CENTRIFUGAL PUMP HYDRAULICS FOR LOW SPECIFIC SPEED APPLICATIONS

by

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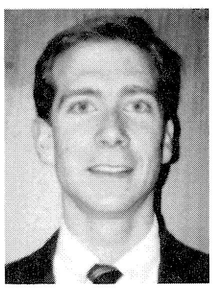

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ABSTRACT

The hydraulic coverage in the low specific speed region has been ignored by most centrifugal pump manufacturers because of unfavorable hydraulic characteristics, and manufacturing obstacles such as small, difficult to cast hydraulic passages. A test program was developed to address the characteristics of curve shape, head coefficient, pump efficiency, and impeller axial thrust. A number of impeller and volute collector designs were tested in an attempt to optimize these performance characteristics. The combination of a high solidity, patent-pending impeller design and a constant velocity volute design provided low specific speed hydraulics with a constant rise-to-shutoff head characteristic, competitive pump efficiency, and low impeller axial thrust.

INTRODUCTION

Centrifugal pumps for the American Petroleum Institute (API) market have consistently utilized radial and Francis type impeller designs to satisfy the head and flow requirements of these pumping applications. These impellers, when run at synchronous motor speeds, are typically applied with design specific speeds above 500. Specific speed, or N_s, is defined as:

$$N_s = \text{rpm}^*(\text{gpm})^{.5}/((\text{ft tdh})^{.75}) \qquad (1)$$

where rpm is the pump speed in revolutions per minute, gpm is the best efficiency point (BEP) flowrate of the pump in gallons per minute, and ft tdh is the total developed head of the pump in feet at BEP.

The hydraulic coverage in the low Specific Speed (approximately less than 500 N_s) region has been ignored by virtually all centrifugal pump manufacturers. The following reasons contribute to the lack of centrifugal pump options in this region:

- *The hydraulic characteristics are unfavorable.* High parasitic losses, such as impeller disk friction and wear ring leakage, significantly decrease overall pump efficiency. The flow *vs* head characteristic is also prone to a "non-rising" curve toward shutoff. This can result in system stability problems at flowrates near the drooping portion of the head *vs* flow characteristic of the pump.
- *Low specific speed hydraulics require small hydraulic passages.* The dimensional repeatability of these passages are usually inconsistent using commercial metal casting technology.
- *A conventional product line of pumps requires a multitude of impeller and casing combinations.* Since hydraulic coverage follows a logarithmic relationship, the same number of pumps are needed to cover the range of 10 to 100 gpm as needed to cover 100 to 1000 gpm. A significant outlay in tooling and part inventory is thus required to support a relatively small portion of the hydraulic market.

APPLICATIONS REQUIRING LOW SPECIFIC SPEEDS

Conventional Low Specific Speed Applications

Consider the hydraulic coverage afforded by a representative line of single stage, direct drive overhung process pumps shown in Figure 1. Each envelope on the chart indicates a different pump size and is marked by the location of the peak or best efficiency point (BEP). For example, the envelope on the lower left side of Figure 1 is identified as "1 × 7" indicating a nominal one inch discharge pipe diameter and a seven inch nominal impeller diameter. The BEP conditions are 80 gpm and 150 ft TDH at 3500 rpm pump speed. The specific speeds used in the hydraulic range of Figure 1 vary from about 400 to 2300, covering approximately 100 to 1000 gpm and 100 to 700 ft of total developed head.

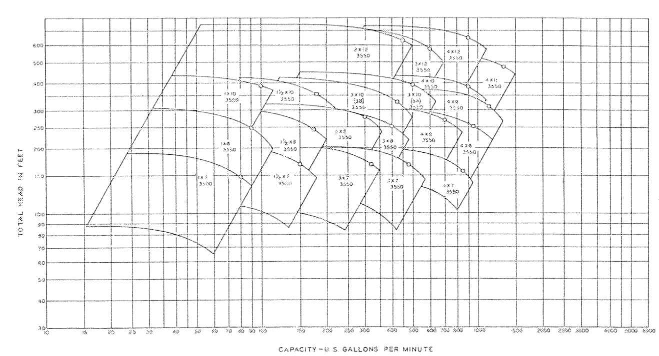

Figure 1. Hydraulic Coverage of a Typical Line of Direct Drive, Single Stage Process Pumps.

