OVERVIEW OF INDUSTRIAL AND ROCKET TURBOPUMP INDUCER DESIGN

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ABSTRACT

High performance inducers for rocket turbopumps and industrial low suction pressure pumps have been designed frequently during the past century. Some design lore has evolved to guide this design process; additionally, some detailed flow observations have been made which shed light on the basic flow process. Design methods have been recommended during the past decades to guide inducer design. This paper reviews some of the design methods, some of the flow observations, and some of the design practice which is used for inducer design. The purpose of this review is to bring together a good portion of prior art and focus it from a modern designer's perspective. Some suggestions for future improvements are provided. A useful overview of current design practice is given which should provide guidance to current inducer pump design.

1. INTRODUCTION

The inducer is an important part of many industrial pumps and rocket turbopumps for which very low inlet pressure is available. In these pumping applications, it is necessary to draw the fluid into the pump with very little ambient pressure, hence, greatly increasing the probability of a critical reduction in pressure below the vapor pressure of the fluid. In this situation, cavitation occurs in many different modes. Examples of cavitation can be found in the books by Brennen (1997) and Japikse et al., (1997) which detail examples of suction side, pressure side, tip vortex, and numerous other types of cavitation formation. The initiation of cavitation is usually not a particularly severe process, but as the inlet pressure is further reduced, the zone of cavitation grows substantially and a large region of blocked flow occurs in the inducer. When this regime of blocked flow grows substantially, usually at the point where it exits the entire stage, there is a total breakdown in head and the pump stage no longer can complete its task of delivering flow at a desired head. Most designers are concerned with the point where the head breaks down substantially; other investigators are concerned where maximum damage occurs or even where the first initiation of cavitation might arise. Associated with these problems is the additional consideration of stability. Some of the cavitating flows are mildly stable whereas others are inherently unstable. Due to the fact that the cavitation process is random in character with continuous formation and collapse of cavitation bubbles, there is nothing in the process which is inherently stable. However, a variety of specific cavitation instabilities can occur, such as rotating stall, cavitation surge, and even large system instabilities which have been referred to as POGO. Against this background, it may be noted that designers have developed methods to execute appropriate designs which have covered a very wide range of industrial requirements and space propulsion requirements. Such designs, however, do press the limits of acceptable performance and significant research studies are ongoing with the objective of extending the design methodology. This investigation simply presents a review of the historical elements of inducer design and then outlines some of the important factors which should be, and, in fact, are being investigated.

2. HISTORICAL OVERVIEW OF DESIGN PROCEDURES

A wide variety of parameters must be considered in the design of an advanced industrial or turbopump inducer. These include the optimum inlet eye diameter, the inlet blade angle, (particularly at the inducer tip), the design point incidence value, leading edge shape, and the blade number as well as the blade turning angle and structural design parameters. These are reviewed on an issue-by-issue basis herein. The first three (the inlet eye diameter, the blade angle, and the incidence level) are, however, closely coupled together and should receive detailed consideration first.

As early as 1962 attention was given to the inlet design problem. Stipling (1962) presented calculations relating to the suction specific speed of an inducer pump versus an optimum flow coefficient at various blade angles. His results are shown in Figure 1. To reach the highest levels of suction specific speed, which are sometimes considered for rocket turbopumps today, it may be necessary to extrapolate beyond the range of results presented. Fortunately, the equation describing Brumfield's criteria is presented in the paper so the extrapolation can be conducted rigorously. Brumfield's equation is:

