THE EFFECT OF SPECIFIC SPEED ON THE EFFICIENCY OF SINGLE STAGE CENTRIFUGAL PUMPS

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 Centifugal 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.S. 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-Application Engineering, Chief Design Engineer and 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 Electrical Systems for Buildings. *He co-edited the* Pump Handbook.

ABSTRACT

A chart relating the efficiency of centrifugal pumps to the specific speed and capacity at the maximum efficiency capacity has been in use in the pump industry for almost forty years. While it has been useful to both the manufacturer and the user of centrifugal pumps its initial formulation was based on some generalizations that require a more precise definition today. For example, the data for the chart reflect hydrodynamic designs before 1945. Improvements in both the design procedures and in the manufacturing process have been achieved in the intervening years. An additional reason for a diminishing utility of the chart is that with the introduction of high speed pumps, the use of capacity alone as a criterion of pump size is no longer valid.

The procedures outlined herein are an attempt to resolve some of these discrepancies while, at the same time, retaining the basic simplicity of the original chart. To say this is not to discredit the value or reliability of the original chart, but rather to enhance its ability to predict pump efficiencies based upon today's procedures and capabilities.

INTRODUCTION

In 1947, George F. Wislicenus published his classic Fluid Mechanics of Turbomachinery [12]. He introduced a chart relating efficiency, specific speed and capacity for single stage pumps (Figure 1). The basic data for the chart was a statistical average of commercial pumps that he and others had access to at the time. The chart has survived for almost forty years and has become something of a doctrine or principle of faith in the pump industry. After forty years, it now seems appropriate to review and to amend, where necessary, the structure and the numerical values of the chart.

The concept of specific speed as applied to centrifugal pumps and its relation to efficiency goes back to the early 1900s. For example, Daugherty states that "The efficiency of a centrifugal pump is a function of the capacity, head and speed, but the most important in its effects is the capacity. The three factors

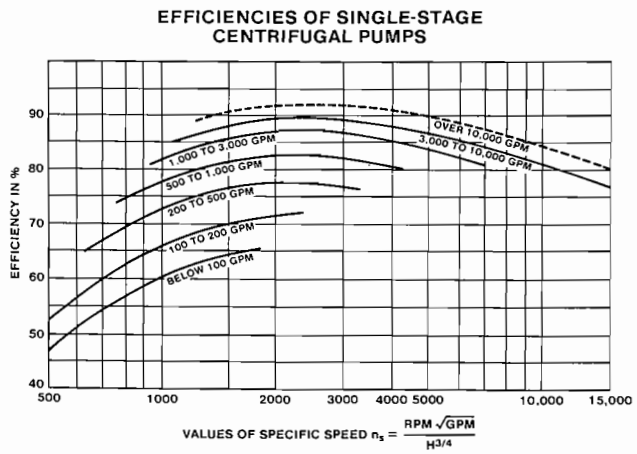

Figure 1. Efficiencies of Single-Stage Centrifugal Pumps [1].

together really involve the type"[2]. The term "type" was an early reference to a relation known as specific speed.

In 1944, Church stated that "From the standpoint of performance the efficiency depends upon capacity, head and speed. Therefore, it would be expected that there would be some relation between the specific speed and efficiency"[3]. Along with some work by Hollander in 1937 [4] and Stepanoff in 1942 [5] the groundwork was laid for the Wislicenus chart of 1947.

The chart has survived to the present day because it serves to answer in a simple and unambiguous way the question of what efficiency can be expected for any given set of conditions of service. Obviously, there must be many shortcomings to such a chart, as it says nothing about the actual design, the suction specific speed, whether there is a shaft through the eye, or where the suction and discharge recirculation capacities are located as a ratio of the best efficiency capacity.

Jekat carries the concept a step further when he says:

> The utility of specific speed as a significant group for the prediction of efficiency rests upon the premise that geometrically similar pumps operated at the same specific speed, have geometrically similar velocity triangles. Consequently, the ratios of the flow velocities to the impeller peripheral speed are the same. Were it not for the effect of scale, their relative losses would also be the same. Scale affects losses because of its influence upon flow friction and leakage losses. A chart such as Figure 5, *(i.e., the specific speed chart but not reproduced herein)* is intellectually disappointing; to represent scale, a dimensional parameter—the capacity Q—appears in an otherwise dimensionless plot (η versus N_s). It would seem more appropriate to use a dimensionless parameter, perhaps a Reynolds number. But the situation is complicated by the fact that surface roughness and clearances are not usually scaled properly. It is an emperical fact that of all known attempts to use a single parameter for scale, the use of capacity has correlated the test results best [6].

