JUDGE THE PUMP HYDRAULIC DESIGN THROUGH NUMBERS BEFORE YOU BUY

by

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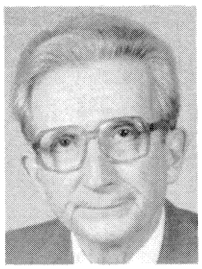

Hans Spring graduated as a Mechanical Engineer in Switzerland (1951) and came shortly thereafter to this country. He received additional graduate education at Brown, Lehigh, and Northeastern University.

In his first industrial position, he became interested in centrifugal pumps, especially with regard to hydraulic design. In the late 1950s, he apprenticed hydraulic design under Alexey Stepanoff.

In 1968, he joined the Delaval Turbine Division as a Hydrodynamics Specialist. In this capacity, he designed some of America's largest circulating pumps and was responsible for the hydraulic design of single stage reactor feed pumps with power requirements as high as 16,000 BHP.

Mr. Spring authored four ASME papers and authored and coauthored two International Pump Users Symposium papers, all dealing with hydraulic design of pumps. In 1992, he retired from Delaval and continues to work as a consultant. He is a registered Professional Engineer.

ABSTRACT

A few well established hydraulic design parameters define the internal quality of an impeller design. Becoming familiar with those parameters and understanding their limits will enable pump buyers to make better pump buying decisions. Most of these design parameters are not reflected in the pump characteristic curve nor in the pump efficiency. These parameters allow buyers to compare the hydraulic design from different manufacturers. Software is available to calculate and display these design parameters.

INTRODUCTION

Buyers of centrifugal pumps are responsible to their companies that the pumps they buy will perform well. If the users' pumps or parts must often be replaced, then the service is either very rough, or the user is buying the wrong pumps. Buyers of process pumps select their pumps based on certain mechanical and hydraulic criteria that are usually sanctioned by the Hydraulic Institute. They rely on their experience and their relationship with the manufacturer. For larger power plant pumps, the Electric Power Research Institute (EPRI) has issued a purchasing guide that attempts to help buyers in that area. Unfortunately, the Hydraulic Institute and the EPRI guidelines do not address one of the big problems at the source: *The hydraulic design of the impeller*. On a worldwide basis, purchasing activities for large utility pumps have been dormant for more than 15 years. Most of the people that were in responsible industry positions for buying and installing those pumps in the seventies have largely retired. The manufacturers of large pumps are in a similar position. The lack of business has stifled research and development. Designers also have retired. Younger people who will be responsible for power plant design in the future face a big problem. If a large utility pump was ordered today, especially a repeat design, the customer stands a very good chance of getting a 30 year old design.

Twenty-five years in the design of high energy pumps has given the insight that a factory pump performance test and a vibration test do not guarantee success. Every large pump that failed in the field during the last 30 years had the benefit of these tests. Increasing the impeller Gap B has helped, but drooping shutoff heads and "saddle" performance curves are still here. A saddle type head curve is defined as having a low point between shutoff head and bep flow. Some pumps still vibrate too much. Small and large pumps are still suffering from NPSH problems. So what is the solution?

During the last 12 years, computer codes have been developed and tested on pumps. They can predict, with sufficient accuracy, the flow inside an impeller. These codes run on PCs and are available to all pump manufacturers at a reasonable cost level. They do not take too much time to set up and run. From the analysis of the impeller internal flow, certain hydraulic design parameters can be deduced that accurately classify an impeller. All impellers look the same, even to the pump specialist, but these hydraulic design parameters reveal the essence of the impeller's design. They will show whether the impeller will produce a saddle curve or a drooping curve. They indicate whether the impeller will produce strong fluid pulsation or will be prone to generate strong radial shaft vibrations. The pump buyer, after studying these hydraulic parameters, can much better understand what kind of pump he will be buying, before it is running in his plant. He can compare impellers of different designs and different manufacturers.

An impeller with softer loading conditions can be larger than an inferior impeller for the same pumping condition. As such it could cost more, but the buyer can make an intelligent decision. Often a better impeller does not need to be larger in diameter, if it is designed with up to date technology. A new design technique makes it possible that an impeller can significantly be improved in the same space envelope it presently occupies [1].

Two existing impellers running in many power plants will be used to demonstrate the meaning of these hydraulic design parameters.

Not all pumps are "bad." Some pumps can operate with relatively higher loading parameters, others cannot. A person who must select and purchase a pump for a new installation is well advised to ask the manufacturer for the information discussed herein. If the suction specific speed is required to be higher than 8000 or 9000, additional information besides performance curves can be made available by the manufacturers to show the suitability of the impeller.

