

AND WHAT OF THE FUTURE?

by

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Mr. Karassik has written numerous articles on centrifugal pumps and steam power plants for technical publications. He is also the author of the books Centrifugal Pumps—Selection, Operation and Maintenance, Engineers Guide to Centrifugal Pumps and Centrifugal Pump Clinic, co-author of Pump Questions and Answers, and co-editor of The Pump Handbook. Mr. Karassik received the B.S. and M.S. degrees from Carnegie Institute of Technology. He is a Life Fellow of ASME and is a Professional Engineer in the State of New Jersey. He is also a member of Tau Beta Pi, Pi Tau Sigma, and Sigma Xi.

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.

INTRODUCTION

Two years ago I delivered the Welcoming Address to the 5th International Pump Users Symposium. It would be unrealistic for us to expect that in two short years such progress would have taken place in the direction I had suggested that our task would be almost done. It is true that in a number of laboratories experiments in flow visualization have developed new insights into the problem of cavitation damage to impellers. But these insights have provided qualitative rather than quantitative results, because more time is still needed to obtain solid evidence for the latter.

Likewise, it is too early to expect that the Hydraulic Institute would have published charts of commercially attainable efficiencies or guidelines for the setting of realistic and safe minimum flows.

But I can single out one area where major progress is being made and that is the education of pump users. This is what the Pump Symposium you are attending is all about.

I came back today for two purposes. First, I want to repeat and reexamine the list of questions to which we still have no definitive answer. These questions deal mostly with the behavior of centrifugal pumps from the point of view of their hydraulic performance. But I also decided to take the bit in my teeth and

venture some predictions of what the centrifugal pump of the future will look like and to suggest to you why it will be so.

In a sense, one can say that the subject "And What of the Future?" truly expresses the theme of this, our 7th International Pump Users Symposium, if not even of all these Symposiums. For at these meetings, in the lectures, tutorials, and short courses, an opportunity is presented to all pump users to examine and evaluate this future and to prepare to participate in it most effectively. The attendees can even see some of this future, since many of the exhibits which they can visit include the first appearance of certain innovations.

There is still one more reason why the subject "And What of the Future?" is most appropriate in this year 1990. This date marks the 150th anniversary of the development of the direct acting steam pump by Henry R. Worthington, an invention which laid the foundation of the entire pump industry. Maybe this anniversary will serve to unleash an effort to take a quantum leap in our attempt to understand what remains to be learned and to be applied in practice.

WHAT REMAINS TO BE LEARNED

As I said two years ago, much remains to be learned about the hydraulic behavior of centrifugal pumps. This is particularly true about this behavior at flows below that at the best efficiency point. For instance:

- We still do not know how to calculate the ratio of the required NPSH at 0 percent drop in head and at the incipient cavitation to the value of NPSH at three percent drop in head. We know superficially that it is affected by a long list of factors, such as:
 - the thermodynamic characteristics of the liquid pumped, and particularly the ratio of the specific volumes of the vapor and of the liquid at the pumping temperature along with the effect of the subcooling derived from the partial flashing of the liquid.
 - the homogeneity of the liquid.
 - the presence of dissolved and entrained gases and their volume by percentage.
 - the specific speed of the pump.
 - the suction specific speed of the pump.
 - and, probably, the exact configuration of the impeller vanes at the inlet.

But we do not know how to distinguish between the individual effects of each one of these factors.

- Most papers devoted to the subject of cavitation indicate that the general shape of the curves of NPSH required at 0 percent drop in head and at the conditions of incipient cavitation depart significantly from the typical shape of the curve of NPSH at three percent drop in head. The ratio between the values for the first two does not remain constant over the entire range of pump flows and increases significantly and abruptly at some partial flow condition. The abrupt increase is very pronounced in the case of incipient cavitation and actually forms a spike at this partial flow, the value of which may even exceed the NPSH value at best efficiency. I have a nagging suspicion that when we examine these curves, we are really seeing a combination of

NPSH required and the effect of internal recirculation at the suction. But, we have not yet determined how we can differentiate between these two separate effects.

- Two years ago I mentioned that the user is not overly concerned about theoretical considerations that determine the required NPSH; all he wishes to know is a reasonable prediction about the life of an impeller operating, as it does so frequently under cavitating conditions. I mentioned that an attempt to provide such a prediction had been made in a paper by J. H. Doolin [1]. The author presented a list of the factors which affect impeller life:

- excess of available NPSH over the required NPSH
- effect of the thermodynamic properties of the liquid
- corrosiveness of the liquid pumped
- effect of different materials
- effect of operating speed
- operation at off-best-efficiency flows
- effect of suction specific speed
- pump duty cycle

The author provided curves and tables for multipliers for each one of these factors. The product of the eight multipliers would then give an overall multiplier which could apply to a pre-selected "standard" life for the impeller.

