A VISION AND MISSION
FOR PUMP R&D OVER THE NEXT 25 YEARS

by

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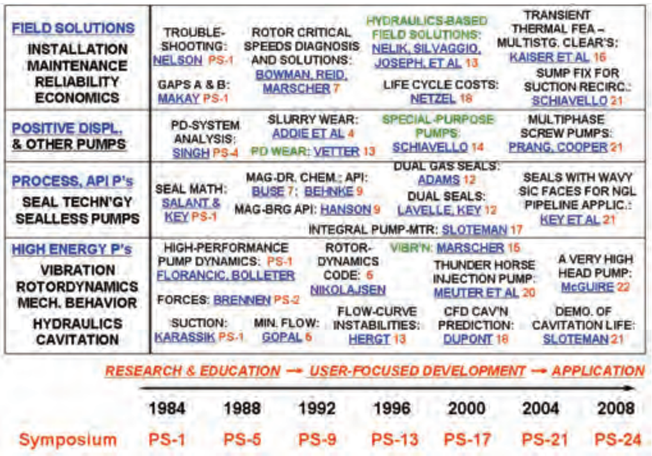
ABSTRACT

Energizing research and development on pumps in today's emerging global and environmental context is envisioned as a major feature of the pump industry in the next quarter century. The mission for how to realize this vision is proposed as a major thrust forward, the point of departure being past technical successes in improving the efficiency and reliability of both pumps and drivers. Examples are given for implementing this mission, including variable geometry, integrating pump and motor, and internal sealing that dramatically reduces internal leakage. These technologies are seen as impacting fields such as carbon capture, oil exploration, and microsize applications. In a new role for both academia and industry, high-technology research and technology (R&D) institutes are proposed for already practicing pump engineers and scientists to implement the mission and realize the vision.

INTRODUCTION

Last year, in the welcoming address for the Twenty-Fourth International Pump Users Symposium, we looked back at the 24 International Pump Users Symposia that had occurred since 1983 and put forward the notion that these symposia had driven the technology of the pump industry worldwide for 25 years (Cooper, 2008). Table 1, taken from that address, was offered as proof of that idea.

Table 1. Pump Symposium Technology Evolution. (Tutorial titles shown in green.)



FIELD SOLUTIONS INSTALLATION MAINTENANCE RELIABILITY ECONOMICS	TROUBLE-SHOOTING: NELSON PS-1 GAPS A & B: MAKAY PS-1	ROTOR CRITICAL SPEEDS DIAGNOSIS AND SOLUTIONS: BOWMAN, REID, MARSCHER 7	HYDRAULICS-BASED FIELD SOLUTIONS: NELIK, SILVAGGIO, JOSEPH, ET AL 13 LIFE CYCLE COSTS: NETZEL 18	TRANSIENT THERMAL FEA – MULTISTG. CLEAR'S: KAISER ET AL 16 SUMP FIX FOR SUCTION RECIRC.: SCHIAVELLO 21
POSITIVE DISPL. & OTHER PUMPS	PD-SYSTEM ANALYSIS: SINGH PS-4	SLURRY WEAR: ADDIE ET AL 4 PD WEAR: VETTER 13	SPECIAL-PURPOSE PUMPS: SCHIAVELLO 14	MULTIPHASE SCREW PUMPS: PRANG, COOPER 21
PROCESS, API P's SEAL TECHN'GY SEALLESS PUMPS	SEAL MATH: SALANT & KEY PS-1	MAG-DR. CHEM.; API: BUSE 7; BEHNKE 9 MAG-BRG API: HANSON 9	DUAL GAS SEALS: ADAMS 12 DUAL SEALS: LAVELLE, KEY 12 INTEGRAL PUMP-MTR: SLOTEMAN 17	SEALS WITH WAVY SIC FACES FOR NGL PIPELINE APPLIC.: KEY ET AL 21
HIGH ENERGY P's VIBRATION ROTORDYNAMICS MECH. BEHAVIOR HYDRAULICS CAVITATION	HIGH-PERFORMANCE PUMP DYNAMICS: PS-1 FLORANCIC, BOLLETER FORCES: BRENNEN PS-2 SUCTION: KARASSIK PS-1	ROTOR-DYNAMICS CODE: 6 NIKOLAJSEN MIN. FLOW: GOPAL 6	VIBR'N: MARSCHER 15 FLOW-CURVE INSTABILITIES: HERGT 13	THUNDER HORSE INJECTION PUMP: MEUTER ET AL 20 CFD CAV'N PREDICTION: DUPONT 18 A VERY HIGH HEAD PUMP: McGUIRE 22 DEMO. OF CAVITATION LIFE: SLOTEMAN 21

RESEARCH & EDUCATION → USER-FOCUSED DEVELOPMENT → APPLICATION

	1984	1988	1992	1996	2000	2004	2008
Symposium	PS-1	PS-5	PS-9	PS-13	PS-17	PS-21	PS-24

Major research and development results had been motivated and presented at these events, which we shall refer to both collectively and individually as the Pump Symposium. Here, pump design features were promoted that would enhance reliability and economy in operation. Hand-in-hand with this R&D reporting went the training that users needed—from selection, installation, and maintenance to the new understandings about flow phenomena such as recirculation and cavitation and how they affected operational reliability, durability, and economics. High-energy multistage pumps were largely demystified, mechanical seals met the new emission standards by leaking less and lasting longer, and sealless and "smart" variable-speed pumps appeared. As could be expected from the objectives of the Pump Symposium Advisory Committee, these developments eventually became more user focused. Then in the more recent years, massive global consolidation throughout the industry led to less emphasis on the R&D itself and more on the application of these learnings. In fact, R&D was cut back, which was accompanied by an attrition of the industry's technologists. A new generation of engineers was expected to implement the new design features in the course of their projects, utilizing the new methods that by this time had been commercialized, such as rotordynamics and computational fluid dynamics (CFD) codes.

From a broader perspective, the rise and fall of pump R&D appears to run in cycles that are dictated by national or global needs. At least two of these cycles occurred in the 20th century.

First was the need for a worldwide infrastructure of advanced mechanical equipment in the first half of the century. In that cycle, distinguished pump engineers and academicians contributed the necessary knowledge and designs that fueled the rise of the pump industry (Cooper, 1996). Some of the outstanding examples from that era are these:

- Den Hartog showed in a 1924 ASME paper that significant pressure pulsations can arise if the number n_b of impeller blades (or a multiple of n_b) differs by one from the number n_v of diffuser vanes (or a multiple of n_v);
- Fischer and Thoma reported in 1932 their observations of rotating stall and backflow in an impeller as they reduced the flow rate toward shutoff;
- Blom described in 1950 his design of the Grand Coulee irrigation pumps with their 13 ft diameter impellers and over 90 percent efficiency;
- Knapp reported in 1955 that cavitation damage increases with the sixth power of velocity; and
- Stahl and Stepanoff enunciated the thermodynamic effect on cavitation in 1956, demonstrating that net positive suction head required (NPSHR) is reduced when pumping fluids with more favorable vaporization characteristics.

That first cycle was obviously energized and redefined by the technological needs of the two world wars and the postwar infrastructure development, but soon after that, new developments in many fields fell behind, leaving the U.S. at a technological disadvantage until R&D was massively reenergized by the space race, which began in earnest after the flight of Sputnik in October 1957. This next cycle made tremendous demands on pump technology. The results are typified by:

- The 53,000 hp (40 MW) package that combined the oxygen and RP1 (kerosene) pumps for each of the five engines of the Saturn V moon rocket's first stage (Figure 1), and
- The high-energy 28,000 hp (21 MW) oxygen pump and 77,000 hp (57 MW) hydrogen pump on each of the three space shuttle main liquid-propellant rocket engines. Most of the space technology R&D was done by large R&D organizations, most notably NASA and their contractors, which included academic institutions. Actual hydrodynamic design of the pumps was done by engineers employed by the manufacturers; e.g., R. B. Furst did the space shuttle pumps for Rocketdyne (Karassik, et al., 2008).

