Experimental Characterization of Two-Phase Flow Centrifugal Pumps

Abstract
In this experimental work, the pressure distribution was measured in a rotating, partially shrouded, open, radial impeller, inlet diffuser and volute under a wide range of air-water two-phase flow conditions. To obtain these pressure measurements, small-diameter pressure-tap holes were drilled through the casing of the radial pump. High speed photography was the vehicle to determine the flow regime of the air-water mixture through the vane and in the volute. An analytical model was developed to predict the radial pump single- and two-phase flow pressure distributions. The distribution for the latter was compared with the test data for different suction void fractions. The physical mechanism responsible for pump performance degradation was also investigated.

Previous Pump Test Programs
Experimental data have been taken by the General Electric (G-E) Company in steady-state, high-pressure, steam-water mixtures [1, 2]; by the Babcock and Wilcox Company in air-water testing [3]; and by the Aerojet Nuclear Company in testing on the Semiscale Centrifugal Pump [4, 5]. In addition, the Electrical Power Research Institute (EPRI) supported two additional pump-research projects. One project, entitled “Phenomenological Understanding and Modeling of Pumps”, was undertaken at Creare Inc. It involved performing an experiment on a 1/20-scale model pump, which was tested in air-water and steam-water flow [6, 10]. The other EPRI-sponsored project, executed at Combustion Engineering (C-E) Power Systems, was entitled “Two-Phase Pump-Performance Program” [8]. This involved the testing of a larger, 1/5-scale, model pump in steam-water flow.

Two earlier areas of interest have also contributed to the knowledge of pump two-phase flow performance. The first area is in pumped hydropower generation and storage, which usually involves radial-flow machines [7, 8, 9, 10, and 11]. Cavitation in hydropower machines has also been widely studied in these references.

Semi-Empirical Correlations
Based on the tests conducted to date, various empirical curves were constructed to characterize the pump performance operating under two-phase flow conditions. Such curves include those of Babcock & Wilcox [3], Semiscale (see Creare [6]), C-E [8], and Creare [7]). All these curves are basically similar in that the head and this expressed by only one parameter, i.e., α, the inlet void fraction.

<table>
<thead>
<tr>
<th>Company</th>
<th>Scale</th>
<th>Pump Type</th>
<th>Specific Speed</th>
<th>Test Conditions</th>
<th>Flow Rate (LPM)</th>
<th>Speed (RPM)</th>
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<tr>
<td>B&amp;W</td>
<td>1/3</td>
<td>Mixed</td>
<td>4317</td>
<td>Air/Water</td>
<td>42297</td>
<td>3580</td>
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<td>Byron Kastner</td>
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<td>6705</td>
<td>Steam-Water</td>
<td>1.1-92.7 bar</td>
<td>11905</td>
</tr>
</tbody>
</table>

Table 1. Review of past two-phase pump testing programs.

Zakem [16], as well as Hench and Johnston [17], used a one-dimensional control volume method for two-phase diffuser flow simulations. Zakem applied the method to rotating machinery and identified...
a non-dimensional parameter in a simple way.

Furuya [14] used the same method proposed by Zakem and developed an analytical method, that revealed several features related to the mechanism of two-phase pump head degradation. His analytical model incorporated void fraction, pump impeller geometry, slip factor, and flow regime into the continuity and momentum equations for the gas-and-liquid flow through the impeller.

The approach undertaken by MIT [15] was based on the single-phase ideal Euler’s head equation coupled with various empirical head-loss data for both single- and two-phase flow media. In this work, a head loss ratio parameter was developed.

**Single-Phase Pump Impeller Theory**

Let us create a control volume around the rotating pump impeller under the platform of one dimensional flow. We can apply the basic laws for this control volume for the case of steady flow with no heat transfer and no chemical reaction.

