THE EFFECT OF DESIGN FEATURES ON CENTRIFUGAL PUMP EFFICIENCY

by

Eugene P. Sabini

Engineering Product Manager

Worthington Pump Division

Dresser Industries, Incorporated

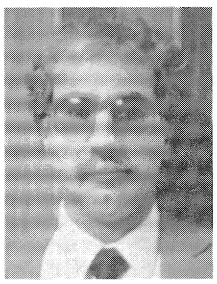
Harrison, New Jersey

and

Warren H. Fraser

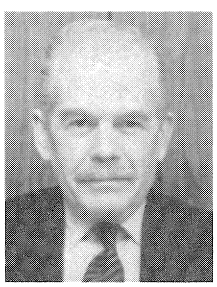
Consulting Engineer

Mountainside, New Jersey



Eugene P. Sabini has been Engineering Product Manager for Centrifugal Pumps, and Hydraulic Specialist for the Worthington Pump Division, Dresser Industries, Incorporated, since 1978. He spent ten years as a Product Engineer for Centrifugal Pumps. He is responsible for product development, including mechanical and hydraulic design.

Mr. Sabini received B.S.M.E. and M.E. degrees from Stevens Institute of Technology and is an associate member of ASME.



Warren H. Fraser is a Consulting Engineer, located in Mountainside, New Jersey. He was with Worthington Corporation from 1956 until 1984, and held positions of Assistant Chief Engineer, Manager-Applications Engineering, Chief Hydraulic Engineer—all in the Centrifugal Pump Section.

Mr. Fraser has a B.M.E. degree from Cornell University. He is a member of ASME and has served on the Power Test Code Committee and the Model Testing and the Performance Test Committee. He has had a number of articles published, along with a book, Centrifugal and Positive Displacement Pumps—Handbook of Mechanical and Electrcal Systems. *He co-edited the* Pump Handbook. *Mr. Fraser was awarded the ASME Henry R. Worthington Medal in 1986.*

ABSTRACT

Charts relating the efficiency of a single stage centrifugal pump to the specific speed were presented at the Texas A&M University Third International Pump Symposium in April 1986 [1]. These charts attempted to resolve some of the basic discrepancies of the original chart published forty years ago, and to introduce a technique for incorporating speed as an indicator of pump size.

These charts defined what we believed to be the highest efficiencies that could be achieved provided clearly defined standards of manufacture and specified operating conditions were applied. The previous paper was an attempt to define these constraints as well as to indicate some of the possible penalties that might be incurred if these constraints were not observed.

The purpose of this presentation is to present techniques that will permit a determination of the effect on efficiency when deviations from the specified constraints are considered.

INTRODUCTION

The authors presented in their original paper, "The Effect of Specific Speed on the Efficiency of Single Stage Centrifugal Pumps" [1], efficiency charts based on eleven constraints of design, manufacture and application. Any economic evaluation of a pump selection must include the initial cost of the pump and driver, the operating costs and a prediction of the maintenance costs. To do this requires some measure of the anticipated range of operation and the stability of the head capacity characteristic of the pump selected to operate within this range. In general, the broader the range of the anticipated operation the lower will be the pump efficiency for a design that will operate throughout this range with a minimum of operating problems.

The purpose of this presentation is to broaden the application of the charts presented in the original paper [1] by giving quantitative corrections that can be applied for deviations from the selected constraints. For convenience the variables that are included in the efficiency calculations and the constraints placed upon them are reproduced below from the original paper [1].

- Single stage pumps only.
- Finish and dimensional fidelity comparable to precision cast impellers with a one percent plus or minus tolerance on all dimensions of the vanes and hydraulic passages.
- A surface roughness of all hydraulic waterways of the impeller and casing to be 0.000002 per in of impeller diameter or better.
- Standard commercial diametral clearances of all wearing rings—i.e., approximately 0.0015 of the ring diameter.
- A suction specific speed value not to exceed 8500 for single suction impellers or for double suction impellers based on one-half of the total pump capacity. Suction specific

speeds are calculated on a NPSH value corresponding to a three percent drop in the total head produced.