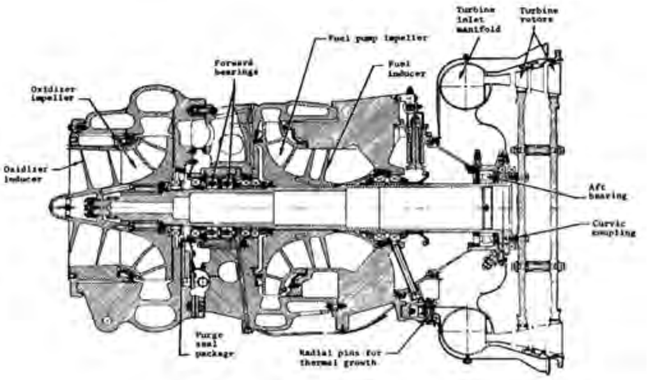


Figure 1. Turbopump on Each F1 Engine of the Saturn V Moon Rocket. Inducers with 16-inch-diameter inlets feed the oxygen pump impeller (left) and the RP1 fuel pump impeller (center), enabling these pumps to operate at a suction specific speed N_{SS} of 35,000 ($\Omega_{SS} = 13$). Rated speed is 5490 rpm. Package size is small for the rated combined shaft power of 53,000 hp (40 MW). (Courtesy of Karassik, et al., 2008, pp. 12.398-12.400)

The space-race mindset infused the whole technology landscape, especially in the U.S.A., the universities turning out more and better-trained scientists and engineers who by contributions to their own organizations and at professional meetings and conferences maintained the high technological level that gave the Pump Symposium a favorable reputation early on. However, as illustrated in Table 1, the aforementioned global consolidation led to the end of what we might call "those technology glory days" and effectively the end of this cycle.

Now it must be asked how we as an industry can move forward in the next 25 years if we stay as we are technologically. Will we really be offering the same product lines a quarter century from now? Undoubtedly a revival of R&D will be needed if we are to meet global needs then. It is probably no coincidence that the automobile industry is suffering from the lack of R&D that could make them more responsive to today's environmental demands. That industry seems also to be at the end of the same cycle. Is another cycle in view? Most of us would answer in the affirmative. Indeed there is a new elephant in the living room, and that is the emerging market for goods and services to address the environmental challenges that have been placed before us. While there is disagreement about the magnitude of this challenge or whether there is even a problem at all, it appears that global market forces are emerging that will demand appropriate R&D efforts to meet this "green" challenge and the accompanying necessity of becoming less dependent on external sources of supply.

If we of the pump industry accept the fact that we must indeed meet this challenge with the appropriate technical developments, we will respond as all responsible organizations do these days; namely, we must:

- Set a *vision* to strive for, and
- Energize a *mission* for ultimately achieving that vision.

This task can be facilitated through the guidance of examples, some of which have already been alluded to in this introduction. Then we can look at some specific developments that can be expected to command the attention of the pump R&D community as it strives to implement that mission.

VISION

Elements

The process of enunciating what we want the pump industry to be like 25 years hence leads to the vision we wish to realize. Words are needed that we can live up to over that period of time. We could begin with a "clean sheet of paper approach" to pumping. To do this requires an exercise in imagination and idealism, as might happen in a brainstorming session. Elements to consider as parts of our vision might then be these:

1. The U.S. pump industry is a player on the global scene, contributing in a spirit of professional leadership, cooperation, and perhaps partnership with others in the industry worldwide.

2. The global pump industry supplies products and auxiliary equipment that enable infrastructure systems at all levels of society to function efficiently.

3. Pump products and related auxiliary equipment will themselves operate near 100 percent efficiency at all load levels.

4. Installation and startup of pumps, their drivers, and related equipment require one-tenth of the time and effort that are now expended.

5. Pumps, their drivers, and related equipment require zero maintenance over a lifetime of at least 25 years.

6. Pumps meet all requirements as quoted with no field adjustments being required; moreover, pump output automatically adjusts to efficiently match the actual system in which it is installed, for which it might otherwise have been over- or undersized.

7. Pump manufacturers and academic institutions jointly focus on pump issues and continuously train personnel for all job functions, according to far-reaching upgrading and succession plans.

8. Lead times for orders are measured in days, not months or years.

9. Causes of accidental shutdown are automatically determined, remedied, and recovery to full operation as intended is also automatic and rapid enough to prevent significant loss of output from the process being served by the pump and related system.

10. Pump manufacturers and academic institutions continually update their vision and mission statements, fielding the necessary joint R&D and educational projects in product design, manufacturing, marketing, and industrial management to implement the mission.

The next step is to review critically each of these elements for realism and then expand, contract, combine, or otherwise modify them. The objective here is to end up with a clear vision that is neither too cumbersome nor too superficial but rather is balanced and challenging enough to inspire an entire organization(s) to work toward it. We can be aided in this critique by:

- Considering past experience and
- Acknowledging emerging competitive, regulatory, political, and other pressures.

A common thread running through all these elements suggests that if they could be achieved, pumps would be commodities that are easily procured and put into operation—much like computers are becoming. Both computers and many pumps at different times in the past were regarded as high-technology products that required much professional attention at every step of the way from selection through design, manufacturing, and deployment. Moreover they were temperamental and required experts to maintain and debug them frequently. Even since the first Pump Symposium in 1984, we have seen a transition that points quite clearly toward the industry attaining commodity status at some time in the future.

However, with respect to element #8 on lead time, the commodity concept applies less well to the really large and/or technologically complex machines or more specifically those pumps that would be in that category 25 years from now. Certainly there are small pumps available today that are commodities already and can be procured immediately. So we would need to modify at least element #8 to allow for such differences, realizing at the same time that pressures will always exist to push us in the “commoditizing” direction. We will not attempt that at this point. Market forces may cause this to happen anyway.

Vision Statement

We now choose for our vision the first seven elements as listed, modifying them in view of the above considerations together with other aspects of realism from our collective experiences—finally combining them into a single *vision statement* as follows:

“The global pump industry contributes highly efficient, reliable, and durable products—often in partnership with other entities—to the building up of environmentally optimum processes worldwide. Maximum and continuously improving economies pervade all phases of the business from selection, procurement, product design, and installation to field service, and maintenance, which is at or near zero over lives of 25 years for pumps, drivers, and related equipment. Highly trained human resources are applied through a partnership of manufacturers, consultants, and educational institutions that is focused throughout all these phases on the needs of the pump industry.”

Does past experience make this sound like an impossible and unrealistic vision statement? One needs only to look at the sweep of the Pump Symposium as summarized in Table 1 to be encouraged in this respect. The papers (lectures) and tutorials listed there are representative of many more like them, all of which convey unavoidable evidence that the march toward this vision is already well established. This trend runs the gamut from the knowledge gained in rotordynamic behavior, reliability improvements in off-design operation, and cavitation life extension of high-energy multistage utility and American Petroleum Institute (API) pumps to the impressive jump in mechanical seal durability and operating envelope that have all occurred over the life of the Pump Symposium. Moreover, glimpses of the educational aspect of the above vision statement are to be had in the partnership of university, consultant, and manufacturer that have produced and animated such an enduring institution as the Pump Symposium. To be sure, there is a long way to go, but our vision gives us another 25 years to get there. And get there we must; otherwise, we risk becoming outmoded and irrelevant in the drive of the emerging market forces already identified.

MISSION

Real progress toward realizing the vision can be achieved only if we define how to do it in a mission statement. Concrete programs are needed in order to be sufficiently definitive. Response to this need in the past has always resolved itself into two distinct and major activities; namely:

a) Advancing the technologies required through a well-defined and professionally managed program of R&D projects and

b) Ensuring the success of (a) with a new and effective training and educational system.

Technologies

With respect to (a), what R&D projects should be undertaken? At this point it appears that the vision would be realized by efforts in the following areas:

i) Reducing the size and first cost of the pump and driver by means of durable higher-speed machines with demonstrated durability.

ii) Maximizing the efficiency of the pump and driver package and minimizing the life-cycle cost of procuring it and operating it in the system in which it is placed. This applies to all operating conditions.

iii) Simplifying installation and maintenance through innovative design of the entire package of pump, driver, and related equipment.

iv) Developing the materials and manufacturing processes necessary to support all areas.

Pump Speed, Size and First Cost

The higher speeds called for in area (i) imply pumps of higher energy or power density. This in turn is accompanied by more intense mechanical and hydraulic behavior, which has already evoked a considerable amount of R&D aimed at maintaining reliable and durable operation, as alluded to in Table 1. For example, a component of this R&D has involved inducers, which had been deployed successfully in the rocket engine pump of Figure 1. This essentially tripled the pump speed over conventional practice, greatly reducing the size and weight of the package, which of course was absolutely necessary in that application. Only shortly before that had the phenomenal suction capabilities of inducers been discovered and understood (Ross and Banerian, 1956). Since then, there has been limited use of inducers in commercial pumps, and again they are seen as essential for the emerging liquefied natural gas (LNG) market (Figure 2).

