AIR MODEL TESTING TO DETERMINE ENTRANCE FLOW FIELDS

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ABSTRACT

The designer of centrifugal pump impellers requires knowledge of the velocity flow field entering the impeller for several reasons. These include an assurance that a reasonably uniform field exists, that the leading edge vane angles are matched to the flow field, and that the amount of prerotation affects the pump head.

Some velocity distributions resulting from various inlet configurations can be calculated by various techniques. However, since these techniques assume mostly axisymmetric, non-viscous flow under ideal conditions, the real velocity distribution caused by side inlets truly remains unknown until an experimental program is instituted to probe the flow field and quantitatively describe the velocity distribution.

The experimental technique of determining entrance flow fields of pumps having side inlets as opposed to axial inlets is described in this paper. The procedure of building models of the flow path preceeding the impeller and testing these models with air as the working fluid is discussed. A description of how the flow field is probed with three-dimensional velocity probes to quantitatively "map out" the full 360° velocity flow field, as well as flow visualization procedures to qualitatively give insight into the flow patterns is described in detail. A case history of the influence of the inlet flow on NPSH tests follows the discussion of the experimental methods.

INTRODUCTION

Of uppermost importance to the centrifugal pump designer is the quantitative knowledge of the flow field existing at various locations along the pump internal flow path. The first location of importance is at the entrance to the pump impeller. This paper refers to a plane perpendicular to the shaft centerline in the unvaned space (cross-under) in the impeller entrance eye area as the P-plane and is considered as the entrance to the impeller. The location of the P-plane is shown in Figure 1. This paper addresses the method of experimentally determining the flow field in the P-plane from model tests using air as the working fluid [1,2].

Various design considerations are discussed for designing the models for use in determining these flow fields from experiment. Model configurations for single stage single suction pumps as well as single stage double suction pumps (both with side inlets) are discussed.

The experimental techniques used to determine the velocity flow fields with three-dimensional velocity probes are presented. A qualitative procedure, using flow visualization to show how the flow fields are generated, is also presented. A procedure to determine model inlet losses is discussed. The last section of the paper is a case history of the influence of the inlet flow fields on NPSH tests.

DESIGN CONSIDERATIONS FOR DETERMINING IMPELLER ENTRANCE FLOW FIELDS

Model Design Considerations

Certain design considerations should be evaluated in determining the model configuration used for air model tests. In all cases, the model flow channels should duplicate the actual geometric shape of the pump flow channels that exist from the inlet flange to the impeller and should also include the impeller hub and shroud contours. It is assumed that the flow entering the actual pump inlet flange is uniform and has no vorticity. Therefore, the model is tested with a uniform, vortex free flow free field at the model inlet flange. Actual pumps are sometimes installed in piping systems that have a tortuous path preceeding the pump. A tortuous path can set up flow patterns that are not uniform, and which can contain irregular, pulsatile flow fields. It is recommended that care is exercised in field installations to ensure that a uniform, vortex free flow enters the pump inlet flange.

The configuration of the model is influenced by the actual pump. The complete pump inlet is modelled for a single stage side inlet pump configuration. The model contains the complete inlet flange (full circle) and the flow channel leading to the impeller, including the impeller hub and shroud contours as shown in Figure 1. The hub and shroud curvatures have an effect of the P-plane flow field in addition to the upstream inlet design. The P-plane is the plane perpendicular to the shaft centerline in the unvaned space (cross-under) in the impeller entrance eye area.

In the case of a single stage double suction pump configuration, only one side of the pump inlet is modelled. The model can consist of one half of the inlet flange (half circle) and the flow channels leading to the impeller, including the impeller hub and shroud contours.

Model Size and Construction Considerations

The size of the model is determined by calculating the pump Reynolds number at critical areas, such as the inlet flange and the P-plane and comparing them with the model Reynolds number at these same critical areas. Since models are normally run with air as the working fluid, the Mach number should be kept less than .3 to keep compressibility effects to a minimum. The pump and model Reynolds numbers should be compared to ensure that both are operating in the same flow regime which in most cases is the turbulent regime. As a general note, the model size factor (model size: pump size) should be kept at some simple size ratio such as 1:2, 1:3, 1:4, 1:10 for ease of model construction.