This corresponds to the following suction recirculation values:

500 to 2500Ns—55%—single suction pumps
63%—double suction pumps
2500 to 10000Ns—71%—single suction pumps
76%—double suction pumps

See Fraser [2] for method of calculation.

- For single stage pumps with a shaft through the eye the shaft to eye diameter ratio is sufficiently low to preclude blockage in the fluid passages of the impeller inlet.
- The discharge recirculation value is not less than nor more than 80 percent to 90 percent of the maximum efficiency capacity.
- A uniform velocity profile of the fluid entering the impeller inlet. This requires an evaluation of the piping or channel flow at the pump inlet to assure that a uniform velocity profile is achieved at the rated flow conditions.
- Fluid pumped is clear water at 150°F or less.
- Efficiencies are based on maximum impeller diameters. Cut down impellers usually result in a two to three point loss in efficiency.
- Wet pit pump efficiencies are based on impellers with no back rings or balancing holes.

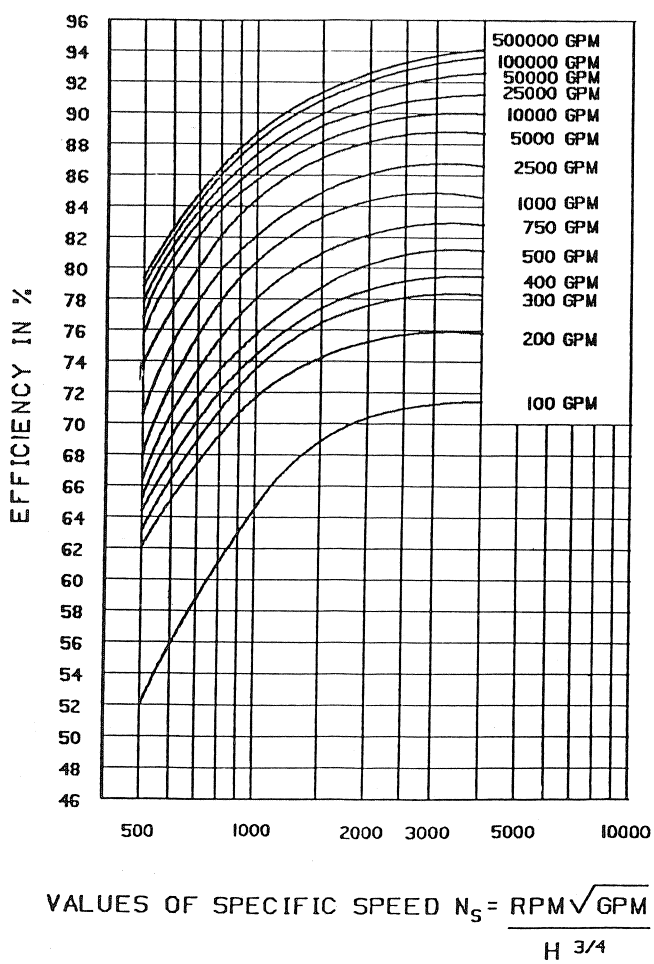


Figure 1. Efficiencies of Single Stage End Suction and Double Suction Centrifugal Pumps.

PROCEDURES

In most cases, the main concern of the user is to select the most cost effective pump which will have a long trouble free life. The eleven constraints built into the specific speed-efficiency chart (Figure 1), as presented in the original paper [1], were developed to maximize both efficiency and reliability. Any deviation from these constraints will affect both the life and the pump performance.

