ENGINEERS' GUIDE TO CENTRIFUGAL PUMPS

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PREFACE

At first glance, the question-and-answer format of this book may appear somewhat primitive and artificial. I feel therefore that I owe an explanation to its readers, an explanation which is really quite simple. The material presented in this book appeared first in the form of articles in a number of technical magazines. It was taken from actual correspondence dealing with various aspects of centrifugal-pump application and operation.

Many years ago, when engineers and users of centrifugal pumps first began to write me about problems they encountered in the field, I noticed that a number of my correspondents had the same problems in common. I decided to save this portion of my correspondence for the purpose of publishing some of it in the form of articles. The first few of these appeared in the magazine *Plant Engineering*, after which the series began to be published in the magazine *The Plant* under the overall title of "Let's Learn More about Centrifugal Pumps." Finally, the series was transferred to *Southern Engineering* (formerly *Southern Power and Industry*) where it is still running under the title "Centrifugal Pump Clinic." This, incidentally, had been the title which I had chosen for these articles initially, but which had not been used until *Southern Engineering* decided to adopt it.

As soon as the articles started to appear, they generated a host of new questions from the readers. In almost all cases, these readers permitted me to use their questions and my answers in new articles. The questions I chose to utilize were mainly intended to acquaint the readers with various ways in which they could improve the operation of their equipment or modify this equipment in order to save operating or maintenance costs. In many cases they were also intended to illustrate the diagnosis of various operating difficulties, which, for one reason or another, arise to plague pump operators.

The articles were later issued in reprint form and distributed by Worthington Corporation. It rapidly became apparent that the material had great interest because an increasing number of engineers requested to be placed on the mailing list for these reprints. Time and again, readers would write me that one or another of these articles would appear providentially in time to solve some particular problem that had been puzzling them. Further interest was evidenced when technical magazines in India, Argentina, and Spain started republishing these articles abroad. They are soon to appear as well in Italy and in Japan.

Many of my readers had been writing me with the suggestion that I choose the most significant of these articles and publish them in book form. I had already written a book, "Centrifugal Pumps, Selection, Operation, and Maintenance," published by the F. W. Dodge Division of the McGraw-Hill Book Company, but I knew that there would be no conflict between this book and the Centrifugal Pump Clinic article series. If anything, the two books would be supplementary.

When I embarked on the project of choosing the questions to be used in this book, I realized that some material from another series of my articles, published under the title of "Steam Power Plant Clinic" in the magazine *Combustion* could be incorporated very usefully in the same book. Once the material had been selected, I proceeded with the task of rearranging the questions into a logical sequence and of grouping similar topics to facilitate easy reference.

The questions touch on practically all the pertinent problems faced by engineers who are involved in one way or another with pumping equipment. While heavy emphasis is placed on operating and maintenance problems, the book embodies some general information that should be considered by engineers selecting centrifugal pumps for fluid handling systems. The first chapter deals with application problems and is followed by a chapter on design and construction questions. This latter chapter is considerably shorter than all the others. This is rather natural, since I did not want to include material which can be found in my book "Centrifugal Pumps." The third chapter deals with pump installation and the fourth with operation. Proper attention to these areas will serve to reduce considerably the cost and annoyance of service interruptions and pump repairs. The fifth chapter deals with questions on preferred methods of pump maintenance. Finally, the last chapter answers questions on field difficulties. This, unfortunately, is a chapter which cannot ever be completed. Certainly my own experience has taught me that, while certain types of difficulties recur only too frequently, one constantly meets completely new and sometimes mysterious difficulties which must be diagnosed and solved without benefit of previous experience.

I have hopes that this material will be found useful by three basic groups of engineers. First, the plant engineering and operating group, facing many of the problems discussed in this book in their daily job, may find this a good working tool. Second, design engineers in consulting engineering firms and in the central engineering staffs of large companies should find it a useful reference when tackling pump application problems. Finally, recent engineering graduates entering into their careers should take advantage of previous experience to avoid many of the pitfalls in the field of pumping problems that others have already encountered and solved.

I wish to thank *Southern Engineering, Combustion, Plant Engineering,* and *The Plant* for their kind permission to utilize the material from the articles which originally appeared on their pages.