Obviously, specific speed and capacity alone are not sufficient to determine the efficiency. Take the case of an end suction single stage pump rated at 500 gpm, with a specific speed of 1000. From Figure 1, the indicated efficiency is 83 percent. Even without further analysis, it is evident that the efficiency of a given design will not increase from 72.5 percent to 83 percent, merely by increasing the speed. The missing term in the formulation is speed as an indicator of pump size.

A revised specific speed-efficiency chart for single stage pumps, based on the maximum efficiencies that can be achieved today consistent with a clearly defined quality of manufacture and range of acceptable operating conditions is presented herein. In view of the more recent developments in the hydrodynamics of turbomachinery, it is now possible to do this. The charts presented are based on the latest hydrodynamic design procedures and loss analysis techniques, validated by the highest efficnecy pumps produced today. It is also a desirable thing to do as it permits an evaluation of the cost of efficiency in terms of the quality of the machine, and the limitations that must be imposed in the operation and installation to achieve these efficiencies.

PROCEDURES

In order to determine the efficiency for any given set of operating conditions, it is obvious that a complete description of the impeller and casing design and those external conditions that have an influence on the performance is needed.

The concern here is not how to do this or the effect of each variable on the efficiency. This will be left to the specialist versed in the application of the basic principles of hydrodynamics to the technology of centrifugal pump design. The interest here is in presenting an efficiency specific speed chart based on the same principles, but with constraints on the following variables that will simplify the complexity of the efficiency calculations:

- 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 inch 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:

$$\begin{aligned} 500 \text{ to } 2500N_s &- 55\% - \text{single suction pumps} \\ &\quad 63\% - \text{double suction pumps} \\ 2500 \text{ to } 10000N_s &- 71\% - \text{single suction pumps} \\ &\quad 76\% - \text{double suction pumps} \end{aligned}$$

Stepanoff presented a method of calculation [5].

- 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.

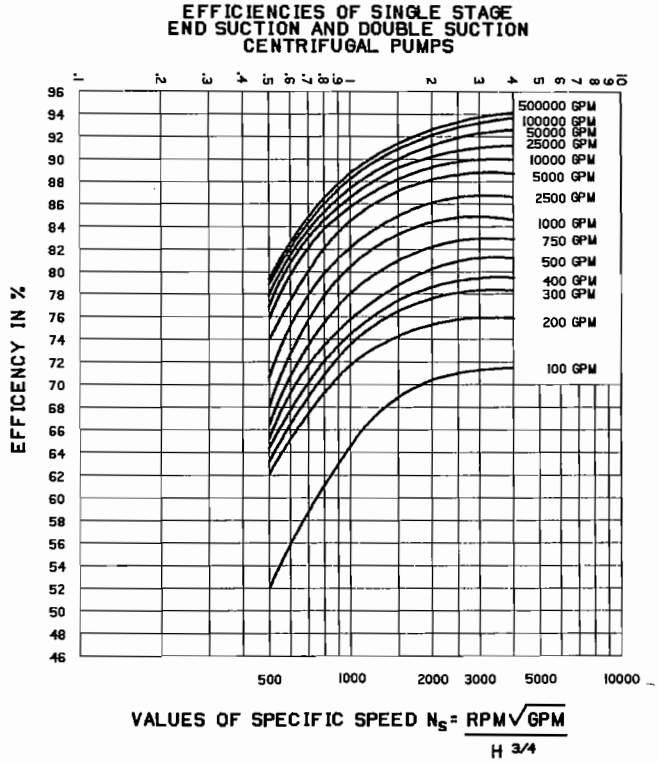

Figure 2. Efficiencies of Single-Stage End Suction and Double Suction Centrifugal Pumps.