One of the simplest and least expensive overhung process pumps is of the vertical in-line design depicted in Figure 2, with a closed impeller mounted with wear rings and rigidly coupled to a vertical motor. Each pump shares parts wherever possible, but at a minimum, a unique casing and impeller design is used for each pump size. Unfortunately, two factors make the design characteristics of a conventional single stage, overhung, closed impeller pump line unsuitable for low specific speed applications. First, the closed impeller and fixed casing geometry require very small and difficult to cast hydraulic passages. Secondly, to adequately support the targeted low specific speed range, a new series of pump sizes would be needed. The log-log plot in Figure 1 shows about twenty pump sizes used to cover the 100 to 1000 gpm flow range. Thus, a comparable number of pumps would be required for the 10 to 100 gpm flow range. Since a conventional closed impeller pump line is limited by these two factors, a unique design approach is necessary for a low specific speed product line.

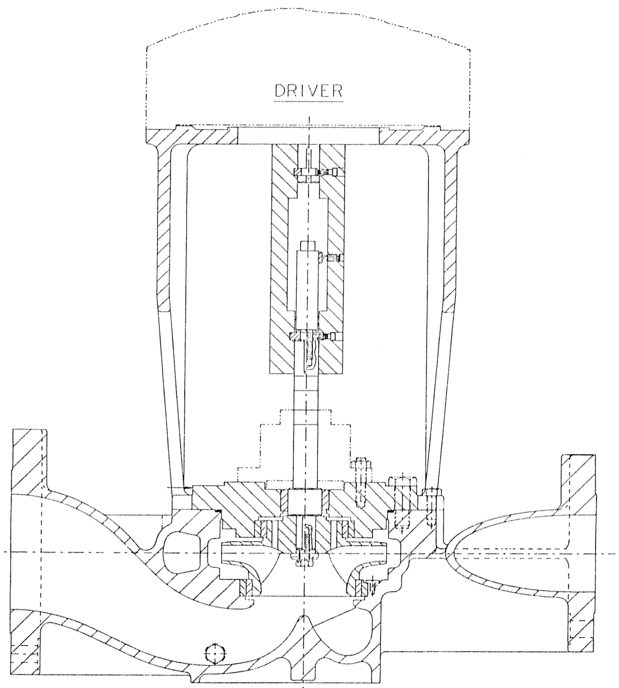

Figure 2. Cross-Section of a Typical Vertical Inline Process Pump for API Service.

Hydraulic Development Needs

A hydraulic development program was conducted to specifically address the drawbacks of low specific speed hydraulics. The three primary features of the plan are presented below.

Unfavorable Hydraulic Performance Characteristics

Low specific speed hydraulics have been largely avoided because of non-rising curve shape characteristics and low efficiency. A recent study [1] indicated that a rising curve shape could be achieved with high solidity (large blade number) designs. Consequently, a number of designs of increasing blade number were planned to validate their effect on curve shape and efficiency. The study would include variations in inlet and exit impeller blade angles to measure their effect on performance.

Impeller Axial Thrust

Historically, low specific speed hydraulics utilize semi-open impellers to simplify manufacturing limitations at the expense of greater hydraulic axial thrust potential. This type of impeller, characterized by a thin width and large diameter cross-section, is prone to significant pressure differences between the front and back of the impeller shroud, manifesting itself in a significant axial thrust. Consequently, tests were required to minimize the effect of the resultant axial thrust without adversely effecting pump efficiency.

Multitude of Impeller/Casing Combinations

As mentioned, the need for small hydraulic passages virtually eliminates the opportunity to use cast closed impellers and cast casing volutes. One alternative to conventional casting technology is precision machined hydraulic parts from wrought or near net shape components. This technique ensures accurate hydraulic passage dimensions in contrast to the significant variations which are often found in the small passages of sand castings. Thus, the subsequent test program was organized primarily to investigate the hydraulic performance and axial thrust issues. The choice of semi-open impellers and precision machined casing volute passages, instead of their alternative sand cast counterparts, was not challenged as part of the test program.

LOW SPECIFIC SPEED HYDRAULIC TESTING

Radial Impeller Design and Testing

Philosophy of Barske Hydraulics

A literature search revealed that one of the first discussions of low specific speed impeller applications was done by Barske [2, 3]. His research was predominantly motivated by the benefits of achieving high speed operation (>3600 rpm) to satisfy the needs of low flow and moderate/high head operations. In so doing, he discovered the inherent benefits of radial, straight bladed, open impeller design.