THE DEFINITION OF THE HYDRAULIC DESIGN PARAMETERS

The Impeller Diameter

In 1984, the author published information on how to select an impeller inlet eye [2]. It was shown that the eye diameter is independent of specific speed or ratios like D1 ÷ D2 *vs* NS. Software is available to the manufacturers for selection of an impeller eye based upon the flowfield in front of the impeller.

Much has been written about the danger of large impeller eyes at low flows. This is quite true. It is not so much the large eye that is at fault, but the large eye prevents the proper design of the impeller shroud line.

What is a large impeller eye? There are pump designs so large that a man can crawl through the impeller. Do these pumps have large eyes? Not necessarily. In making this judgment, the human eye cannot be relied upon, because it mostly will give the wrong answer. This is probably the only pump area where the buyer can make his own calculation with the help of the inlet flow coefficient. This calculation (and other hydraulic limits discussed later) must be made at the pumps best efficient point (bep) and not at the duty point.

The inlet flow coefficient is defined as:

$$\Phi 1 = \frac{CM1}{U1} \tag{1}$$

$$CM1 = \frac{gpm \times 0.321 \times (1 + LK)}{\text{Eye Area} \times MU1} \tag{2}$$

$$\text{Eye area} = \frac{\pi}{4} \times (D1^2 - DH^2) \tag{3}$$

$$U1 = D1 \times rpm \div 229.18 \tag{4}$$

where

CM1 = Meridional velocity is defined by Equation (2).

U1 = Peripheral speed is defined by Equation (4).

gpm = Flow is measured in gallons per minute.

LK = Leakage is defined as wearing leakage divided by pump flow at bep.

MU1 = Impeller exit area contraction is caused by vane thickness.

D1 = Impeller inlet diameter

DH = Shaft or sleeve diameter at the impeller inlet area.

rpm = Rotations per minute.

The leakage LK and the contraction MU1 are included, because the exit flow coefficient includes these quantities, also. For simplicity assume LK = 0.02 and MU1 = 0.94.

The range of the inlet flow coefficient should be between 0.22 to 0.33. Inlet Flow Coefficients smaller then 0.22 lead to impeller eyes that are too large. An inlet flow coefficient of 0.22 is permissible for a correctly designed suction impeller. The same number for an interstage multistage impeller would be too large.

A normal stage impeller eye will have a **Φ1** of close to 0.3. An eye with **Φ1** = 0.35 is small and will result in a large inlet angle. Large inlet angles (over 24 degrees) in the shroud are not suitable for low flow conditions.

The Lift Coefficient

The lift coefficient has long been used in aerodynamics, but could not be applied to pumps. In 1979, Wesche published a suitable expression for the lift coefficient on pump impellers [3]. He also defined a range for the LC as a function of the specific speed. It is recognized that, for a given NS, there can be hundreds of different hydraulic designs for the same size pump, which probably alters the range of the permissible LC. However, Wesche has planted some permanent makers in an otherwise murky field and pointed the way. In the author's experience, almost all troublesome pumps he looked at were among other items deficient in the LC.

The LC is defined as:

$$LC = \frac{2 \times \Gamma}{ZI \times VLS \times W_{AVG}} \tag{5}$$

$$\Gamma = (R2 \times CU22 - R1 \times CU1) \tag{6}$$

$$W_{AVG} = \left(\frac{W1 + W22}{2}\right) \tag{7}$$

where

ZI = Number of vanes

VLS = Lengths of vane

W1 = Inlet relative velocity in the streamline

W22 = Exit relative velocity in the streamline

Γ = Circulation in impeller, which corresponds to impeller head.

For the buyer of a pump it is really not necessary to understand Equations (5) and (6), except that he should know that Equation (5) exists. The exact meaning of Equation (5) is explained by Wesche [3] and Equation (6) in Spring's earlier article [1].

Wesche specified the LC as a function of NS. Therefore, Equation (5) requires that the number of vanes, the vane lengths, and the internal velocities for the specified head must stay within a certain quantity. Since the range of the permissible vane number is small, the vane length and the average relative velocity are controlled. The upper and lower LC given for each NS define a minimum and a maximum dimension of the impeller.

In the low to mid NS range, the impeller shroud line carries usually the largest hydraulic loading. For that reason, Wesche recommended Equation (5) be applied to the shroud line. The author has found that impellers in the range of NS = 5000 are often loaded highest in the hub line. For these impellers, the LC limits should be considered at the hub or at least at the centerline.

The calculation of the LCs must be left to the manufacturer, the buyer would not have enough resources to do this. The buyer should, however, request to see the LC numbers for the three major impeller streamlines (hub, center, and shroud) and Wesche recommended LC number.