$$\frac{NPSH}{U_{1t}^{\ 2}/2g} = \frac{2\phi^2_{opt}}{1-2\phi^2_{opt}} \tag{1}$$

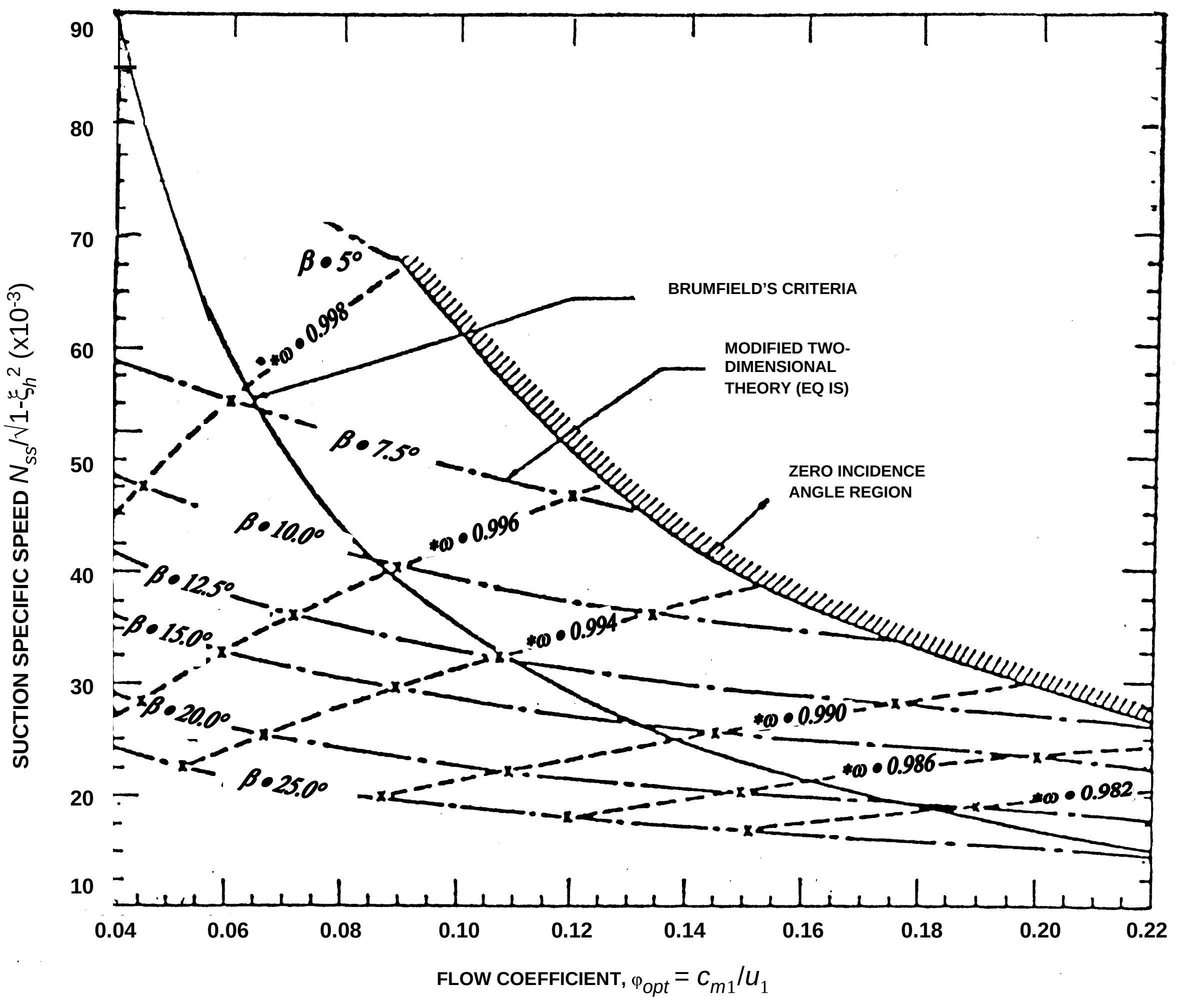

Figure 1. Suction specific speed versus optimum flow coefficient for various blade angles (adapted from Stripling).

Unfortunately, the original reference to Brumfield's work has evidently been lost and today only a name is associated with the above equation. Stripling concluded that the Brumfield equation was well supported with his cascade theory.

The next important contribution was by Furst and Desclaux (1989). Their work presented a relationship for the suction specific speed for an inducer pump, but employed an empirical coefficient that is not easily obtained. Also, a mathematical error in the equation required correcting. When the relationship was employed, the results of Figure 2 were achieved. In this case the recommended empirical coefficient of 1.2 was employed and a design flow coefficient of 0.055, typical of certain turbopumps, was utilized. Operation at a variety of different coefficients is displayed in the figure. In this figure, four different calculations have been compared and the locus of maxima roughly resembles the Brumfield correlation of Figure 2. For an inlet blade angle of 2.8°, it is possible to achieve

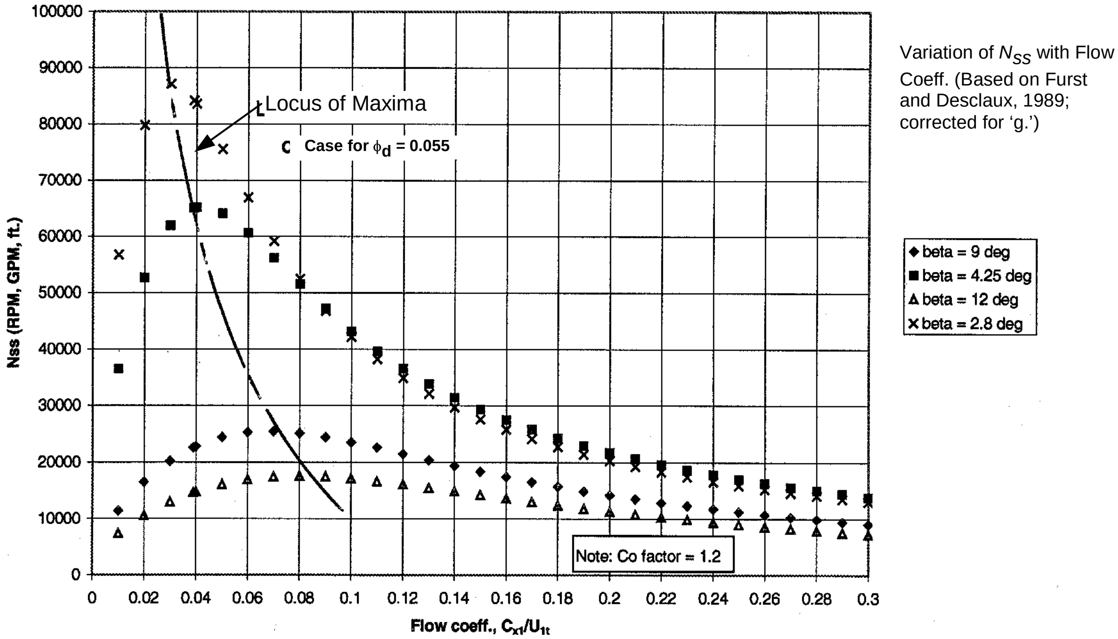

Figure 2. Evaluation of suction specific speed as a function of inlet blade angle and impeller flow coefficient according to the model of Furst and Desclaux, 1989.

N_{ss} = 85,000. At a blade angle of approximately 4.25°, the theory indicates that an N_{ss} = 65,000 can be achieved. These results are interesting both from the qualitative character and from the quantitative character as well; they differ strongly from the Brumfield/Stripling approach (2.8° versus 4.8° at N_{ss} = 85,000 and 4.25° versus 6.0° at N_{ss} = 65,000). Incidentally, this model is based on a specific perception of certain possible flow phenomena (perhaps speculative) without large amounts of validating data.

The flow models used by Stripling, and Furst and Desclaux assume basically a two-dimensional flow along simplified flow lines or pseudo-streamlines which would be parallel to the hub and shroud surfaces of the passage. In this presumed smooth flow direction, a cavitating sheet is presumed to develop and to form a thick region of blockage much like a classical boundary layer development problem, but on a much thicker scale. The Furst et al. model is based on the idea of this region of blocked flow, assuming a two-dimensional wake-like or boundary layer-like development of the sheet of cavitating flow. The assumption always is of sheet cavitation.

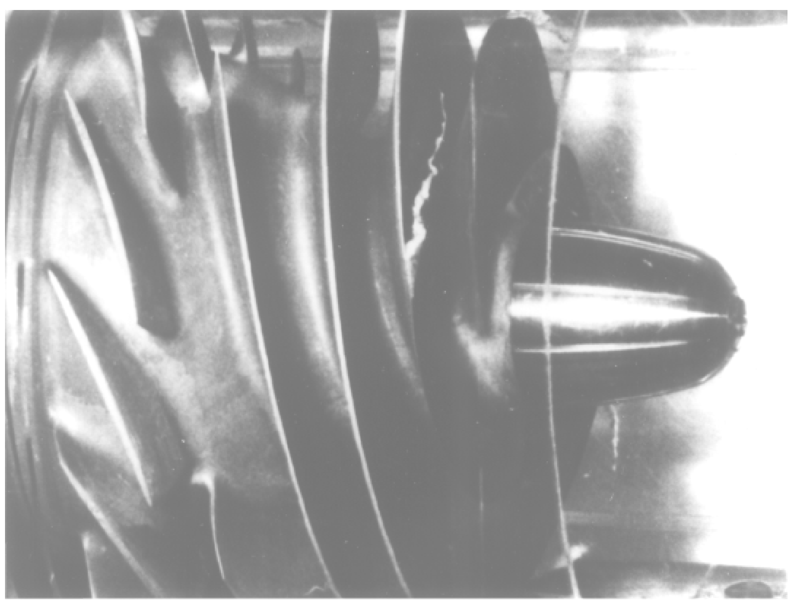

Figure 3. Classical Part Span or Tip Vortex. Brennen (1994)