1. **Continuity**:
For steady one-dimensional flow at the inlet and outlet of the vane, we can write:

$$\rho_1(V_1)A_1 = \rho_2(V_2)A_2$$

where $A_1$ and $A_2$ are the vane inlet and outlet areas, respectively. For incompressible flow Equation (1) becomes

$$\rho_1(V_1)_n A_1 = \rho_2(V_2)_n A_2 = Q$$

2. **Moment of momentum**:
It is very useful to use the moment of momentum equation taken around the axis if the shaft, which for steady flow leads to

$$\oint_{CS} \overline{rT_\theta} dA + \iint_{CV} rB_\theta dV = \iint_{CV} \rho V \cdot dA$$

where:
- $T_\theta$ - torque transmitted through the control surface
- $r$ - radius of the infinitesimal control volume
- $B_\theta$ - body forces (vector)

$V$ - velocity of the control volume (vector)
$V_0$ - fluid directional velocity

Appreciable shear stress occurs at the inlet and outlet surfaces and torque ($Tqid$) is transmitted through the control volume. The one-dimensional ideal torque for inlet and outlet flow is given by:

$$Tqid = -\overline{r(V_\theta)_1}[\rho_1(V_1)_n A_1] +\overline{r_2(V_\theta)_2}[\rho_2(V_2)_n A_2]$$

For incompressible flow, it reduces to:

$$Tqid = [r_2(V_\theta)_2 - r_1(V_\theta)_1] \rho Q$$

where:
- $Tqid$ - ideal torque
- $V$ - fluid absolute velocity (1 in, 2 out)
- $V_\theta$ - fluid directional velocity (1 in, 2 out)
- $Q$ - fluid volumetric flow
- $r$ - impeller radius (1 inlet, 2 outlet)

3. **First law of thermodynamics**:
For steady flow with one-dimensional inlet and outlet flows and no heat transfer this is described as follows:

$$\frac{V_1^2}{2} + gz_1 + h_1 - \frac{dW_s}{dm} = \left(\frac{V_2^2}{2} + gz_2 + h_2\right)$$

For incompressible flow this is reduces to:

$$\frac{V_1^2}{2} + gz_1 + \frac{p_1}{\rho} \cdot \rho Q - \frac{dW_s}{dt} = \frac{V_2^2}{2} + gz_2 + \frac{p_2}{\rho} \cdot \rho Q$$

By combining the first law of thermodynamics with the moment of momentum, Equation 5 can be written ($M_\omega = -dW_\theta$) as

$$\frac{dW_s}{dt} = [\overline{r_1(V_\theta)_1} - \overline{r_2(V_\theta)_2}] \cdot \rho Q \omega$$

Now substituting Equation 8 into Equation 7 and rearranging, this is gives:

$$-\rho g \Delta H = [\overline{r_1(V_\theta)_1} - \overline{r_2(V_\theta)_2}] \cdot \rho Q \omega$$
$C_D = \text{drag coefficient (0.13 for turbulent flow)}$

During that the wheel peripheral velocity is $U_2 = \omega \cdot r_2$, therefore, $T = \frac{m}{g_0} (U_2 \cdot \omega) C_{\theta 2}$, or:

$$\omega T = \left( \frac{m}{g_0} \right) \cdot U_2 \cdot C_{\theta 2}$$

where $C_{\theta 2}$ is the impeller tangential discharge velocity and $\omega T$ is the total energy input (power) to control volume which appears entirely in the energy increase in energy of the fluid between the inlet and outlet. For the case of no energy dissipation in the impeller and single phase-fluid, the energy increase per unit mass flow can be described as:

$$\Delta E_{\text{vane}} = \frac{\Delta p}{\rho} = \frac{\Delta p}{\rho} + \frac{1}{2g_0} \Delta C^2$$

where:

- $\rho$ = total pressure
- $p$ = static pressure
- $C$ = absolute velocity
- $\Delta$ = change between the inlet and the outlet

