A MAP OF THE FOREST . . . UNDERSTANDING PUMP SUCTION BEHAVIOR: WHERE DO WE GO FROM HERE?

by

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In 1980, he was awarded the first ASME Henry R. Worthington Medal. In 1981, he was elected an Honorary Member of the Russian-American Engineers Association, a distinction he shares with other scientists-engineers, such as Dr. Igor Sikorsky, the inventor of the helicopter, Prof. S. Timoshenko and Dr. V. K. Zworykin, the inventor of the TV iconoscope tube.

Mr. Karassik was employed by Worthington in 1934 and was engaged in research and design work on single and multistage pumps. By 1936, he had specialized in the application of multi-stage high pressure pumps, especially for steam power stations. From 1956 to 1959, Mr. Karassik was Consulting Engineer and Assistant to the Vice President, Harrison Division. From 1959 to 1963, he was Consulting Engineer and Manager of Planning of the Pump and Heat Transfer Division. In 1963, he became General Manager of the Advance Products Division, and in 1969-70 he was President of the Pioneer Products Division. From 1971 through 1976, he was Vice President and Chief Consulting Engineer of Worthington Pump, Incorporated. He was appointed to his present position on his retirement in December 1976.

Starting in 1936, as of December 1983, I. J. Karassik has written four hundred seventy (470) articles on centrifugal pumps and allied subjects for technical publications. Counting translations and the reprinting in English in foreign countries, a total of 1,495 magazine issues have carried his articles, in one hundred fifty (150) different magazines, in twenty-two (22) countries and in ten (10) languages.

Mr. Karassik is also the author of the books Centrifugal Pumps—Selection, Operation and Maintenance, Engineer's Guide to Centrifugal Pumps *and* Centrifugal Pump Clinic, *co-author of the book* Pump Questions and Answers *and co-editor of the* Pump Handbook. *Some of these books have also been translated into Spanish and Japanese.*

INTRODUCTION

Just over fifty years ago I started my career with Worthington and became involved with centrifugal pumps. Maybe it is time to take stock of what other engineers and I in the industry have learned about centrifugal pumps and to examine what still remains to be learned.

I would like to restrict myself to one particular area of it, and that is the phenomena which take place in and around the suction portion of the centrifugal pump impeller. Even this small portion of the total subject is almost too broad an area to do it justice in the short time available to us.

There is no dearth of literature published about the complex processes which underlie the behavior of centrifugal pumps with respect to their suction characteristics. Why is still one more paper being presented on that same subject? After all, it should be comforting and tempting to accept the printed word as gospel truth. The misfortune here lies in the fact that printed words on this subject contradict themselves, and I believe that we have said much but have explained too little.

I think that the most important fact I can list under this category is that our knowledge is far from being precise. We know reasonably well or can measure, for instance, the NPSH required by a given pump running at a selected speed and delivering a specified capacity of a particular liquid, and noting that the total head will not be reduced by more than 3 percent of the head, the pump develops with considerably more NPSH. We do not know whether the NPSH we measure represents any more than the average required NPSH for the bulk flow delivered by that pump. We do not know what the NPSH required is for any single discrete streamline entering the impeller.

When pumping liquids other than cold water, the required NPSH appears to diminish in accordance with some imprecise relationship to the characteristics of these liquids. Our attempts to reduce the required NPSH too drastically by altering the conventional configuration of the impeller can introduce serious dangers from a new quarter, namely the unfavorable effects of internal suction recirculation.

The more facets of these bits of knowledge I examine, the more questions I find to ask. Set against what we still have to learn, that of which we now have confirmed knowledge is trivial.

I should add, parenthetically, that this is not only the case with our knowledge of the behavior of centrifugal pumps. It is equally true of most of man's technology of the 20th century. I was never as impressed by the truth of this observation as when I recently read a suspense-mystery novel [1], the prologue of which started as follows:

> What oil is, and how it is formed in the first place, no one quite seems to know. The technical books and treatises on this subject are legion—and they are largely, so I am assured, in close agreement—except when they come to what one would have thought was a point of considerable interest: How, precisely, does oil *become* oil?

Here, I must make one more important observation. This world of ours can be readily divided into two categories of technical personnel: those who make pumps and those who use them. It is indisputable that there are considerably more of the latter than of the former. It is with this majority's interests that we really concern ourselves. We should remember that in the normal course of events, the user has no ready access to the design details of the pumps he will install or is already operating. He is primarily interested in interpreting the relationship between facts he knows or can ascertain—such as the characteristics of the liquid he is pumping—and the changes in performance that are created by these facts. The information that still remains to be developed should be such that would better satisfy these interests.

This paper, therefore, is intended to examine those areas of our subject which still remain clouded in some obscurity and to provide a starting point from which we can ultimately produce a map of the forest we need to traverse.