To permit some measure of the effect of the deviations on life and performance, the following should be considered:

- The original charts were developed for single stage pumps. A properly designed volute or diffuser multi-stage pump can achieve efficiencies comparable to a single stage end suction design of the same size and specific speed provided the design incorporates the effect of the shaft, and an allowance is made for balance flow losses.
- A cross-over return channel design that is not optimized can reduce the maximum efficiency by two or more points. Much of the loss can be attributed to improper turning of the fluid that induces prerotation at the eye of the subsequent impeller. As a result separation occurs at the inlet vane tip with a loss in efficiency. Lobanoff and Ross [3] cite an example of the effect on performance of a crossover shape that is deliberately distorted (Figures 2 and 3). In the example, they show a two point loss in efficiency for a commercial unit with sharp turns in the crossover volutes.
- Multi-stage pumps, unlike single stage end suction designs, have a shaft through the eye of the impellers. This creates design problems as the size of the pump and the total input power increase. The large shaft to eye diameter ratio at the impeller inlet produces flow distortions and blockage in

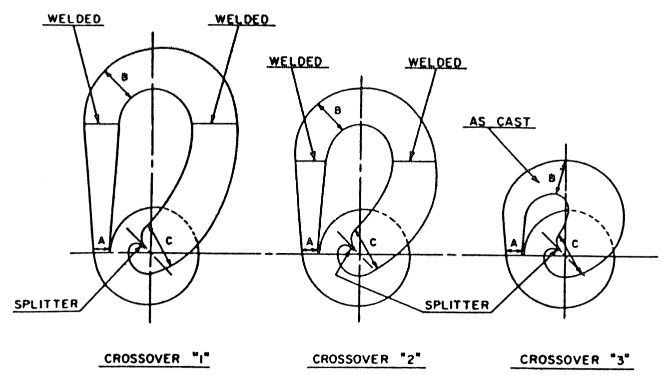


Figure 2. Configurations Evaluated During Crossover Performance Study.

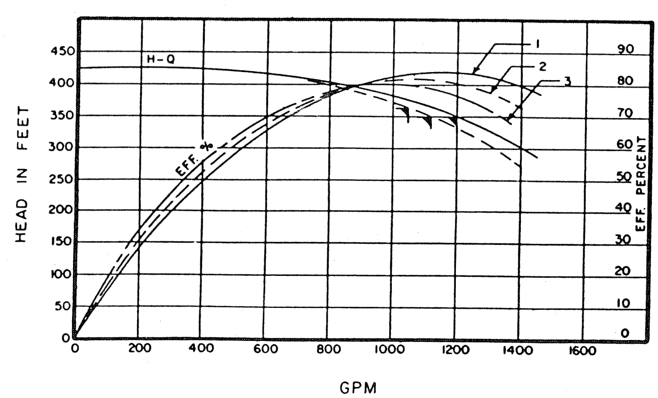


Figure 3. Results of Crossover Performance Study.

the fluid passage that lead to a loss of efficiency. Hydrodynamic design techniques, in conjunction with flow visualization testing, permit the designer to minimize the effect of the distortions by reducing the incidence between the fluid flow anqle and the metal angle of the impeller vane, along with reducing any gradient of the velocity profile and of pressure distributions within the hydraulic passages. This, in turn, leads to some sacrifice in the NPSH performance, i.e., the larger the ratio of the shaft to the discharge diameter of the impeller, the lower is the suction specific speed required to maintain optimum efficiency. As an example, a 1400 Ns impeller with a shaft to impeller diameter ratio of 0.35 will have a maximum of suction specific speed of only 5000 to 6000 to achieve maximum efficiency.

There is a benefit that can be realized, however, from a design that incorporates a low suction specific speed design impeller. The range of operation can be improved as low suction specific speed impellers are inherently more stable at low flows.

When it becomes necessary to apply a multistage pump for low NPSH performance, the first stage can be designed for a higher suction specific speed value than the subsequent stages. The fact that the first stage is lower in efficiency than the remaining stages has very little effect on the overall efficiency of a five or six stage pump.

• The end user is usually not concerned with the finish or dimensional fidelity of the hydraulic passages, nor does he have any control over them. Since he expects to receive castings of the highest quality, no corrections are included in this paper.