Equating $m \cdot \Delta E$ to the expression for the total power input $\omega T$, results in

$$m \cdot \Delta E = \omega \cdot T_{\text{qid}} = \frac{m}{g_0} U_2 C_{\theta 2}$$

$$\Delta E = \frac{U_2^2}{g_0} \cdot C_{\theta 2}$$

Thereby, Equations 16 and 17 can be combined and rearranged, giving

$$\Delta P_{\text{vane}} = \rho \left( \frac{U_2}{g_0} C_{\theta 2} + \frac{C_2^2 - C_1^2}{2g_0} \cdot \frac{r_2}{r_1} \right) + C_1$$

From Figure 1, the velocity vector diagram at the impeller discharge yields the relation,

$$C_{\theta 2} = \sigma_2 \cdot U_2 - C_{m2} \cdot \tan(\beta_2)$$

where:

- $\beta_2$ = impeller-discharge -blade angle measured to radial direction
- $C_{m2}$ = radial-discharge velocity
- $\sigma_2$ = impeller slip factor

Consequently by continuity, $Q = C_{m2} \cdot A_2$

where:

\[ \]
\[ Q = \text{total volumetric flow rate} \]
\[ A_2 = \text{impeller-discharge flow area} \]
then:
\[ \Delta P_{\text{vane}} = \rho \left[ \frac{U_2}{g_0} \left( \sigma_2 U_2 - C_{m2} \tan \beta_2 \right) + \right. \]
\[ \left. \frac{1}{2g_0} \left( C_2^2 - C_1^2 \right) \right] \frac{r_2}{r_1} + C1 \]

Alternatively by defining the head rise coefficient of the vane:
\[ \Delta P_{\text{vane}} = \frac{U_2^2}{2g_0} \Psi \] (20)

**Pressure Rise at the Pump Suction**

Figure 2 shows a straight pipe prior to an impeller of radial pump. At the suction station AB,

![Figure 2. Hydraulics and energy gradient along the suction pipe [18].](image)

the pressure \( p_1 \) is uniform through the cross section of the pipe. At section CD the pressure \( p_2 \), as measured at the pipe wall, is higher than the one at section AB where the pressure profile is paraboloid, due to rotation of the stream. On the basis of the pre-rotational speed, the centrifugal pressure rise inside the pipe can be calculated by:
\[ \Delta P_{\text{prerot}} = \rho \left( \frac{U_{\text{prerot}}}{2} \right)^2 \frac{4}{1000} \] (21)
Likewise, it can be correlated by:
\[ \Delta P_{\text{prerot}} = A \left( \frac{n}{60} \right)^2 + B \frac{n}{60} \] (22)

where:
\[ A = 0.04525714 \]
\[ B = 0.00342857 \]
\[ n = \text{rotor revolution [rpm]} \]
\[ \Delta P_{\text{prerot}} = \text{centrifugal pressure rise [kPa]} \]

A straight tapered diffuser is the best choice in every respect for single-inlet impeller. Such a diffuser, the area of which gradually increases toward the impeller eye, has a definite steadying effect on the flow and assures a uniform liquid feed to the impeller. The pressure rise through this short distance can be calculated by:
\[ \Delta P_{\text{diff}} = \frac{\left( C_{m1} - \frac{V_{\text{pipe}}^2}{g_0} \right)}{2} \rho \] (23)

\[ V_{\text{pipe}} = \frac{q}{A_{\text{pipe}}} \], \[ C_{m1} = \frac{q}{A_{\text{inlet}}} \] (24)

where:
\[ q = \text{flow rate [m}^3/\text{sec]} \]
\[ A_{\text{inlet}} = \text{impeller inlet area [m}^2 \] [3]
\[ A_{\text{pipe}} = \text{suction pipe cross-section [m}^2 \]

Finally, the pressure rise at the pump suction is:
\[ \Delta P_{\text{suction}} = \Delta P_{\text{diff}} + \Delta P_{\text{prerot}} \] (25)

**Pressure rise in the Volute**

The pump volute casing is determines much of the environment in which the impeller works, and sometimes has a profound effect on the impeller performance.