HISTORICAL BACKGROUND

Let us first review the historical events which brought us to where we are today. Quite early after the commercial introduction of the centrifugal pump, both manufacturers and users became aware that a great portion of field troubles experienced could be traced to inadequate suction conditions. Limitations on permissible suction lifts were imposed on strictly empirical grounds. The understanding of the phenomenon of cavitation was imperfect, thus, these limitations were sometimes over conservative, but more often over optimistic.

The need for a more exact understanding became more pressing as the requirements imposed on these pumps grew in magnitude. In 1922, at the Hydroelectric Conference held at Philadelphia, H. B. Taylor and L. F. Moody first presented the concept of a parameter, Sigma, to facilitate the description of the conditions under which cavitation occurs. Sigma was defined as:

$$\text{Sigma} = \int = \frac{H_s}{H}$$

where

H_s = Net Positive Suction Head

and

H = Total Head

D. Thoma was developing the same concept in Germany at about the same time and, therefore, "Sigma" has since been known to centrifugal pump designers as the Thoma-Moody parameter.

Means were now available to relate the operating conditions of a centrifugal pump—its capacity, head and rotating speed—to the minimum net positive suction head required for satisfactory operation. Commercial pressures seem to have outweighed sound engineering judgement much too often in the 1920s. The number of companies manufacturing centrifugal pumps had proliferated without necessarily a corresponding increase of knowledgeable and experienced designers. Spurred on by the advantage of offering a higher operating speed than competition or guaranteeing satisfactory operation with higher suction lifts, some companies made installations which had disastrously expensive consequences for users and manufacturers.

Codifying Suction Conditions

The Hydraulic Institute appointed a technical committee to investigate centrifugal pump suction problems. This committee proceeded to collect information on centrifugal pump installations in which cavitation troubles had been experienced, as well as on satisfactory installations. As early as in 1905, pump designers had started classifying the performance characteristics of centrifugal pumps by using the concepts of Specific Speed,

$$N_s = \frac{NQ^{1/2}}{H^{3/4}}$$

It was found that in order to avoid difficulties, for any given total head and suction lift or suction head conditions, the Specific Speed of the pump should be kept below a certain value. The conclusions of the committee were published in October 1932 in the Hydraulic Institute Standards, in the form of charts that have since become commonly known among centrifugal pump engineers and users as "Specific Speed Limit Charts."

It is important to remember that these charts were strictly empirical. They did not indicate that pumps built for the limit allowed were the best design, nor that pumps built to lower limits were not more economical in certain cases. All that these charts were intended to indicate was that for a given set of head, capacity and suction conditions, a certain maximum rotative speed should give some assurance that the pump would be capable of giving satisfactory service.

As experience was accumulated on better designs than described in these first charts of 1932, revised charts were prepared and published by the Hydraulic Institute. Charts were ultimately provided for several varieties of pump design, such as:

1. Double-suction pumps;
2. Single-suction pumps with shaft through the eye of the impeller;
3. Single-suction overhung impeller pumps;
4. Single-suction mixed and axial flow pumps;
5. Hot water pumps, single suction;
6. Hot water pumps, double suction;
7. Condensate pumps with shaft through the eye of the impeller.

Suction Specific Speed

The application of the Thoma-Moody concept or of the Specific Speed Limit Charts as they were originally developed had an important shortcoming: satisfactory suction conditions were tied directly to the total head developed by the pump. The performance of an impeller from the point of view of cavitation cannot be affected too significantly by conditions existing at its discharge periphery, which are the prime factors in determining the total head that the impeller will develop.

If an impeller exhibits certain suction characteristics, cutting down its diameter within reasonable limits and thus reducing its head should have no influence on its suction capabilities. Since the total head H is changed, a strict interpretation of the Specific Speed Limit Charts would indicate that unless the suction lift were to be commensurately altered, the maximum permissible specific speed must be changed. To maintain a fixed value for the Thoma-Moody parameter, a reduction in head by cutting the impeller diameter would be followed by a proportionate reduction in the net positive suction head.

I dwell in some detail on the developments that took place in connection with the elimination of this inconsistency, because I was intimately connected with these developments, and it might be interesting to have on record the manner in which the problem was eventually solved.

In 1937, two of my colleagues and I developed reasonable evidence that some sort of relationship did exist between Sigma (the Thoma-Moody parameter) and specific speed. However, the head term, present in both index numbers, was stubbornly refusing to disappear from the relationship. We knew that for a certain range of specific speeds, conditions at the impeller discharge could not be affecting suction conditions.

The steps that finally led us to the solution were reasonably simple. All that was required was a mere algebraic manipulation. We saw that if instead of trying to relate Sigma to the specific speed, we looked for a relation between the specific speed and the three-quarter power of Sigma, the total head disappeared very conveniently from the relation.

Thus was born the concept of Suction Specific Speed. If you look up the first two papers that were presented on this new method of representing pump suction characteristics, you will find that a much more sophisticated derivation of the Suction Specific Speed was developed. It can be derived using either similarity considerations or dimensional analysis. The interesting fact is that it was first stumbled upon by much simpler means.