A typical radial impeller of the Barske and high solidity type is shown in Figure 3. The term "Barske" is used to classify impellers with straight radial impeller vanes. This design is usually comprised of vanes with inlet angles, β_1, and exit angles, β_2, equal to 90 degrees. The BETA angle (β) is the angle described by the tangent vector on the surface of the impeller vane relative to the vector pointed in the circumferential direction at the same point on the impeller vane. The label, "high solidity" refers to a

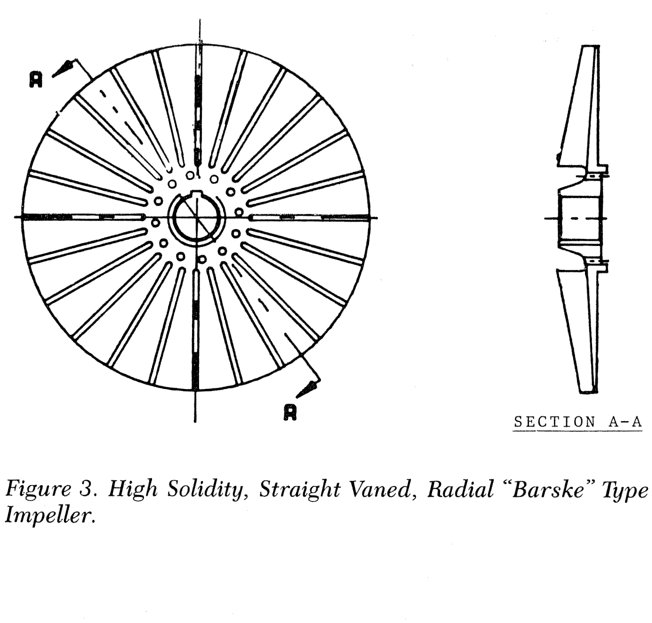

Figure 3. High Solidity, Straight Vaned, Radial "Barske" Type Impeller.

larger than normal number of impeller vanes. Solidity is the ratio of total blade length to impeller circumference, where

$$\text{Solidity} = (L)(Z)/(\pi D_2) \qquad (2)$$

where L is the impeller vane length, Z is the number of impeller vanes, and D_2 is the outside impeller diameter. Barske recommended between two and six impeller vanes for adequate performance [2], but current art shows that the vane number, often greater than 20, has a significant positive effect on performance.

Interestingly, Barske did recognize the unconventional design practice he was professing by stating [2], "To a skilled designer the pump which forms the subject of this paper will, at first glance, appear most unfavorable and may well be regarded as an offense against present views of hydrodynamics."

It was this unorthodox design that led to the notion of a "partial emission" pump, referred to as a P.E. pump by Gravelle [4], which is particularly well suited for low specific speed applications. A partial emission pump is distinguished by its tall, straight radial impeller vanes fit inside a concentric bowl casing. A "full emission" pump, designated as an F.E. pump, is a more typical design and is characterized by its closed impeller with backward swept vanes and constant velocity volute casing.

The P.E. concept introduced by Barske provided important insight into the potential for this type of design in the low specific speed hydraulic range. The general lack of published design data on the P.E. pump made it imperative to pursue a study of this type of design.

Test Rig Design

A modular test rig was designed and built to permit rapid testing of a multitude of hydraulic combinations (Figure 4). The test rig featured provisions for testing a variety of hydraulic variables within a single unit. In particular, the following options were available on the test rig:

- Multiple impeller designs
- Multiple casing insert designs
- Multiple inducer designs
- Six "bolt-on" suction bell designs
- Two "bolt-on" discharge nozzle designs
- Variable suction bell to floor clearance
- Variable impeller to casing wall clearance

Due to the great number of test variations available with this test rig, the subsequent discussions in this paper are limited to variations of the casing insert and impeller.

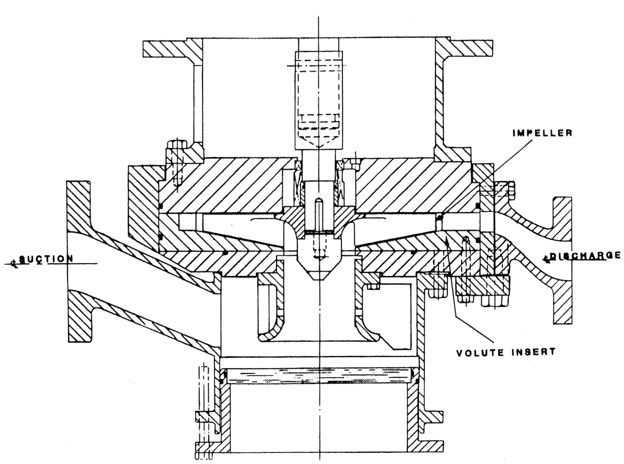

Figure 4. "Modular" Test Rig with Provisions for Interchangeable Hydraulic Components Such As Impellers and Volute Inserts.

Phase I—Impeller Testing

The Phase I impeller tests were designed to measure the effect of solidity, blade number, and blade angle. This was accomplished by testing four alternate impeller designs in the same test rig with the same casing design. The common cross section shown in Figure 5 is of the impellers used in Phase I tests. The basic dimensional data used in each of the impeller designs [A], [B], [C], and [D] are reflected in Table 1. A number of common variables exist between each of the impellers so that the effect on performance of changing an isolated geometric dimension can be compared between two different impellers.