The flow in the curved channel is governed by equilibrium between centrifugal forces and the pressure gradient by:
\[ \frac{1}{\rho} \frac{d}{dR} \frac{V_T^2}{R_c} \]

Where \( R(\Theta) \) is the inside radius of the curved channel, \( R_c(\Theta) \) is the curvature radius of the channel from the pump axis, and \( V_T(\Theta) \) is the tangential velocity of the
Substituting the variables into Equation 26 and integrating it, the swirl pressure gradient can be written by:

$$ \Delta p_{\text{vol1}} = 7.25 + \log(0.00 \, lb + c) $$

(27)

$$ \Delta p_{\text{vol2}} = 1.84 - \log(2 \pi b + c) $$

(28)

$$ (\Delta p_{\text{vol}})_{12} = (\Delta p_{\text{vol1}} + \Delta p_{\text{vol2}}) \cdot \Gamma + C_1 $$

(29)

$$ \Gamma = \frac{a^2 b \rho}{d^4 c g 10^3} $$

Where:

- $C_1 = 0.048532n^3 + 0.314544n^2 - 58.4971n + K$
- $K = 377.7143$
- $a = q/(2 \pi)$, $q$ - flow rate [m$^3$/sec]
- $b = A_{\text{vo}}/(4\pi)$, $A_{\text{vo}}$ - volute discharge area [m$^2$]
- $c = d_2/2$, $d_2$ - impeller outlet diameter [m]
- $d = \pi d^2 A_{\text{vo}}/(4\pi)$

If it is assumed that the volute throat does not create local disturbances to the flow, the velocity varies according

$$ u = c_{\theta 2} \cdot \frac{r_2}{r} $$

(31)

where $c_{\theta 2}$ is the whirl velocity at the impeller outer radius. The mean velocity in the throat is

$$ c_3 = \frac{1}{r_{\text{vt}}} \cdot \int_{r_2}^{r_{\text{vt}}} u \, dr $$

(32)

that is

$$ c_3 = \left. \frac{\ln(1 + 2r_{\text{vt}}/r_2)}{2r_{\text{vt}}/r_2} \right|_{r_2} $$

(33)

$$ c_{\theta 2} = \frac{gH}{u_2} $$

(34)

The flow in the volute related to the head generated by the volute “characteristic” is

$$ \frac{q}{u_2 d_{\text{vt}}^2} = \frac{gH}{\eta u_2^2} \left( 1 + \frac{2d_{\text{vt}}}{d_2} \right) $$

(35)

And finally the volute characteristic pressure $\Delta p_{\text{vch}}$ is:

$$ \Delta p_{\text{vch}} = \eta_h \frac{\rho}{1000} \left[ \frac{u_2 q}{d_{\text{vt}}^2} \left( 1 + \frac{2d_{\text{vt}}}{d_2} \right) - C_1 \right] $$

(36)

The total pressure distribution in the volute is:

$$ \Delta p_{\text{vol}} = \Delta p_{\text{vch}} + (\Delta p_{\text{vol}})_{12} $$

(37)

The total pressure distribution through the pump is:

$$ \Delta p_{\text{pump}} = \Delta p_{\text{preat}} + \Delta p_{\text{vane}} + \Delta p_{\text{vol}} $$

(38)

where:

- $\eta_h$ - hydraulic efficiency
- $q$ - fluid flow rate
- $\rho$ - fluid specific weight
- $u_2$ - impeller peripheral velocity
- $d_{\text{vt}}$ - volute throat diameter
- $d_2$ - impeller outside diameter
- $C_1$ - speed-dependent constant

**Two-Phase Flow Pump Head Calculation**

The total mass flow rate through the pipe is equal to the sum of the mass flow rates of the gas and the liquid:
\[ m = m_1 + m_2 \]  
(39)

The two phase–flow quality is:
\[ x = m_1 / m \]  
(40)

The void fraction can be calculated by the Lockhart-Martinelli model [20]
\[ \alpha = [1 + 0.28(1 - x)^{0.64}(\rho_g / \rho_l)^{0.36}(\mu_l / \mu_g)^{0.07}]^{-1} \]  
(41)

where:
- \( x \) - two-phase flow quality
- \( \alpha \) - void fraction
- \( \mu_l \) - dynamic viscosity of the liquid
- \( \mu_g \) - dynamic viscosity of the gas
- \( \rho_l \) - specific weight of the liquid
- \( \rho_g \) - specific weight of the gas