I would like also to acknowledge the help of W. C. Krutzsch, Manager of Engineering, Pump and Heat Transfer Division of Worthington Corporation, who co-authored several Centrifugal Pump Clinics which have been incorporated in this book.

Igor J. Karassik

CONTENTS

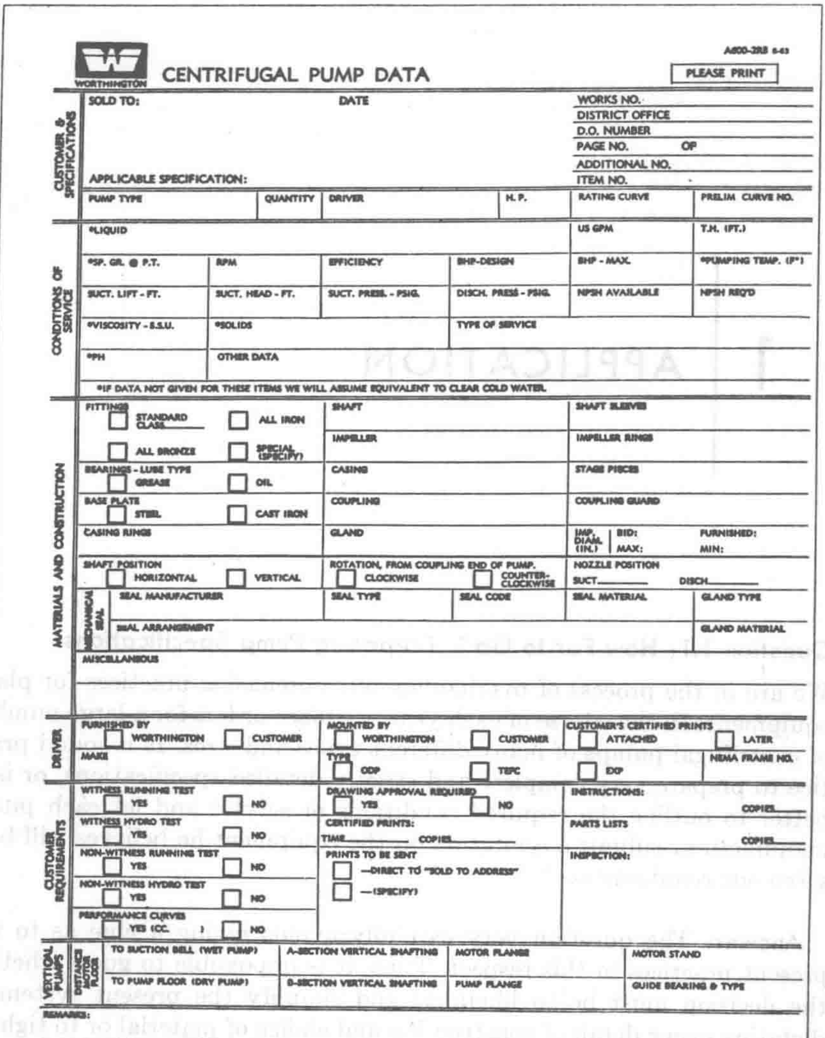
1 APPLICATION

Question 1.1: How Far to Go in Preparing Pump Specifications

We are in the process of overhauling our purchasing practices for plant equipment. In the course of each year, we place orders for a large number of centrifugal pumps of many different types and sizes. Is it sound practice to prepare very complete and strictly detailed specifications, or is it better to outline the required conditions of service and let each pump manufacturer submit a quotation on the equipment he believes will best serve our requirements?

Answer. The question very carefully avoids giving a clue as to the present practices in this respect. Thus, it is impossible to guess whether the decision must be to liberalize and simplify the present system of dictating every detail of construction and choice of material or to tighten the reins on loose methods of purchasing whatever manufacturers' representatives recommend as desirable or suitable.