• In the previous paper [1], an efficiency correction for surface finish was presented.

• It is possible to obtain higher efficiencies than those indicated by reducing the specified ring clearances. This reduces the leakage loss of the impeller and increases the overall efficiency of the unit. This reduction in clearance can, and often does, lead to a reduction in component life. Conversely, an increase in ring clearances increases component life, but with reduced efficiencies.

Viewing Figure 4 will enable the user to determine the effect of increasing or decreasing ring clearances from the stated 0.0015 ratio of ring clearance to ring diameter.

As an example, take the following 1000 Ns pump:

3560 cpm
1000 gpm
542 ft
5.63 in ring diameter

From Figure 1 for 1000 Ns and 1000 gpm, the chart efficiency is 80.5 percent.

If ring clearance is increased from the standard (0.0085) in to API (0.0175), which in this case, coincides with two times the standard, enter Figure 4, for 1000 Ns and for two times the standard clearance to find a 4.5 point loss in efficiency, or

$$80.5\% - 4.5\% = 76.0\%$$

It is possible to calculate the yearly extra cost of operating with increased clearances as follows:

Calculate the power requirements
Calculate the power requirements

$$\text{BHP1} = 1000 \times 542 \times 100 \;/\; (3960 \times 80.5) = 170.0$$

$$\text{BHP2} = 1000 \times 542 \times 100 \;/\; (3960 \times 76.0) = 180.0$$

Assume a motor efficiency of 91%

$$\text{DELTA KW} = (0.7457 \times 180.0 \;/\; 0.91) - (0.7457 \times 170.0 \;/\; 0.91) = 8.2$$

Assuming 8000 hours per year operation and a $0.05 KW-Hour electricity cost, the additional cost is:

$$\text{Extra cost} = 8000 \times 12.29 \times 0.05 = \$4916.00\text{/year}$$

With the same type of calculation, the user can make an economic determination of the point at which the ring clearances should be renewed.

• It must be realized that the recirculation values shown in Figures 5 and 6 are the capacities at which a flow reversal first occurs. The recommended minimum flows in actual operation will depend on the size of the pump as well as the fluid pumped. As a general rule for pumps rated at 2500 gpm and 150 feet of head or less the minimum operating flows can be reduced to fifty percent of the recirculation