Forty-seven years after the event I will not be criticized for divulging that a very bitter controversy sprang up immediately within the Hydraulic Institute over the validity or necessity of this concept. The controversy was short-lived and Suction Specific Speed was accepted as the most convenient parameter for comparing the suction capabilities of centrifugal pumps without consideration of their size or of the total head they produced.

The Hydraulic Institute Specific Speed Limit Charts were revised several times since they were first adopted as a guideline for centrifugal pump suction conditions. Until 1983, they were still based on the erroneous premise that the total head developed by the pump plays a part in determining the maximum permissible rotative speed for a given set of suction conditions. Until 1983, the Standards continued to define suction capability limits through the use of suction lifts and suction heads on 85°F water, rather than of required NPSH, thus necessitating a conversion for other temperatures and other liquids.

Finally, in 1983 (14th edition), the Standards were revised considerably. They are based on a Suction Specific Speed of 8500 for both single- and double-suction impellers. The recommended values are now expressed in terms of NPSH, significantly simplifying their use. Charts for special applications, such as boiler feed or condensate service have been eliminated.

Of course, nothing is ever perfect! The 14th Edition introduced a strange and false concept: that of the Available Suction Specific Speed. Such a term is utterly irrelevant and it is hoped that it will be eliminated in the next edition.

As early as in the 1932 edition of the Standards, the critical suction condition that was being codified was defined as that which produced a 3 percent drop in total head. We can be sure that the choice was either arbitrary or assumed to be one in tune with the accuracy of instrumentation available at the time.

I do wish to mention, however, that this definition would return to haunt us in later years. Hydraulic Institute Standards made vague references in later editions to the fact that under some circumstances a more prudent approach was to use a 1 percent head drop as the criterion for pump tests. Coupled to the vagueness of the reference was the fact that no guidelines were indicated for the relationship between the values of NPSH required for different values of head drop.

However, the die was cast. The 3 percent drop in head was set in concrete for at least fifty-some years and this afforded me and many other engineers the pleasures of dissent and criticism for that same long period. We would not be robbed of our pleasures and so, we never protested loudly enough to win the day for our own preferred definitions. I use the plural advisedly.

The Standards give recognition to certain side issues. For instance, an extra value to be added to the NPSH when handling hot water was introduced in the 8th Edition (1948). It was intended to afford some added protection to pumps on boiler feed service in steam power plants with open feedwater cycles against the unfavorable events which follow a sudden load reduction. For some unexplained reasons, the chart giving this extra NPSH was eliminated in the 10th Edition (1961). Again, bowing to the inevitable, the Standards recognized that when handling hydrocarbons and certain other liquids, the required NPSH appeared to diminish and a correction chart was introduced in the 9th Edition (1951). It was later revised in the 13th Edition (1975) to provide a better correlation between the guidelines and the observed facts.

DEFINITION OF REQUIRED NPSH

Some explanation is necessary here of the reasons for my disenchantment with a definition of required NPSH which rests on an observed measurement of a 3 percent drop in head. When the technical committee of the Hydraulic Institute was assigned the task of codifying centrifugal pump section conditions, it prescribed a "suppression test" as the means of establishing required NPSH. This test consists of operating a pump at constant speed and constant capacity and of progressively reducing the available NPSH. The total head produced by the pump at that speed and capacity is plotted against the available NPSH. A drop in head is then considered to be an indication of cavitation. Because of the difficulty of determining the exact conditions when this change takes place, the Hydraulic Institute defined required NPSH as that value where a drop of 3 percent in head will have taken place (Figure 1). The test is then repeated over a range of capacities and the curve, or required NPSH against capacity, is produced in this manner.

It is this definition of required NPSH which creates some of the ambiguities of our problem. Certainly, a deterioration of 3 percent in the total head produced by a pump indicates that cavitation is taking place and obviously will take place even under somewhat higher values of available NPSH. Since guidelines were intended to assure that no cavitation should take place, the present definition is really not effective in producing cavitation-free operation. It creates one more serious incongruity. Practice in the United States, unlike that followed in Europe, accepts no negative tolerance for either head or capacity guarantees. By implication, if one accepts a 3 percent drop in head when defining required NPSH, one automatically ignores this prohibition against negative tolerances.

One could claim that there are commercial and even legal impediments created by this definition. If, under the guaranteed conditions of required NPSH, a pump suffers a 3 percent reduction in head, under a strict interpretation of the accepted guarantee code it follows that full legal and binding contractual obligations will not have been met. While I have never heard of someone having taken legal recourse in this connection, the paradox nevertheless remains.

A more suitable definition must be developed. Several possibiliies present themselves for consideration:

1. One may choose a lower percentage of head degradation than 3 percent, such as possibly 1 percent or even 0 percent.

2. Considering that some cavitation probably takes place even with a 0 percent head drop, one might wish to choose an NPSH value which would suppress even "incipient cavitation", that is the creation of even a few small vapor bubbles within the confines of the impeller.