In construction of the model, the materials for construction are selected based on the type of model studies to be conducted. If only velocity traverses using three dimensional probes are planned, with potentially some surface flow pattern visualization, models constructed of wood are satisfactory and are easily constructed at minimal cost. However, if complete flow visualization of the entire flow channel is required, the model can be constructed of transparent plastics. Transparent models are required when using smoke for tufts for flow visualization or when using laser-doppler anemonmetry for determining velocity fields. Models constructed of transparent plastics are relatively expensive for pump inlets because of the three dimensional curved surfaces comprising the inlet.

EXPERIMENTAL TECHNIQUES

Quantitative Tests

When testing the models with air as the working fluid to quantitatively determine the flow fields at various locations along the pump internal flow path, the tests should be run with Mach numbers less than 0.3 so that the compressibility effects are insignificant. Also, the test Reynolds numbers should be as close as possible (but not higher) to the actual pump operating Reynolds numbers which is in the turbulent range for most pumps. Through the use of three-dimensional probes, an excellent determination of the flow field can be made.

This discussion is confined to the P-plane, as shown in Figure 1. In the process of planning for the test, the P-plane should be mapped out to show the exact location of each data point with its assigned number for use in test data reduction by a computer. This is also exemplified in Figure 1.

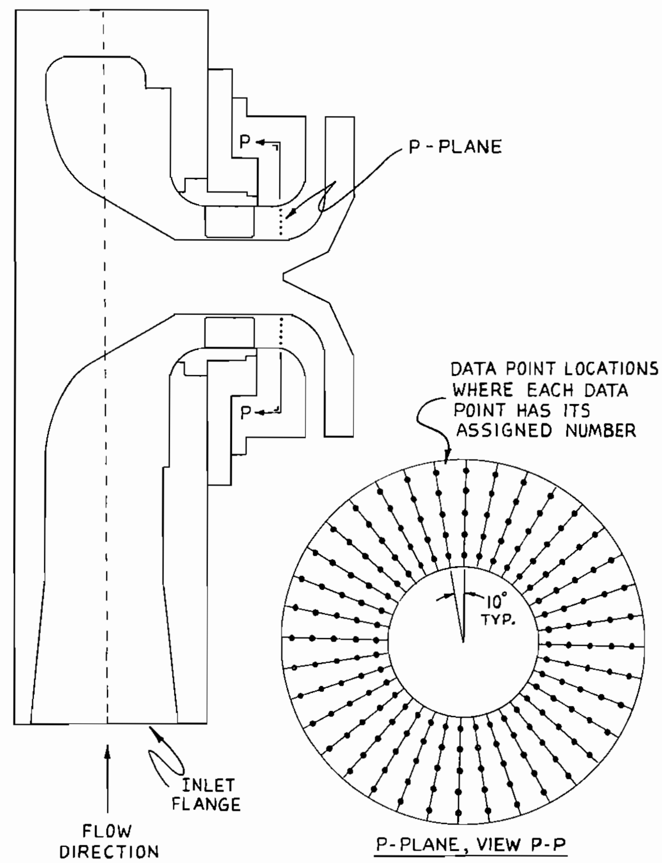

Figure 1. Typical Model Configuration of a Single Suction Side Inlet Pump.

The mass flow based on the test Mach number must be determined before the test. The quantitative description of the velocity field in the P-plane can be determined using three-dimensional velocity probes. These probes must be calibrated by the supplier. The data taken by these probes must be corrected based on calibration characteristics and blockage effects before the velocity vector can be calculated for each data point. This task is easily completed through the use of a computer with all probe calibration characteristics stored in its memory. The same computer program splits the velocity vector into meridional, tangential, and radial components for documentation and further processing.

An example of the three-dimensional probes used for this study is shown in Figure 2. The probe contains five pressure tap measuring locations that, combined with calibration characterisitics, give the total and static pressures for the velocity calculation, as well as yaw and pitch angle determination.