Modern investigation has shown that sheet cavitation is, of course, common for certain pump configurations, but not for others. A careful examination of the references given herein will show cases of sheet cavitation, but usually when the leading edge is a straight line that is quasi-orthogonal to the actual streamlines passing through the passage. In other cases, particularly when the leading edge is swept back, sheet cavitation develops late in the process, and sometimes perhaps not at all. Instead, the actual flow mechanism is that of a rolled-up vortex near the leading edge which, in fact, can be calculated with common CFD methods, see Figures 3 and 4. Thus, the cavitation mechanism for many of the highly swept leading edge inducer designs is fundamentally different from the assumed pattern accompanying the above two models.

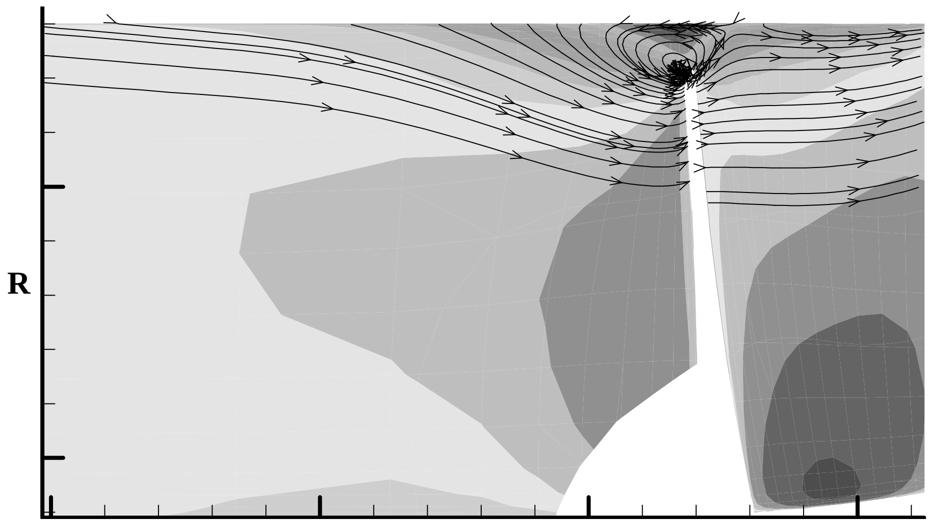

Figure 4. Detailed CFD Showing Inlet Part Span or Tip Vortex.

The third approach* taken is comparatively simple and is based on an old industrial design criterion. This criterion was introduced in the reference by Dixon (1978) and expanded in a subsequent reference by Japikse (1997). Dixon quotes the industrial inlet blade design method (for cavitation consideration), which gives:

$$p_{1t} = p_{\upsilon} + \sigma_b\left(\frac{1}{2}\rho W_1^2\right) \text{ (this relationship defines the blade cavitation coefficient, } \sigma_b\text{)} \quad (2)$$

$$N_{ss}^2 \Big/ \left(1 - \frac{r_{h1}^2}{r_{t1}^2}\right) = 3.420 \Big/ \left(\sigma_b\left(1+\sigma_b\right)^{0.5}\right) \quad (3)$$

And this concept was extended by Japikse to provide a general specification of r_{1t} for an inducer:

$$r_{1t}^2 = r_{1h}^2 + \left(\frac{2\,AK(1+\sigma_b)}{\sigma_b}\right)^{1/3}\left(\frac{Q\sin\phi 60}{\pi^2 2N}\right)^{2/3} \quad (4)$$

where $AK = C_{m1t}/C_{m1m}$ and ϕ is equal to 90° - the local streamline inclination angle. Clearly, r_{1t} becomes quite large as σ_b becomes small (for typical σ_b values, see Figure 5 legend).

When the σ_b relationships are utilized with different values of blade cavitation coefficient, then trends much like the previous figures are obtained as shown in Figure 5. For the blade cavitation coefficient of 0.003 it is possible to achieve 85,000 N_{ss} and for a blade cavitation coefficient of 0.0053, one can achieve 65,000 N_{ss}. The

*Note: This method is implemented in the design code PUMPAL® which is available from Concepts NREC.

theory suggests that it might even be possible to reach much higher suction specific speeds if it is technically possible to realize the very low blade cavitation coefficients. Of course, there is little evidence as to how far one can go on blade cavitation coefficient and the actual values of achievable σ_b must truly be dependent upon the art of leading edge design and other aspects of the inducer layout technique. Using Figure 5, one may select ϕ = 0.04-0.05 at N_{ss} = 85,000 and ϕ = 0.05-0.06 for N_{ss} = 65,000 and, allowing 3.0° to 3.5° for incidence, then one gets β_{1bt} = 5.3° to 6.2° for N_{ss} = 85,000 and β_{1bt} = 5.9° to 6.9° for N_{ss} = 65,000. These numbers fall close to the Stripling/Brumfield relationship. Of course, for any of the examples shown in Figures 1, 2, or 5, it is a straightforward exercise to compute from the flow coefficient to an inlet eye radius (essentially Equation 4, above). The information in Figures 1, 2, or 5 can be used to assemble reasonable expectations for achieving the inlet eye diameter, the inlet blade angle, and the inlet flow angle. These different theories, which evidently derive from different points of view, suggest some commonality in the basic criteria and indicate that there must be a fundamental inlet inducer eye diameter and a blade or flow angle which are necessary, but not sufficient, conditions to achieve desired inducer performance.

It should be noted that there is no specific physical model (sheet cavitation, rolled-up leading edge vortex, tip vortex, etc.) accompanying this equation set, as was the case for the previous two examples.