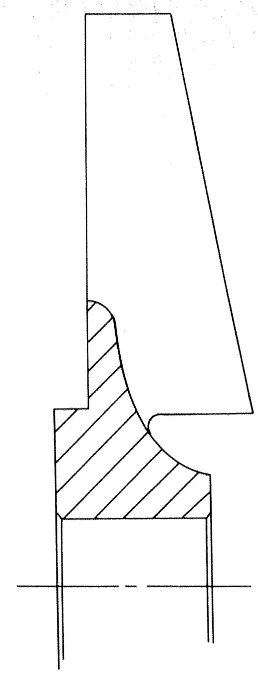

Figure 5. Impeller Cross-Section Used for all Phase I Impeller Tests.

Table 1. Phase I Testing—Impeller Geometric Dimensions.

Impeller	(units)	[A]	[B]	[C]	[D]
D2	(in.)	12.00	12.00	12.00	12.00
D1	(in.)	3.63	3.63	3.63	3.63
Z	(–)	10	14	16	24
Beta,1	(–)	32	90	90	90
Beta,2	(–)	45	45	90	90
L	(in.)	6.63	4.75	4.13	4.19
Solidity	(–)	1.76	1.76	1.76	2.67

The performance curve of flow *vs* head is shown in Figure 6 for each of the four impeller designs. All of the flow *vs* head characteristics show a "nonrising" curve between the best efficiency flow and the shutoff flow with the peak TDH occurring at about 40-50 percent of the BEP flow. Significant differences in the amount of total developed head generated by each impeller did exist, however.

The performance indices for each test are shown in Table 2. These indices, when compared with the dimensional data in

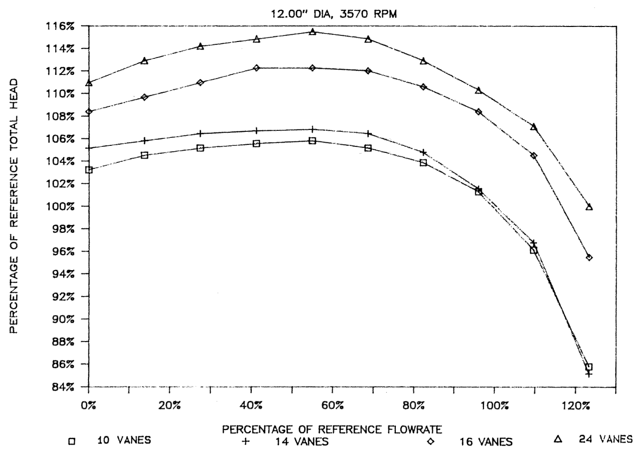

Figure 6. The Effect of Impeller Vane Number on the Flow Vs Head Characteristic for the four Phase I Impellers.

Table 1, offer insight into the effect of various hydraulic parameters on overall performance. For example, impellers [A] and [B] have the same vane solidity of 1.76 and exit blade angle of 45 degrees, but a different blade number and inlet blade angle. The best efficiency point is virtually identical for the two impellers but the [B] impeller lost 1.5 points in efficiency compared to the baseline [A] impeller. The shut-off head of the [B] impeller was also higher than the [A] impeller.

Table 2. Performance Data.

Impeller	(units)	[A]	[B]	[C]	[D]
Speed	(rpm)	3576	3577	3568	3569
Q,bep	(gpm)	365	370	385	400
H,bep	(feet)	775	775	820	835
H,so	(feet)	800	815	840	860
PSI,bep	(–)	0.71	0.71	0.76	0.77
PSI,so	(–)	0.73	0.75	0.77	0.79
N,s	(–)	465	468	457	460
NPSHR @ 365gpm	(feet)	27	29	30	35
NPSHR @ BEP	(feet)	27	30	32	39
N,ss	(–)	5768	5368	5203	4574
Delta Effy	(–)	X%	X%−1.5%	X%−4%	X%−1.5%

Impeller [B] differs from [C] in both blade number and discharge blade angle but the inlet angle and vane Solidity remain the same. A notable increase in head coefficient is seen as PSI,bep increases from 0.71 to 0.76. Note that PSI,BEP is defined as,

$$\text{PSI,BEP} = gH/U_2^2 \qquad (3)$$

where g is the gravitational constant, H is the TDH of the pump, and U_2 is the tip speed at the exit vane of the impeller. A substantial drop in overall efficiency, 2.5 points relative to the [B] impeller, accompanies the increase in head. This drop in efficiency may be attributed to the aggressive 90 degree exit angle which promotes the generation of more head but with greater impeller hydraulic losses.