The Velocity Load Coefficient

At the bep, there is a certain velocity distribution on the vane surfaces. These velocities vary greatly between the pressure and the suction surface of the vane. They are also different for the three major streamlines of the impeller. These velocities are not within the boundary layer near the surfaces, but are the active velocities next to the vane surface. This velocity distribution is shown schematically in Figure 1 for a streamline. The velocity load coefficient (VLC) is defined as:

$$VLC = \frac{\text{Max. Vel. Diff.}}{\text{Avg. Rel. Vel.}} \tag{8}$$

The Maximum Velocity Difference and the Average Relative Velocity are defined in Figure 1.

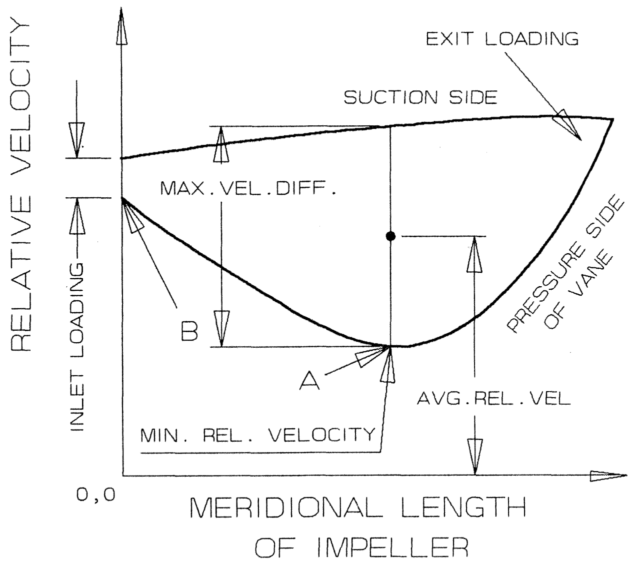

Figure 1. Vane Surface Velocity Diagram for an Impeller.

For high energy diffuser pumps the VLCs should be lower than for volute pumps to prevent saddle head curves.

For diffuser pumps:

NS	1000	1500	2000
VLC	1.5	1.3	1.2

Buyers may see VLCs as high as 2.0 between NS 1000 and 2000 on pumps. If they accept such high numbers, they will find that the pump will exhibit more lateral shaft vibrations. With an NS of 1400 and above, diffuser pumps also will have saddle curves. Pumps that gave problems were always associated with high VLCs.

For high energy volute pumps the VLCs can be somewhat higher, because volute pumps generally do not produce 'saddle' curves.

For volute pumps:

For volute pumps:	NS	1000	1500	2000	5000
	VLC	1.5	1.4	1.3	0.7

The VLC does not change with rpm on a given pump, just as the curve shape obeys the affinity laws.

The Minimum Relative Velocity
On the Pressure Side of the Vanes

The minimum pressure side velocity is also indicated in Figure 1. To obtain steady rising head curves on high energy pumps, these minimum velocities must be at least 30 ft/sec. Many smaller pumps have zero or "negative" minimum velocities. This indicates that the rotating stall already begins at the bep or even at higher flows. These pumps will have drooping curves. Today, new design technology exists that permits better vane design in the same impeller profile. (The impeller diameter and the impeller weight may not have to be increased for a better design.) Smaller pumps should have a minimum velocity of about 10 ft/sec.

The Vane Surface Diffusion Rate.

With points A and B in Figure 1 the pressure side diffusion rate can be defined as:

$$\text{Diff. Rate} = \left(\frac{W_{MIN,(PT.A)}}{W_{MAX,INLET,(PT.B)}}\right)_{PRESSURE\ SIDE} \quad (9)$$

$W_{MIN,(PT.A)}$ is the minimum relative velocity at point A on the pressure side of the vane in Figure 1.

If the minimum pressure side velocity increases, then the diffusion rate improves, also. The minimum diffusion rate should not be less then 0.25 for NS = 1500 pumps. Higher numbers would be better to produce a higher shutoff head.

For pumps in the area of NS = 5000. The diffusion rate should not be less then 0.7, because of the small VLCs used there. For a more detailed discussion of the diffusion rate see Spring's earlier article [1].

The Inlet Loading

The leading edge inlet loading is also indicated in Figure 1. For normal stage impellers, where $NPSH_R$ is of no concern, this should never be more then 0.4 times the maximum velocity difference for the leading edge. For impellers with a demand for good NPSH performance, the two velocity lines at the inlet, (pressure side and suction side) must be close together for a short stretch, not just at the LE. This is shown in Figure 2. For an extensive discussion of good and 'bad' suction impellers see Spring's 1989 article [4].