The next step is to find a Flow number that can indicate a certain value where the pressure starts to drop across the entrance chamber, the vane, and the volute. This flow number can be cast in a general form,
\[ F = \frac{\rho_{sp}}{\rho_l} \left[ \frac{x}{1 - x} \frac{\rho_l}{\rho_{air}} \right]^{-\alpha} \]  
(42)

where \( \rho_{sp} \) two-phase flow density can be determined by the correlation given by:
\[ \rho_{sp} = \left( \frac{x}{\rho_l} + \frac{1 - x}{\rho_{air}} \right)^{-1} \]  
(43)

The declining effective flow-path-length of the vane is the basic mechanism of the pressure degradation of the pump impeller. The volume of the vane can be calculated by:
\[ Vol = C_i (r_2^2 - r_1^2)xh \]  
(44)

The height of the vane is:
\[ h = (r_2 - r_1) \frac{\tan(\Theta)}{2} + h_2 \]  
(45)

where:
- \( r_1 \) = impeller inlet radius (0.043 m)
- \( r_2 \) = impeller outlet radius (0.1225 m)
- \( r \) = variable radius from \( r_1 \) to \( r_2 \)
- \( h_2 \) = height of the impeller outlet (0.0308 meter)

\( C_i \) = effective volume multiplier (0.74)
\( \theta \) = vane inclining angle from \( r_2 \) to \( r_1 \)
(\( \theta = 11.66^\circ \))

The normalized pressure \( (dpvane / \pi^2) \) will be compared with the Pressure number. The pressure number definition is
\[ P = \frac{dpvane_{sp} Vol \cdot n}{n^2 Q_{meff}} \]  
(46)

where:
- \( dpvane_{sp} \) = single-phase vane pressure rise at the appropriate water flow rate [kPa].
- \( n \) = pump speed [1/sec]
- \( Vol \) = effective vane flow volume [m³]
- \( Q_{meff} \) = the most efficient single-phase flow rate [m³/sec]

Pressure loss prediction in the entrance chamber can be determined with a simple experimental model. When the normalized pressure-versus-Flow number was plotted (Figure 4), the pressure started to drop at 1.4 and become zero at 0.9. Therefore, the pressure in the inlet chamber is equal to the single-phase pressure rise if the Flow number larger >1.4, and equal to zero when the Flow number is less than 0.9. Between 1.4 and 0.9, the pressure drop can be modeled by
\[ \tan \beta = \frac{dp9_{sp}}{(1.4 - 0.9)} \]  
(47)
\[ dp9 = \tan \beta \cdot [F(i) - 0.9] \]  
(48)

where:
- \( F(i) \) =Flow number from 1.4 to 0.9
- \( B \) =angle of pressure loss
- \( dp9 \) =pressure distribution in the inlet chamber

Figure 5 shows the overall characteristics of the vane and volute differential pressure distributions. The trend is similar, therefore, it is logical to use the shape or volume function (Eq. 44) to develop an experimental equation to predict the two-phase flow pressure distribution in the volute as follows,
\[ dpvolum = c_i (dpvolsp) (nvol) (nFn) - c_2 \]  
(49)
Figure 4. Normalized vane and inlet chamber pressure vs. Flow number.

where:
- $c_1$: speed dependent multiplier = $0.00142 \times (n/60) + 0.0018$
- $c_2$: speed dependent constant = $0.864 \times (n/60) - 10.6$
- $n$: speed [rpm]
- $dpvo/ls$ single-phase volute differential pressure
- $nvo$: normalized volume function (Eq. 44)
- $nFn$: normalized Flow number (Eq. 42)

The input parameters required to predict the single- and two-phase flow pump differential pressure distributions are:
- Liquid flow rate [lit/min]
- Gas flow rate [slit/min]
- Pump suction pressure [kPa]
- Temperature of the mixture at suction [$^\circ C$]
- Pump impeller parameters ($r_1[m], r_2[m], \beta_1[rad], \beta_2[rad]$)
- Pump impeller speed [rpm]

The Test Loop

A schematic flow diagram of the stainless-steel test loop is shown in Figure 6. Water flows from the 182 gallon air-water separator tank via a control valve to the second, small, square separator tank mounted near the top of the primary tank, and is discharged to the atmosphere through an air turbine meter. Water accumulating in the tank is periodically drained.