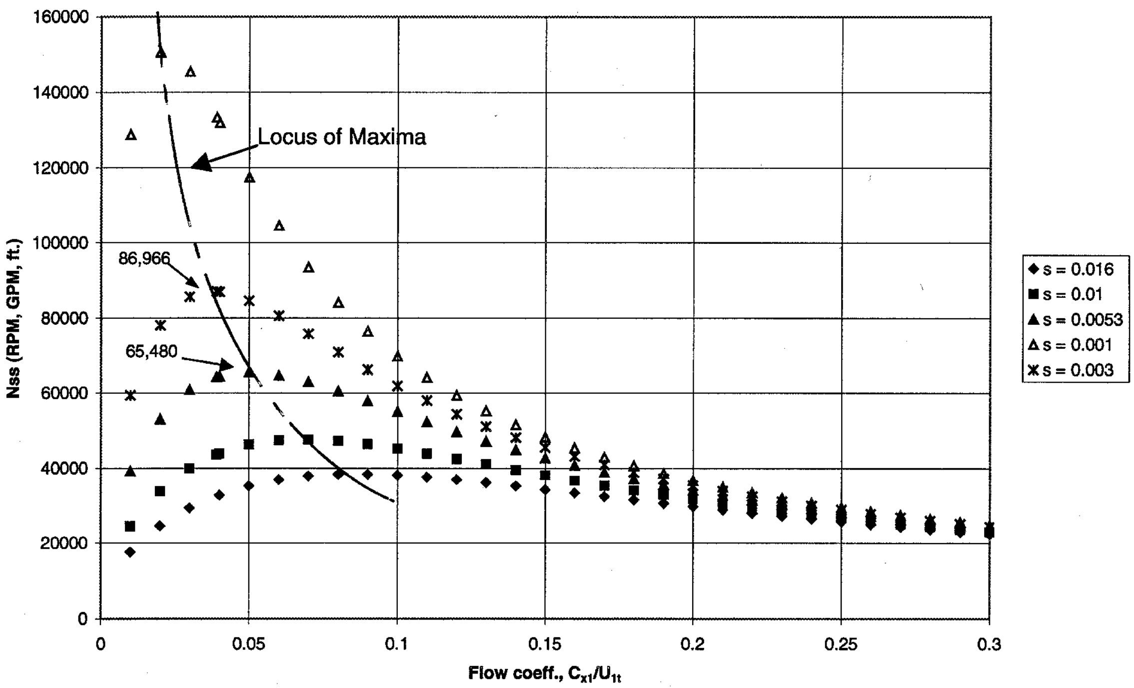

Figure 5. Evaluation of suction specific speed versus flow coefficient at different levels of inlet blade cavitation coefficient. The legend gives $\sigma_b = s$ at typical levels.

The reader must also be aware of the resulting performance trade-off. As N_{ss} is increased (Equation 3), r_{1t} must be increased (Equation 4) and, hence, U_{1t} increases rapidly. With U_{1t}, W_{1t} rises quickly, and at inflated levels of W_{1t}, the performance will deteriorate quickly. Assuming the same (constant) diffusion levels, a change in N_{ss} from 30,000 to 85,000 may be expected to drop the efficiency by up to six points.

Finding the appropriate value of σ_b is solely an empirical matter, having no analytical foundation. The value of σ_b that is used for a given design is based on inlet flow conditions, including inlet blade sweep angle and the detailed conditions of blade leading edge shape at the inlet. There is no universal table of such values available to guide a designer. However, some values are summarized in Japikse et al. (*op. cit.*), pp. 7-27 to 7-29. When conventional design practice is utilized, as detailed herein, the values of σ_b in Figure 3 may be applied at the appropriate level of suction specific speed. It is assumed, however, that diligent application of all of the design rules in this paper are used.

The data provided in Figures 1 and 2 are sufficient to determine incidence according to the curves represented in those figures. This has been computed and displayed in Figure 6. The dark lines correspond to the Stripling evaluation; the dotted lines correspond to the Furst and Desclaux correlation. The blade angle is shown as the upper two curves with the flow angles below them. The incidence is shown for Stripling as triangle symbols, and for the Furst and Desclaux case with solid squares. There is a difference in design point incidence. At high suction specific speed the difference becomes small. Nonetheless, it is clear that the inducer design incidence varies from moderately large numbers such as seven or ten degrees at low suction specific speed up to modest numbers such as two to four degrees at high suction specific speed. The variation over this range is suggested to be continuous. In the subsequent section, an alternative view is gleaned from historical design experience.

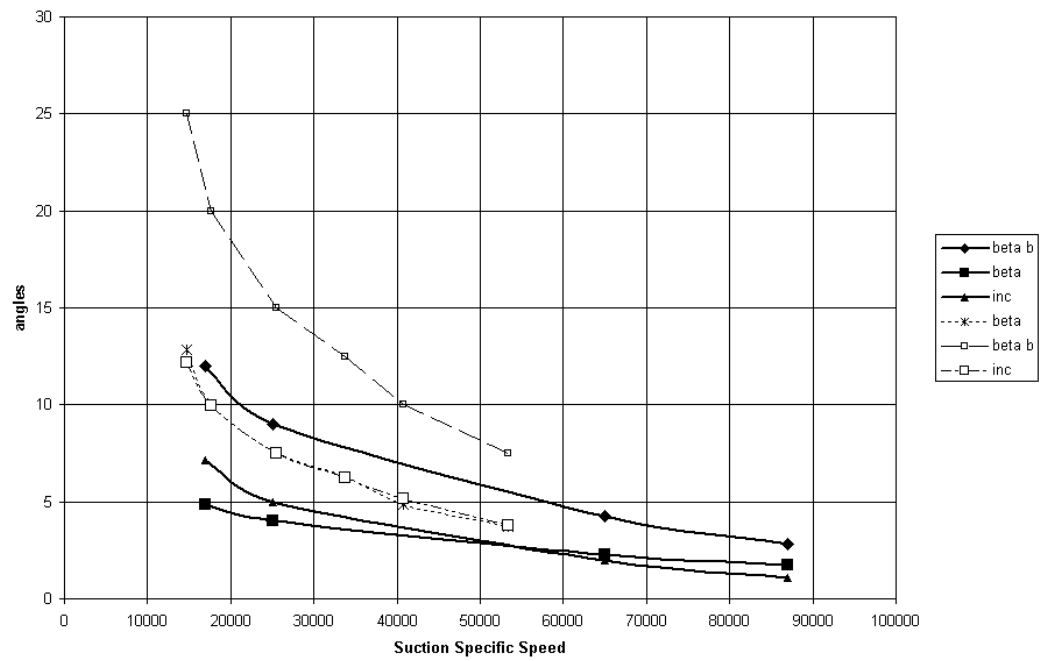

Figure 6. Design values of incidence with corresponding blade and flow angles according to the Stripling (solid lines above, refer to Figure 1) and Furst and Desclaux (dashed lines above, refer to Figure 2) inducer design models.