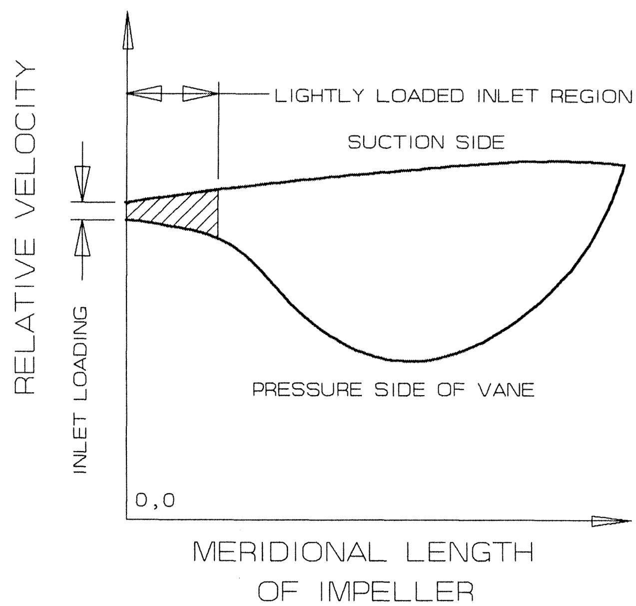

Figure 2. Vane Surface Velocity Diagram for a Suction Impeller.

Impeller Gap B

The least effect of a narrow Gap B will be increased noise levels for the pump and the worst effect will be destruction of the impeller. A Gap B too wide will flatten the head curve at low flow or even cause a drooping curve. In extreme cases, the efficiency will suffer also. Choosing a sufficient Gap B will correct the above conditions, but will not change the previously discussed design parameters. The correct Gap B is only part of good design practice.

The normal practice of describing the Gap B as a fixed percentage of the tongue diameter to the impeller OD is not sufficient. The Gap B requirements change depending on impeller OD, average impeller flow discharge angle and actual rpm of the pump. The Gap B index can best describe the requirements. The Gap B index is:

$$BGI = \frac{C11 \times B2}{ZI \times TRAJ} \quad (10)$$

where

C11 = Absolute velocity at tongue entrance
B2 = Impeller discharge width
ZI = Number of impeller vanes
TRAJ = Average length of the trajectory for a fluid particle from impeller to tongue circle. This depends on gap chosen and fluid absolute exit angle.

Gap B Index for pumps should be below 18. Gap B index for very high energy pumps (6000 BHP/stage and up) should be 8.

Maximum Velocity Difference Between Suction and Pressure Side

The maximum velocity difference between suction and pressure side of the vane is also indicated in Figure 1. This quantity and the Gap B index are the only parameters that increase with increasing rpm for the same impeller. For high energy impellers, the maximum velocity difference should not exceed approximately 160 ft/sec. On some impellers, 200 ft/sec have been observed, which is too high. When this occurred, other design limits on these pumps were always exceeded. For smaller pumps, this difference is not anywhere so large, nor is the Gap B a problem. Then main concern there is that the minimum surface velocity on the pressure side is at least 10 ft/sec.

APPLICATION

Existing Impeller A

Q = 5836 gpm
H = 2058 ft
N = 5500 rpm
NS = 1357

With the information given so far, it is possible to get a close look at an impeller. In Figure 3, the skeleton view of an existing interstage boiler feed impeller is shown. (This is not a first stage or suction impeller.)Even for an expert, it is not possible to draw any conclusions about the quality of design from this drawing. However, an analysis of this impeller tells the story. This impeller runs in many power plants, and there is nothing obviously wrong with it. The performance curve is steady rising. The pump efficiency is acceptable. The pump vibrations appeared to be within acceptable limits. The fluid pulsation measured at the suction and discharge flanges were acceptable. Sometimes during the shop tests, the pump vibrations were higher or lower then on the day before.

After startup in the field, there were the same observations. When the shop test was of concern, the field tests were fine, or visa versa. This was a condition nobody could do anything tangible about it, nor did it cause any immediate wrecks. One had just to live with it.