Interestingly, test results [A], [B], and [C] support the orthodox view of the "skilled designer" that higher exit angles and blade numbers tend to incur additional separation and friction losses, respectively, within the impeller resulting in lower overall pump efficiency. Test [D], however, reverses the trend. The only difference between test [C] and [D] is the blade number. The BEP head coefficient rose from 0.76 to 0.77 and the shutoff (S.O.) head coefficient rose at a higher rate from 0.77 to 0.79. The overall pump efficiency of [D] improved 2.5 points compared to impeller [C] in this case even though the blade number was increased from 16 to 24.

The suction performance of each impeller is also given in the NPSHR data shown in Table 2. The NPSHR at the BEP flowrate of each of the impellers is shown as well as the NPSHR at 365 gpm, which is the flowrate of the baseline [A] impeller BEP. The data indicates a trend of increasing NPSHR from impeller [A] through [D] caused by the additional blockage at the eye of the impeller with the increasing blade number. Suction specific speed, N_{ss}, is also shown in Table 2 and is defined like N_s (Equation (1)) except TDH is replaced by NPSHR. A decreasing N_{ss} is especially apparent when utilizing higher blade numbers such as impeller [D] which requires 30 percent more NPSH than the baseline [A] impeller. This low N_{ss} performance highlights the need for impeller inlet modifications to reduce the NPSHR of the high solidity designs.

Thus, the Phase I impeller tests indicate that a straight radial bladed impeller does have potential as an alternative to a conventional backward swept impeller design. The trend indicates that even higher efficiencies and head coefficients can be achieved with higher solidities. An added benefit of the [D] impeller is the reduction in impeller diameter required to meet the same head and flow condition. For example, an 11.50 in diameter [D] impeller can generate the same amount of head as a 12.00 in [A] impeller. By correctly sizing the casing for an 11.50 in diameter impeller, one might expect a comparable efficiency between the [A] and [D] impeller due to the associated reduction in disk friction with the smaller impeller diameter.

Impeller Axial Thrust Testing

Phase II—High Solidity Impeller Axial Thrust Testing

The Phase I impeller tests indicated that high blade numbers had a positive effect on head coefficient, curve shape, and overall pump efficiency. However, a continuously rising curve was not achieved with any of the four basic impeller designs. The goal of the Phase II testing was to take advantage of the characteristics of high solidity impellers to achieve (1) a continuously rising head characteristic while (2) simultaneously reducing the hydraulic axial thrust and (3) improving the NPSH performance.

A new concept radial bladed impeller (Impeller [E]) was designed to address these issues. Geometric changes in the impeller design successfully disposed of issues (1) and (3). Regarding issue (2), further axial thrust related development was required.

As discussed, the semi-open impeller design of impeller [E] is preferred over a closed impeller design for ease of casting small hydraulic passages and for greater machining flexibility. The primary drawback is the increased impeller axial thrust potential of the semi-open impeller, especially with a full back shroud design. To summarize the nature of hydraulic axial thrust, it is caused by the pressure difference between the front and back sides of the impeller. When an impeller is closed (has a front shroud), the pressure distributions are nearly the same on the front and back shroud from the impeller outside diameter down to the wear rings. On semi-open impellers, the pressure difference can be significant since the nature of the pressure distribution on the front and back of the shroud is different.

In order to predict the impeller thrust of a semi-open impeller, a pressure distribution must be assumed on the fore and aft side of the back impeller shroud and integrated over the effected shroud area. For tall, high solidity radial bladed impellers, the total static pressure rise can be modeled as a function of the impeller blade by assuming a forced vortex distribution inside the impeller from the inlet tip to the outside diameter. The pressure at the back of the impeller is assumed to be governed by a forced vortex distribution, also. Classic tests have shown that the forced vortex for a fully back shrouded impeller is one-half the rotational speed of the impeller [5].

To understand the significance of impeller thrust, consider the theoretical thrust generated by a semi-open impeller similar to [E]. Assuming a forced vortex distribution is acting on the front of the impeller shroud and a one-half speed forced vortex distribution is acting on the back of the impeller shroud, a calculated theoretical thrust of 8200 lb is anticipated. (The details of the hydraulic thrust calculation are beyond the scope of this presentation.) This indicates that a typical medium thrust vertical motor rated for a two year life (17500 hr) with a thrust load of 2000 lb would have a life rating of just over 250 hr with a thrust of 8200 lb.

In an effort to reduce the significant axial thrust which exists on a semi-open impeller, a test program was initiated to reduce its effect. The method of lowering the thrust in the Phase I impellers was by "scalloping" the back shroud. This involves the removal of portions of the shroud area such that a pressure difference cannot occur across the shroud. An alternative approach is the use of balance holes to promote radial flow down the back of the impeller, and hence, decrease the net thrust on the back of the impeller [6]. This was the approach investigated in Phase II of the test program, whereby the axial thrust was measured for different balance hole configurations.