The fourth approach to the inducer design optimization problem is reflected in past experience. A nominal amount of past experience has been assembled in Figure 7, which shows the design blade angle for a series of different inducer pumps, the expected incidence for the impellers, and the nominal flow angle (simply the arc tan of the flow coefficient).

With the important issue of inlet inducer tip radius, incidence and blade angle having been resolved, or at least sensibly focused, other issues still remain for inducer optimization, but they appear to be somewhat secondary by comparison. Blade number is next on the list. Inducers have been manufactured with anywhere from one to four blades (occasionally more) in the inlet section. However, a count of two, three, or four blades is most prevalent. In this area, conventional experience or conventional wisdom has dominated current design practice. Although it is possible to have stability problems with *any* blade count due to cavitation occurring on one blade, but not on adjacent blades (and, hence, a rotordynamic unbalance), it has been found from practical industrial experience that a three-bladed inducer tends to have fewer dynamic stability problems than either two- or four-bladed inducers (where alternating cavitation is more prevalent). The three-bladed inducers tend to dominate industrial and aerospace design today.

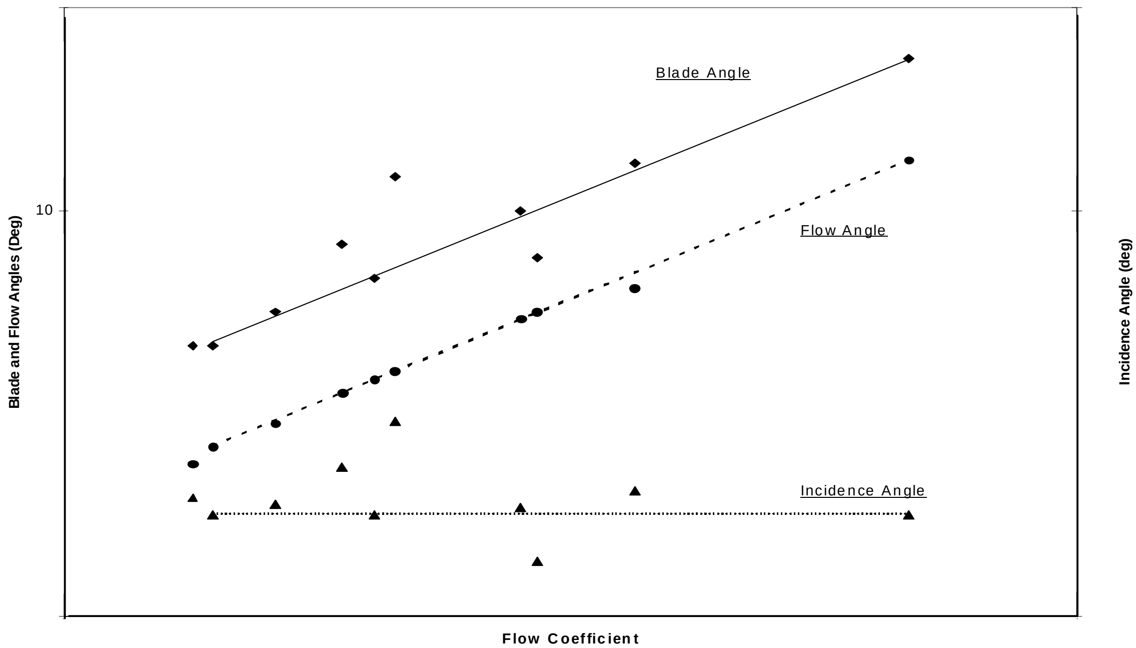

Figure 7. An historical study of incidence, blade angle, and flow angle for a set of ten different pump inducers.

The details of leading edge shape are also important, but the technical literature has not revealed any confirmed quantitative procedure. Design practice of the last several decades has evolved so that a swept-back leading edge shape is highly preferred today for controlling the location of the sheet of cavitation bubbles, which occur in the inducer and may be eventually eliminated by pressure rise. This leading edge shape provides both structural and fluid dynamic relief. The structural relief comes about by eliminating metal in the inducer tip region (hence, stiffening the blade) and the fluid dynamic relief comes by reducing the velocity component orthogonal to the vane leading edge. There may also be important effects (see subsequent discussion) concerning the development of a leading edge tip vortex, see Figure 3. The basic vortex structure can be calculated via CFD and leading edge studies can be made to reduce the vortex strength; Figure 4 gives an example. Common practice is to allow a nominal degree of leading edge sweep back so that the leading edge shape can be faired into the shroud line. An additional facet of this leading edge approach is worthy of comment. By creating the meridional shape of the impeller leading edge by a lathe turning operation, prior to milling the blade shapes (a ‘lathe-mill’ or L-M leading edge), it is possible to obtain a meaningful reference blade leading edge configuration. This technique is also far more economical to produce from a manufacturing point of view. The resulting leading edge shape is that of a sharp wedge, which would be much more difficult to manufacture by any other machining technique. However, the wedge angle of the blade leading edge is generally greater than some published standards. Nonetheless, it is a good starting shape (very economical) for systematic evaluation with subsequent leading edge reshaping to be considered for a specific application.

Information was obtained in the technical literature concerning the leading edge wedge shape. Furst (1997) suggests a wedge angle equal to 1/2 or 2/3 of the inlet blade angle. Hence, values of approximately 3° wedge angle are recommended for higher N_{ss} designs. As indicated in the preceding paragraph, this tight angle does not naturally occur when using the simple leading edge fabrication procedure (turning and subsequent milling), which gives a leading edge wedge angle equal to the blade angle).

A further design consideration involves the amount of flow angle turning from the leading edge up to the throat region of the inducer. Concrete values have not been widely discussed in the literature, but guidance can be taken from a similar problem. The fundamental issue is minimizing the diffusion loading on the boundary layers along the blade surface. The exact same problem is encountered for transonic compressors between the vane leading edge and the throat. The problems are identical in the sense that one must not add additional sources of loading (i.e., excessive turning) to these fragile boundary layers. They already have enough distress due to local diffusion. Limits for transonic compressor design are clearly known: one should minimize the turning from the blade leading edge to the throat to less than one or two degrees. Rarely have higher numbers been allowable; the practice is generally to keep the turning to one degree or less. It is considered probable that this angle could be important for centering the cavitation sheet in the middle of an inducer passage, so that the bubble sheet does not collapse directly onto an inducer blade surface, hence assuring reasonable inducer life. An additional criterion is the overall wrap angle for the complete inducer. Again, information from the technical literature cannot be obtained. This parameter does, after all, control the amount of loading on the individual vane surfaces. In recent design experience of the author, the amount of turning has been reduced and the surface loading has been increased without any evidence of deleterious effects.