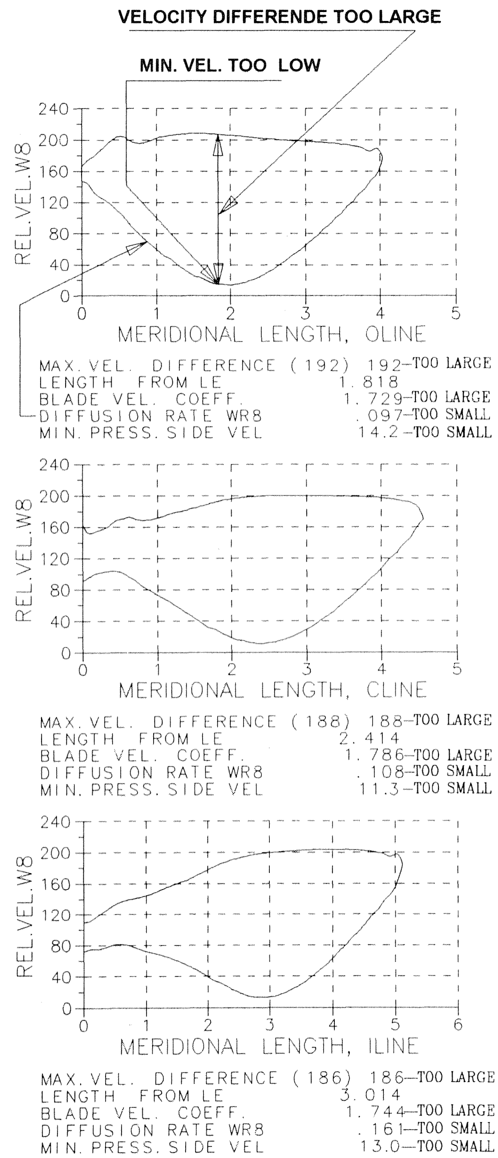

Figure 4. Vane Surface Velocity Diagram for Impeller A.

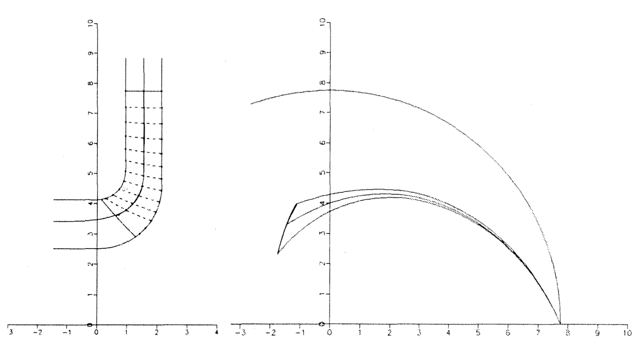

Figure 3. Skeleton View for Existing Impeller A.

In Figure 4, the vane surface velocities for the ILINE (Hub), the CLINE (centerline), and the OLINE (Shroud) streamlines are shown. From these diagrams the following conclusions can be drawn:

Velocity Load Coefficient: The highest number seen is almost 1.8 for the centerline. All three numbers shown are in excess of the guidelines.

Velocity Load Conditions: The lowest number shown is 11 ft/sec. This is much too low for a pump speed of 5500 rpm. The rotating stall will already begin close to the bep flow, and will be very strong.

Shroud Diffusion Rate: The lowest number is shown for the shroud and is 0.099. This is below the guidelines also. This diffusion rate is related to the high loading indicated above.

Maximum Velocity Difference: The highest number shows in the shroudline as 192 ft/sec, which is excessive.

Inlet Loading: The diagrams indicate that the inlet loading is generally fine. If this impeller was a suction impeller, then the shroud loading near the inlet would be excessive.

It becomes quite clear that many of the design parameters discussed depend on the shape of the blade velocity diagrams. That is why a pump's behavior can be judged from the velocity diagram shape and the numbers extracted from it. None of these quantities would be reflected by the pump's efficiency. For a buyer, it is worthwhile to become acquainted with these definitions.

With available software, it is possible to generate one single page, where all the design attributes of an impeller are displayed. Such a display is shown in Table 1. Although most of the data there are intended for the pump designer, all the information a buyer should be interested in is also shown. The information for the buyer is highlighted with comments.

Table 1. Summary of Analysis for Existing Impeller A.

Best Efficiency Point and Shutoff Data:

Flow	**5683**	**gpmf**	Cap.Coeff: CM2÷U2	0.093	-
Head	**2061**	**ft**	Head Coeff.	0.479	-
Speed	**5500**	**rpm**	Pre.Rot.Gam.	38.3	Deg.
Eff. Calc.	.877	-	Hydr. Eff.	0.921	-
NS (Full)	1356	-	Slip P	0.390	-
Dia. D2	15.500	in	Slip Coeff. A	0.708	-
Dia. DT	15.875	in	B2/D2	0.077	-
Gap B	**2.4**	**too small**	B4/B2	1.155	-
Gap B Index	**32.4**	**too high**	Head Rise	30.39	%
B2	1.190	in	Shutoff Hyd. Eff.	0.868	-
And.Ar.Ratio	1.83	-	Shutoff Hd. Coeff.	0.624	-
D1	8.250	in	**Inl. Fl. Co: CM1÷U1**	**0.295**	**very normal eye**
DH	5.000	in			

Impeller Loading Data:

	Iline	Cline	Oline	
Circulation	777.638	785.979	788.395	
Theo. Static Head Rise	1586.5	1591.7	1581.6	
OMF (Vane Wrap, Deg.)	126.4	113.4	105.2	
Vane Length	12.103	11.482	11.066	
Meridional Length L	5.124	4.540	4.0030	
Beta1	22.40	24.45	18.22	
Beta2	22.77	22.83	22.42	
Flow Devi.at LE, Degrees	**3.89**	**−0.38**	**6.23**	**check deviation**
Average Solidity	1.772	1.709	1.645	
Free Flow Diffusion	1.000	0.731	0.591	
Vane Press.Side Diffusion	**0.161**	**0.108**	**0.099**	**too low**
Max.Blade Vel. Difference	**186**	**188**	**192**	**too high**
Min.Vel. on Pressure Side.	**13.0**	**11.3**	**14.2**	**too low**
Max.Blade Vel. Load Coeff.	**1.744**	**1.786**	**1.729**	**too high**
Max.Press. Load Coefficient	3.509	3.576	3.486	
Max.Diff.Blade Head In ft	**622**	**619**	**668**	**too high**

Wesche's Recommendation for Number of Vanes and Lift Coefficient is Based on NS and Circulation:

Lower Number of Vanes	7.0	6.8	**6.3**	
Upper Number of Vanes	8.8	8.5	**7.9**	
Actual Vane Number is	**6**			
Lift Coefficient	1.921	1.854	**1.730**	**too high**
Wesche's Upper Lift Coeff.	**1.636**			
Wesche's Lower Lift Coeff.	**1.312**			
Calculated NPSH NODROP (Based on Vane Pressure Distribution)	87	127	168	

The Inlet Flow Coefficient is listed as approximately 0.3. This is about as normal as it can be.

The Gap B is listed as 2.4 percent and the resulting Gap B Index is 32. These are numbers that were used up to the early 1970s and resulted in impeller breakage. The Gap Bs have long been increased to five percent and resulting Gap B indexes are below 18. The excessive noise of narrow gaps and the pump damage have been alleviated with that change. *However, it is important to note that increasing the Gap B does not improve any of the other parameters discussed above! This is the very reason why sometimes large Gap Bs do not eliminate certain conditions.*

In Table 1, the lift coefficient in the shroud line exceeds Wesche's recommendation. Higher LCs tend to increase fluid pulsation. This is exactly the problem with this impeller. During a shop test, the fluid pulsations are measured at the suction and exit flanges. At those locations, the pulsations are already diminished. The actual conditions are not well represented. In combination with a large stall force, which this impeller also has, high LCs can lead to unstable running. High LCs and high velocity load coefficients can generate higher radial shaft vibrations which destroy the wearing clearances faster. This type of impeller requires earlier replacement of wearing clearances than impellers with lesser loading parameters.

Inlet Flow Deviations: Check the deviation between flow angles and inlet vane angles. In this example the largest deviation is 6.2 degrees, which is larger than desirable. However, it is not always easy to determine this difference. If numbers of more than 10 degrees are noticed, ask why.

The last line of Table 1 indicates the NPSH requirements at the NODROP condition, based on calculated fluid pressure on the vanes. The indicated 168 ft of NPSH reflects an SS of 8885. Therefore, the $NPSH_R$ at three percent dropoff would be less than 168 ft.

With available design technology, it is now possible to improve this impeller to the point where it would be acceptable. If the resulting numbers still exceed the recommended limits, a larger impeller diameter would have to be selected.

Field Experience

It would have been nice to show the actual field results of an improved impeller of this type. The design technology to improve this impeller exists now. But the product design life cycle of these pumps is probably 40 years or more. Therefore, a pump designer has only one chance in a lifetime, the first time, to design an impeller. The author had very few chances for a second try. Fortunately, the design technology had improved to a point where a much better chance for success existed. Several actual cases were reported [1].

However, a large source of field experience is provided by other pumps with their impellers that have been running for many years. The author has examined many such pumps and found some amazing results. With the design parameters discussed above, it was possible to understand the behavior of such pumps. With today's new design technology it is possible to clone this good behavior into a new or revised impeller. For how to accomplish, this see the literature [1].

Existing Impeller B

Q = 10,700 gpm

NS = 1627

H = 2214 ft

SS = 14,500 at 3.0 percent head drop

N = 5075 rpm

This impeller is a suction impeller and is shown in Figure 5. This impeller type has been in the field more than 20 years of field experience and has stood the test of time. It also was reported in the literature {4} with information slanted for designers. However, the author would like to show the users what they could learn from the design parameters, if they wanted to buy a suction impeller. Although these impellers operate in power plants, the dominant features for suction impellers for refineries or water works remain the same.