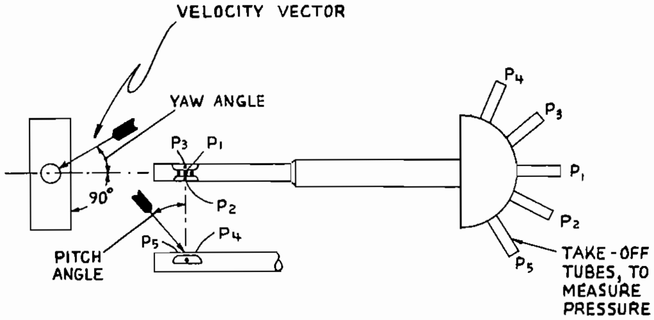

Figure 2. Typical Three-Dimensional Velocity Probe. Used for Transversing the Flow Fields.

The studies discussed in this paper did not use the advanced, more complicated method of laser-doppler anemometry for velocity field determination. This advanced technique is gaining popularity but is very complicated, time consuming, and expensive to use. It is not indicated for use in these relatively simple flow fields.

Qualitative Tests

At certain times in a test series, when it is desirable to know how the flow field is developing upstream of and through the flow measuring plane, it is very advantageous to use flow visualization techniques to get a visual picture of the flow. Although tufts and smoke can be used, both of which require a transparent model, a very valuable technique with wooden models is by the use of lampblack and oil. The surfaces of the flow passage are first coated with a mixture of lampblack and oil. The model is then run and disassembled for observation. The lampblack and oil will give a streaked patern on all the flow surfaces, which is an indication of how the flow is behaving on the flow channel surfaces. This gives insight into the flow pattern development and suggests changes for a more desirable distribution. An example of a flow visualization test using a coating of lampblack and oil is shown in Figure 3.

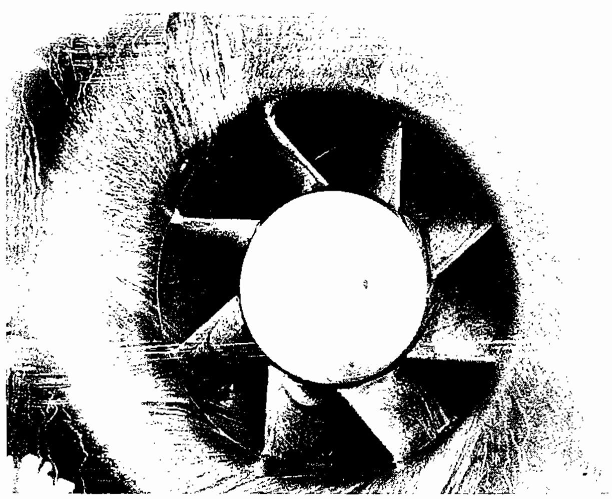

Figure 3. An Example of a Flow Visualization Test Using a Coating of Lampblack and Oil, Showing the Inlet Cascade Used in the Tests.

INLET LOSSES

Additional valuable information can be easily obtained about inlet losses if the static pressure is measured at the model inlet, and the flow going through the model is measured from which the velocity head and, hence, the total head can be determined. The model is instrumented with static pressure taps at the inlet to measure the inlet static head, H_{SI}. Using the inlet area of the model, A_I, and the model flow rate, Q_I, the average inlet velocity V_I can be calculated as follows,

$$V_I = Q_I/A_I \qquad (1)$$

Using V_I, the velocity head, H_{VI}, at the model inlet can be determined from,

$$H_{VI} = V_I^2/2g \qquad (2)$$

where g is the acceleration of gravity. The total head, H_{TI}, at the model inlet is,

$$H_{TI} = H_{SI} + H_{VI} \qquad (3)$$

The total head at the P-plane can be determined by an integration procedure. This procedure mass averages the total head, H_T, in the P-plane to obtain the mass averaged total head, H_{TP},

$$H_{TP} = \frac{\int H_T \cdot \gamma \cdot CM \cdot dA}{\int \gamma \cdot CM \cdot dA} \qquad (4)$$

where,

H_T is the local total head
γ is the local density
CM is the local meridional velocity
dA is the differential area

The inlet loss coefficient, LC, is determined from the equation,

$$LC = \frac{H_{TI} - H_{TP}}{H_{VI}} \qquad (5)$$

The loss coefficient is a measure of the inlet losses and, in turn, gives a measure of the expected pressure loss between the pump inlet and the impeller P-plane.