A first order design issue is recognized in the turbopump industry concerning whether an inducer should be designed as a separate element, which then precedes a secondary pump impeller that develops most of the desired stage head, or whether a single piece inducer and impeller ought to be designed. Throughout the past half century each method has been used, although the separate inducer has, by far, been the most common. The author's design experience covers both areas. The history of the design activity shows that the same cavitation performance can be achieved by either approach. A separate inducer is not necessary in order to achieve high suction performance. On the other hand, issues of clearance flow and thrust balancing have not yet been resolved. For lower N_{ss} designs, the single piece configuration is surely a safe one. More investigation under the very high head conditions is necessary.

Structural design is also of considerable significance. The technical literature broadly indicates that stresses become very high for high specific speed inducer designs. Additionally, and perhaps related to the structural considerations, is the observation that hub separation will exist in all of these inducers when the design value of N_{ss} is pushed to moderately high levels. These observations show up in a number of previous technical papers. Indeed, stress limitations may be the eventual limiting factor for the high level of suction specific speed that might be desired for a given application. Careful structural calculations must be conducted for any basic design process.

The specific references mentioned above form an explicit component of the design technology employed in current inducer-pump design activity. However, many other references are available for either background information or specific examples. Some of the principal documents that were particularly helpful are listed as follows. The work by Engeda and Rautenberg (1989) provided relevant background data. Likewise, the paper by Bario, et al., (1991) entitled *Air Test Flow Analysis of the Hydrogen Pump of Vulcain Rocket Engine,* also provides some background insight. This particular inducer (Bario) is characterized by an s-wall hub contour. The consideration of rotor dynamic forces by Franz et al., (1989) may be particularly useful. The issue of impeller backflow influencing stage performance is developed further by Tanaka (1980) in his study, *An Experimental Study of Backflow Phenomena in a High Specific Speed Propeller Pump*. The surveys by Brennen (1978) (*The Unsteady, Dynamic Characterization of Hydraulic Systems with Emphasis on Cavitation and Turbomachines*) and also by Acosta (1992), *Flow and Inducer Pumps, an Aperçu* were helpful for general orientation. Additional data on cavitation-induced oscillations are provided by Natanzon et. Al., (1974). Further information on cavitating inducer instabilities was provided in 1977 by Kamijo, et. al., (1977). Of course, the visualization study of flow and axial flow inducers by Lakshminarayana (1972) is quite helpful. The work by Howard, Almahroos and Roeber (1987), using a laser velocimeter for studying an axial pump inducer at off-design performance, is helpful in that conditions of reverse flow from the impeller have been identified and specifically quantified by laser velocimeter measurements. The survey by Lakshminarayana (1982), entitled *Fluid Dynamics of Inducers – A Review*, is well known by all in the field. It provides a lot of technical detail and much general overview of component performance. The more recent study of alternate blade cavitation by Huang et. al., (1998) also provides good guideline information. An extensive tome of information on cavitating axial inducers was provided by Carpenter (1957) as a thesis. It contains a lot of basic information for the so-called simple helical flat plate type inducers. Kamijo and Yamada (1998) provide further information concerning vibrations induced by backflow activity. Other data concerning dynamic response of liquid oxygen pumps were provided by Shimura and Kamijo (1983) detailing some of the general vibration levels observed. A more detailed analysis of system response, basically a surge type analysis, is provided by Sach and Nottage (1965). A further step in the analysis by Furst and Desclaux (referenced above) was given by Kueny and Desclaux *Theoretical Analysis of Cavitation in Rocket Engine Inducers* (1989).

This paper offers a few more insights concerning possible sheet cavitation formation and parameters that might influence this phenomenon. Some insight to Rocketdyne practice of the past was provided by Furst and Bache (1985) in their paper on the high pressure oxygen pump for the space shuttle. A very good overview with substantial data was developed by Brophy (1975) concerning cavitating inducers as applied to water jet propulsion. They specifically have data concerning loss coefficients for inducers that is worthy of note.

The book by Brennen (1994), *Hydrodynamics of Pumps*, and the book by Japikse, Marscher, and Furst (1997), *Centrifugal Pump Design and Performance*, each contain a wide variety of examples of pump performance data. Additionally, NASA SP8052 *Liquid Rocket Engine Turbopump Inducers*, dated May 1971 and compiled by Furst (1973), and others provides general background information and a wide range of references.

The references just cited provide useful background context for the work conducted. However, there are defects in the historical database, which certainly cloud all present work. First, there is a very academic orientation to these references where problems are observed, and then studied with considerable restriction in a very narrow context. Frequently, the investigators make little effort to eliminate the problem by rational engineering re-design, in deference to studying its mathematical characteristics. Additionally, many (but certainly not all) of the studies are not based on realistic turbopump designs that are intended for a specific engineering application. Frequently simplified laboratory inducers are used that provide interesting background information but may not apply specifically to a meaningful engineering application. Finally, most of the work is context sensitive, meaning that the information may very well be accurate in the specific context in which it was studied, but may not be suitable for broad generalization (e.g., helical or so called flat plate inducers versus 3D inducer pumps). Considerable problems in the latter area are identified in this study.

3. TEST PERSPECTIVES

The theories and models presented in the preceding section of this review article are based, to a large degree, on test results. Consequently, a few remarks concerning the test area are offered herein. Accurate turbomachinery testing is difficult under most all circumstances; when dealing with a cavitating inducer, they are even more complex and demanding. These notes are not intended to give a comprehensive overview, but simply to record a few observations.