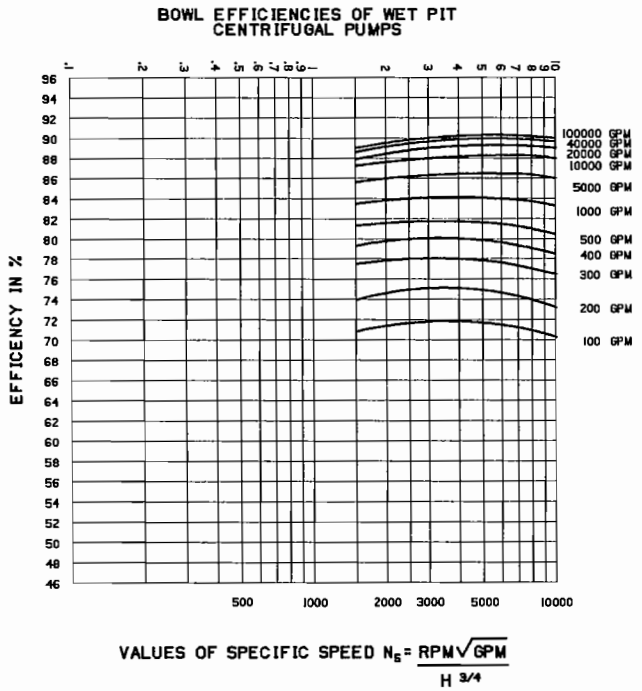

Figure 3. Bowl Efficiencies of Wet Pit Centrifugal Pumps.

• 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.

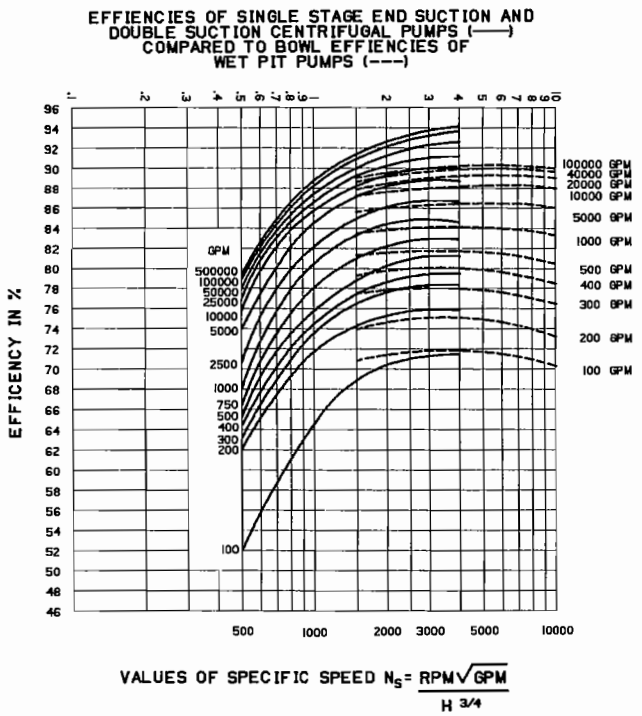

Figure 4. Efficiencies of Single Stage End Suction and Double Suction Centrifugal Pumps compared to Bowl Efficiencies of Wet Pit Pumps.

• Fluid pumped is clear water of 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.

The results of this type of analysis are shown in Figure 2 for single and double suction pumps, and in Figure 3 for wet pit pumps.

A composite curve (Figure 4) of both the volute and wet pit pumps shows clearly that the peak efficiencies of 92 percent and 93 percent are achieved by single stage volute pumps at 4000 N_s. The peak efficiency of the wet pit pumps occurs at approximately 5000 N_s and is just over 90 percent.

To use these curves requires some measure of the pump speed. The curve shown in Figure 5 relates speed to capacity, and is based on a N Q parameter of approximately 115,000, where N is the rotative speed in cpm and Q is the rated capacity in gpm. In practice, first enter Figure 5 with the maximum efficiency capacity of the pump in question and then read off the standard speed from the ordinate. The capacity of the pump is then corrected by the ratio of the speeds. Next, enter Figure 2 or Figure 3, depending on the type of pump, with this capacity and the specific speed determined from the original pump rating. As an example, consider the following conditions of service for a single stage single suction pump.

1100 gpm
225 ft. THD
1750 cpm
1000 N_s

Enter Fig. 5 for speed correction. The corrected speed is 3450 cpm then

$$\frac{3450}{1750} \times 1100 \text{ gpm} = 2200 \text{ gpm}$$

Enter Figure 2 with 1000 N_s and 2200 gpm to determine the efficiency as equal to 81.9 percent.

The charts can also be used as an effective method to predict efficiencies, from a model performance to a full size machine. As an example, consider the 84 in vertical volute single suction, single stage pumps installed by the Bureau of Reclamation in the Tracy Pumping Plant in California. In this case, 0.104 model of the full size 84 in pump was built and tested at the prototype head. The model was tested under laboratory conditions with great care as to geometric similarity between the model and the prototype. The field tests were conducted by the Bureau of Reclamation. These tests provide an excellent opportunity to check the modelling laws for large model to prototype ratios. The model pump was operated under the following conditions of service:

3800 gpm
197 ft. THD
1728 cpm
2025 N_s
89% efficiency

To check this performance against the charts, do the following:

• Enter the speed correction chart Figure 5 at 3800 gpm and read off the speed. In this case, the corrected speed is 1890 cpm.