THE INFLUENCE OF THE INLET FLOW FIELD ON NPSH TESTS: A CASE HISTORY

The Pump

An impeller was designed to operate under low NPSH conditions. This was a single suction impeller in the first stage position of a multistage pump rotor. The pump case was equiped with a side inlet, as is customary with utility type boiler feed pumps. The inlet section immediately before the impeller was equipped with a cascade that provided prerotation (inlet swirl) to the flow. The general arrangement of the inlet is shown in Figure 1.

An NPSH test was performed that gave generally good results, although the locations of the optimum NPSH did not occur at the desired flow. The test was carried out at 4,000 RPM and 130°F. Figure 4 shows the results of this test. The normalized pump head and efficiency are shown covering the standard performance range. The NPSH is represented by the 0 percent, 1 percent and 3 percent head drop curves. In addition, the breakdown or critical NPSH is also shown by the "C" curve. At 0.7 Qn (where Qn is rated flow) there is a rise in all dropoff curves except for the break-down curve. The best NPSH point is located at 0.90 Qn and yields a suction specific speed (SS) of 13,100 at 1 percent head drop and 9,975 at 0 percent head drop, which are good SS values. Beyond Qn all NPSH curves rise sharply. At 1.25 Qn and above, no amount of available plant NPSH (NPSHA) would satisfy this impeller.

As noted before, the best NPSH was found at 0.90 Qn, but was expected at 1.05 Qn. Although this is not a big flow difference, it was decided to investigate why this flow shift existed.

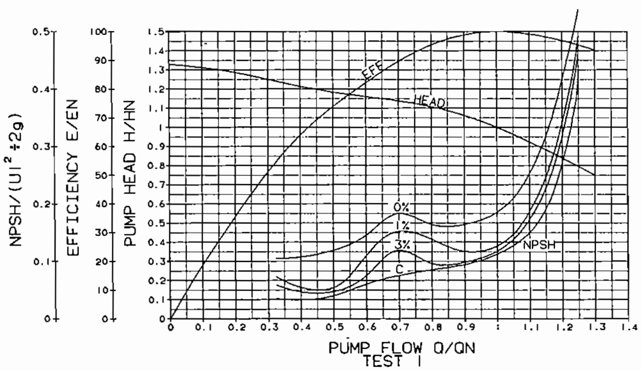

Figure 4. Normalized Pump and NPSH Performance, Test 1.

The Quantification of the Impeller Inlet Flow

A wooden model was built of this inlet to determine the impeller entrance flow field. First, a flow visualization test was made by applying lampblack and oil to internal contours of the model. This test indicated the existence of "horseshoe" vortices at the inlet cascade as shown in Figure 3.

The P-plane flow field was then measured on five stream surfaces. The velocities in the outermost, the center, and the innermost stream surface are shown in Figure 5. It shows that the flow was not axisymmetric. The absolute velocity varied excessively. Small areas of reverse flow also occurred. In addition, there were wide variations of the flow angle. In short, the flow field was not satisfactory.