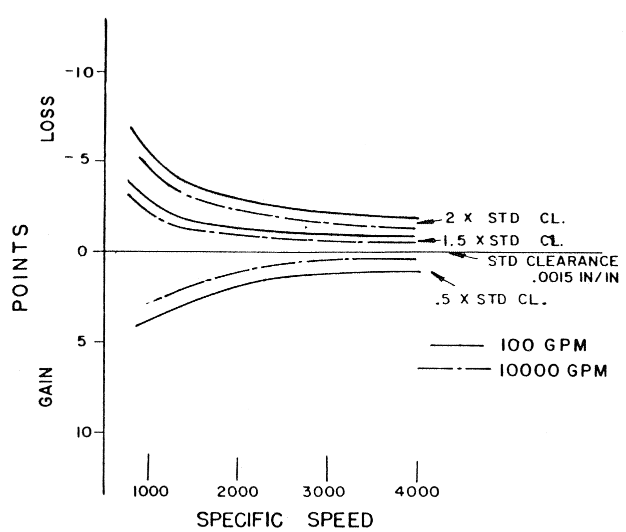


Figure 4. Change in Efficiency vs *Specific Speed Based on Ring Clearance.*

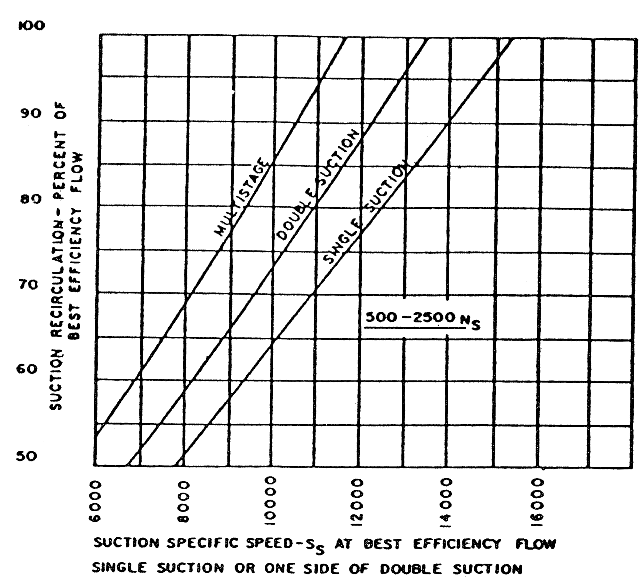


Figure 5. Suction Recirculation—500 to 2500 N_s.

values shown for continuous operation and to twenty-five percent for intermittent operation. Similarly, for hydrocarbons, the minimum flows can be reduced to sixty percent of the values shown for continuous operation and to twenty-five percent for intermittent operation.

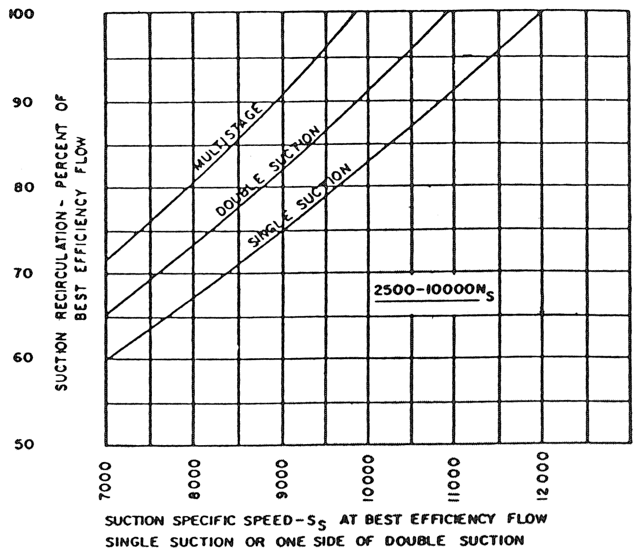


Figure 6. Suction Recirculation 2500 to 25000 N_s.

• The required low NPSH characteristics can be improved by reducing the inlet vane loading. This change in vane loading is accompanied by large gradients in both the velocity and pressure fields within the hydraulic passages. The lower the NPSHR the greater the distortions of the velocity profile and pressure field within the hydraulic passage, and the greater the loss in efficiency.

Viewing Figures 7 and 8 will enable the user to determine the loss in efficiencv associated with an increase in the suction specific speed. Curves for an impeller without a shaft through the eye are shown in Figure 7, and curves for an impeller with a shaft through the eye and a shaft to impeller discharge diameter ratio of 0.2 are presented in Figure 8.

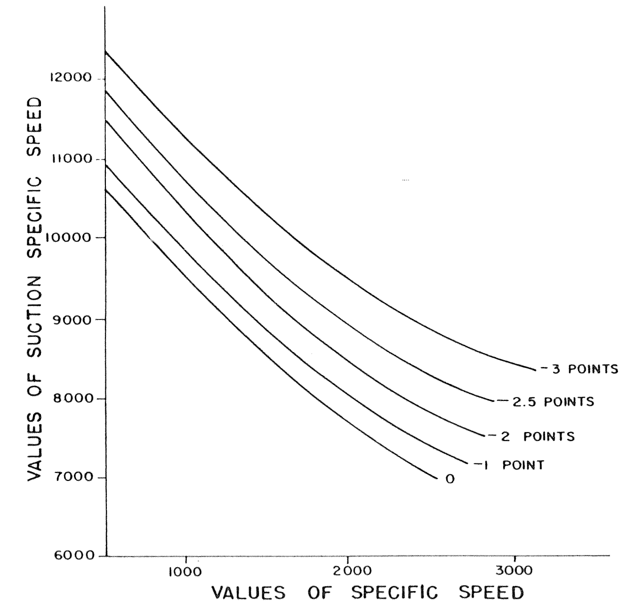


Figure 8. Loss in Efficiency for Suction Speed vs *Specific Speed. Shaft OD ratio = 0.2.*

It is recommended that users review Fraser [2], to determine any potentialproblems that may be encountered with high suction specific speed designs.