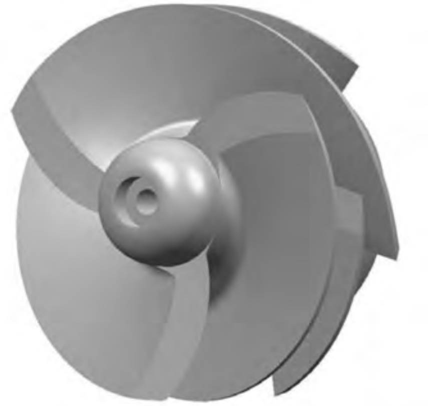


Figure 2. Typical Inducer. Notice the splitter blades. Used in LNG service. (Courtesy of Karassik, et al., 2008, pp. 12.335-12.359)

However, two problems have limited more general acceptance of inducers, namely, off-design, low-flow instabilities and cavitation erosion at high blade tip speeds. Neither of these problems impacted the space applications, because the turbopumps operated near design conditions throughout the rocket burn, which lasted only a few minutes. The stability problem has been the subject of extensive research since the early days of the Pump Symposium (Sloteman, et al., 1984) and most recently by researchers connected with the space programs of Japan and France (Tsujimoto, et al., 2005; Bakir, et al., 2004). Uniquely-shaped impeller blades have eliminated cavitation and the accompanying erosion in high-energy pumps (Sloteman, et al., 2004); however, inducers operate at such low net positive suction head available (NPSHA) that cavitation is always present; although, in line with Knapp's work (Cooper, 1996), erosion is essentially nonexistent at the lower velocities of small pumps. For the higher blade speeds of large and/or high-speed pumps, materials more resistant to cavitation would appear to be the solution. Here there also is knowledge, and advances in this knowledge would be an ideal focus of the support area (iv) above. Success in this effort could open the way for reducing the size and weight of commercial pumps, e.g., condensate pumps. The durability objective is paramount in all of this high-speed, size-reduction R&D. Only then would the corresponding cost savings be possible.

Variable Speed

The subject of cost brings us to area (ii), wherein operational costs are considered in addition to the lower first cost being sought through reduction in pump size. In recent years, life-cycle cost (LCC) has come under scrutiny, especially in the case of American National Standards Institute and International Organization for Standardization (ANSI and ISO) pumps ("Pump Life Cycle Costs," 2001). Assuming the validity of our vision, LCC analysis will play an increasingly prominent role in the pump procurement process as the future unfolds. Impacting the operational costs have been the efficiencies of both pump and driver, as well as any mismatch of pump capacity—in terms of head and flow rate—with that of the system in which it is installed. Variable pump speed is a major development that is saving operational costs by better matching the pump head and flow rate capabilities to those of the system. This enables an oversized pump (often specified by engineering contractors as providing a "safe" margin) to run nearly as efficiently in the system as if it had been correctly specified. This results in considerable savings of energy that would otherwise be wasted by throttling constant-speed pumps. Because they are always running at full speed, such constant-speed pumps are more prone to suffer from violent response to recirculation at flow rates well below that of the best efficiency point (BEP). Further matching improvements are possible with variable geometry, which will be addressed further on.

Pump Efficiency

In addition to how well the pump is matched to the system, LCC analysis also involves the pump efficiency and that of its driver. Guidance as to how a given pump is performing relative to "the state-of-the-art" can be had from efficiency charts that many authors have presented. An example is found in Figure 3, derived from H. H. Anderson's work with water pumps. As such it has limitations with respect to pumping more viscous fluids; also it seems to be too conservative at the higher specific speeds, the dashed line being more representative for high-specific-speed pumps. On the whole, however, it has proven to be a reasonable guide, not least because an equation can be used in place of the chart. Pump size is also taken into account through the parameter "gpm/rpm," major size effects being the relative roughness of the passages and the ratio of clearances of the impeller neck-ring seals to pump size. Therefore, pumps with hydraulically smooth passageways and essentially zero internal leakage are worthy R&D subject matter.

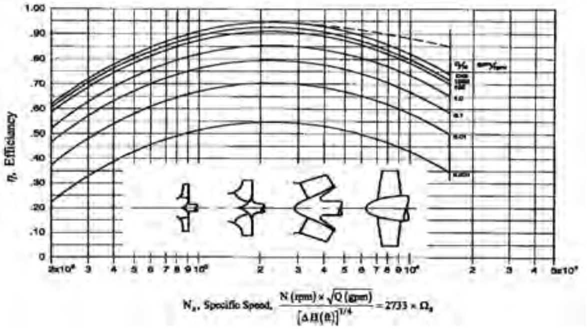


Figure 3. Pump Efficiency Versus Size and Specific Speed. Plotted from H. H. Anderson's empirical equations, as summarized in the source. (Courtesy of Karassik, et al., 2008, p. 2.25)

Motors

Next comes the efficiency of the pump driver, which is generally an electric motor. Assuming this to be the case together with the fact that AC induction motors are the most ubiquitous of such machines, we note from Figure 4 that both efficiency and power factor fall off precipitously at reduced loads, such as might occur at low flow rates on some centrifugal pumps. Low efficiency directly translates into more shaft power to carry the load; and low power factor requires more current and therefore larger generating equipment, wires, and so on. Nevertheless, it must be admitted that the normal load range of most pumps is within the range for good motor efficiency and probably acceptable power factor. This is evident from Figure 4 and from the fact that minimum pump shaft power (at shutoff) is usually no less than 30 to 40 percent of BEP power or maximum power, assuming the essentially constant speed of an induction motor over the whole flow range.

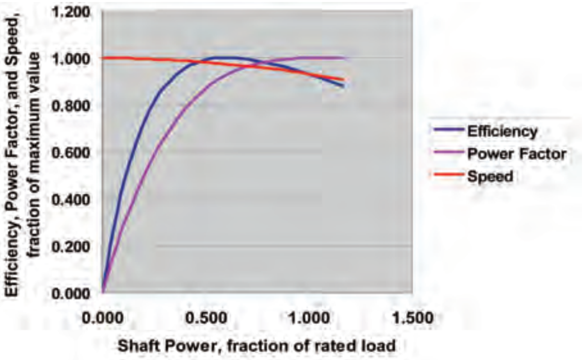


Figure 4. Typical AC Motor Performance.

However, an alternative to the alternating current (AC) induction motor has been gaining prominence over the life of the Pump Symposium, and that is the permanent-magnet brushless direct current (DC) motor, or PM brushless motor. For a given power output and speed, the PM brushless motor is about two-thirds the size of an AC induction motor and less than half the weight. However this motor requires an electronic box called a "drive" that first rectifies the AC input to DC (typically 600 VDC for a typical 460 Volt input) and then, by means of special switching transistors called insulated gate bipolar transistors (IGBTs), inverts the DC back to a variable frequency AC. This AC wave at a given frequency is made up of pulses whose time duration or width varies with the load or torque. This pulse-width-modulated wave powers the stator coils that interact magnetically with permanent magnets in the motor rotor. The result is an inherently variable-speed, synchronous machine with 100 percent power factor and usually more than 90 percent efficiency at all loads or torques (Karassik, et al., 2008, pp. 9.10-9.13). The drive can be programmed for a soft start, thereby reducing mechanical wear and tear and replacing the starter needed for induction motors. Moreover the speed can be computer controlled, based on, say, inputs from pump head or flow rate sensors, the drive having circuitry that relates the speed and torque of the motor and establishes the pulse width needed to create torque demanded by the pump at the required speed and corresponding frequency.

The power rating of PM brushless motors has been climbing over the years from the very small sizes needed for high-speed computer disk drives measured in Watts to as much as 10 MW (Smith and Watson, 2006). Successful development has depended on progress in electronics, mainly the technology of IGBTs that have had to switch ever larger currents. Costs of these motors of a practical size for most pumps have been high in the past, but they have been coming down at the same time that the reliability of the electronics and other elements of the motor has been increasing. It would therefore appear that the PM brushless motor can be expected to become a basic building block of future pump packages needed to realize our vision. Thus one of our mission objectives must be to identify and conduct the appropriate motor R&D to reach that goal.

Packaging, Materials, Manufacturing

This brings us to area (iii) identified at the beginning of this mission section, in which we speak of "innovative design of the entire package of pump, driver, and related equipment." Several R&D projects could be envisioned in this category, especially in view of the preceding discussions. Identifying them all is a challenge that involves the applications envisioned, the manufacturing organization, etc. Rather than attempt to do this here, we will present a compelling example in the next section as we try to get some glimpses of how our mission might be implemented. The support area (iv) would then develop naturally as the materials and manufacturing processes involved in the other areas are identified.

Education and Training

To be effective, however, all of these R&D initiatives need to be enabled by the focused and disciplined expertise of a new brain trust of personnel who are skilled in the requisite mechanical, hydraulic, and electronics areas in terms of both academic training and experience in the pump industry. This leads to definition of the other main activity (b) of our mission; namely, "ensuring the success of (a) with a new and effective training and educational system." Already, today's engineers have begun the move toward our vision by applying the knowledge gained in the most recent cycle of R&D activity.