Balance Hole Testing

A new design Specific Speed of 270 was selected for the [E] impeller using another 12 in impeller diameter and a new casing volute insert. The first configuration of impeller [E], identified as [E1] and "Balance Hole 1," had a balance hole configuration commonly found on commercial semi-open impellers located radially inward of the vane tips. The performance test resulted in a curve shape much steeper than the first four impeller tests as shown in Figure 7. As anticipated though, the measured axial thrust was higher than desired for the typical medium thrust in-line vertical motor. The measured axial thrust of 8310 lb reflected in Table 3 correlated well with the 8200 lb predicted with the axial thrust model described earlier.

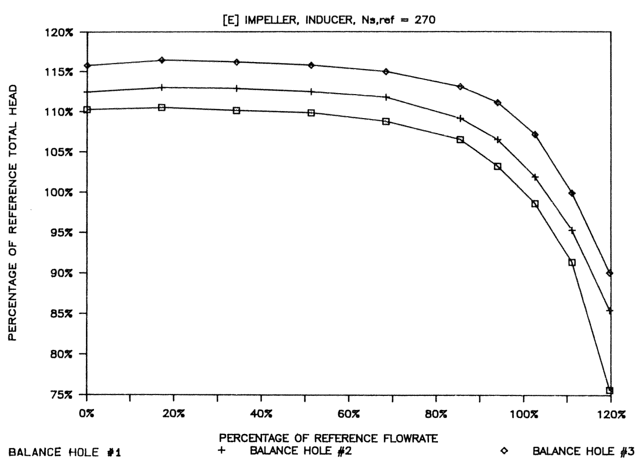

Figure 7. The Effect of three Different Phase II Impeller Balance Hole Configurations on the Flow Vs Head Characteristic.

Two alternative balance hole configurations to the standard [E] impeller were designed and identified as impeller [E2] using "Balance Hole 2" and impeller [E3] using "Balance Hole 3" configurations, respectively. The "Balance Hole 2" and "Balance Hole 3" configurations simply utilize larger balance hole areas, compared to "Balance Hole 1." The composite head *vs* flow characteristics for each of the 3 [E] impeller tests are shown in Figure 7. A lower axial thrust was expected and achieved in each successive test but, ironically, the overall head coefficient also increased with each test. More significantly, the pump efficiency did not decrease as a consequence of the balance hole modifications. In conclusion, by simply modifying the balance hole configuration, the [E3] impeller achieved a higher head coefficient, better curve shape, and a fraction of the axial thrust of the [E1] impeller without sacrificing efficiency.

Table 3. Phase II Testing—Measured Axial Thrust.

Impeller	Measured Axial Thrust (lbs)	Error (%)
[E1]	8310	8
[E2]	4880	7
[E3]	1160	38
[E4]	1960	9

Staggered Balance Hole Test

A variation of the [E3] impeller was designed, called the [E4] impeller, which utilized a "Staggered" Balance Hole arrangement. The test results of this design are shown in Figure 8 and compared to the original [E1] impeller. A dramatic improvement in curve shape and rise-to-shutoff is evident between the [E1] and [E4] impeller. A reduction in axial thrust of more than four times was also achieved bringing the total impeller axial thrust within acceptable motor bearing load limits.

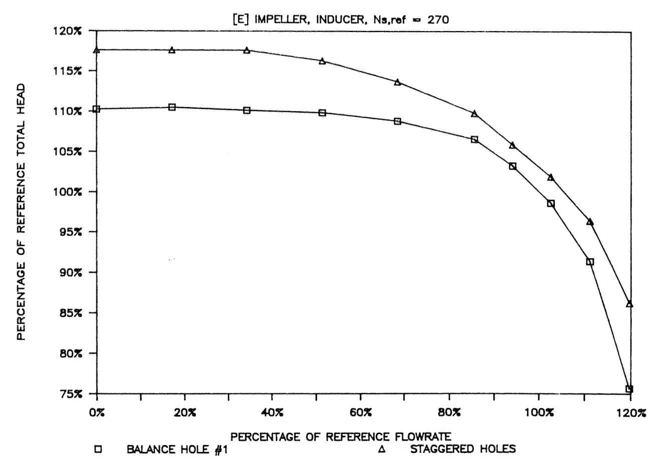

Figure 8. Performance Comparison Between the Standard "Balance Hole 1" Vs the "Staggered Hole" Impeller Configurations.