To illustrate this point, consider a 2000 Ns end suction pump with a suction specific speed of 10000. A loss of two points in efficiency is indicated in Figure 7. Compare this to a pump of the same suction specific speed and the same specific speed with a shaft to impeller discharge ratio of 0.20. A loss of 3.25 points in efficiency is indicated in Figure 8. The capacity at which suction recirculation starts increases from 55 percent to 64 percent of BEP for the end suction pump, and from 63 percent to 73 percent of BEP for the shaft through the impeller eye pump. See Fraser [2] for calculation techniques.

• The change in efficiency is related in Figure 9, for a range of discharge recirculation values for any given specific speed. For maximum life of both the wetted and mechanical components of the pump, operation below the discharge

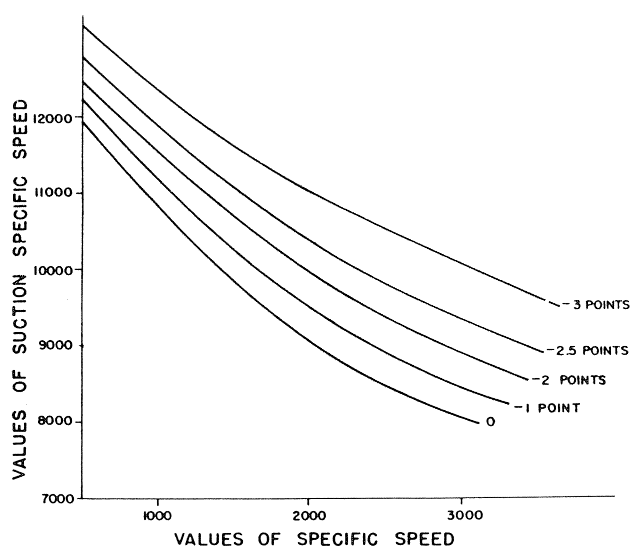


Figure 7. Loss in Efficiency for Suction Speed vs *Specific Speed. Shaft OD ratio = 0.*

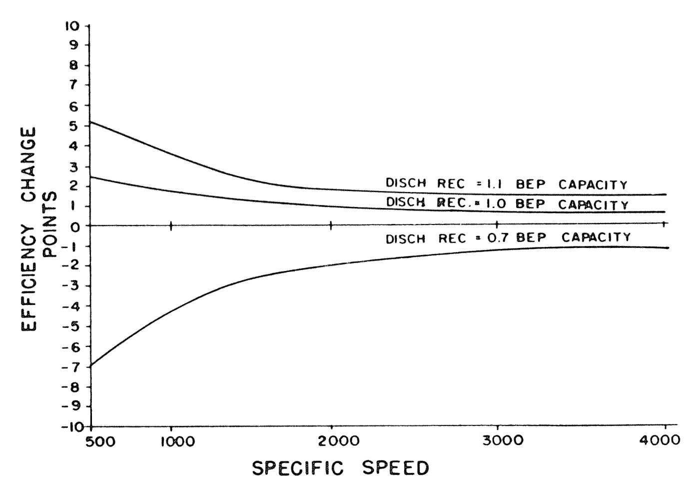


Figure 9. Change in Efficiency. Discharge recirculation recirculation vs *specific speed.*

recirculation capacity should be avoided. See Fraser [2] for these effects and procedures for the determination of discharge recirculation. As an example, to find the loss in efficiency associated with reducing the discharge recirculation to 70 percent of BEP for a 2000 Ns pump, enter Figure 9 and read off that for a 70 percent discharge recirculation capacity the loss in efficiency is two points.