How can we ensure that the thread of these learnings will be picked up and exploited to the full? An example can be had from the business world, wherein prominent academic institutions have combined the resources of their engineering and business schools in intensive "Executive Master of Science in Technology" programs. Seasoned, proven engineers have gone through such programs in a couple of years while still on the job, in one case being on the campus on weekends, beginning on Friday morning (Sloteman, 1994). These are high-profile programs that expose the students to the latest technologies and experts in these fields, infusing the whole with a viable commercial outlook. Enlightened managements have begun to include this type of program in their training budgets, thereby investing in the future of both the person and the company. For the future of the pump industry as defined in our vision, this concept could be adapted so as to focus more specifically on pump technologies. This would yield immediate value to the company by way of the infusion of high technology into its products. The difference is that this would not be just another master's degree; rather, a prerequisite for the program would be that the student must have experience in the industry; specifically, the participant must already have the functional equivalent of a master's degree.

As proposed in the next section of this paper, new focused partnerships between academic institutions and pump manufacturing companies would be needed to realize this part of our mission. As an aid to formulating such partnerships, established relationships exist that are precedents for this new educational product. For example, Texas A&M University is well positioned to move into this sphere of educational activity. They would come from a position of strength in turbomachinery and pump leadership, supported by an enviable record of participation with industry not only via the Pump and Turbomachinery Symposia but also in a series of involvements and courses covering all aspects of the industries involved. This is the legacy of Texas A&M's user orientation.

Mission Statement

Having explored the ideas proposed at the beginning of this section on mission, we can conclude by distilling them all—with some augmentation as needed to maintain fidelity to our vision—into the following six-part mission statement:

1. *Develop the technologies that improve efficiency of both pumps and drivers over their entire flow and load ranges and that realize the energy savings achieved through better matching of this equipment to the systems and processes they are a part of.*

2. *Conduct the applied research required to deal with the hydraulic, mechanical, and electronic phenomena that accompany and challenge efforts to reduce size and cost.*
3. *Develop cost-reducing innovative packaging as well as the ability to accommodate higher speeds, incorporating design features and materials that limit and control all challenging phenomena.*

4. *Accept only those concepts and developments that substantially increase the reliability and durability of the equipment involved relative to existing products.*

5. *Identify and conduct the R&D projects necessary to accomplish these objectives as well as economies in procurement, design, testing, manufacture, and installation that lead to less field service and maintenance.*

6. *Establish new and focused partnership arrangements with educational institutions (and consultants as needed) that provide the effective advanced and more rapid training needed for experienced and promising pump industry personnel to ensure quality and enduring performance in pursuit of the vision.*

This mission statement is therefore proposed as at least a preliminary roadmap to follow. It will very likely need to be revised a few times over the next quarter century. Perhaps others with further insights and experience may want to revise it immediately.

In all cases, the test of the relevance of any mission statement will always be how well it can move us as an industry toward achieving our vision. Moreover, changing needs of the global society and the related markets for the pump industry could even require us and/or our successors to rethink this vision.

IMPLEMENTATION

To implement the foregoing mission or any other that strives to realize the vision, one needs only to refer to the statement itself to recognize that continuous R&D and complementary training are major features of the life of the pump industry in the years that lie ahead. While many examples of the required effort come to mind, a few that stand out as being representative in setting the stage for the future are now presented.

Variable Geometry

Matching and low-flow stability—In the discussion that led up to the mission statement variable speed was cited as enabling the match of pump to system. This provides an excellent match of impeller pumps to pure-friction systems, wherein the head increases as the square of the flow rate. Thus if the pump is matched to this system at one flow rate, it will be matched quite well at all other flow rates so long as the speed is adjusted to make the pump head the same as the system head. However, for the elevation-dominated system where the head changes very little with flow rate, the speed cannot be reduced very much as flow rate is reduced; and so the pump is then operating at flow rates far lower than that of the BEP. It is well known that at such flow rates, recirculation and the accompanying instabilities occur that can have deleterious effects on pump reliability and durability, especially those pumps with high pressure rise (high energy) or high suction energy. Figure 5 is a reminder of the internal flow patterns in the pump under such conditions. In this case, some of the surrounding diffuser passages are stalled, meaning that not all diffuser passages have the same pressure rise. This imposes a radial load on the impeller. Moreover, such separated flows are notoriously unsteady, so that the stall patterns move or rotate to the adjacent passage. The mechanical consequences can be imagined if the pump has a high pressure rise. The final blow is that the pump efficiency under such conditions is very low.

Figure 5. Stalled Flow Visualized Within Pump Impeller and Diffuser at Flow Rates Well Below the BEP. Flow rate is near the rotating stall condition. Whole passages in the diffuser are stalled; others are flowing full. (Courtesy of "Pump Hydraulics—Advanced," 1999)

High specific speed: mixed-flow machines—Such confused flow conditions and low efficiency would not occur if the impeller and diffuser passageways could be closed down (like Venetian blinds)—as the flow rate is decreased—by rotating the impeller blades and the diffuser vanes about axes located at mid-chord—as seen in Figure 6, which is a cross-section of the mixed-flow Deriaz turbine (Warnick, et al., 1984). Having such fully-variable geometry, this type of turbine has very high efficiency over most of its flow range, as does the higher-specific-speed axial-flow Kaplan or propeller turbine, both of which are represented in Figure 7 by the curve marked "adjustable blade." The Deriaz turbine is of particular interest to pump engineers, because:

- It was developed as a reversible pump turbine for use in pumped storage applications, and
- It approximates a wide range of mixed flow pumps.

The benefits of this same variable geometry in such pumps are immediately obvious. The usually high power consumption at low flow rates of fixed-geometry mixed- and axial-flow pumps yields low efficiency and oversized motors, such machines routinely drawing the highest power (at a given speed) as the flow rate approaches zero (shutoff). (Of course, such pumps operating in pure-friction systems could realize top efficiencies at all flow rates through speed variation alone; however, such an ideal system is not always available.)

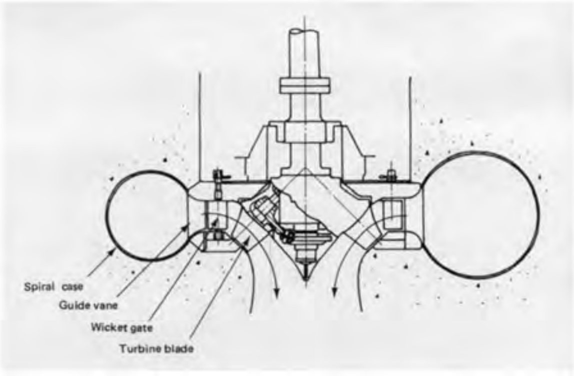


Figure 6. Deriaz Turbine, Showing Adjustable Blades and Wicket Gates. Deriaz turbines were developed for use as reversible pump turbines. (Courtesy of Warnick, et al., 1984, p. 18)

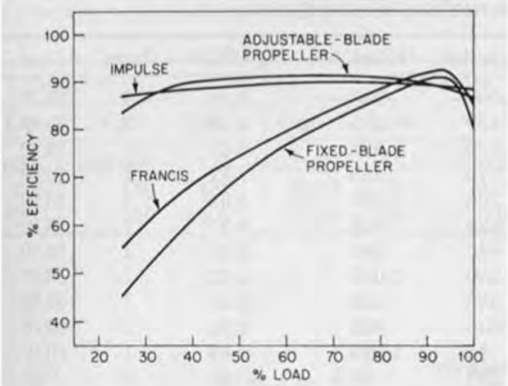


Figure 7. Effect of Variable Geometry on Efficiency of Hydraulic Turbines. Both the Francis turbine and fixed-blade propeller turbine have adjustable wicket gates. Adding this variability to the runner, as with the adjustable-blade propeller, maintains maximum efficiency at all load levels. (Courtesy of Karassik, et al., 2008, p. 9.81)

All of this is not to say that no variable-geometry pumps exist. Some quite large axial-flow flood control pumps have adjustable blades on their propellers. Today, the cost of doing this on the more typical smaller pumps is prohibitive; and the reliability of the blade- and vane-adjusting mechanisms can be an issue. The efficiency improvement goals of our mission, however, clearly indicate that variable geometry is a prime candidate to satisfy this need and that therefore it will quite likely find its way into the more typical impeller pumps going forward.