• Correct the rated capacity for the speed change as follows:

$$\frac{1890}{1728} \times 3800 = 4156 \text{ gpm}$$

• Enter Figure 2 with a specific speed of 2025, and 4156 gpm and read off the efficiency. In this case, the efficiency will be 87.8 percent, as compared to the model test efficiency of 89 percent. This is very close agreement when it is realized that in a model pump of this quality the wearing ring clearance is less than commercial clearances of 0.0015 of the wearing ring diameter.

Now consider the field performance of the full size machine. The tests showed the following:

350,000 gpm
197 ft. THD
180 cpm
2025 N_s
92% efficiency

To check this performance against the charts, proceed as follows:

• Enter the speed correction chart Figure 5 at 350,000 gpm and read off the speed. In this case, the corrected speed is 205 cpm.

• Correct the rated capacity for the speed change as follows:

$$\frac{205}{180} \times 350{,}000 = 398{,}600 \text{ gpm}$$

• Enter Figure 2 with a specific speed of 2025 and 398,600 gpm and read off the efficiency. In this case, the efficiency will be 92.5 percent as compared to the field test efficiency of 92 percent.

A similar calculation can be made for a wet pit pump. Consider the following conditions of service:

5000 gpm
72 ft. THD
1750 cpm
5000 N_s

• Enter the speed correction chart Figure 5 at 5000 gpm and read off the speed. In this case the corrected speed is 16500 cpm.

• Correct the rated capacity for the speed change as follows:

$$\frac{1650}{1750} \times 5000 = 4714 \text{ gpm}$$

• Enter Figure 3 with a specific speed of 5000, a capacity of 4714 gpm and read off the expected bowl efficiency of 86.5 percent. Note that this is the bowl efficiency and additional calculations would have to be made for column and shaft losses

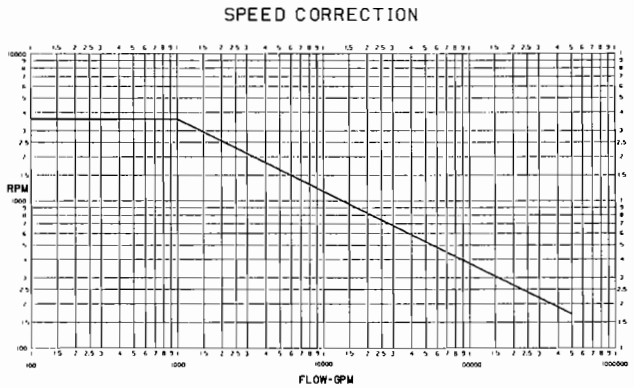

Figure 5. Speed Correction.

to determine the pump efficiency. The 86.5 percent bowl efficiency is for a single stage pump. Depending on the design some increase in efficiency (1 to 2 points) may be achieved with multistaging.

The effect of changing hydraulic surface roughness from 0.000002 per inch of impeller diameter to 0.00001 per inch of impeller diameter is illustrated in Figure 6. This is necessary to permit an economic evaluation of the effect of surface roughness on pump efficiency.

To illustrate the use of this correction chart, consider the first example presented. In this particular example, the impeller diameter is 16 in.