Whatever the present policies, it appears that changes are contemplated. But this is an extremely controversial question and rather difficult to answer quite objectively. I prefer, therefore, to permit my personal feelings on this subject to color my answer to some extent, and since I preface it with this warning, it should not cause anyone to take violent exception. The points covered here are basically those contained on the manufacturer's typical data sheet (Fig. 1.1), the data entered thereon



A600-2R8 6-63

WORTHINGTON

CENTRIFUGAL PUMP DATA

PLEASE PRINT

CUSTOMER & SPECIFICATIONS

SOLD TO: | DATE | WORKS NO.
DISTRICT OFFICE
D.O. NUMBER
PAGE NO. OF
ADDITIONAL NO.
APPLICABLE SPECIFICATION: | ITEM NO.

PUMP TYPE | QUANTITY | DRIVER | H. P. | RATING CURVE | PRELIM CURVE NO.

CONDITIONS OF SERVICE

*LIQUID | US GPM | T.H. (FT.)
*SP. GR. @ P.T. | RPM | EFFICIENCY | BHP-DESIGN | BHP - MAX. | *PUMPING TEMP. (F°)
SUCT. LIFT - FT. | SUCT. HEAD - FT. | SUCT. PRESS. - PSIG. | DISCH. PRESS - PSIG. | NPSH AVAILABLE | NPSH REQ'D
*VISCOSITY - S.S.U. | *SOLIDS | TYPE OF SERVICE
*PH | OTHER DATA
*IF DATA NOT GIVEN FOR THESE ITEMS WE WILL ASSUME EQUIVALENT TO CLEAR COLD WATER.

MATERIALS AND CONSTRUCTION

FITTINGS: STANDARD CLASS ____ / ALL IRON / ALL BRONZE / SPECIAL (SPECIFY) | SHAFT | SHAFT SLEEVES
| IMPELLER | IMPELLER RINGS
BEARINGS - LUBE TYPE: GREASE / OIL | CASING | STAGE PIECES
BASE PLATE: STEEL / CAST IRON | COUPLING | COUPLING GUARD
CASING RINGS | GLAND | IMP. DIAM. (IN.) BID: MAX: | FURNISHED: MIN:
SHAFT POSITION: HORIZONTAL / VERTICAL | ROTATION, FROM COUPLING END OF PUMP. CLOCKWISE / COUNTER-CLOCKWISE | NOZZLE POSITION SUCT.____ DISCH.____
MECHANICAL SEAL: SEAL MANUFACTURER | SEAL TYPE | SEAL CODE | SEAL MATERIAL | GLAND TYPE
SEAL ARRANGEMENT | GLAND MATERIAL
MISCELLANEOUS

DRIVER

FURNISHED BY: WORTHINGTON / CUSTOMER | MOUNTED BY: WORTHINGTON / CUSTOMER | CUSTOMER'S CERTIFIED PRINTS: ATTACHED / LATER
MAKE | TYPE: ODP / TEFC / EXP | NEMA FRAME NO.

CUSTOMER REQUIREMENTS

WITNESS RUNNING TEST: YES / NO | DRAWING APPROVAL REQUIRED: YES / NO | INSTRUCTIONS: COPIES____
WITNESS HYDRO TEST: YES / NO | CERTIFIED PRINTS: TIME____ COPIES____ | PARTS LISTS COPIES____
NON-WITNESS RUNNING TEST: YES / NO | PRINTS TO BE SENT: —DIRECT TO "SOLD TO ADDRESS" / —(SPECIFY) | INSPECTION:
NON-WITNESS HYDRO TEST: YES / NO
PERFORMANCE CURVES: YES (CC.) / NO

VERTICAL PUMPS

DISTANCE MOTOR FLOOR: TO SUCTION BELL (WET PUMP) | A-SECTION VERTICAL SHAFTING | MOTOR FLANGE | MOTOR STAND
TO PUMP FLOOR (DRY PUMP) | B-SECTION VERTICAL SHAFTING | PUMP FLANGE | GUIDE BEARING & TYPE

REMARKS:

FIG. 1.1. Data sheet from the author's company which lists the information required to select and build a pump.

being the minimum amount of information required to select and build a pump.