Low specific speed: centrifugal machines—Variable geometry, however, is presently a challenge for closed-impeller centrifugal pumps. However, diffuser vanes stand a better chance of being so adapted. An example that comes from within the experience of some pump manufacturers is the adaptation of centrifugal (radial-flow) pumps for use as turbines to recover the energy of pressure let-down that otherwise occurs across throttling valves. These are called hydraulic power recovery turbines (HPRTs). Rules have been developed for deploying centrifugal pumps running in reverse as HPRTs (Karassik, et al., 2008, pp. 2.180-2.184). Such machines have also been optimized as turbines, with the appropriate changes in impellers (now runners) and diffuser vanes (now nozzles or wicket gates) thereby achieving greater efficiency in the turbine mode (Nelik and Cooper, 1984). Making the positions of the nozzles or wicket gates variable has been done for both single- and multistage HPRTs (Gopalakrishnan, 1986). Large, hydropower Francis turbines (Warnick, et al., 1984) all have variable wicket gates, and the performance is indicated in Figure 7, for which the head across the turbine is constant. If these gates were not variable, the flow rate—corresponding approximately to the "load" in Figure 7—of such a constant head machine could not be changed.

Flow range improvement—But the way this works for centrifugal pumps, where flow rate is the independent variable, is that the efficiency at half the BEP flow rate is typically about 80 percent of the BEP efficiency for fixed geometry. However, if the diffuser vanes could be closed up as flow rate is decreased, this 80 percent point would be achieved at lower flow, say a quarter of BEP, giving the pump a higher flow range at good efficiency. Moreover the confused diffuser flow patterns would not exist, yielding a corresponding improvement in stability, reliability, and durability of such a variable-geometry pump.

HPRT/Francis turbine—An example of an HPRT with variable wicket gates is shown in Figures 8 through 10. The rotatable gates or nozzles vanes are seen in Figure 8; although, the runner—similar to a centrifugal impeller—is not shown. As can be seen in Figure 9, the whole machine at first sight differs little from pumps that are normally produced by the manufacturer (Miller, 2008); which explains why they are pump products and are treated in pump books (Karassik, et al., 2008, pp. 2.180-2.184). However, looking more closely at Figure 9, one sees part of the mechanism required to vary the openings between the wicket gates of Figure 8. This is basically a low-specific-speed Francis turbine, and its performance map is shown in Figure 10 in typical pump (and HPRT) coordinates of head versus flow rate. A comparison of this machine with the much larger, hydropower Francis turbine curve of Figure 7 is possible by finding the efficiency change at constant head in Figure 10 as the flow rate is reduced.

Figure 8. Adjustable Wicket Gates of a Hydraulic Power Recovery Turbine (HPRT). Built by a pump manufacturer as an optimized, variable-geometry turbine instead of a fixed-geometry pump running in reverse as a turbine. (Courtesy of Flowserve Corporation, Pump Division, Taneytown Operation)

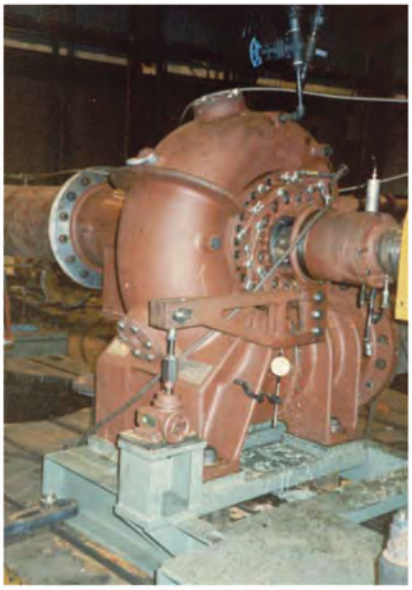


Figure 9. Shop View of the HPRT. The mechanism that actuates the wicket gates can be seen. Otherwise this machine is very much like a pump. (Courtesy of Flowserve Corporation, Pump Division, Taneytown Operation)

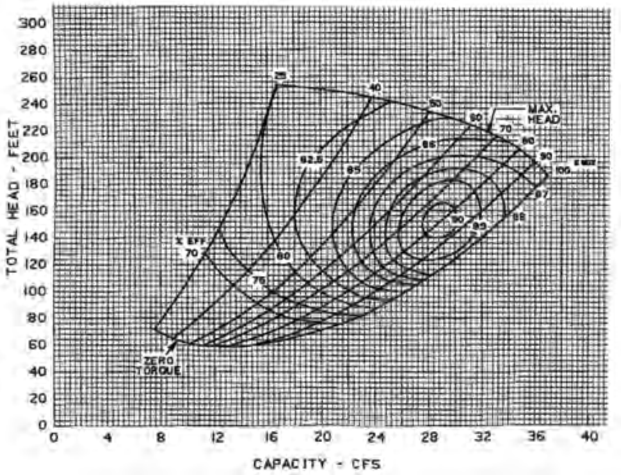


Figure 10. Performance Map of the HPRT. The parabolic curves are of turbine head versus flow rate (or capacity) for various wicket gate openings (given in percent of maximum opening), and the islands are loci of constant efficiency (shown as ranging from 70 to 90 percent). Speed is constant at 1210 rpm. (Courtesy of Flowserve Corporation, Pump Division, Taneytown Operation)

Centrifugal impellers—Here again and finally, our main interest is in what variable geometry can do for centrifugal pumps; and to this case the discussion above still applies; namely, about achieving 80 percent of the maximum pump efficiency at much lower flow fractions of the BEP flow. The real challenge for the future of such pumps is to go for fully variable geometry, i.e., also moving the impeller blades as is done with the higher-specific-speed machines such as the Deriaz pump turbine (Figure 6). Creative mechanical development of a reliable, durable, and cost-effective innovation in this direction can be expected to occur further into the future, and the benefits of this effort will move the entire pump closer to being capable of running virtually at shutoff—both smoothly and at very low power levels. Imagine this being achieved with high-energy multistage boiler feed pumps as well as process pumps of similar energy levels! Contrast such an achievement with past history in which the geometry of these machines was kept fixed. That single limitation could be viewed as having generated a quarter-century or more of intense research efforts aimed at making these pumps do almost the impossible in terms of off-design operation.

Integral Motor Pumps

Just as our mission statement calls for projects to substantially improve efficiency, reliability and durability—as we have predicted that variable geometry can surely accomplish—so also does it add the need for the further efficiencies and cost savings in all activities of the pump industry. A promising way to accomplish this is by integrating the pump impeller with the drive motor into a single compact unit to create an "integral motor pump" package, which is a concept that promises to combine product performance improvements with revolutionary economies in product first cost, installation, and maintenance—also specifically called for by the mission statement.

Sealless pump legacy—Sealless pumps and permanent-magnet brushless DC motors form the background and building blocks for integral motor pumps (Sloteman and Piercey, 2000). Sealless pumps burst on the scene in the 1990s as the way to avoid problems of seal leakage and satisfy tougher emission standards (Cooper, 2008). Pump and motor were still separate units, but the pump was hermetically sealed and was driven through a cylindrical magnetic coupling by a standard AC induction motor—all typically mounted on a bedplate. The pump bearings were typically made of silicon carbide, because they had to run in the fluid being pumped (the pumpage)—with cooling flow being sent to them from the higher pressure zone of the pump and returning to a lower pressure zone. The concept was attractive in that it promised zero emissions. However the hoped-for takeover of the ANSI and API pump markets by the purveyors of this new product did not occur because:

- The bearings overheated and failed when pumping certain volatile fluids containing dissolved gases (Wood, et al., 1998), and
- The seal industry responded vigorously with a sustained program of performance and reliability improvement of mechanical seals (Cooper, 2008; Sloteman and Piercey, 2000).

Nevertheless, careful analysis and testing of the internal flow path for cooling the bearings were performed by the pump manufacturers, which provided design guidance for eliminating the bearing failures or at least identifying when they might occur (Wood, et al., 1998; Guinzburg and Buse, 1998). This understanding of product-lubricated bearings and the related cooling and lubrication circuit became an important feature of integral motor/pump technology.

Incorporating a canned PM brushless motor—Probably the closest relative of the integral motor/pump is the hermetically sealed canned motor pump, wherein both pump and motor are still separate entities; however, as opposed to sealless pumps, both pump and motor run in the pumped fluid. Here again, the bearings run in the fluid. But canned motor pumps are expensive, usually operate at constant speed, and serve a niche market (Karassik, et al., 2008, pp. 2.349-2.361). To use the canned motor pump concept cost-effectively in all respects would indicate the use of a PM brushless motor, which is:

- Highly efficient at all loads,
- Smaller in size, and
- Can be matched to the system via its inherently variable speed.

The most compact way of integrating this motor with the pump is to configure it as an axial-field motor. The result is the sealed centrifugal axial-field motor pump (SCAMP) shown in cross-section in Figure 11. The armature of this motor consists of the necessary array of permanent magnets sealed into the back side of the impeller, and the stator coils are sealed into the adjacent stationary wall. So, the magnetic circuit is directed axially between the rotor or armature and the stator—rather than radially as in the more typical cylindrical rotor configuration (Sloteman and Piercey, 2000). The motor is cooled by some of the pumped fluid being bled from the impeller outside diameter (OD) discharge zone, then taking various paths through the motor, and finally rejoining the main flow stream at the inlet or suction area of the impeller (Sloteman, et al., 2000). Special materials have been utilized to ensure reliability, and durability was established by continuous running of the unit in the laboratory for more than five years without a failure (Sloteman, 2008).

