pressure for the discharge regions of the casing was 2850 psig at 450°F. This was just enough to accommodate the new hydraulic design's 28 percent head rise (from rated to bypass) plus maximum suction pressure.

Once the basic hydraulic design was selected and a decision made to proceed, the critical steps in executing the rerate were:

- Material selection
- Material delivery
- Final pump rating
- Casing restoration and hydrotest
- Element assembly
- Pump test

To meet the owner's unit turnaround date, completing these steps in time became an exercise in concurrent engineering. That meant starting manufacture of the various parts before detail engineering was completed, then issuing engineering changes as manufacture progressed. With close cooperation between the owner, the manufacturer, and its suppliers this was achieved. The net result was that the tested pump shipped from the manufacturing plant 17 weeks after the owner authorized the first step in manufacture (production of waxes for the precision cast impellers).

At the point of being ready to start final machining of the hydraulic parts, process design settled on a pump rating of 2300 gpm, 5750 ft, 3450 hp at 3570 rpm, slightly below that used for the initial design. Figure 10 shows the rerated pump's "as-built" performance.

The unit went back online with the rerated pump in late May 1994. It has run since at rates up to the permitted 70,000 bpd and without any operating incidents attributable to the charge pump. The pump has demonstrated the ability to run out to higher rates, meaning that its rating is conservative, and when checked after 39 months operation showed no reduction in hydraulic performance.

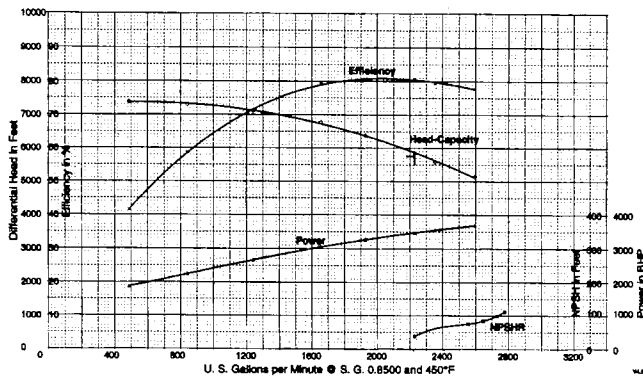


Figure 10. "As-Built" Shop Test Performance of Pump in Figure 9.

Latin American Refinery—Hydrocracker

The hydrocracker unit was to have two reactors operating in parallel for a total capacity of about 57,000 bpd. It was equipped with two full-capacity double-casing 13-stage charge pumps, each rated 1828 gpm, 8440 ft, 4530 hp at 3570 rpm. Rated capacity was at 79 percent of pump BEP.

For reasons not known, only one of the reactors was built, which meant that the charge flow required was about half the pumps' rating or about 40 percent of BEP. It was recognized that prolonged operation at such a low fraction of BEP would lead to shorter pump MTBR and so higher maintenance costs. To avoid this, the unit started up circa 1971 with a permanent bypass of 870 gpm to raise the operating charge pump's gross capacity to rated. From 1971 through 1993, the charge pump realized acceptable MTBR. To quite an extent, this was offset by very short MTBR, on the order of three months, for the bypass valve. This came to light in 1993 when the owner sought a multiple pressure reducing orifice to lower the bypass valve's pressure drop and so its erosion rate. Discussion of the problem suggested that it might be more economical over the long term to rerate the pumps rather than install a more durable permanent bypass.

The owner's process designer set the desired capacity for a rerated pump at 1140 gpm, based on a reasonable margin over the unit's actual operating capacity. Of the available hydraulic designs, the best fit for the new rating had rated capacity at 93 percent of BEP (Figure 11). Using \$0.04 per kWh for the cost of electricity, a low value, and assuming continuous operation and 83 percent unit availability, the minimum savings in energy cost was estimated at \$315,000 per year. This was sufficient to pay for a rerated inner element in just under nine months. The rerate required new impellers and diffusers, the former mounted on the existing shafts, and latter installed in the existing stage pieces.

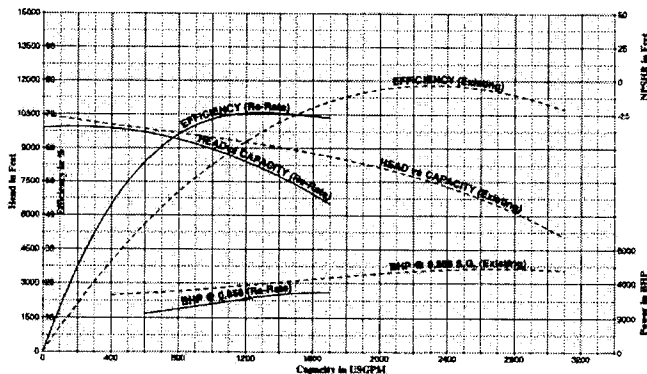


Figure 11. Original and Proposed Rerated Performance of Hydrocracker Charge Pump.

Because the shaft diameter at the impeller, d_1 , was an unusually large fraction of the impeller outside diameter, D_2 (ratio $d_1/D_2 = 0.38$), care was needed with the impeller design. A search of the company's hydraulic database established that there were successful versions of such designs and therefore provided guidance on the design for the rerate. The section of the pumps with the new hydraulic components is shown in Figure 12.

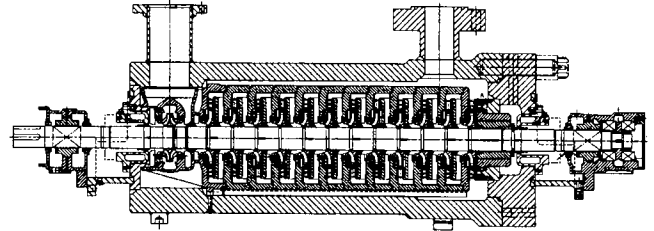


Figure 12. Section, 8 Inch Discharge, 13-Stage Pump Hydrocracker Charge Pump.

The first replacement element, produced by modifying the spare element, was completed in late 1994. It was not tested because both the casings were needed at the refinery. This element was installed and put into service in early 1995. Initial field testing showed capacity was well below expected. A check of the system piping found the 4 inch bypass line had been left in place to warm up the standby pump. Once the warmup flow was reduced to the correct value, the operating pump's capacity was as expected.

The second element was shipped in early 2000 and started up later the same year.

CONCLUSIONS

Rerating an existing pump can:

- Lower the capital cost of a unit revamp for higher rate thereby improving its financial viability,
- Reduce energy consumption when the actual operating capacity is far from the pump's design capacity, and
- Raise the MTBR of a pump whose reliability is poor.

The keys to a successful rerate are:

- Determining the actual system head for the pump as now operating,
- Checking the pump's operating history,
- Establishing the new rating for the pump,
- Careful review of the required rating and operating history by the OEM or a competent manufacturer of similar pumps,
- A simple purchase specification, and
- Competent project management.

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