This second possibility does not appear to be practical. I strongly urge that it be eliminated from consideration. Whether we should choose 1 percent or take the final step and proceed directly to a 0 percent drop in head criterion, I prefer to leave the decision to others. For the sake of brevity, I shall assume that the decision shall fall on the 0 percent drop.

That is not to say that once we alter our definition of required NPSH:

1. We cannot use any of the accumulated data, which are almost in their entirety based on tests with 3 percent drop in head, or

2. We will have to carry *all* future tests on the basis of 0 percent drop in head.

Once the relationship between the NPSH required at 0 percent and at 3 percent drop in head is established with a reasonable degree of accuracy, existing and new test data can be converted at will from one basis of definition to the other.

Incidentally, to distinguish between various definitions of required NPSH, I suggest that we adopt an auxilary nomenclature as follows:

1. We should use the subscripts (3), (1), (0) and (i) to denote conditions with 3 percent, 1 percent, 0 percent drop in head and for incipient cavitation, respectively.

2. The subscript (w) added to other subscripts will refer to the performance of a pump on cold deaerated water.

3. These subscripts will apply equally to the required NPSH (e.g. H_{s3w} or H_{s0w}) and to the corresponding Suction Specific Speed values at BEP (e.g. S_{3w} and S_{0w}).

If we change the present definition of required NPSH, we must engage in a series of other changes as well. The guidelines provided in the Hydraulic Institute Standards for recommended NPSH values would have to be replotted to reflect the new definition of acceptable suction conditions. The value of the Suction Specific Speed for any given pump would have to be recalculated and so would be any recommended values of this Suction Specific Speed as, for instance, maximum values to avoid the unfavorable effects of internal recirculation.

AS THROUGH A GLASS, DARKLY

Considering the immensity of the literature on our subject, it is remarkable that there might still remain anything important to discover about the forest of facts related to pump suction performance. Is some of this material made up of hypotheses, plain surmises or even fiction? I say this because when I come to those things that we do not know, the list of questions I pose to myself and to my professional colleagues is more extensive than one might suspect. For instance, we still do not know too precisely the following facts:

1. We do not know the relationship between H_{s0w} and H_{s3w}. Each article that I have read gives a different ratio between these two values, with variations of as much as 2 to 1 or more. I submit to you that such a range of values indicated that we do not know what the relationship is. Nor do we know the relationship between H_{siw} and H_{s3w}.

2. We are not quite certain about the effect of pump speed on any of the required NPSH values.

3. We do not know with sufficient precision the facts that create the impression that the NPSH required for hot water or for hydrocarbons is appreciably less than for cold water.

4. We do not know whether the values for H_{s0} and of H_{si}—contrary to the value of H_{s3}—are equal for all liquids. I think that the answer to this question is "yes" but we have not enough proof for this answer.

5. And while we are at it, what is a proper definition for H_{si}, the NPSH for incipient cavitation?

6. What is the precise effect of dissolved or entrained gases or air on the required NPSH?

7. How and exactly where does the collapse of the bubbles created by cavitation take place as the two-phase flow proceeds through the impeller? Do the bubbles which collapse along a streamline in the middle of surrounding liquid play any part in creating the damage we so frequently observe? Or are the bubbles only those which collapse in the immediate vicinity of the surface of the impeller vanes? We do not know, nor do we know the velocities of the liquid which rushes in to fill the voids created by the collapse of the bubbles.

8. Does liquid surface tension play any part in the suppression of flashing? Or does it play such an insignificant role that we need not concern ourselves with it?

9. What effect, if any, does liquid viscosity have on the required NPSH?

10. When I spoke about the relationship required for 3 percent and 0 percent drop in head and for incipient cavitation, I was referring to NPSH values at the best efficiency point. But we also need to know the NPSH required at other capacities. What is the shape of the curves for H_{s3}, H_{s0} and H_{si}?

11. Why does the required NPSH of cryogenic pumps first drop with a reduction in capacity and then, reversing itself, climb upwards so as to apparently exceed the NPSH at best efficiency flow?

12. What is the magnitude of the NPSH available that is required to suppress the unfavorable effects of the problem of internal recirculation at the suction? This simply eliminates physical damage to the impeller (damage to the pressure side of the vanes, that is the invisible side), but not necessarily eliminate the hydraulic surges and pulsations?

I realize that this list is not complete. Maybe I have raised a sufficient number of questions for us to examine what are the philosophical reasons for the fact that they have remained unanswered. I shall re-examine some of these questions in greater detail.

All the rules we have enunciated in the past as governing the performance of centrifugal pumps have been derived empirically. Once theoretical relationships have been set up between the physical configuration of a pump and its performance, various coefficients are established from tests to correlate the actual and the theoretical performance. These coefficients are only exact for the particular pump which has been tested. Experience has shown that they can be applied to other pumps sufficiently similar to the test pump—at least within the limits of commercial accuracy.