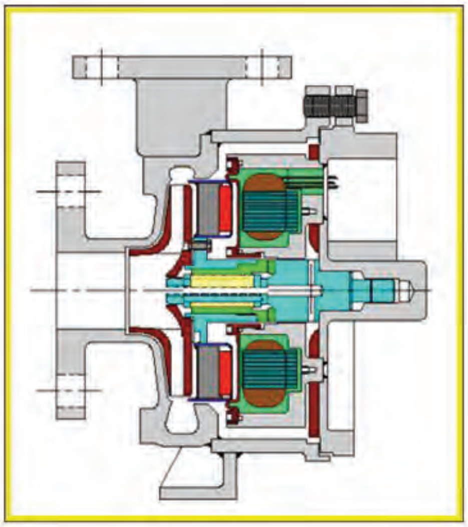


Figure 11. 3.6 HP Sealed Centrifugal Axial-Field Motor Pump (SCAMP). This unit has a 4-inch impeller/rotor diameter and a PM brushless motor. Speed is variable up to 5600 rpm, and the unit delivers 60 gpm at a head of 120 ft. (Courtesy of Sloteman and Piercey, 2000)

SCAMP applications—The SCAMP concept, like the more conventional sealless pumps, has yet to be widely accepted. While one of its major application targets was originally and still is the ANSI/ISO pump market, in the meantime it has attracted the attention of the U. S. Navy, who have been slowly adopting it for surface ship application. In a recent order to the present owner of this design and related patents, they plan to add 20 more units. The Navy also has written a military specification for permanent magnet-driven integral motor pumps like SCAMP as well as the double-suction version (shown in Figure 12 and described next). They are actually replacing general service pumps with these special, sealless variable-speed machines (Sloteman, 2008). This is an early indicator of what the future may hold in terms of the mission and vision we have proposed for the industry.

Larger versions—With the success of the original SCAMP concept of Figure 11, it is not hard to imagine larger and more powerful integral motor pumps. Two versions of such machines have been designed, built, and tested; one being a 240 hp "Super-SCAMP" with a larger (14.5-inch) single-suction impeller. The other is the 400 hp "double-suction SCAMP" of Figure 12 (Sloteman and Piercey, 2000; Cooper and Bulson, 1992). Here the permanent magnets of a large, axial-field PM brushless motor were embedded in both shrouds of a standard 18-inch double-suction impeller. This impeller had extended wear ring lands that became the bearing journals and, as such, were surrounded by a multitude of silicon-carbide tilting-pad bearings. Immediately adjacent to each side of this double-suction impeller, the sealed PM brushless motor coil assemblies were configured (as was done adjacent to the back side of the single-suction impeller of Figure 11.) The bearing housings of the accompanying standard pump casing were sawed off and replaced by the end covers seen in Figure 12. The first query by the casual observer of this incredible machine would most likely be, "Where is the motor?" In fact there is no driven shaft in this pump! The installation and maintenance requirements of our mission are obviously met with this configuration, no alignment or bed plate being involved. It is not hard to imagine this as the API pump of choice before another 25 years have passed.

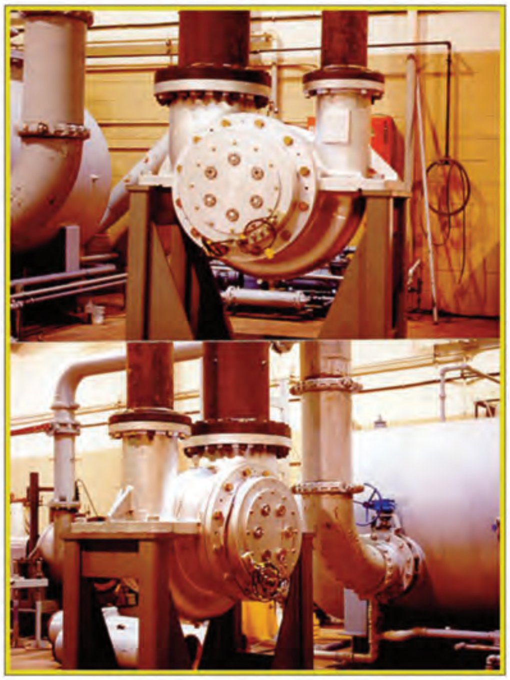


Figure 12. 400 HP, Double-Suction Sealless Integral Motor Pump. This unit has an 18-inch double-suction impeller/rotor diameter and a correspondingly double-sided PM brushless motor. The unit delivers 3000 gpm at 350 ft. Speed is variable up to 2450 rpm, which is 40 percent greater than normal for this flow rate. (Courtesy of Sloteman and Piercey, 2000)

Efficiency—The efficiency of the pump of Figure 12 is 66 percent, which is below the 80 percent found in Anderson's chart (Figure 3) for the stated flow rate and rpm and the specific speed of about 1700. This is at least partly caused by the additional fluid friction drag of the oversized impeller shrouds, which are wider than normal because of the embedded magnets and 18 inches in diameter as compared to the impeller blade trailing edges, which are at 16 inches. Smaller motor diameter would therefore need to be sought in a future R&D project.

Internal seals—Another way to make an improvement in pump efficiency that has often commanded attention of researchers is the internal leakage past the impeller wear rings. This is especially important at low pump specific speeds, because this leakage depends mostly on pump head and not on the main flow through the pump (Karassik, et al., 2008, p. 2.48). Thus at lower flow rates, a significant gain in efficiency could be realized if such leakage could be eliminated. To this end, advances in noncontacting seal technology have reduced internal leakage by an order of magnitude or more (Gardner, 1999). New developments in this area involve compliant annular seals that can maintain ⅓ mil (0.0003 in) radial leakage clearance and yet accommodate shaft or impeller ring land movement of ± 10 mils (± 0.010 in) while simultaneously maintaining most of the strong Lomakin support of traditional smooth bushing seals (Gardner, 2008). The multiple tilting-pad product-lubricated bearings already described as supporting the wear ring lands of the double-suction SCAMP of Figure 12 could most likely be configured to achieve this advantage.

Scope of products and applications—Combining both pump and motor within a sealed package is actually an old concept, because pumps, compressors, and turbines are found in applications that demand total sealing from the environment. Refrigeration and air-conditioning systems have achieved high reliability and many years of life with hermetically sealed turbomachinery. A few examples for pumps follow, running the gamut from microsized units to large subsea multiphase pumps:

- Micropumps and -motors are expected to find a host of applications in energy conversion and power production, e.g., miniature Rankine cycles and rocket engines. An impeller diameter of 8 mm is typical of the machines in this growing micro-electro-mechanical systems (MEMS) field, in which integration with the driver is certainly an option (Epstein, et al., 2000).
- Biomedical applications such as in-vitro heart pumps are currently a step up from MEMS in size. In a working unit, the rotor is magnetically suspended in a sealless centrifugal pump configuration that is otherwise simply a miniature version of the SCAMP integral motor pump of Figure 11 (Akamatsu, 2000).
- Multiphase twin-screw pumps are being designed for subsea oil and gas production, which requires that the entire pump and submerged-motor package be hermetically sealed from the subsea environment (Karassik, et al., 2008, pp. 3.115-3.119). A key future development would eliminate the need to supply clean oil from the surface to lubricate the bearings and timing gears and to cool the motor. Instead, the integral motor pump concept of using the pumpage to perform these functions would be introduced, thereby eliminating the four large mechanical seals that are typically deployed in these pumps (Cooper and Prang, 2002). Further, utilizing the synchronous feature of PM brushless motors to electronically phase two of these motors—one for each screw—would also eliminate the timing gears (Sloteman, 2001).

Limitations—It will be recognized, however, that extending the concept of the sealless integral motor pump to the larger pump installations is unrealistic. R&D projects in this sphere would need to identify where the concept has merit at every point in time through the years ahead. For example, we can confidently expect that the present-day trains of high-energy multistage pumps and drivers will still be with us a quarter century hence. The same could be said for single- and multistage vertical pumps except possibly for the smaller units such as the electric submersible pumps (ESPs) used in the oil field. This fact does not exempt these or any other kind of pump from being subject to our mission for the industry. For example, the CO2 pumps needed in the systems being developed for carbon capture at coal-fired power plants may be excellent candidates for the efficiency improvements available with both variable geometry and variable speed. Nevertheless, for the majority of smaller pumps, the innovative packaging illustrated in Figures 11 and 12 and developed further as needed should yield the advantages we seek relative to overall costs, performance, and life.