To begin with, I do not believe that centrifugal-pump specifications can ever be too explicit or too detailed where the operating conditions are concerned. There are frequently circumstances which appear unessential to the purchaser which can affect the ultimate life of the equipment very materially. I will cite one specific instance: The specifications of a general-service vertical wet-pit pump that once came to my attention had been extremely detailed in all respects except in indicating the range of operating capacities. The inference had been that the pump would operate constantly at or about the design capacity. After the pump had been installed and very shortly after its initial operation, it was discovered that a serious shaft whip prevented satisfactory behavior of the stuffing-box packing. Two months from initial operation, the pump shaft broke right under the impeller. Investigation disclosed that 80 per cent of the time the pump was operated in a range of flows from 25 to 10 per cent of design conditions. This particular style of pump had a decided radial thrust in this range of capacities and was not suitable for operation except in the general range of 75 to 120 per cent of rated flow. Had attention of the manufacturer been drawn to the actual type of operation, a pump suitable for this service could have easily been selected at very little higher cost to the user and the serious difficulties avoided.

While we cannot treat in too great detail all the items which must be thoroughly investigated and made known to the prospective bidders, it may be advisable to list some of the essential information which should be stipulated in the specifications:

1. What is the nature of the liquid to be pumped? Is it
   a. Fresh or salt water, acid or alkali, oil, gasoline or slurry, etc.?
   b. Cold or hot and if hot, at what temperature? What is the vapor pressure of the liquid?
   c. What is its specific gravity?
   d. Is it viscous or nonviscous?
   e. Clear and free from suspended foreign matter or dirty and gritty? If the latter, what is the size and nature of the solids, and are they abrasive? If the liquid is of a pulpy nature, what is the consistency expressed in percentage or in pounds per cubic foot of liquid? What is the suspended material?
   f. What is the chemical analysis, pH value, etc.? And what variations are expected in this analysis?
2. What is the required capacity as well as the minimum and maximum amounts of liquid the pump will be called upon to deliver?
3. What are the suction conditions? Is there a suction lift or a suction head? What variations are expected in these conditions?
4. What are the discharge conditions?
   a. What is the static head? Is it constant or variable?
   b. What is the friction head?
   c. What is the maximum discharge pressure against which the pump must deliver the liquid?

5. Is the service continuous or intermittent?
6. Is the pump to be installed in a horizontal or vertical position? and if the latter,
   a. In a wet pit?
   b. Or in a dry pit?
7. What type of power is available to drive the pump, and what are the characteristics of this power?
8. What space, weight, or transportation limitations are involved?
9. Location of installation. This should include reference to elevation above sea level, geographical location with its effect on spare parts recommended, and immediate surroundings which might affect accessibility.
10. Are there special requirements or marked preferences with respect to the design, construction, or performance of the pump?

It is this last item which can easily become a very controversial issue. I believe that these special requirements or preferences should be listed but that the purchaser avoid too rigid an attitude toward a manufacturer who deviates from them in his quotation, provided that a sound explanation is given for the deviation. Remember that some of these preferences may be based on insufficient knowledge of the most modern practice or designs.

On the other hand, they may as likely originate from experience with pumps operating under the same conditions that the new pumps will have to meet. Much valuable information may be made available to the pump manufacturer, which will enable him to furnish that type of equipment which will give the longest and most reliable service.

But I firmly believe that the customer's recommendations should be limited to his experience with pumps operating under similar conditions lest his preferences result in the purchase of very special equipment. Whenever the manufacturer's standard construction can be used, it is preferable to specially built units, both from the point of view of initial cost and that of repair parts in later times.

To be sure, there are circumstances which may force the customer to write very "tight" specifications. This is the case, for instance, with municipal or Federal agencies. The legal problems which arise with the purchase of any equipment by these agencies dictate the use of such specifications lest the agency be constrained to purchase unsatisfactory equipment merely because the bidder's price is the lowest of all those submitted. But where such a situation does not exist, as in the case of private concerns who are not forced to buy the lowest bid, excess zeal in circumscribing the possible offerings into very narrow limits and in demanding special construction, special materials, and special tests where these are not needed does not lead to the selection of the most economical equipment.

Many customers, such as consulting engineering firms or large-volume buyers, find it advisable to develop "prefabricated" specifications for a variety of centrifugal-pump services such as boiler feed, heater drain, condensate, ash sluicing, circulating water, etc. This practice generally leads to a reduction in the cost of specification preparation, especially if the customer frequently avails himself of the help that can be given him by pump manufacturers in reviewing these standard specifications in the light of the latest developments and experience.