Why can we not establish these coefficients from strictly theoretical considerations? The theory of centrifugal pump performance is derived on the basis of two-dimensional flow, while the real flows are three-dimensional. Even three-dimensional calculations of pump performance depart materially from the true state of affairs. Empirical verification of coefficients is actually the most reliable means at our disposal, because we do not pump ideal fluids, because flow patterns within the impeller and the casing are extremely complex. We are interested in the "total average" performance of a pump—not with the theoretical performance of any one single streamline. A test is the best means to establish this average performance.

Even such a test may be deceiving. Assuming that there has been no conscious and intended deviation in the modelling of a pump from its prototype, a variance between predicted and actual performance may still exist. The degree of this variance will depend on:

1. The exactness with which the new patterns reproduce the calculated dimensions;
2. The accuracy of the casting process;
3. The relative finish of the castings of the internal

machined surfaces;

4. And not the least, the relative accuracy of the tests conducted.

The deviation in performance may be wrongfully ascribed to the scale factor or to the differences in the characteristics of the liquids pumped.

I really cannot overemphasize the empirical character of our understanding of all the phases of the performance of centrifugal pumps. This holds true for the performance with any given liquid, but even more so for deviations caused by such diverse factors as the thermodynamic or viscosity characteristics of different liquids, or by the presence and amount of entrained or dissolved gases.

In the pursuit of our assignment to produce a map of the forest, let us bring our theodolite into closer focus on the questions I have raised.

H_{s0} *vs.* H_{s3}

Let us consider first the relation between the required NPSH at 0 percent head drop and that at 3 percent head drop, that is:

$$\frac{H_{s0}}{H_{s3}}$$

I have been intrigued by reading the large number of articles that have appeared on the general subject of centrifugal pump suction conditions and which give discrete numerical values for this ratio. From the collection of data on this relationship, one might imagine that very definite information is available, that the subject has been exhausted, that everything that needs to be known is known, and that my claim that our technology in this area is not a precise science is not warranted.

Or is this last statement true? All that is needed to prove that, unfortunately, is to compare some of the published data. You will find ratios as low as 1.25 and as high as 2.5 or even 3.0. I am reminded, perversely, of the saying that if a man has a watch, he knows what the time is; if he has two watches, he no longer knows.

Why, then, should it be so difficult to establish reasonably precise values for the ratio H_{s0}/H_{s3}? It is difficult because they are affected by a large number of factors, of which the following is probably only a partial listing:

- The characteristics of the liquid pumped, including both chemical composition and thermodynamic properties such as the vapor-to-liquid specific volume ratio.
- The pumping temperature.
- The presence of dissolved and entrained gases and their volume by percentage.
- The relative size of the individual vapor bubbles generated by the flashing of the liquid.
- The specific speed of the pump.
- The suction specific speed of the pump.
- The exact configuration of the impeller vanes at the inlet.
- The Reynolds Number, as affected by the pump size and pump operating speed.

As a matter of fact, my friend and colleague W. C. Krutzsch facetiously remarked that I should add one more factor which he expressed thus: "Once in a while, the ratio H_{s0}/H_{s3} will vary in inverse proportion to the desire of the vendor to get the order." He might have been somewhat cynical, but I am afraid that he was right in some cases. I don't imagine this happens too often, but certainly the availability of well documented data on this matter would preclude even these rare occurrences.

The very nature of some of our definitions creates problems. As long as we say that the required NPSH is that value with which the drop in head is 3 percent, our explanations of why the NPSH required for hot water or for hydrocarbons is less than that for cold water will be reasonably correct. However, the moment we speak of NPSH required for 0 percent drop in head, or for incipient cavitation, we can no longer claim any distinction based on the characteristics of the liquid. As a matter of fact, we can deduce by logic that the NPSH for 0 percent drop in head should be the same for all liquids. Every reduction in the degree of cavitation which we ascribe to variations in liquid characteristics requires that at least some cavitation take place before any differentiation can occur. This is true with respect to the effect of the ratio between the specific volume of vapor and that of the liquid. It is equally true of the varying effect of the sub-cooling created by the flashing of even a minute portion of the liquid being pumped.

One more source of difficulty is the sub-conscious mental block created by our choice in setting up this ratio. I find it distracting to express the relationship between the NPSH required for 0 percent drop in head and that for 3 percent drop, as a ratio between the former and the latter. The use of such a ratio implies that the NPSH at 3 percent drop in head is a discrete value determined strictly by the geometric configuration of an impeller, and that the NPSH at 0 percent drop in head varies under the influence of liquid characteristics.

It is the reverse of this implication which is true, and I strongly urge that we express the relationship as the inverse ratio which, of course, will always yield a fractional number less than 1.00. Coupled with the redefinition of NPSH required as that value of NPSH which produces a 0 percent drop in head, this recommendation would go a long way towards clarifying our thinking.

Incipient Cavitation

I find it difficult to adopt an acceptable definition "incipient cavitation." One could argue that the first appearance of a bubble of vapor bears witness to the presence of cavitation. Such a definition would create almost unsurmountable obstacles to establishing easily observable detection and, probably, repeatable observation.