CFD Versus Testing

Cost of design implementation—Rendering these advanced pump design ideas into commercial products is the first major component of the total cost picture. In the past, new pump configurations required a considerable amount of experimental development, and this often had a discouraging impact on the schedule and related development cost picture. Arguably, it has been a major deterrent to R&D. Yet, on a more encouraging note for the success of our mission, this obstacle may soon disappear. For example, over the life of the Pump Symposium we have seen a phenomenal increase in the role played by both mechanical and fluid dynamical analysis in the total design process. Computer codes have been commercialized and are becoming indispensible.

CFD benefits—In addition to the rotordynamics and finite-element (FEA) stress and vibration codes (Cooper, 2008), computational fluid dynamics codes are removing the need for the extensive laboratory experimentation that was previously required to prove out all aspects of the hydraulic performance that had been so mystifying, especially at off-design conditions. This includes the complexities of unsteady and recirculating flows and their impact on the adjacent flow elements such as inlet passageways, impellers, diffusers, and return vanes (Karassik, et al., 2008, pp. 2.97-2.120). Predicting shutoff head and power is no longer a challenge. Even two-phase flows are now routinely computed in detail, and, in this regard, Figure 13 is an impressive example of the ability of CFD to predict the elusive curve of head deterioration versus decreasing available NPSH (Sloteman, et al., 2004). In some recent cases, this capability has virtually eliminated NPSH-testing prior to the final acceptance test still required by the users. It is becoming no longer necessary to maintain a large hydraulic development testing organization to come up with new pump products; one fluid dynamically savvy engineer manning a high-end CFD station with multiple processors (Karassik, et al., 2008, pp. 2.97-2.120) can replace an R&D test laboratory. It is conceivable that such reliability in predicting pump hydraulic performance will become so widely recognized that even acceptance testing could be seen as an unnecessary expense in the procurement process. This is already the case for FEA stress analysis, which has had a generational head start on CFD, for example, FEA results are accepted as confirming compliance with seismic requirements by nuclear pumps (Karassik, et al., 2008, pp. 12.139-12.149).

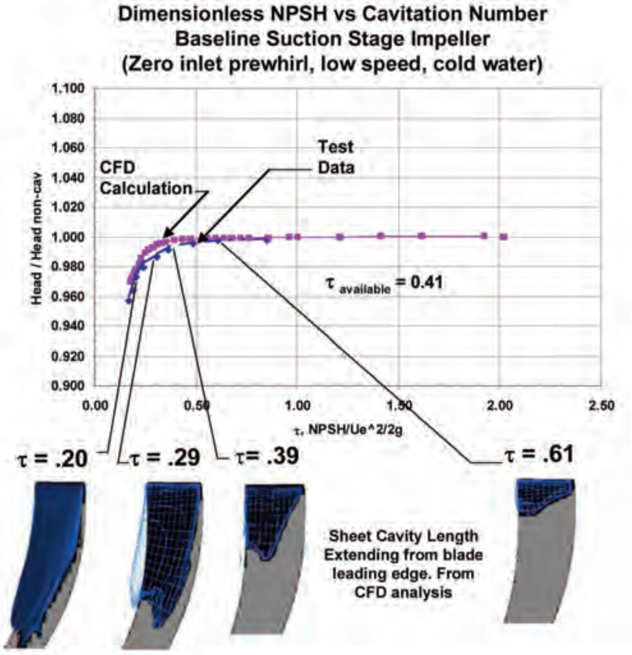


Figure 13. CFD Computation Versus Test for the NPSH-Curve for a Boiler Feed Pump Impeller. The accompanying computed cavities were also observed experimentally. (Courtesy of Sloteman, et al., 2004)

R&D Institutes

Key personnel—Where do the above fluid dynamcally savvy engineers come from? They not only have a normal university education and degree but also the understanding that comes only from experience in the industry, which enables them to apply enlightened fluid dynamical reasoning in a cost-effective manner to the flow geometries they generate, analyze, and modify in order to achieve the final performance needed. It can take a major portion of a person's career to develop such valuable capability. Of course this applies to all areas in the R&D spectrum, not just fluid dynamics or hydraulics.

The advanced-education challenge—At present, the essential capabilities for conducting productive R&D take too long to acquire. Typically, achieving such capability has required 20 years of experience, ideally including time at an R&D center—which most companies have lost or never had (Cooper, 1996)—and/or otherwise setting up an R&D function in a business unit; and going to school to learn how to implement and capitalize. As mentioned earlier under the section *"Education and Training"* in developing our mission statement in the "MISSION" section, schools in the recent past have responded creatively with programs like that of the "Executive Master of Science in Technology," which were designed to form:

- Business leaders and
- Business-informed technologists.

While this has indeed helped develop quality personnel, it has not materially shortened the time for such persons to have the desired effect on the bottom line. Companies cannot afford to take 20 to 25 years to develop an individual with the all-around capability needed to conceive, execute, and capitalize on R&D programs. So, if the goals of our mission are to be viable, both the industry and the academic community need to rethink their relationships. R&D-tasked executives would welcome a focused, guaranteed way of educating their recognized achievers to help push their respective pump companies to prominence in all areas of the business.

Focused partnership—Institutes within universities must be established that enable R&D projects to be born while preparing experienced students. These could be outgrowths of existing programs such as the Turbomachinery Research Consortium at Texas A&M University (Turbomachinery Research Consortium, website) which are proprietary partnerships with pump manufacturers, each with the specific goal of educating one (or more) of their experienced and promising persons. This person would have at least the functional equivalent of a master's degree in terms of pump experience. He/she would interface with assigned academicians, and both would work on one of the R&D projects defined by the company. Together, this team of student and academician would make the headway on the high technology aspects of the project that is needed before the student completes the project and commercializes the resulting product at the company. The emphasis would at first be on the theoretical and analytical aspects of the project but would most likely also include some laboratory testing at the company and/or the school—one or more of these partnerships would form an R&D institute within the relevant university department, be funded by the company; and adhere to a schedule mutually defined by both parties to the partnership. Defined proprietary aspects of the work would be honored by all involved, including what could be published when and where.

Unique academic program—The R&D institute concept would not only encompass all aspects of a classic MS or PhD program, for example, it would probably have less course work, but it would equip individuals in the employ of pump manufacturers to achieve the level of today's qualified pump-industry personnel in the time needed to complete the agreed-upon institute program. This could be of two years' duration and would most likely go hand-in-hand with the student's responsibilities within the R&D department of the company. Institutes would need the best of everything from computer power to the mechanical/electrical/electronics laboratory equipment needed for each project.

Program cost and professional aspect—These partnerships would not be inexpensive, but we need to recognize that more than a decade ago, some managers were paying $25,000 per year for that two-year, after-school/weekends "Executive MS in Technology," in which the students learned the cutting-edge technologies they would need for the next phase of their industrial careers. In this vein, we would have to include some aspects of that type of program in the R&D institute programs. The more focused degree that would be awarded under the aegis of the institute might be "Advanced High Technology Master's Degree in Pump Engineering." Such a program could cost in today's and future terms about $100,000 per year for a two-year commitment and would involve that experienced student with all the great achievers in the field in addition to tasking him/her to do an R&D project that really attracts top management as a path to achieving the company's vision.

CONCLUSIONS

The trends and accomplishments of the pump industry over the past quarter century provide guidance for the future. Taking these into account together with the expected market demands has led to a vision of what the attainments can be after the passage of another 25 years. Envisioned is a global pump industry that is contributing to environmentally optimum processes by being efficient in every aspect of its business from product design and performance to installation and near-zero maintenance.

This vision can be realized by adhering to a mission that is infused with the R&D projects that make it happen. This mission has the following elements:

- Improving pump and driver efficiencies over their entire ranges of flow and load.
- Avoiding the waste of energy due to mismatching pumps with systems in which they are installed and operate.
- Enabling decreases in size and cost through higher speeds and innovative packaging, limiting and controlling the related challenging hydraulic and mechanical phenomena.
- Accepting only those developments that increase equipment reliability and durability.
- Effecting savings in procurement, design, testing, manufacture, installation, and field service that reduce or eliminate maintenance.
- Establishing focused partnerships between industry and academia to effectively and rapidly train experienced and promising personnel to ensure that the vision is realized.

Along with efficiency, the mentality of reducing life-cycle costs is a fundamental characteristic of this mission, and examples for implementing both characteristics are:

1. More widely utilizing *variable speed* to match the pump to the system.

2. Introducing *variable geometry* to improve efficiency and stability at flows well below BEP—at first by closing up the stator (diffuser) vane openings—as has been done by some pump manufacturers with hydraulic power recovery turbines, and (as cost-effectiveness develops in doing so) closing up the impeller blade passages, mixed-flow and axial-flow impellers being the likely first candidates due to their shorter blades that can more easily be rotated as is done with large hydroturbines.