In summary, I feel that centrifugal-pump specifications should be as complete and detailed as possible with regards to "what must be performed by the pump" and as general as possible in restricting the manufacturer within the framework of unnecessary preferences or special treatment.

Question 1.2: Reducing Impeller Diameter to Correct Motor Overload

We have a centrifugal pump which is required to pump 20 gpm of water at 90 psi discharge pressure. The pump has a 7-in.-diameter impeller and is driven by a 5-hp 3500-rpm motor. When tested, it pumped 20 gpm at 100 psi pressure. The motor horsepower developed on this test was 6 hp. Evidently the motor is somewhat overloaded and our discharge pressure is too high. What rules of proportion must be applied to reduce the impeller diameter to proper size?

Answer. A centrifugal pump is a velocity machine. A change in impeller diameter will change the peripheral speed of the impeller directly in proportion to this change. Furthermore, the velocities in the pump impeller and in the casing for *similar points on the characteristic curve* will vary directly in the same proportion. Therefore, the pump capacity, which is a direct function of the velocities, will vary directly as the impeller diameter ratio. The total head, which is a function of the square of the peripheral speed, will vary as the square of the diameter ratio. Finally, since the power consumption varies as the product of the head and of the capacity, the power will vary as the cube of the diameter ratio.

However, it must be remembered that, with a given impeller design, a reduction in impeller diameter will result in a change in the basic design of the impeller and thus may affect the pump characteristics. For this reason, the basic rule can be applied only over a limited range, depending on the type of the impeller. Furthermore, if the impeller diameter is reduced excessively, the pump efficiency will be somewhat affected, so that the power consumption will no longer follow the cube of the diameter ratio relationship.

In the particular case at hand and presuming that the suction lift or suction head is negligible (you give no value for this item), the efficiency of the pump will probably change very little if the impeller diameter is reduced to give 90 psi discharge pressure instead of 100 psi. The shape of the head-capacity curve at 20 gpm is probably very flat at the present operating point, so that the diameter ratio can be *approximated* from the square root of the two pressures. This results in an impeller diameter of about $6\frac{5}{8}$ in. After the impeller is cut, the vane tips should be filed very carefully to approximate the finish of the vanes as they now appear. Presuming that the impeller cutdown does not exceed reasonable limits for this pump (and this would not be likely), the power consumption at 20 gpm would vary directly with the pressure. (Note that in this case the power does not vary as the cube of the impeller diameter, since we do not change the capacity at the operating point.) Thus, the power consumption at 20 gpm after the cutdown can be estimated at 5.4 hp.

Note that, when larger size pumps and operating conditions further to the right of zero flow are dealt with, this approximation is not quite correct. It becomes necessary to determine by trial and error, on the slide rule, the diameter ratio which will step up the desired capacity (directly as the diameter ratio) and the desired total head (as the square of the ratio) to a set of Q and H conditions right on the test curve of the pump. Having established this ratio, it becomes quite easy to plot the resulting head-capacity curve directly from the original one. One simply sets the ratio on the lower scale of the slide rule, and several points on the test curve having been selected, the capacities are stepped down directly on the same scale and the total heads on the square scale.

Question 1.3: Effect of Pump Speed Change

If the pump in the preceding question were belt driven, could the speed be changed to obtain the same result?

Answer. Since it is the peripheral speed of the impeller which determines the total head and the capacity of the pump, obviously it is immaterial whether the peripheral speed is changed by cutting down the impeller diameter or the pump speed. In this particular case, if the pump developed 100 psi net pressure at 3500 rpm, the speed would have had to be reduced in the same proportion as we found it necessary to cut the impeller diameter, in other words to 3320 rpm.