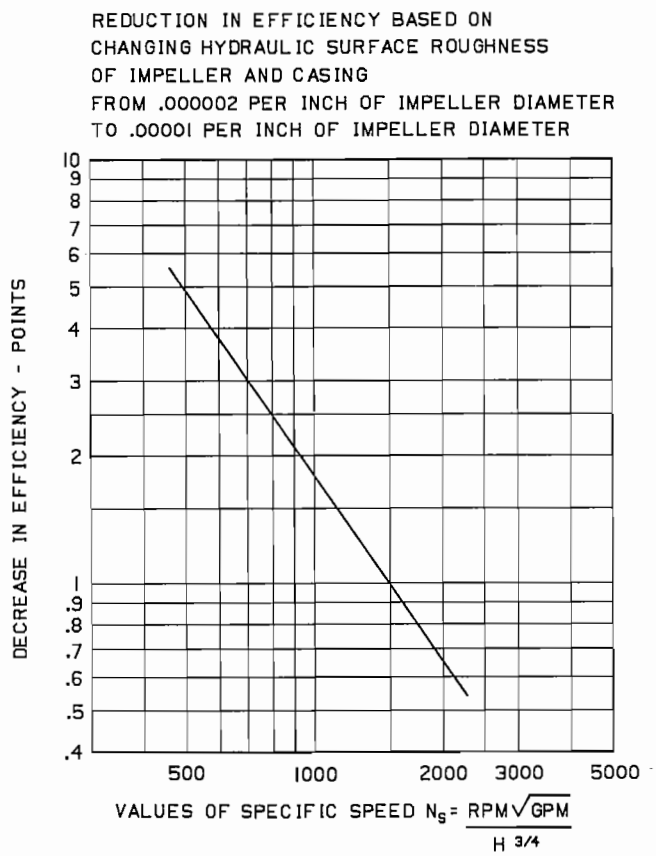

Figure 6. Reduction in Efficiency Based on Changing Hydraulic Surface Roughness of Impeller and Casing from 0.000002 per inch of Impeller Diameter to 0.00001 per inch of Impeller Diameter.

1100 gpm
225 ft. THD
1750 cpm
1000 N_s
81.9% efficiency

Enter Figure 6 with 1000 N_s and determine a decrease of 1.8 points in efficiency, by changing the surface roughness of the wetted hydraulic surfaces from 0.000002(16) = 0.000032 or 32 root mean square (RMS) to 0.00001(16) = 0.00016 or 160 RMS. The reduced efficiency is then:

81.9 − 1.8% = 80.1 percent

CONCLUSION

An expanded version of the original specific speed efficiency chart shown in Figure 1 has been presented. To accomplish this and still retain the simplicity of the original chart, some constraints on the type of design and applications must be applied.

The effect of such factors as the relative roughness of hydraulic waterways or the wearing ring clearance are obvious—an increase in surface roughness or an increase in wearing ring clearances will result in a degradation of the efficiency. For example, API clearances will reduce pump efficiencies from two to five points, depending on the pump size and specific speed. Other factors such as suction specific speed or recirculation capacities are not so obvious, but the effect of each of these design parameters on efficiency would require a detailed design analysis. As a general principle, however, it can be stated that increasing the design suction specific speed values will decrease the maximum efficiency obtainable. Similarly, any reduction in the discharge recirculation capacity will decrease the efficiency.

The application of the specific speed efficiency charts has been limited to single stage pumps. Although the desirability of including dry pit multistage pumps is recognized, the number of design variables involved precludes the use of specific speed and flowrates as the sole basis for efficiency prediction. The efficiency of wet pit diffuser pumps, on the other hand, can be reasonably related to specific speed, because of the simplicity and virtual standardization of single and multistage designs, compared to the dry pit multistage pumps with large diameter shafts and a variety of collector return designs. For these and other design reasons, the dry pit multistage pumps do not show the one to two points gain in efficiency with multistaging that the wet pit pumps exhibit.

Efficiency is particularly sensitive to the design suction specific speed of the impeller. The optimum suction specific speed for a single suction pump without a shaft through the eye is in the 8500 to 9000 range. An increase in the design suction specific speed from 9000 to 11000, for example, not only increases the suction recirculation values by approximately 25 percent, but can also result in a three to five point loss in efficiency.

It is important, therefore, to consider not only the efficiency of any pump selection, but also the effect of component life and mechanical reliability on efficiency. In general, an increase in component life and mechanical reliability means a reduction in the maximum efficiency attainable. For most applications, an economic evaluation should be made of the power costs and maintenance costs over the life of the pump to arrive at an optimum selection.

REFERENCES

1. Wislicenus, G.F., *Fluid Mechanics of Turbomachinery*, New York: McGraw-Hill Book Company (1947).
2. Daugherty, R.L., *Hydraulics*, New York: McGraw-Hill Book Company, (1937).
3. Church, A.H., *Centrifugal Pumps and Blowers*, New York: John Wiley & Sons,Incorporated (1944).
4. Hollander, D., *in Hydraulics*, R.L. Daugherty, ed., New York: McGraw Hill Book Company (1937).
5. Stepanoff, A.J., "Centrifugal Pump Performance as a Function of Specific Speed," ASME (August 1943).
6. Jekat, W., "Centrifugal Pump Theory," *in Pump Handbook*, I.J. Karassik, W.C. Krutzsch, W.H. Fraser and J. Messina, eds., Section 2.1, 2nd Edition, New York: McGraw Hill Book Company (1986).
7. Fraser, W.H., "Flow Recirculation in Centrifugal Pumps," Proceedings of the 10th Turbomachinery Symposium, Turbomachinery Laboratories, Department of Mechanical Engineering, Texas A&M University, College Station, Texas (1981).