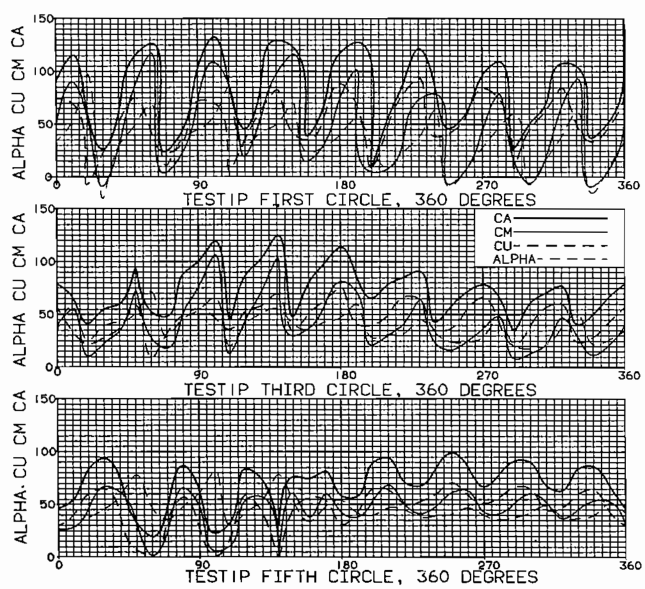

Figure 5. Absolute, Meridional, and Tangential Velocity and Absolute Flow Angle in the First, Third, and Fifth Circle of the P-Plane, Test 1.

The averaged values of the meridional velocities CM, the tangenitial velocities CU, and the radial velocities CZ at the P-plane location are shown in Figure 6. The component velocities were integrated on circles over the stream surfaces and then normalied with the length of the particular stream surface. A large radial velocity component CZ exists which emphasizes the unsatisfactory flow field at the P-plane. We have already noted in Figure 5 that there is a severe variation of all the component velocities around the streamline surface, so one should not be influenced by the apparent "calmness" of the averaged picture.

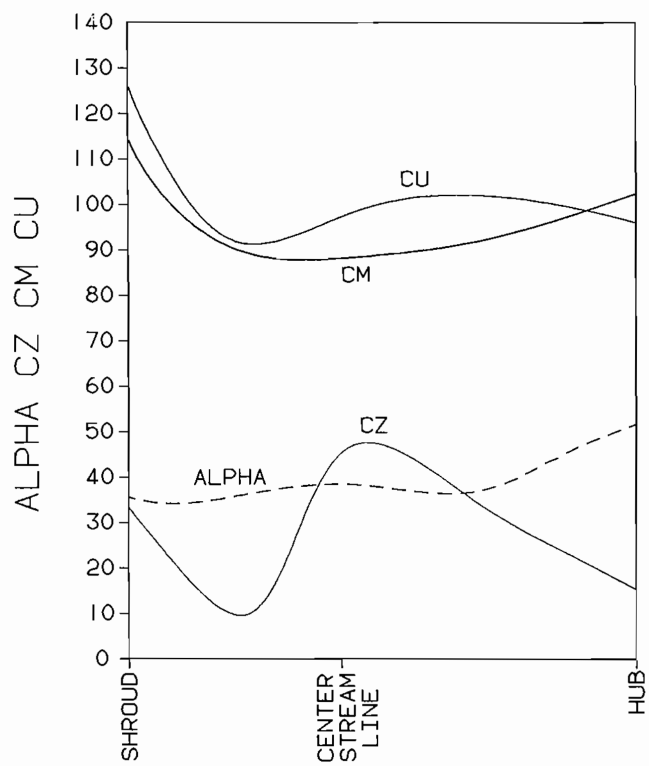

Figure 6. Averaged P-Plane Flow Field, Test 1.

Some further properties of the flow field are displayed in Figure 7. First, observe the quadrant flow in the P-plane. Consider the quadrant flow in the P-plane. Consider the quadrant locations 1, 2, 3, and 4. Ideally, 25 percent of the P-plane capacity should flow through each quadrant. Then, the quadrant deviation would be 0 percent. Note that the deviations in quadrants 1 and 3 are low, but the deviation in quadrants 2 and 4 are high, i.e., 28 percent in quadrant 2. The quadrant flow picture gives another "feel" of a poor flow distribution. Secondly, the relative flow angles and the relative velocities at the outermost stream surface are displayed. The average flow angle is 10.8° and compares well with the local vane inlet angle of 11°. It is interesting to note that the best NPSH area is at the flow where the average flow field matches the inlet vane angle.

A location where the minimum relative flow angle is negative and, in fact, some backflow also occurs as is revealed in Figure 7. At this location, a flow angle deviation of $-12.7°$ exists with respect to the vane leading edge. At another location, a maximum flow angle deviation of $+13.7°$ exists. These flow angle deviations are relatively large.