I do feel uneasy when confronted by a definition based on the observation of a finite length or area of bubble formation, as it has been suggested by some engineers. If for instance, incipient cavitation is defined by the presence of a 5mm trail of bubbles, I have two questions.

1. Why 5 mm and not 4 mm or 6 mm?
2. Is the degree of cavitation which results in the formation of 5mm of vapor bubbles the same for a 12 inch impeller as for a 24 inch impeller?

There have been suggestions made of defining an NPSH value which would guarantee somewhere between 30,000 and 40,000 hours of safe pump operation. I am very reluctant to consider such a proposition. It would require users and manufacturers alike to consider the effect of cavitation on a vast variety of impeller materials and to conduct long duration tests which would be most unlikely to duplicate "real life" situations.

My objections may lead to the unacceptable conclusion that the NPSH required for incipient cavitation may always remain a mysterious, unknown and unknowable number. If this is so, I can offer no words of consolation.

Effect of Dissolved or Entrained Air

When a pump is tested with cold water containing a certain amount of air, the air bubbles pass from the suction towards the entrance to the impeller, into a zone where the ambient pressure is even lower than at the pump suction flange. Air being a compressible fluid, the bubbles will expand in volume inversely as the absolute ambient pressure. They will occupy a volume even greater by percentage than at the suction flange. Whatever that volume by percentage may be, once these bubbles reach the entrance to the impeller, they will occupy space that would normally be filled with liquid. It follows that for the same net capacity of liquid being handled by the pump, the liquid velocities will be higher than if there were no air present.

These higher velocities, in turn, will require a greater transformation of static pressure into kinetic energy and, consequently, the presence of the entrained air will have led to a greater lowering of the ambient static pressure at the entry to the impeller. Thus, I am led to conclude that the presence of entrained air (or gas) tends to increase the required NPSH for all rates of flow.

The key question remains "by how much?" It is a difficult question to answer, because the effect on the NPSH will be masked by the effect of the entrained air on the head-capacity performance of the pump. Our problem stems from the fact that we are confronted by two separate, simultaneous phenomena:

1. The presently accepted definition of required NPSH is that it is that value of NPSH which, for a given capacity and at a given speed, produces a 3 percent drop in total head, as illustrated on Figure 1.

2. On the other hand, it is a well known and documented fact that if as little as 1 percent by volume of air or gas is entrained with the liquid pumped, the head-capacity curve is noticeably reduced. As this percentage by volume increases, the reduction becomes even more drastic, until at about 6 percent we reach a condition when most pumps cease to perform satisfactorily.

There might be a way out of our quandary, though I am not quite satisfied that the results would be fully valid. We could conduct our tests for NPSH, not with deaerated water, but rather with whatever percent of entrained air we wish to use. The head-capacity curve with ample NPSH would be used as our bench-mark. Then, a regular suppression test would be conducted and the required NPSH with that percent of entrained air established. Finally, this last value would be compared with the required NPSH at the tested capacity under deaerated conditions. Again, the results may not completely isolate the effect of the entrained air on the required NPSH.

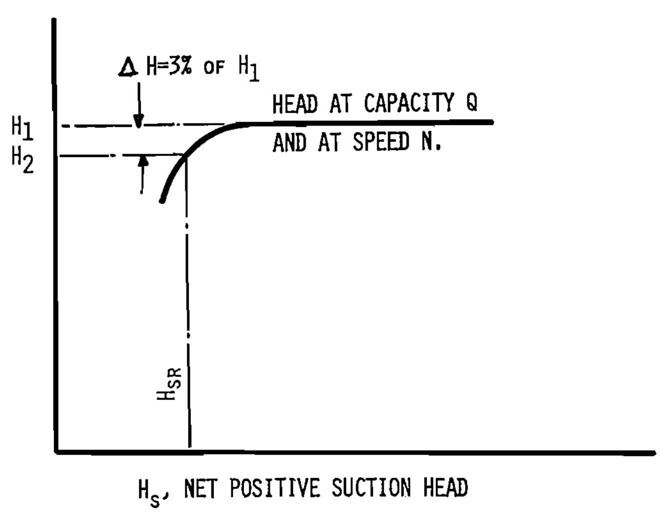

Figure 1. Determination of Required NPSH as Presently Defined by the Hydraulic Institute.

The results may not necessarily be "generic", that is applicable to all pumps of all types. Certainly much more analysis and many more tests would be required before we could feel certain that we know the answer.

Effect of Viscosity

Since the NPSH required includes some hydraulic friction losses between the pump suction flange and the entrance into the passages between the pump suction flange and the entrance into the passages between the impeller vanes, it should follow that pumps handling liquids with a viscosity higher than that of water would have a higher friction loss component of the NPSH. Hence, they would require a somewhat higher NPSH than when handling water. But, by how much? And is this increase significant enough to be measured? Frankly speaking, I do not know, nor do I know of any reference text which has reported authenticated test data on this subject.