The important formulas to remember are, therefore,

$$\frac{\text{gpm}_2}{\text{gpm}_1} = \frac{D_2}{D_1} = \frac{\sqrt{H_2}}{\sqrt{H_1}} = \frac{\sqrt[3]{\text{bhp}_2}}{\sqrt[3]{\text{bhp}_1}}$$

and

$$\frac{\text{gpm}_2}{\text{gpm}_1} = \frac{\text{rpm}_2}{\text{rpm}_1} = \frac{\sqrt{H_2}}{\sqrt{H_1}} = \frac{\sqrt[3]{\text{bhp}_2}}{\sqrt[3]{\text{bhp}_1}}$$

Question 1.4: Pump Capacity vs. Impeller Diameter

I would like to refer to a statement which has appeared in a number of articles on centrifugal pumps and which deals with the effect of cutting down the impeller diameter on the capacity and head produced by a centrifugal pump. It is stated that, for similar points on the characteristic curve, the pump capacity will vary directly as the impeller diameter while the head will vary as the square of this impeller.

This statement seems to be contrary to general belief. I should like to point out that the discharge (or capacity) varies as both the area and the velocity. Reducing the impeller diameter would reduce both the area at the impeller exit and the velocity. Should not the capacity, therefore, vary as the square of the impeller diameter instead of directly as this diameter?

Answer. The functional relationship between a change in impeller diameter and the resulting change in capacity may be readily established in either of the two following ways:

1. Empirically, by analysis of actual test results.
2. Theoretically, by comparison of discharge velocity triangles or other analytical means.

In either case the results substantiate the fact that capacity varies directly with change in diameter, subject to slight variations which will be discussed later. Considering the empirical approach first, the substantiating evidence lies in literally thousands of tests which have been made and are still being regularly conducted by both pump manufacturers and pump users. While a review of any substantial number of such tests covering a wide range of different pump types would undoubtedly turn up some designs which would indicate a capacity change greater than that expected, it would also reveal some showing a smaller change than expected and on the whole would indicate that the large majority performed very closely in accordance with the rule that capacity varies directly with diameter.

As far as theoretical proof is concerned, there would probably be little value in and even less need for a complete theoretical explanation in this book, since this can be readily found in textbooks on pump design. However, I would like to comment briefly on your statement regarding reduction of discharge area with reduction of impeller diameter.

Assuming for purposes of discussion a radial-flow closed impeller with parallel shrouds, it is obvious that *circumferential* area is reduced in direct proportion to any reduction in impeller diameter. However, this area does not represent the effective discharge area of the impeller and therefore does not exert the principal controlling effect on pump capacity. If such an impeller is considered as a series of rotating flow channels (each channel being formed by the upper side of one vane, the underside of an adjacent vane, and the inside surfaces of the shrouds), it is then apparent that the *effective* discharge area is equal to the normal area between any two vanes times the number of vanes. Referring to Fig. 1.2, this area is represented by $A \times B$ for full diameter and by $A \times B'$ at the reduced diameter. In most impellers these two areas will remain very nearly or exactly the same for the range of diameters over which the impeller is normally applied. Since this area remains constant, capacity then varies only as velocity or, in other words, directly with impeller diameter.

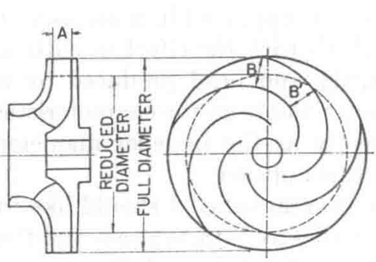


FIG. 1.2. Geometry of reducing impeller diameter.

The fact that variations from this rule are sometimes observed during pump tests does not in any way detract from its validity but rather points up the variations and limitations encountered in practical pump designs. For example, the necessity for obtaining special characteristics of pump performance, such as an unusually flat or unusually steep characteristic curve, may sometimes force the designer to distort the impeller areas or configuration from the optimum, thus affecting similarity relationships. In addition, internal losses in a pump prevent us from measuring the performance of the impeller itself and influence overall performance to vastly different degrees in different designs. These and a number of other similar factors all serve to illustrate why deviations from the general rule are to be expected.