The flow fields that might exist in the inlet of a boiler feed pump having an adverse flow distribution are described in Figures 5, 6, and 7. The air test revealed why the best NPSH occurred at 0.90 Qn. The main reason for this was the existence of "horseshoe" vortices in the inlet cascade which distorted the flow in the P-plane.

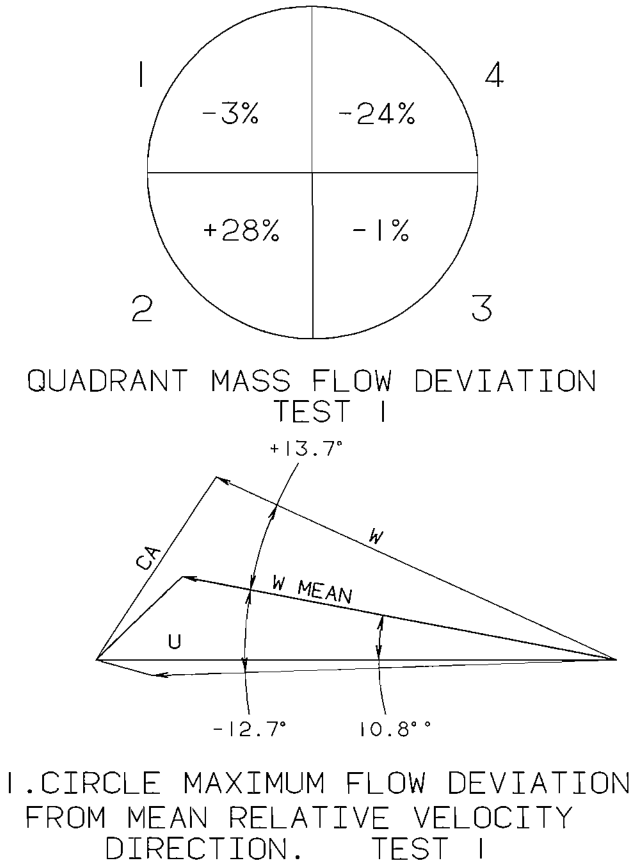

Figure 7. Quadrant Flow Errors and Maximum, Mean and Minimum Relative Velocity in the P-Plane, Test 1.

The Implication

An excellent cavitation test was obtained even though the relatively poor inlet flow field existed. It should be noted that the flow angle deviations shown in Figure 7 are so large that local cavitation can appear at these locations. Experience has shown that local flow disturbances can cause spot cavitation damage.

For this reason, it is our strong conviction that an air test should always precede the design of a large engineered pump with an inlet that could cause flow distortion. In the European pump community, it is frequently necessary to make visual cavitation tests on large pumps. This is done by cutting ports into the pump through which the cavitation can be examined through stroboscopic light. The permissible cavitation cavity length at the leading edge that occurs under a certain suction pressure is then measured and used as a means of specifying the required NPSH over the whole flow range. However, this method is not failproof. Cavitation damage can still occur at a location having a local flow disturbance which is not being observed. This is due to the fact that the stroboscopic light can only cover a small region around the impeller inlet.

The Second Test

The pump inlet was then modified to produce a better flow field. This was accomplished through the installation of a central baffle extending from the inlet flange area to the cascade area. This baffle effectively divides the model inlet into two symmetrical halves. It was expected that this would reduce the quadrant flow deviation. As a result of this measure, the second air test is considerably improved in comparison to the first test, but it is by no means perfect.

The NPSH test with the modified inlet is shown in Figure 8. The best NPSH area had moved out to 1.05 Qn, where the 1 percent drop represents an SS of 10,200.