3. Integrating in a sealless package the impeller with a lighter, more efficient motor than today's induction motors, e.g., axial-field permanent-magnet, brushless DC motors, which also have top efficiency across their entire load range and are inherently variable-speed machines. A host of applications for these *integral motor pumps* includes the range of pump sizes from microsized MEMS to ANSI/ISO pumps and to API pumps of at least 400 hp. *However, the feasibility and cost-effectiveness of integrating large and/or high-energy impeller pumps and most multistage impeller pumps with their drivers is not evident*.

4. Including twin-screw pumps of all sizes with their four mechanical seals and their drive motors into a single hermetically *sealed package* for environmentally acceptable subsea oil and gas production.

5. Adding *inducers* to enable higher pump speeds—together with the means and materials needed for these devices to operate at low flows and to withstand the damaging effects of cavitation.
6. Utilizing *CFD analysis* computer codes in the product design and development process (as well as rotordynamic analysis codes and FEA stress and vibration codes) to reduce or even eliminate the need for costly development testing and eventually eliminating even the acceptance testing on some pumps.

7. Establishing *R&D institutes* that focus on pump technology projects and in which the pump company partners with the educational institution in a unique academic program that equips experienced and promising engineers and others in the employ of the company to take a fast track in technology education and in obtaining R&D results that favorably impact the company's bottom line.

The guiding principle from the vision in the drive to accomplish this mission is to work on and accept only those developments that are efficient in all respects and that are cost effective; establishing the required R&D projects, and supporting them with properly trained people who can and will single-mindedly undertake this work. In so doing, it is believed that the pump industry will be well on the road to achieving an ambitious and laudable vision that responsibly benefits the world over the next 25 years.

REFERENCES

Akamatsu, T., 2000, "Development of Turbo-Blood Pump: A Centrifugal Pump with Magnetically Suspended Impeller and Sealless Rotary Blood Pumps," *Proceedings of the 8th International Symposium on Transport Phenomena and Dynamics of Rotating Machinery (ISROMAC - 8)*, Organized by Pacific Center of Thermal-Fluids Engineering, and Texas A&M University, Mechanical Engineering Department, March, pp. 26-32.

Bakir, F., Rey, R., Gerber, A. G., Belamri, T., and Hutchinson, B., 2004, "Numerical and Experimental Investigations of the Cavitating Behavior of an Inducer," *International Journal of Rotating Machinery*, *10*, (1), pp. 15-25.

Cooper, P., 1996, "Perspective: The New Face of R&D—A Case Study of the Pump Industry," *ASME Transactions ASME, Journal of Fluids Engineering*, *118*, December, pp. 654-664.

Cooper, P., 2008, "25 Years of Driving Technology," Welcoming Address at the Twenty-Fourth International Pump Users Symposium, Published in the Proceedings of the Twenty-Fifth International Pump Users Symposium, Turbomachinery Laboratory, Texas A&M University, College Station, Texas, 2009.

Cooper, P. and Bulson, L. J., 1992, "Integrated Centrifugal Pump and Motor," U.S. Patent No. 5,158,440, October 27.

Cooper, P. and Prang, A. J., 2002, "Sealless Multiphase Screw-Pump-and-Motor Package," U.S. Patent No. 6,457,950 B1, October 1.

Epstein, A. H., Jacobson, S. A., Protz, J. M., and Frechette, L. G., 2000, "Shirtbutton-Sized Gas Turbines: The Engineering Challenges of Micro High Speed Rotating Machinery," *Proceedings of the 8th International Symposium on Transport Phenomena and Dynamics of Rotating Machinery (ISROMAC - 8)*, Organized by Pacific Center of Thermal-Fluids Engineering, and Texas A&M University, Mechanical Engineering Department, March, pp. 1-11.

Gardner, J. F., 1999, "Development of a High Speed, High Temperature Compressor Discharge Seal," Paper No. AIAA 99-2684 of the 35th AIAA/ASME/SAE/ASEE Joint Propulsion Conference and Exhibit, Los Angeles, June 20-24.

Gardner, J., 2008, "Concepts for noncontacting, low-leakage, compliant, annular internal seals for pumps," Exeter, Rhode Island, Private Communication, October.

Gopalakrishnan, S., 1986, "Power Recovery Turbines for the Process Industry," *Proceedings of the Third International Pump Symposium*, Turbomachinery Laboratory, Texas A&M University, College Station, Texas, pp. 3-11.

Guinzburg, A. and Buse, F. W., 1998, “Computer Simulation of the Flowpath in Magnetic Sealless Pumps,” *Proceedings of the Fifteenth International Pump Users Symposium*, Turbomachinery Laboratory, Texas A&M University, College Station, Texas, pp. 1-9.

Karassik, I., Messina, J., Cooper, P., and Heald, C., 2008, *Pump Handbook*, Fourth Edition, New York, New York: McGraw Hill.

Miller, A., 2008, “Data on a variable-geometry hydraulic power recovery turbine built at the Flowserve Pump Division,” Taneytown, Maryland, Private Communication, November.

Nelik, L. and Cooper, P., 1984, “Performance of Multi-Stage, Radial-Inflow Hydraulic Power Recovery Turbines,” ASME Paper 84-WA/FM-4.

“Pump Hydraulics—Advanced,” 1999, Paul Cooper, Short Course 8, Sixteenth International Pump Users Symposium and Short Courses, Turbomachinery Laboratory, Texas A&M University, College Station, Texas.

“Pump Life Cycle Costs: A Guide to LCC Analysis for Pumping Systems,” 2001, (Team of Editors), Published jointly by the Hydraulic Institute, http://www.pumps.org/, and Europump, http://www.europump.org/.

Ross, C. C. and Banerian, G., 1956, “Some Aspects of High Suction Specific Speed Inducers,” *ASME Transactions*, *78*, pp. 1715 et seq.

Sloteman, D. P., 1994, Private Communication regarding the Executive MS in Technology Program at the University of Pennsylvania: Degree earned in 1994.

Sloteman, D. P., 2001, “Compact Sealless Screw Pump,” U.S. Patent No. 6,241,486 B1, June 5.

Sloteman, D. P., 2008, Private Communication. Curtiss-Wright Engineered Pump Division, Phillipsburg, New Jersey, December.

Sloteman, D. P. and Piercey, M., 2000, “Developing Sealless Integral Motor Pumps Using Axial Field, Permanent Magnet, Disk Motors,” *Proceedings of the Seventeenth International Pump Users Symposium*, Turbomachinery Laboratory, Texas A&M University, College Station, Texas, pp. 53-67.

Sloteman, D. P., Buse, F. W., and Piercey, M., 2000, “Centrifugal Pump with an Axial-Field Integral Motor Cooled by Working Fluid,” U.S. Patent No. 6,012,909, January 11.

Sloteman, D. P., Cooper, P., and Dussourd, J. L., 1984, “Control of Backflow at the Inlets of Centrifugal Pumps and Inducers,” *Proceedings of the First International Pump Symposium*, Turbomachinery Laboratory, Texas A&M University, College Station, Texas, pp. 9-22.

Sloteman, D. P., Robertson, D. A., and Margolin, L., 2004, “Demonstration of Cavitation Life Extension for Suction-Stage Impellers in High Energy Pumps,” *Proceedings of the Twenty-First International Pump Users Symposium*, Turbomachinery Laboratory, Texas A&M University, College Station, Texas, pp. 103-115.

Smith, J. S. and Watson, A. P., 2006, “Design, Manufacture, and Testing of a High Speed 10 MW Permanent Magnet Motor and Discussion of Potential Applications,” *Proceedings of the Thirty-Fifth Turbomachinery Symposium*, Turbomachinery Laboratory, Texas A&M University, College Station, Texas, pp. 19-24.

Tsujimoto, Y., Origuchi, H., and Qiao, X., 2005, “Backflow from Inducer and its Dynamics,” *Proceedings of the ASME Fluids Engineering Division Summer Meeting and Exhibition*, June, Paper FEDSM2005-77381.

Turbomachinery Research Consortium (TRC), http://turbolab.tamu.edu/research/researchConsortiumTRC/default.aspx.

Warnick, C. C., Mayo, H. A., Jr., Carson, J. L., and Sheldon, L. H., 1984, *Hydropower Engineering*, Prentice-Hall, pp. 16-21.

Wood, D. W., Hart, R. J., and Marra, E., 1998, “Application Guidelines for Pumping Liquids That Have a Large Dissolved Gas Content,” *Proceedings of the Fifteenth International Pump Users Symposium*, Turbomachinery Laboratory, Texas A&M University, College Station, Texas, pp. 91-98.