Question 1.5: Narrow and Wide Impellers

I have read somewhere that the capacity of a centrifugal pump varies approximately with the width of the impeller. That is, if the impeller is $\frac{3}{4}$ in. wide and the head capacity is like that shown in Fig. 1.3, then if the impeller were made only $\frac{5}{8}$ in. wide on the outside diameter, the head curve would fall off in the ratio of $\frac{5}{8}:\frac{3}{4}$, or of $\frac{5}{6}$. Now I would like to know if the efficiency curve for the $\frac{3}{4}$-in.-wide impeller is as indicated on the sketch, what will it be for the $\frac{5}{8}$-in.-wide impeller? Will the whole efficiency curve shift to the left, or will it remain the same

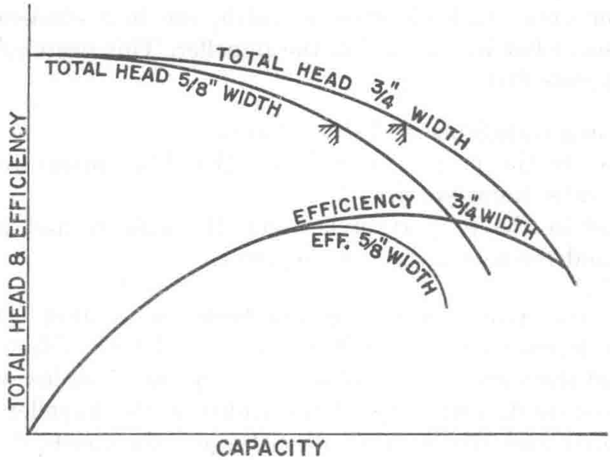


FIG. 1.3. Assumed effect of narrowing impeller width on pump performance.

as for the $\frac{3}{4}$-in.-wide impeller but fall off in the higher capacity range as I have indicated (see Fig. 1.3)?

Answer. The problem involved here is somewhat complex, and it is difficult to make general rules that would apply to any and all designs. While theoretically the head-capacity curve would fall off in the ratio of the impeller widths as you indicate, this would be true only within a narrow range of widths. Should too narrow an impeller be used in a given pump casing, there could take place an excessive amount of turbulence and shock losses, which not only would tend to reduce the capacity of the pump but also may react unfavorably on the head generated by the pump. Therefore, the head at shutoff or zero delivery may be less than that produced by the full-width impeller. If the reduction in width is not excessive, however, the head at shutoff will be approximately the same, as it can be controlled within certain limits by proper design of vane exit angles and other design factors.

The best efficiency point of a narrow impeller will move to a lower capacity approximately in the same proportion as the head-capacity curve, *provided* that other portions of the impeller are properly adjusted in design. In other words, it is also necessary to reduce the inlet area between the vanes. Another way of expressing this is that a factor less than 1.0 is applied to the entire impeller, after which the impeller is extended to its original diameter. The casing design may also affect the behavior of the impellers. If, for instance, the wider impeller is being severely throttled in the casing itself, the narrow one may show very little change in performance.

Except for very small changes in width, the best efficiency will be decreased somewhat by narrowing the impeller. This decrease is caused by three separate factors:

1. Increasing turbulence and shock losses.
2. Increase in the proportion between the disk horsepower and the useful water horsepower.
3. Increase in the proportion between the leakage and mechanical losses and the useful water horsepower.

The disk horsepower is that power required to drag the impeller through the liquid surrounding it and is caused by the friction of the liquid against the shrouds (or walls) of the impeller. This loss will remain essentially constant, regardless of the width of the impeller, assuming that the speed and the impeller diameter remain constant. However, the decrease in the pump capacity for any given total head will decrease the net output of the pump, and therefore, the proportion of the losses to the net output will increase, lowering the pump efficiency.

In the same manner, with the same differential pressure across the wearing rings, the leakage losses remain constant. So do the mechanical losses in the bearings and at the stuffing boxes. The overall effect can best be visualized by examining the formula for pump efficiency:

$$e = \frac{\text{water hp}}{\text{brake hp}}$$

where brake hp is made up of the sum of the following elements:

Water horsepower.
Hydraulic losses.
Disk horsepower.
Leakage losses.
Mechanical losses.

If we reduce the water horsepower but keep the disk horsepower, leakage losses, and mechanical losses constant, the efficiency will obviously be reduced.

The aforementioned should not be interpreted to mean that narrow impellers should not or are not used. As a matter of fact, in many cases a standard line of pumps will include two groups of impellers: One (called