The problem is that we have not yet agreed what this "standard" life should be. Nor do we have sufficient data on hand to assign multiplying coefficients that would accurately reflect the effect of each one of these eight factors. It remains for us to produce such data.

- After W. H. Fraser released his seminal paper on internal recirculation in 1981 (which for the first time provided a means to calculate the onset of internal recirculation at the suction and at the discharge), several other engineers published their own theories and their own conclusions - each of them differing from Fraser's approach and from each other. I suggest that we do not delay putting all these methods of prediction to the test. Several impellers could be selected and tested under identical conditions. One of these methods will probably prove to be more accurate than all the others. And yet, it is possible that the differences will be so slight that no distinction need be made and that the choice of the preferred method can be left to anyone's personal preferences. But the tests *must* be carried out.

I am certain that I have not exhausted the list of what remains to be learned about the hydraulic behavior of centrifugal pumps. But this is already a formidable list and we better get on with the task of reducing it.

THE CENTRIFUGAL PUMP OF THE FUTURE

The most vexing problem faced by the user of centrifugal pumps is not a major and catastrophic failure of his equipment—such occurrences are rare. It is, instead, the fact that the life of many pump components is not as long as he would wish. This is manifested by such circumstances as short packing life, frequent replacements of mechanical seals, bearings, wearings rings, etc. Thus, a major task of the pump industry must be an effort to constantly extend the MTBF (mean time between failures) of centrifugal pumps. Thus, if we want to imagine what the centrifugal pump of the future will look like, we need to examine in some detail the various factors which affect the MTBF unfavorably and the means whereby we can eliminate them or, at least, reduce their effect to a satisfactory level. Let us then list some of the weakest points of the centrifugal pump of today. These are:

- The life of the impeller under cavitating conditions. I have already spoken of this problem in the first part of this presentation.
- The seal area.

- The bearings.
- The wear at the internal running joints.

THE SEAL AREA

The pumping of water—whether in small or large quantities—does not present any major constructional problem, save in the case of such pump designs in which internal product lubricated bearings must be used. In these designs, the problem is created by the inability of the bearings to run dry, as they will during the starting and stopping periods of operation. But even this problem appears to be capable of solution, as described in the paper, "Dry Start Bearing for Vertical Pumps," by Satoh and Takeda, presented at the 1989 Pump Symposium.

So, it is to the pumping of toxic or inflammable liquids that we have to direct the major thrust of our efforts. Of course, there is no doubt that the commercialization of highly effective mechanical seals in the 1940s played an important role in making it possible to pump a variety of such liquids without incurring significant dangers. This, in turn, facilitated the development of many important processes which contributed to the rapid progress of our technological civilization.

But we have not achieved this without paying a constantly mounting price, in money, in effort, or in a deterioration of our environment. And it is this last kind of price which now threatens not only to interfere with further progress, but even to force us to take some steps backwards.

We must remember that a mechanical seal does not eliminate leakage. It only provides us with a considerably reduced leakage in the form of a vapor which escapes into the atmosphere surrounding our centrifugal pump installation. And while this leakage may consist of an inert hydrocarbon when a multiple seal is used, emerging governmental regulations will soon make even this small amount of leakage illegal and, therefore, unacceptable.

Thus, whether we are willing or not, a major change will have to take place in the near future. Technological processes change because something becomes impractical if we continue to do it in the old way. They seldom change just before that for strictly accidental or providential reasons.

What, then, are the possible solutions to the problem we are about to face? I see three such solutions:

- One could theoretically develop a triple seal such that the injected liquid between the last two seals would be water, the leakage of which is quite acceptable. But this is hardly a viable solution. To begin with, some of this sealing water would migrate into the circuit carrying the "inert" hydrocarbon, from which it could further migrate into the pumped fluid, and that is unacceptable. In addition, this solution would increase the pump shaft span and require a most complicated sealing circuit system. Definitely, we must look elsewhere.

- We could enclose the entire unit of pump and driver into a tight plastic dome housing, with the ambient atmosphere being constantly evacuated and passed through some sort of filter which would remove the traces of inert hydrocarbon vapor. I suspect that this approach would be found unacceptable by the user for a variety of obvious reasons.

- We will have to use hermetic pumps, either of the magnetic coupling variety or with canned motors.