Shape of the NPSH Curves

No less variety is offered to us when we examine the shape of the NPSH curves plotted against capacity which have appeared in the literature on this subject. Figure 2 is taken from one of my own articles and illustrates my own indecision by showing variations with question marks in the range of lower capacities. I have carefully refrained from providing these

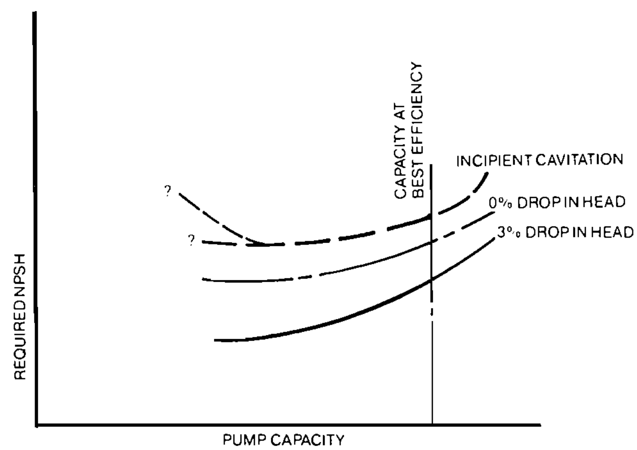

Figure 2. Probable Shapes of Required NPSH Curves for 3 Percent and 0 Percent Drop in Head and for Total Suppression in Incipient Cavitation.

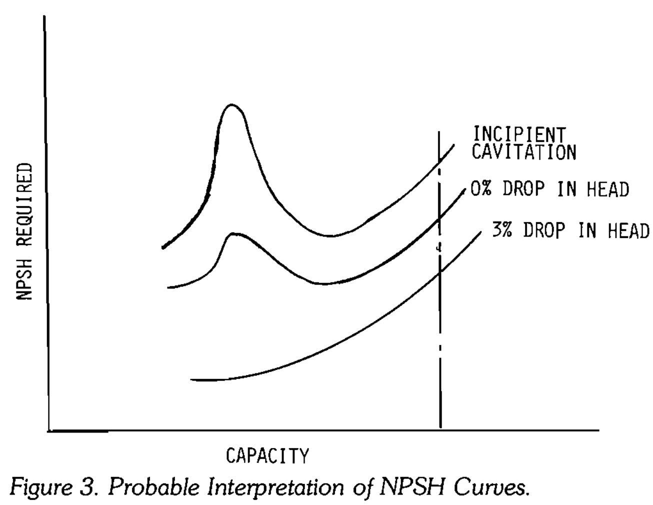

Figure 3. Probable Interpretation of NPSH Curves.

curves with any scale. I have indicated, qualitatively, that the ratio between the NPSH rquired at 3 percent and at 0 percent drop in head does not remain constant over the entire range of pump flows, and that the variation in ratio is even more pronounced in the case of NPSH for incipient recirculation. A somewhat different interpretation, typical of those which one can find in several recent papers is shown in Figure 3.

I suspect that some of the curves that resemble Figure 3 combine the effects of classical cavitation phenomena and those created by internal recirculation at the suction. If this is so, I would prefer something more specific, that is two separate curves, each one illustrating one of the effects. The curve showing the energy required to suppress the unfavorable effects of internal recirculation must, by virtue of its definition, remain equal to zero at all flows above the onset of internal recirculation, as shown on Figure 4. The exact shape of this curve is unknown, as indicated by the question marks.

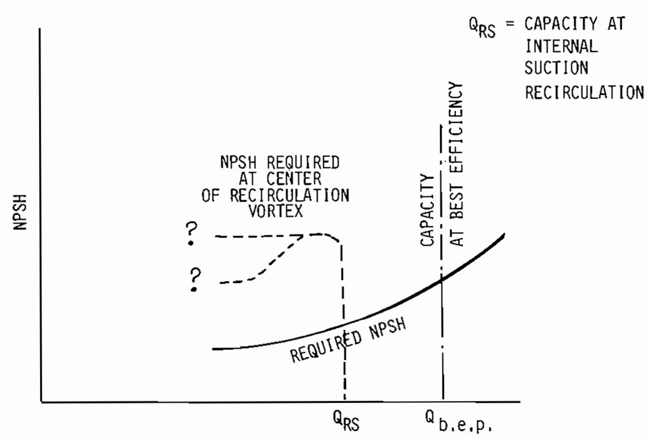

Figure 4. Possible Shape of NPSH Curves at 3 percent and 0 percent Drop in Head and at Incipient Recirculation.

NPSH of Cryogenic Pumps

It have been frequently claimed by users that the required NPSH curve of cryogenic pumps has a marked increase as the capacity falls to some 20 or 30 percent of design conditions. These claims appeared to fail the test of logic, particularly as reference was made to NPSH values with a 3 percent drop in head. Now, I think we may unravel this mystery.

The observed facts are strictly apparent and not real. What one sees in a test for NPSH of a cryogenic pump is an error in measuring the NPSH available and not an increase in NPSH required.