[E4] vs *[D] Vane Impeller Comparison*

A comparison test was made between the new [E4] impeller and the 24 bladed [D] impeller design tested during Phase I. The [D] impeller was modified to have the same profile dimensions as the [E4] impeller such that both impellers could be tested with identical test rig hardware. The head *vs* flow characteristic is shown in Figure 9 indicating more desirable performance characteristics from the [E4] impeller. The continuously rising curve shape and the rise-to-shutoff of about 17% is rather

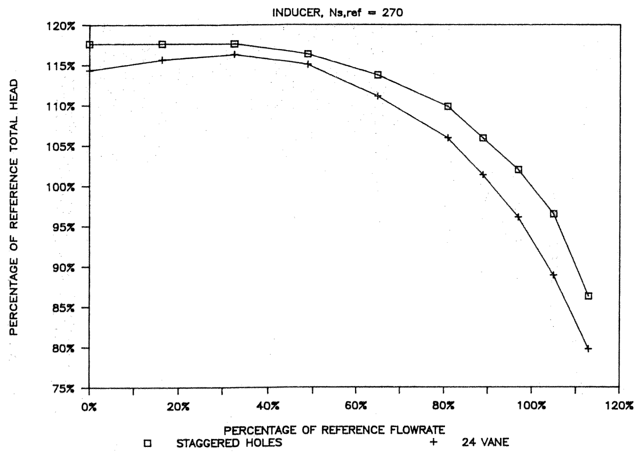

Figure 9. Performance Comparison Between the Staggered Balance Hole [E4] Impeller and the 24 Vane [D] Impeller.

good for a pump with an N_s of less than 300. A two point efficiency gain was also realized with the [E4] *vs* the [D] impeller. Although a 270 N_s version of the original [A] impeller configuration was not tested, the conclusion at the end of the Phase I testing was that the [D] impeller operates with a comparable efficiency to the [A] impeller for a given flow, head, and speed. Thus, one can assume that the new [E4] impeller would perform with a higher efficiency than the [A] impeller at 270 N_s.

Note that the balance hole testing identified in Figures 7, 8, and 9 were each conducted using an identical inducer. This was done primarily to establish an improved level of NPSHR in light of the high NPSHR levels experienced during the Phase I tests. Since the inducer will alter the shape of the head *vs* flow curve, a comparison of the [E4] impeller operating with and without an inducer is shown in Figure 10. The inducer increases the total developed head of the pump throughout the flow range with a slightly greater head rise occurring near low flow. The rising curve shape of the [E4] impeller is still achieved even after removing the inducer. Also, the new design had less inlet blockage, and even without the inducer this had a positive effect on NPSH performance. The [E4] impeller N_{ss} of 10,540 compares favorably to the [D] impeller N_{ss} of 4620 (for the N_s = 270 hydraulics), both without an inducer.

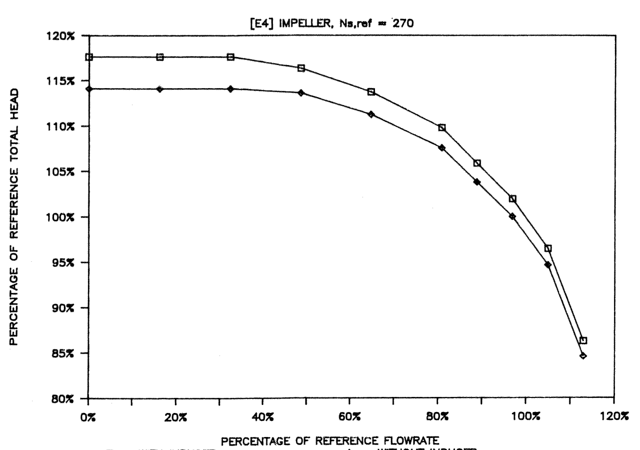

Figure 10. Performance Comparison Between the Staggered Balance Hole [E4] Impeller Operating with and without an Inducer.

Patent Pending [E4] Impeller

While attempting to reduce the impeller axial thrust of a semi-open impeller, serendipity resulted in an impeller design which addressed many of the drawbacks of low specific speed centrifugal hydraulics. Primarily, a continuously rising head curve was attained with a high solidity, high head coefficient impeller design. The design also achieved acceptable commercial levels of axial thrust and NPSHR. The unique blade configuration and staggered balance hole arrangement eventually led to the declaration of a patent application and ultimately a patent-pending impeller design.