Figure 8 displays a typical head breakdown characteristic under the influence of cavitation. It comprises data for a specific test stage under a variety of different case variations. Following classical perception, the head remains constant for a given flow and speed or operating condition until substantial cavitation develops within the stage. Industrial clients frequently limit the acceptable operating regime to a 3% head breakdown, which occurs just after the knee in the curve of Figure 8. For rocket turbomachinery, where the inducer is followed by a subsequent impeller, or is part of a moderate or high head stage, a greater breakdown is generally permissible. For the inducer tested for rocket turbomachinery alone, one usually looks for at least 10% head breakdown and it has not been uncommon to consider a 50% head breakdown for an inducer when it is known to be followed by a complete stage. The test data displayed correspond to a condition approaching the 50% head breakdown. Once the head breakdown starts, it proceeds very rapidly so some of the distinction is academic. Considerable data have been taken for the Figure 8 case just prior to head breakdown. A range of auto-oscillation has been noted. This is very common for inducer pumps. In this regime, a chugging flow condition or oscillation exists where cavitating flow from one passage is interacting with cavitating flow on the adjacent passage and a significant oscillation is triggered (auto oscillation). It is clear that this is influenced by various secondary geometric parameters which have been varied from Case A through Case D. It is hoped in the future that more will be learned about the actual conditions of auto-oscillation and how it can be mitigated by simple design variations. Preparing a Figure 6 type characteristic is not too difficult once an appropriate test facility is set up. However, these facilities are not cheap; they are not trivial to design and they are comparatively few in number. Anyone wishing to work in the field must take considerable pains to develop a very good test facility with considerable flexibility in order to have the freedom to carry out the appropriate tests.

The objective of studying the cavitation breakdown plus auto-oscillation characteristics shown in Figure 8 is, of course, to understand the level of suction head that is required at the eye of the impeller. When this head is converted to a suction specific speed, as shown on the ordinate of Figure 9, then the design issue of low inlet suction head is related to the other characteristics of design including speed and flow rate. It is a goal of modern industrial pump design, in certain niche market areas, plus the turbopump industry, to reach ever-increasing values of suction specific speed. This has been displayed in Figure 8 in its classical form. Presently, the author and his colleagues are involved in a multi-year long-term investigation on the characteristics of a large family of inducer pumps from moderate to very high suction specific speed characteristics. The trace shown in Figure 9 is simply an example of the behavioral characteristics of one member of this family under a variety of different related geometric conditions.

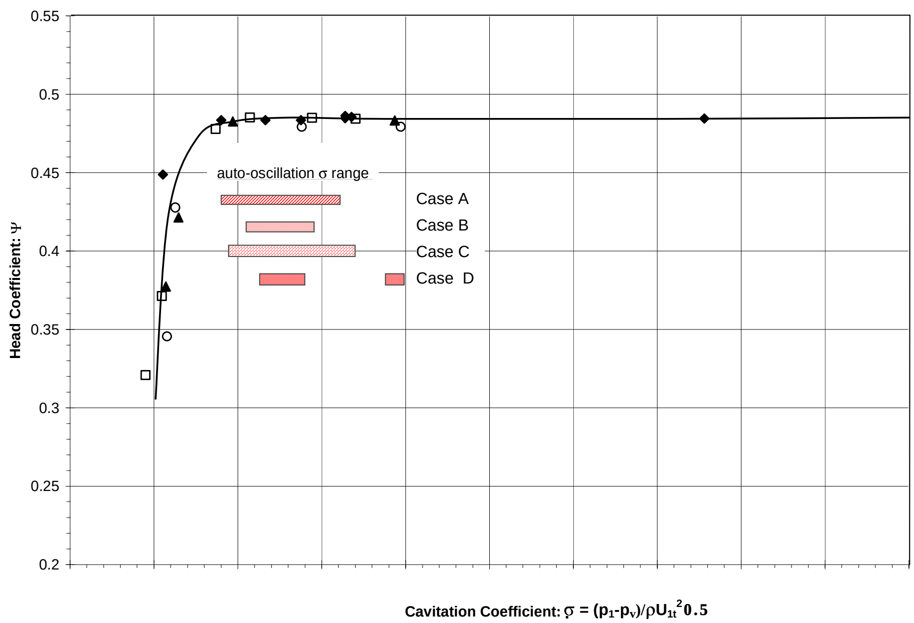

Figure 8. ψ versus σ for various conditions. (Note: p_1 is the inlet pressure, p_v is the fluid vapor pressure, and U_{1t} is the inlet tip wheel speed.)

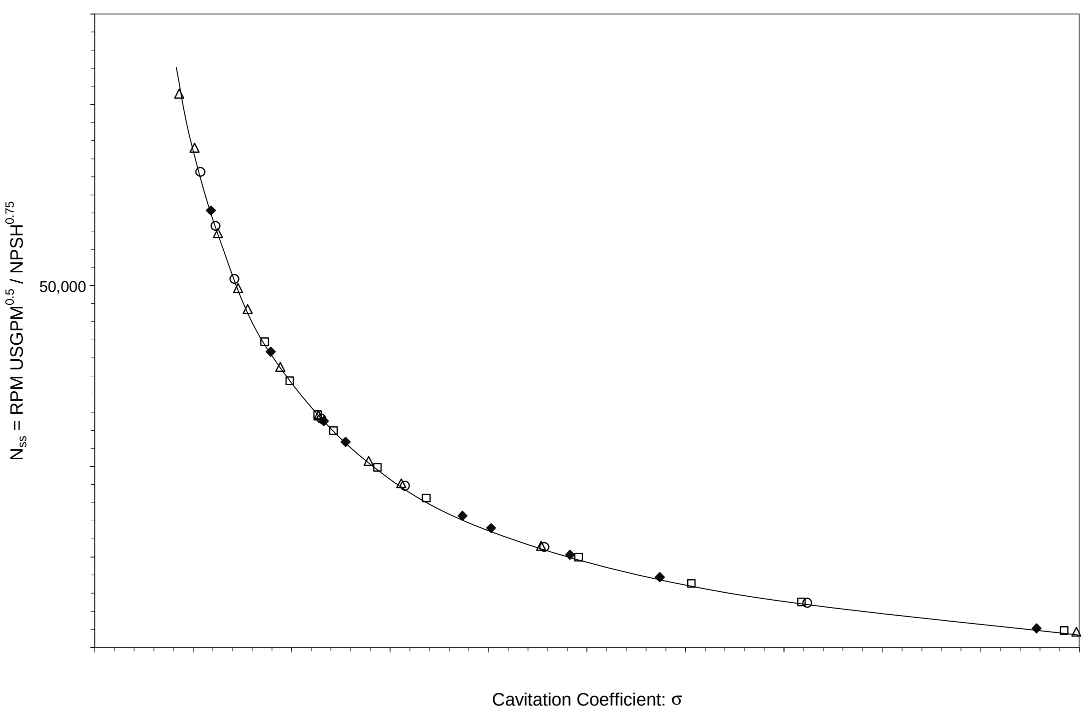

Figure 9. N_{ss} versus σ comparison for an inducer pump with various inlet designs at 100°F (H_2O test).