As a result of recent investigations it is now possible to estimate the life of impellers designed for high S-values. Doolin [4] states:

> Pumps designed for high S-values will have shorter life. Even at the best efficiency flow, where no recirculation should occur, a high S-value impeller will suffer from high peripheral speed at the inlet. The reason is that high S-value impeller require large inlet areas and diameters to operate effectively. This reduces life.

Figures 10 and 11 are reproduced from Doolin's paper and are useful in estimating the impeller life. These curves permit a comparison of the life of a double suction impeller with a 10000 S-value operating at 75 percent of BEP as against a double suction impeller with an 8000 S-value operating at BEP. For example:

	10000 S-value	8000 S-value
Fig. 10 Relative life factor	0.77	0.99
Fig. 11 Erosion Rate	1/(1.5)	1/(1.0)
Product of above	0.513	0.91

This means that a 10000 S-value impeller operating at 75 percent of BEP will have an impeller life of 56.4 percent as compared to a pump with an S-value of 8000 operating at BEP.

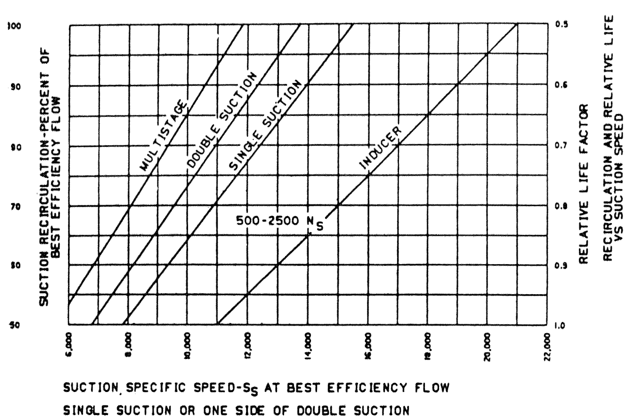


Figure 10. Relative Life Factor.

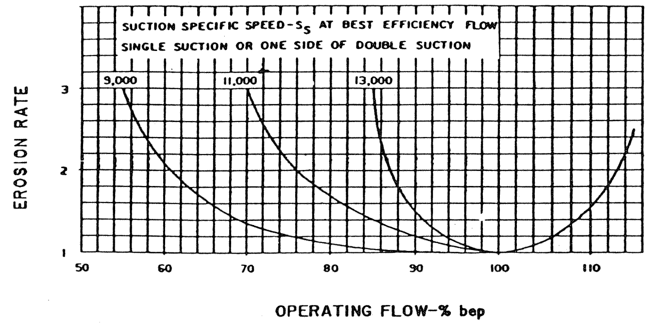


Figure 11. Erosion Rate.

CONCLUSIONS

The expanded version of the specific speed efficiency chart shown in Figure 1 has been enhanced by the addition of efficiency correction charts. It is now possible to correct pump efficiency for ring clearance, suction specific speed, shaft size and discharge recirculation that will enable the user to evaluate the effect of these variables on efficiency.

REFERENCES

1. Sabini, E. P., and Fraser, W. H., "The Effect of Specific Speed on the Efficiency of Single Stage Centrifugal Pumps," *Proceedings of the Third International Pump Symposium,* Turbomachinery Laboratories, Department of Mechanical Engineering, Texas A&M University, College Station, Texas (1986).
2. Fraser, W. H., "Flow Recirculation in Centrifugal Pumps", *Proceedings of the Tenth Turbomachinery Symposium,* Turbomachinery Laboratories, Department of Mechanical Engineering, Texas A&M University, College Station, Texas (1981).
3. Lobanoff, V. L., and Ross, R. R., *Centrifugal Pumps Design and Application,* Houston, Texas: Gulf Publishing Co. (1985).
4. Doolin, J.H., "Judge Relative Cavitation Peril with Aid of these Eight Factors," Power, October 1986.