Casing Collector Testing

Hydraulic designers have argued the merits of various casing collector designs on overall pump performance. In the low specific speed region, little hydraulic information exists on collector designs. The trade-off has been primarily the choice between a P.E. type of design utilizing a Barske impeller mated with a concentric volute, or the F.E. design using backward swept impeller vanes and a constant velocity (CV) volute casing. The CV type of volute is characterized by an inner volute (or casing) wall which grows proportionally from the cutwater tip around the circumference and into the diffuser inlet. This type of volute design is typically used in a specific speed range above 500-600. The concentric casing is characterized by a volute inner wall which is concentric with the impeller, and is usually applied in a specific speed range less than 500-600 without any unusual loss in performance. The more cost effective collector is the circular volute design when compared to a more conventional constant velocity volute. Since all of the Phase I and Phase II testing utilized the more conventional CV volute, a comparison test between the two alternatives was required to evaluate the performance differences and ultimately justify the more cost effective design.

A performance comparison is shown in Figure 11 of a circular volute and a constant velocity (CV) volute design. Both tests utilized the same test rig, the same [E] design impeller, and similar volute throat areas. One can see that the two options vary distinctly in overall performance. The CV volute design had a lower peak efficiency flow and head but achieved a more desirable rise-to-shutoff ratio of about 15 percent. The circular volute design peaked at a higher flow and head and achieved a one point efficiency advantage. However, the falling head curve at flows below about 60 percent of the BEP flow is undesirable and usually unacceptable. The preferred design is the CV design,

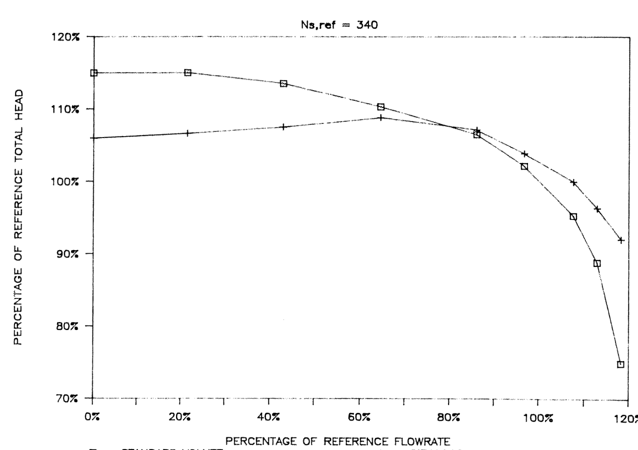

Figure 11. Performance Comparison Between a Standard CV Type Volute with a Circular Volute, Each with the Same Impeller.

simply because it is functionally superior over the entire head and flow range of the pump curve.

DISCUSSION

A test program was developed to find a solution for many of the problems which have prevented greater applications of low specific speed hydraulics by manufacturers in the API market. Phase I of the program was a survey of impellers, all of the semi-open design, which were tested to compare efficiency, curve shape and head coefficient. The trend was for higher head coefficients as a function of higher blade number but all designs exhibited undesirable "drooping" flow *vs* head characteristics.

Phase II testing concentrated on gaining higher efficiencies, improving curve shape and suction performance, and reducing the high axial thrust inherent with semi-open impellers. A design was ultimately developed which exhibited a constantly rising curve, maintained a high head coefficient at a comparatively high pump efficiency, and limited axial thrust to acceptably low levels.

Additional testing concentrated on the effect of casing volute design on curve shape and pump efficiency. Circular volute *vs* standard volute options were tested and revealed a better performance characteristic when using the standard volute design.

CONCLUSIONS

Thus, this test program addressed three areas which previously limited pump applications using low specific speed hydraulics. First, high blade number semi-open impellers used with standard machined volute designs are good alternatives to current >500 specific speed designs running at off-peak conditions. The efficiency is higher than the comparable off-peak alternatives and a rising curve shape is achievable. This is also a better alternative to the Barske "partial emission" hydraulic concept which does not exhibit the same high rise-to-shutoff characteristic. Second, the casting difficulties are avoided by simple radial bladed semi-open impellers and accurately machined casing volutes. Third, an entire low specific speed line can be designed by machining matched sets of impellers and volutes from a minimum number of raw castings and wrought blanks. This permits the rated conditions to be set at or near the design condition of the pump hydraulics optimizing curve fit and efficiency.

NOMENCLATURE

Symbol	*Description*
BEP	best efficiency point
beta	Impeller vane angle
D	impeller diameter
g	gravitational constant
gpm	pump flowrate, gallons per min.
H	pump total developed head
L	length of one impeller vane
N_s	specific speed
N_{ss}	suction specific speed
$NPSH_R$	net positive suction head required
PSI	head coefficient
Q	pump flowrate
rpm	pump speed, rev. per minute
tdh	pump total developed head
U	peripheral impeller velocity
Z	number of impeller vanes

Subscripts

1	at the impeller inlet
2	at the impeller exit
u	in the tangential direction
bep	at bep
s.o.	at shutoff

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