I do not mean to imply that we have no hermetic pumps today. But they are still not as reliable nor as efficient (in the case of canned motor pumps) as would be necessary for them to obtain the share of the pump market that this concept merits. The main problem, as far as reliability is concerned, is that they all have to rely on bearings, wherein the fluid being pumped acts as the lubricant. As I have already mentioned, product lubricated bearings are ineffective during the startup and shutdown of the

pump, and they rapidly lose their effectiveness as wear increases the internal clearances.

This brings me to the conclusion that magnetic bearings—either the “activated” or the “passive” variety—are the most logical direction to explore. Activated bearings are provided with proximity probes to monitor the centering of the rotor; feedback from these probes constantly adjusts the magnetic field around the periphery of the bearings. It is possible that passive magnetic bearings may provide the required levitation during the startup and shutdown conditions and reduce the cost of the construction by eliminating the need of the electronic controls required by activated bearings.

This is not pie-in-the-sky that I am talking about: you can see canned motor pumps, magnetic couplings, and magnetic bearings in the exhibit area today. You could have seen them there in several of the previous Symposiums. You can also attend a tutorial which will be devoted to “sealless” pumps. I am merely saying that our efforts must be directed towards increasing the efficiency and reliability of these designs and—in the case of magnetic bearings—towards reducing their cost by at least one order of magnitude.

THE BEARINGS

At least for the immediate future, many if not the majority of centrifugal pumps will continue to be provided with external bearings, mostly of the antifriction type. It is the MTBF of these bearings that we must learn to extend. Again, I must remind you that I had spoken of this in my Welcoming Address two years ago. I refer specifically to the fact that Heinz Bloch, a member of the Advisory Committee of the Symposium has made an almost single-handed crusade to convince pump manufacturers and users alike of the necessity to develop a so-called Upgraded Medium Duty Pump, a design whose major thrust addresses itself to the prolongation of the MTBF of pump bearings. This is to be accomplished by the use of essentially impermeable bearing housing seals to eliminate lubricant contamination, reduced shaft deflections and the use of better antifriction bearings. You can see some Upgraded Medium Duty Pumps in the exhibit area, but I suspect that their greater proliferation is hampered by the fact that ANSI has not yet decided to incorporate their features as available options in their standards.

WEAR AT THE INTERNAL RUNNING JOINTS

Two years ago I spoke of the need to use hard coatings more extensively than we do now. I want to expand on this recommendation and to suggest certain directions that the pump industry should and probably will take. Hard coatings of the metal surfaces at the internal running joints need not be as expensive as they are presently. The major portion of their cost is incurred from the grinding operation which, on reflection, is completely unnecessary. After all, the deposition of the coating can be made

within a tolerance of plus or minus one thousandth of an inch. The rougher surface of the deposit, which has not been ground, is conducive to higher friction losses and, hence, to a reduction to the internal leakage past wearing rings, interstage bushings, and balancing devices. Once this is understood, the expense of hard coating these surfaces can be significantly reduced.

One can even predict that the centrifugal pump of the future may dispense with renewable wearing parts as it may be more economical to deposit the hard coating directly on either the stationary or rotating surfaces, or both. If the thickness of the deposit is such that five or ten years will elapse before it is completely worn off, it would be an acceptable life span to the user.

I would like to make two more observations:

- It has been shown that the spraying of hard coatings on the visible surfaces of the inlet of a pump impeller will prolong its life materially under cavitating conditions.
- Sprayed and ground shaft sleeves will frequently have a life of five times as long as non-hardcoated sleeves.

And in connection with hard coatings, I suggest that you visit the exhibit area where several companies specializing in this process can discuss with you the benefits of this means of prolonging the life of wearing parts.

CONCLUSIONS

I have spoken to you today of innovation and change. But I have frequently pointed out that the need for technological changes can be foreseen and forecast by extrapolating present trends and establishing “limits of practicability.” Once these limits are established, a certain quantum of time is going to be required so that technology can be developed to obviate the impossible. Ideas must be incubated; a feasibility study must be carried out. Tests may have to be run. A prototype must be built or a pilot plant operated. And, finally, production models must be put into manufacturing.

This quantum of time is a reasonably definable item, but it need not be such a short time as to require a “panic approach.” It takes time not only to make a product technically feasible, but also to make it economically feasible. This quantum of time, moreover, is subject to both compression and expansion forces. Compression forces, because technology changes more and more rapidly nowadays. Expansion forces, because of the strange workings of the scale factor. The ratio between the full scale equipment and the typical laboratory size model is growing constantly.

What I am trying to say is that there should always be enough time to develop the necessary innovations in an orderly fashion, without benefit of crash programs and the concomitant blood, sweat, and tears *if we start early enough*.

I am now 78 years old. If I am to see the results of my suggestions and of my predictions, we better start cracking now!