The air-water mixture flows through a baffle plate used to break up and mix the two streams. A twelve-foot long, horizontal, 4-inch-diameter transparent pipe conveys the air-water mixture to the test pump. The two-phase mixture is routed back to the primary air-water separator tank through 3-inch diameter piping. A 3-inch control valve in the pump discharge line is used to establish the required pump inlet pressure at the desired flow rate. The discharge of the air-water mixture into the storage tank allows the water to release the entrained air, and an approximate half-minute settling time before the water enters into the suction pipe of the loop pump. A test-section bypass line and a loop heat exchanger are located in this line. A mixer bypass line is also installed for small test-section flows.

The test pump installation is shown in Figure 7. The horizontal orientation of the pump in the air-water test loop and the relatively long, straight section of the inlet separator tank flows via a control valve to the second, small, square separator tank mounted near the top of the primary tank, and is discharged to the atmosphere through an air turbine meter. Water accumulating in the tank is periodically drained.

Figure 5. Normalized vane and volute differential pressure vs. Flow number.
piping permit both single-phase and stratified two-phase flow to enter the pump with only negligible rotational effect.

The test pump instrumentation consist of standard pump head, torque, and speed measurement, which are used in conjunction with loop measurement to determine overall pump performance.

In addition to the standard pump measurements, special measurements made on the pump are:

1. 14 differential pressure measurements detailing the pressure variation in the impeller and volute region of the pump;
2. five-optical probe ports also giving information in the impeller and volute flow regime of the pump;
3. optical information about the two-phase Flow regime in the Plexiglas pump suction pipe and at the pump discharge;
4. a six-beam, low-energy gamma densitometer mounted to the four-inch pipe size pump suction flange.

An essential requirement in the test pump specification was the use of an open impeller. With such an impeller configuration, both pressure and optical measurement can be made in the impeller flow channels once penetrations are made through the pump casing wall.

Two-Phase Flow Experimental Results and Comparison of Model Predictions

Observation of the flow in both the pump suction and discharge sections through transparent piping lines showed flow regimes dependent principally on void fraction. In the air-water two-phase flow loop, the pump suction line was horizontal and the discharge section configuration was vertical. In the horizontal inlet line, the flow had the tendency to become stratified. In the bubbly flow regime, the bubbles were concentrated at the top of the four-inch transparent pipe. Stratified and wavy flow regimes were clearly separated, having only air at the top and only liquid at the bottom of the pipe. For higher inlet void fraction, the flow appeared to become slug with this effect increasing with higher void fractions. We did not have annular flow regimes.

The changes from one flow regime to another were not, of course, sudden. In bubbly flow, as the bubbles become more numerous, coalescence of smaller bubbles produces larger plug-type bubbles, which flow in the upper portion of the pipe. This is referred to as plug flow regime. Then, with continued increase in void fraction, the stratified flow developed waves, the peaks
eventually coming in contact with the pipe wall and becoming slug flow.

The correlation of slip-velocity ratio with gas-super-ficial-velocity shows slip-ratio values generally close to unity.

However, the slip factors and flow regimes found in the inlet and outlet section of the pump do not give any guide as to the flow regimes occurring within the impeller and the volute of the pump itself. The flow within the pump is modified from that occurring in the suction line by the two powerful but conflicting mechanisms. One is the chopping action of the rotating blades, which tends to make the flow homogeneous, and to drive the slip ratio toward unity. The second is the intense centrifugal field in three components (in each principal plane), which tends to stratify the flow and thereby allow a relatively high slip ratio.