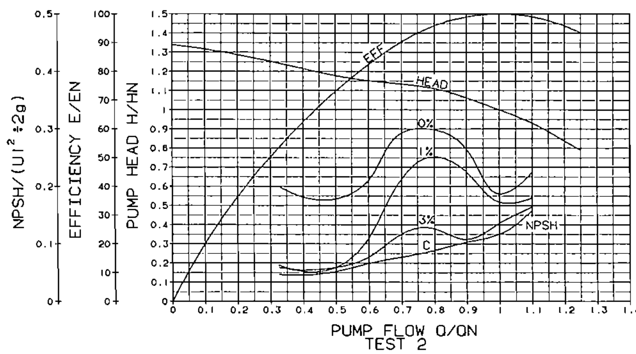

Figure 8. Normalized Pump and NPSH Performance, Test 2.

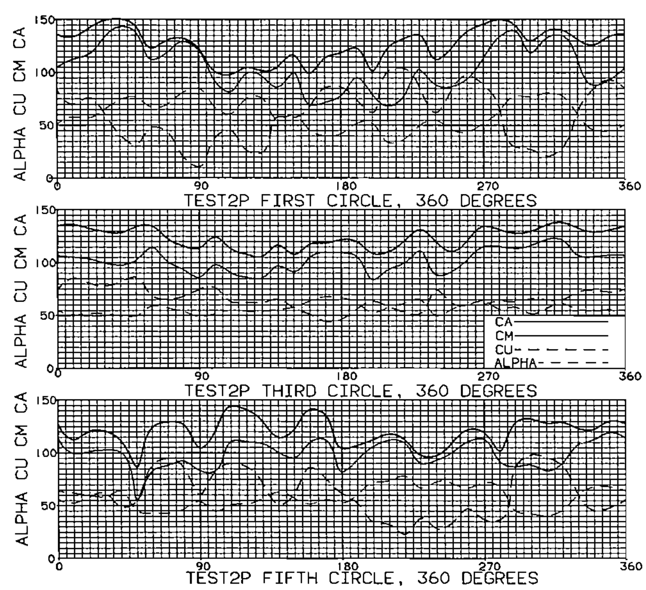

Figure 9. Absolute, Meridional, and Tangential Velocity and Absolute Flow Angle in the First, Third, and Fifth Circle of the P-Plane, Test 2.

The velocities on three stream surfaces at the P-plane are shown in Figure 9. The flow is still not axi-symmetrical, but more uniform than in the first test. The averaged values of the meridional, tangential, and radial velocity components are shown in Figure 10. In addition, the averaged flow angle is shown. The resulting meridional velocity distribution is the result of the tangential velocity distribution over the P-plane. This is a specialized case of a so called Beltrami flow named after the Italian mathematician Beltrami, who first solved certain cases of this flow type.

The quadrant flow deviation, the relative velocities, and the flow angles at the P-plane are represented in Figure 11. The results are also better than in the first test. The highest mass flow variation is in quadrant 4 with an 8 percent deviation. The average relative flow angle is again very close to the vane angle for the best NPSH point, which occurs at a higher flow, 1.05 Qn. Again, the flow location of the best NPSH was determined by the matched flow angle. The deviation angles are now $-3.5°$ and $+3.6°$ for the minimum and maximum flow angle in regard to the leading edge angle.