The end objective of the laboratory evaluation is to confirm that a given pump can operate stably under very low inlet suction conditions as represented in Figures 8 and 9. As indicated above, it is extremely difficult to set up all of the necessary control conditions to have truly excellent data, but once such a facility is available, then it is not too difficult to obtain useful data, as revealed in these figures. Flow visualization has proven to be extremely important. In the author's opinion, no advanced inducer development work should be conducted without direct flow visualization. The ability to look directly into the flow field and observe the characteristic of the cavitating flow is extremely important.

4. SUMMARY

This survey has provided a brief overview of design considerations for modern industrial rocket turbopump inducer design. When the guidelines given herein are employed, the author's experience shows that very good inducers can be designed. To be sure, there are many questions which will need further evaluation in future studies. CFD, although not detailed herein, has shown considerable capability in identifying a number of the single-phase flow phenomena that are important for good inducer design. CFD should always be employed in any modern design problem.

Future research is under way into a variety of inducer related issues. The specific details of the leading edge shape, including both meridional profile and actual leading edge wedge shape are under scrutiny at this time. Various inlet design modifications which might impact the inducer stability are also under investigation. Additionally, investigations are beginning to study the dynamic forces associated with the operation of various rocket turbopump inducers.

REFERENCES

Acosta, A.J. "Flow and Inducer Pumps, An Aperçu," Proc. 4th International Symposium on Transport Phenomena and Dynamics of Rotating Machinery, Honolulu, Vol. A., pp. 1-13, 1992.

Bario, F., Barral, L., and Bois, G., "Air Test Flow Analysis of the Hydrogen Pump of Vulcain Rocket Engine," *Journal Of Fluids Engineering*, Dec. 1991, pp. 654-659.

Brennen, C. E., *Hydrodynamics of Pumps*, Concepts ETI, Inc. and Oxford University Press, Wilder, VT, 1994.

Brennen, C., "The Unsteady, Dynamic Characterization of Hydraulic Systems with Emphasis on Cavitation and Turbomachines," Symposium on Design and Operation of Fluid Machinery, IAHR, ASME, ASCE, 1978, pp. 97-107.

Brophy, M. C., "A Theoretical Study of Cavitating Inducers with an Application to Water-Jet Propulsion," *Naval Ship Research and Development Center,* Bethesda, MD., February 1975.

Carpenter, S.H., "Performance of Cavitating Axial Inducers with Varying Tip Clearance and Solidity," 1957.

Dixon, S. L., *Fluid Mechanics, Thermodynamics of Turbomachinery, 3rd Edition,* Pergamon Press, England 1978.

Engeda, A., and Rautenberg, "Investigation of Semi-Open Impellers," Pumping Machinery - 1989, ASME FED-Vol. 81, 1989, pp. 55-62.

Franz, R., Acosta, A.J., Brennen, C.E. and Caughey, T.K., "The Rotordynamic Forces on a Centrifugal Pump Impeller in the Presence of Cavitation," Proc. ASME Symp. Pumping Machinery, FED-81, 205–212, 1989.

Furst, R., "Liquid Rocket Engine Centrifugal Flow Turbopumps. NASA SP-8109, 1973.

Furst, R., and Bache, G., "The Development of Inducers for the Space Shuttle High-Pressure Oxygen Pump," presented at the ASME Winter Annual Meeting, Miami, 1985.

Furst, R., and Desclaux, J., "A Simple Procedure for Prediction of NPSH Required by Inducers," Pumping Machinery 1989, ASME FED-81, ASME, New York.

Howard, J.H.G., Almahroos, H.M.H., and Roeber, T. W., "LDA-Measurements of the Velocity Field Within and Ahead or an Axial Pump Inducer at Off-Design Flow Rate," *ASME/JSME Thermal Engineering Joint Conference*—Vol. 2. 63-69, 1987.

Huang, J., Aoki, M. and Zhang, J., “Alternate Blade Cavitation on Inducer, *JSME International Journal* Series B 41, No. 1, 1998.

Japikse, D., et al., *Centrifugal Pump Design and Performance*, Concepts ETI, Inc., pp. 8-13, 1997.

Japikse, D., Marscher, W. D., and Furst, R., *Centrifugal Pump Design and Performance*, Concepts ETI, Inc., Wilder, VT, 1997.

Kamijo, K., Shimura, T., and Watanabe, M., “An Experimental Investigation of Cavitating Inducer Instability.” *ASME Paper 77-WA/FW-14*, 1977.

Kamijo, K., Yamada Hitoshi, Y., “Rotor Vibrations of Turbopump Due to Cavitating Flows in Inducer,” *Proceedings of US-Japan Seminar: Abnormal Flow Phenomena in Turbomachines*, , Osaka, Japan, November 1-6, 1998.

Kueny, J. L., and Desclaux, J., “Theoretical Analysis of Cavitation in Rocket Engine Inducers,” *Pumping Machinery* ASME FED-Vol. 81, pp. 11-18, 1989.

Lakshminarayana, B., “Fluid Dynamics of Inducers—A Review,” J. Fluid Mech., 104, 411-427, 1982.

Lakshminarayana, B., “Visualization Study of Flow in Axial Flow Inducer.” *ASME J. Basic Eng.*, 94, 777–787, 1972.

Natanzon, M.S., Bl'tsev, N.E., Bazhanov, V.V., and Leydervarger, M.R., “Experimental Investi-gation of Cavitation-Induced Oscillations of Helical Inducers,” *Fluid Mech., Soviet Res.*, 3, No.1, 38–45, 1974.

Sack, L.E. and Nottage, H.B., “System Oscillations Associated with Cavitating Inducers.” *ASME J. Basic Eng.*, 87, 917–924, 1965.

Shimura, T., and Kamijo, K., “Dynamic Response of a Centrifugal Liquid Oxygen Rocket Pump,” *ASME Paper* No. 83-FE-24, 1983.

Stripling, L. B., and Acosta, A. J., “Cavitation in Turbopumps – Part I,” *ASME J. Basic Eng*., **84**, 326-338,1962.

Tanaka, T., “An Experimental Study of Backflow Phenomena in a High Specific Speed Propeller Pump,” *ASME Fluids Eng*, 80-FE-6, 1980.