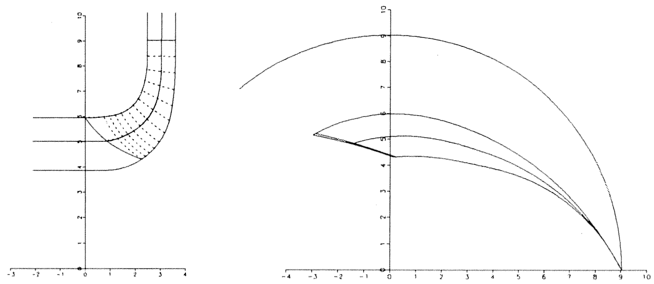

Figure 5. Skeleton View for Existing Impeller B.

The blade surface velocities are given in Figure 6. *The most notable characteristic is the closeness of the pressure and suction side velocities in the OLINE (shroud).* Not only is the LE lightly loaded, the light loading extends almost one inch into the meridional length. This light loading is, therefore, in effect for the first few inches of the vane length. A shroudline with this characteristic will always produce a good suction impeller.

The CLINE (centerline) is lightly loaded also, but the light loading does not extend much into the impeller.

The ILINE (hub) by contrast is highly loaded. When the impeller was visually tested for cavitation inception, the first cavitation bubbles showed up on the ILINE! The light loading is most important in the OLINE, because the internal velocities there are higher than in the ILINE. Even though this is an outstanding suction impeller, it could have been designed much better by reducing the LE loading in the ILINE. The recommended limit for LE loading is not to exceed one-fourth of the maximum velocity difference on the streamline for any impeller. For the buyer of pumps, it is important to know that impellers with reduced inlet loading can be designed.

The impeller has a headrise to shutoff of 26 percent and has a steady rising performance curve. It is a diffuser pump. The impeller has seven vanes, a feature that usually contributes to drooping head curves. There is no curve saddle in spite of the NS.

The reason for this can be found in the design parameters in the OLINE:

- The Velocity Load Coefficient is 1.0
- The Diffusion rate is 0.30
- The Minimum Pressure Side Velocity is 81 ft/sec
- The Maximum Velocity Difference is 166 ft/sec

Most of these design parameters are better than the recommended limits. The design parameters for the CLINE and the ILINE are close or better than the recommended limits.

Impellers have to be designed for the bep point. If the hydraulic design parameters are within limits or better, then impeller loading and rotating stall forces are less. Whenever this is the case, the impeller performance at flow ranges other then the bep is better also.

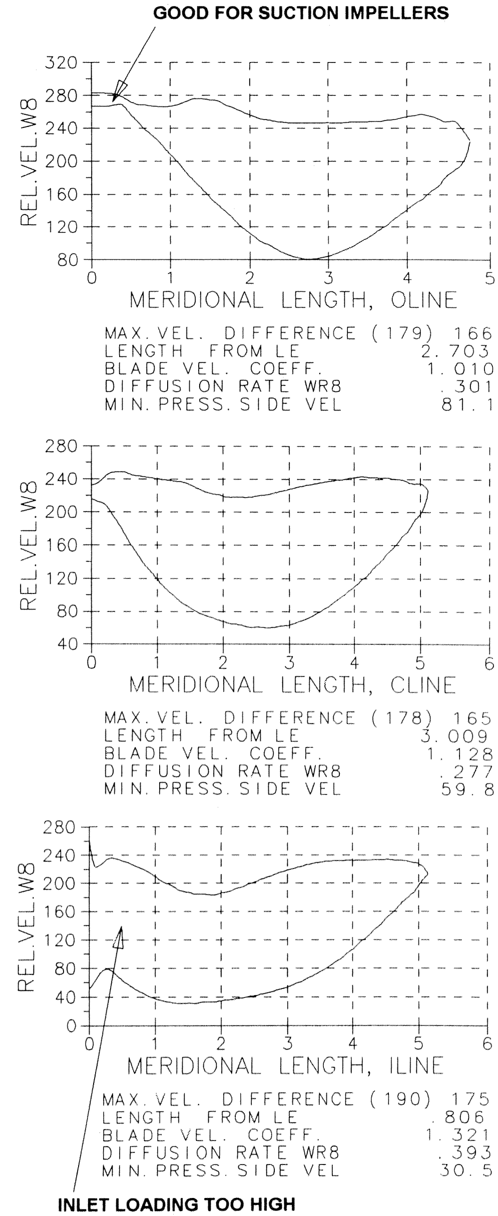

Figure 6. Vane Surface Velocity Diagram for Impeller B.