One does not measure the NPSH required, one measures the available NPSH and then determines the required NPSH by observing the test values and calling that NPSH available at which a 3 percent drop in head occurs the required NPSH. The available NPSH is stated to be equal to the energy over and above the vapor pumping temperature *at the pump suction flange.*

At or near the best efficiency point, this cannot introduce any significant error, since the temperature rise in the pump is negligible, and the flow past the wearing ring of the first stage is but a diminutive fraction of the flow into the pump. Thus, the temperature at the eye of the impeller does not change from the temperature at the suction flange and the assumed vapor pressure is essentially correct.

But, as the capacity is reduced, the temperature rise increases and the leakage flow increases as a percent of the suction flow. The calculated temperature rise takes place in the discharge passages. Some of the liquid flows back into the suction through the clearance joints and mixes with the incoming liquid.

The temperature at the eye of the impeller is no longer the same as at the suction flange, nor of course is the vapor pressure.

Consider, for instance, the effect of an increase in liquid temperature of 1°F on the vapor pressure of water at 80°F and, say, of methane at the usual pumping temperature of −240°F.

	Temperature	*Vapor Pressure*
For water:	80°F	0.507 psia
	79°F	*0.490 psia*
Δt = 1°F.		Difference 0.017 psi
	or	0.04 feet
For methane:	−240°F	33 psia
	−220°F	*64 psia*
Δt = 20°F		Difference 31 psi
Δt = 1°F.		Difference 1.5 psi
or, m at s.g. of 0.4 = 8.95 feet		

In other words, an increase in temperature of 1°F increases the vapor pressure of 80°F water by 0.04 feet and that of −240°F methane by 8.95 feet.

It should be noted that at some low flow, the effect I have described raises the liquid temperature by 0.5°F at the eye of the impeller, the result increases the vapor pressure by a negligible amount if the liquid is 80°F water, but by as much as almost 4.5 feet if it is methane at −240°F.

This increase in vapor pressure is not normally taken into account when running the NPSH test. The real NPSH available is 4.5 feet less than the apparent NPSH available, if we use the temperature rise I have assumed. Since by definition the NPSH required is that NPSH available which will not cause a drop in total head of over 3 percent, *it appears that the NPSH required has gone up,* but it hasn't really.

NPSH for Internal Recirculation

At this point, I would like to examine a very curious and still unresolved question dealng with the phenomenon of internal recirculation at the impeller suction. Specifically, the question might be posed as follows: What is the NPSH available that will eliminate all the unfavorable effects of internal recirculation? Under certain available NPSH conditions, all symptoms of internal recirculation disappear. Damage hardly ever occurs at the inlet of the second stage of a multistage pump. The value of this NPSH is far from having been established.

The answer to this question may prove to be of strictly academic interest. If one has to provide as much as 4 or 5 times the required NPSH to permit operations of a pump well within the recirculation zone, why not simply select an impeller with an NPSH required 1.5 times greater than the original selection? This will reduce the Suction Specific Speed from, for instance 12,000 to 8,850. The net result in both cases, would be the broadening of the safe operating capacity range and would be achieved with a considerably lesser increase in available NPSH.

REDEFINING OUR PRIORITIES

What are we to make of all these observations on the state of the art with respect to the suction performance of centrifugal pumps? I think that I have given reasonable proof that there still remains a long list of unanswered questions. On the other hand, I have not demonstrated that all these questions can be answered in a practical context. Certainly, some of them present a strictly academic interest—not a commercial one. The progress of our technological civilization will not be measureably impaired if some of my questions remain forever unanswered.

Remember that our task is not limited to satisfying our scientific curiosity. It should really be directed to providing the user of pumping equipment with such information that will permit him to select it, install it and operate it with the greatest degree of reliability at no excessive sacrifice of expenditures. This must be our first and foremost priority—the search for this information.

When all is said and done, it is entirely possible that certain characteristics of a relatively simple machine such as the centrifugal pump can never be predicted with any acceptable degree of precision. Thus, the question remains whether further advances will in fact increase our understanding or whether they will only confuse us further. In turn, the answer to this last question depends on our ability to divorce our observations from the influence of deep-seated preconception. We will have to learn to constantly test old theories against fresh evidence.

Yet, I cannot help thinking that there is invariably an answer to every technological problem. This answer may be impossible to provide at any given moment, because the conceptual work is insufficiently advanced. In the end, however, there is a solution waiting to be found.

CONCLUSIONS

I hope that there is a lesson to be learned from this excursion into a technical territory which has been frequently explored before, but of which no overall map had been drawn up for a variety of reasons. I shall not dwell on what these reasons were, lest I disturb the equanimity of some friends and some acquaintances. Nor shall I try to spell the moral of the lesson, as I am unsure whether it has one or, if it does, that the moral is digestible.

If I am permitted to extend the metaphor of the title I have chosen for this paper, here, then, are a few seedlings I have brought to be planted at the edge of the forest. May the engineers who follow me nurture them to full growth.

REFERENCE

1. Maclean, A., *Athabasca,* New York: Doubleday, 1980.