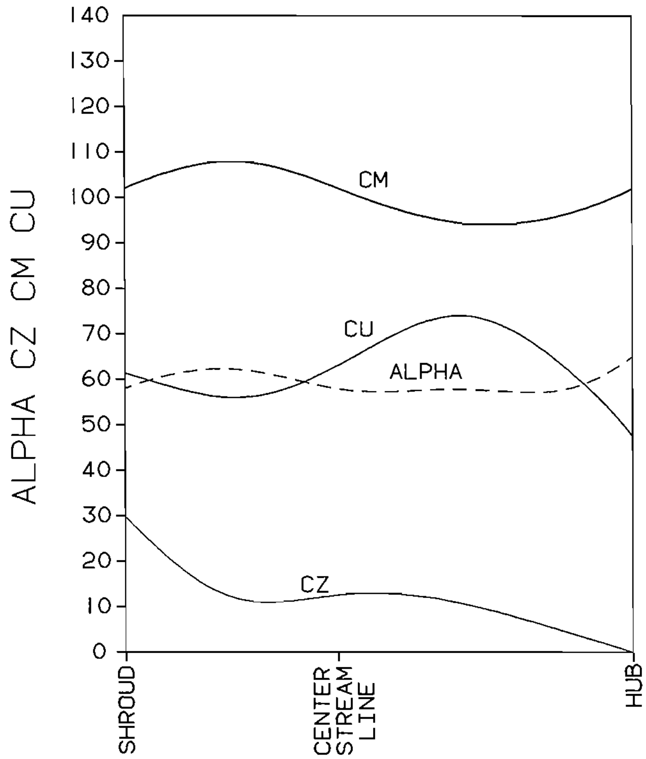

Figure 10. Averaged P-Plane Flow Field, Test 2.

This development project is continuing at this time. The above discussion indicates the importance of having suction impellers operating with a reasonably good inlet flow field.

Additional Uses of Model Air Test

In this test series, the inlet cascade was designed to produce prerotation. With the air test, the tangential and the meridional velocities are obtained. From these two velocity components, the leading edge vane angle can be calculated at the shroud in the P-plane. Similarly, the flow field can be evaluated at the center streamline leading edge and near the hub leading edge. This, in turn, yields the vane angles at those locations.

Also, the overall loss coefficient for the pump inlet can be obtained. This is useful for the pump component loss calculations and the NPSH prediction. The loss coefficients in these tests are referenced to the P-plane area. For the first test, we obtained an LC of 6.98. For the second test, the LC was 1.04, which is a tremendous improvement.

Local loss coefficients at the P-plane were also obtained. The LC in the outermost streamtube was 3.63 for the first test and 0.80 for the second test. The losses to the outer streamtube at the P-plane are 23 percent less than the overall losses. This is a confirmation that prerotation can reduce NPSH requirements of a pump.

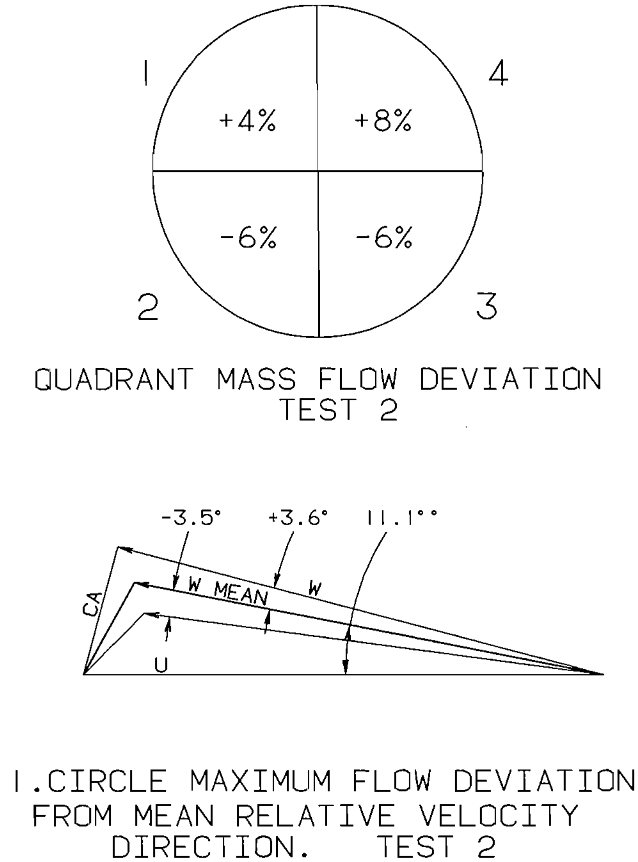

Figure 11. Quadrant Flow Errors and Maximum, Mean, and Minimum Relative Velocity in the P-Plane, Test 2.

CONCLUSION

Inlet flow field air tests of large engineered pumps offer a multiple bonus. The quality of the flow field can be evaluated in regard to uniformity, shape of vortex, and non-linearity of the meridional velocity, quadrant flow and deviation of the flow angles. Loss coefficients can be obtained. Any newly designed engineered large pump with a side inlet should have a model air test performed to assure that no major flow deviation exists.

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