In Table 2, the consolidated results for impeller B are listed. The pertinent items for buyers are highlighted.

The Gap B Index is 18.7, which slightly exceeds the upper limit of 18. That is acceptable here, because all the other parameters are within limits.

The Lift Coefficient in the OLINE is actually below Wesche's recommendations. The performance drawback for this could be reduced stage efficiency. The LCs for the other two streamlines are fine.

The calculated NPSH NODROP in the OLINE is 105 ft. However, the tested NPSH NODROP was 175 ft, which is close to the 187 ft indicated in the ILINE (Hub). The ILINE became the limiting $NPSH_R$ factor here because of its high inlet loading, which is illustrated in Figure 6. If buyers learn to pay attention to the impeller inlet loading, they will be able to buy much better suction impellers in the future. Normally, the highest $NPSH_R$ occurs in the OLINE.

Table 2. Summary of Analysis for Impeller B.

Best Efficiency Point and Shutoff Data:

Flow	**10709**	gpmf	Cap.Coeff: CM2÷U2	0.154	-
Head	**2216**	ft	Head Coeff.	0.445	-
Speed	**5075**	rpm	Pre.Rot.Gam.	0.0	Deg.
Eff. Calc.	.816	-	Hydr. Eff.	0.868	-
NS (Full)	1626	-	Slip P	0.364	-
Dia. D2	18.070	in	Slip Coeff. A	0.733	-
Dia. DT	18.964	in	B2/D2	0.061	-
Gap B	**4.9 ok**	%	B4/B2	1.231	-
Gap B Index	**18.7 ok**	-	Head Rise	26.47	%
B2	1.106	in	Shutoff Hyd. Eff.	0.768	-
And.Ar.Ratio	1.15	-	Shutoff Hd. Coeff.	0.563	-
D1	11.908	in	**Inl. Fl. Co: CM1÷U1**	**0.224**	**on lower side, but still OK**
DH	7.736	in			

Impeller Loading Data:

	Iline	Cline	Oline	
Circulation	976.570	971.796	963.021	
Theo. Static Head Rise	1891.5	1884.8	1873.5	
OMF (Vane Wrap, Deg.)	87.2	106.0	118.9	
Vane Length	10.703	12.786	14.828	
Meridional Length L	5.140	5.117	4.754	
Beta1	**14.97**	**13.94**	**10.78**	**ok**
Beta2	26.46	27.17	28.38	
Flow Devi.AT LE, Degrees	**1.15**	**2.20**	**1.24**	**ok**
Average Solidity	1.769	2.138	2.331	
Free Flow Diffusion	0.509	0.628	0.567	
Vane Press.Side Diffusion	**0.393**	**0.277**	**0.301**	**good**
Max.Blade Vel. Difference	**175**	**165**	**166**	
Min.Vel. on Pressure Side.	**30.5**	**59.8**	**81.1**	
Max.Blade Vel. Load Coeff.	**1.321**	**1.128**	**1.010**	**good**
Max.Press. Load Coefficient	2.888	2.304	2.021	
Max.Diff.Blade Head in ft	779	765	844	

Wesche's Recommendation for Number of Vanes and Lift Coefficient is Based on NS and Circulation:

Lower Number of Vanes	7.6	**6.0**	**4.8**	**ok**
Upper Number of Vanes	9.2	**7.3**	**5.9**	**ok**
Actual Vane Number is	**7**			
Lift Coefficient	**1.529**	**1.213**	**0.970**	**good**
Wesche's Upper Lift Coeff.	**1.413**			
Wesche's Lower Lift Coeff.	**1.160**			
Calculated NPSH NODROP (Based on Vane Pressure Distribution)	187	137	**105**	

The Inlet Flow Coefficient is 0.224, which is near the lower limit. The eye dimension on this impeller does not interfere with head rise, nor does it cause problems at the low flows. The impeller is installed in plants where the pump flow is cycled at least daily from high to low and back.

CONCLUSION

The product design life cycle for a large pump can be as much as 30 or 40 years. The design life for a smaller process pump is not known to the author. If a buyer sends out a bid for a pump today, it is almost certain he would be offered a 30 year old design from any company, from any source. He would be shown an efficiency curve and probably an installation list. The informed buyer must insist on seeing the above discussed design parameters, if he wants to make sure he is purchasing a pump that benefitted from the new design technology that emerged during the last 15 years.

It is up to the pump buyers to ask the manufacturers for the above discussed design information. The manufacturers can make the information available. The supplier is not asked to give any "design secrets," it is merely additional performance data that characterize the pump.

Pump buyers can substantially increase the prospect of purchasing better pumps if they learn to understand what the Blade Velocity Load Diagram tells them about the pump.

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