The distribution of the air bubbles in the radial pump impeller passages was observed photographically by using a strobe-light and high-speed camera. The change of the pump performance was studied in relation to the flow patterns in the impeller. Our speed camera photography indicated that, at low suction void fraction, the first mechanism was dominant. When the injected air quantity was increased into the water stream, the second mechanism determined the flow regime in the impeller passages. The flow regimes are shown in

On entering the pump impeller, air was crushed into finite bubbles by the impeller and the bubbles were scattered uniformly in the streaming water (Figure 8, sketch A). When the void fraction increases, the bubbles start to accumulate at the pressure side of the impeller blade (Figure 8, sketch B). The strong adverse pressure existing in the inlet parts of the impeller blades decelerates the air bubbles and they accumulate there, resulting in a hollow air-filled space (Figure 8, sketch C). If the air quantity increased further, a large, air-filled hollow space would extend from the inlet to the outlet of the impeller (Figure 8, sketch D). Because ~75% of single-phase pressure rise and of course the two-phase flow head degradation occur between the impeller passages, our analysis focuses primarily on the impeller performance.

![Figure 8 Two-phase flow regimes in the radial pump impeller passages](image)

The different test data are plotted against the suction void fraction in Figure 9. It can be observed that the total head and the impeller differential pressure degradation characteristic are very similar, which means that the basic pump head loss is caused by the loss of the impeller pressure generation ability. Similar trends can be observed for different speed ranges. However, the threshold suction void fraction, at which the pump differential pressure starts to degrade rapidly is significantly different for higher than for lower suction pressure. The higher suction pressure requires lower percent gas compression work through the pump.

In the next graph, Figure 10, the cumulative calculated values of the inlet chamber and impeller were compared with the normalized differential pressure between the pump inlet and the impeller discharge. The predicted and experimental results for impeller pressure degradation are in good agreement.

In order to demonstrate the usefulness of this proposed theory, the model to predict the two-phase flow impeller differential pressure degradation was applied for
different pump speeds, mass fluxes, and suction pressures, as shown in Figures 11.

![Figure 9](image-url)

Figure 9. Two-phase flow pump torque and pressure distribution versus suction void fraction.

![Figure 10](image-url)

Figure 10. Pump inlet chamber and impeller normalized differential pressure comparison with the theoretical prediction (N=2000 rpm, G=2750 kg/(m^2 sec)).

The impeller and the volute interact with one another. The pump casing determines much of the environment in which the impeller works. The volute provides a constant pressure working environment to the impeller. This constant pressure is equal to the single-phase pressure of the volute at the appropriate water flow rate and independent of the injected air flow rate. When the air flow rate is increased further, the homogeneous bubble flow starts to separate at the end of the impeller passages. This effect alters the relative flow angle and absolute velocity of the flow. At this stage, the volute loses its pressure-generating ability.

The volute differential pressure distribution can be predicted by Equation 44. The comparison between the measured and calculated differential pressure values versus suction void fraction is shown in Figure 12.

We have arrived at the point where the measured pump total pressure rise can be matched with the calculated pump overall pressure distribution. The theoretical model can predict the pressure distribution of the inlet chamber, the impeller, and volute. The summation of these predictions should be the same as the measured differential pressures between the pump inlet and outlet flanges. These comparison plots are shown

![Figure 11](image-url)

Figure 11. Measured differential pressure between pump suction and impeller discharge comparison with the theoretical prediction (N=2000 rpm, G=2750 kg/(m^2 sec)).
in Figures 13 and 14 versus the suction void fraction.

Figure 12. Calculated and measured volute differential pressure distribution (N=2000 [rpm], G=2700 [kg/(m^2 sec)]).

Figure 13. Calculated and measured total pump pressure distribution versus suction void fraction (N=2000 [rpm], G=2750 [kg/m^2 sec]).

Conclusion

The pump performance under different single- and two-phase flow and within a wide range of speed and flow rate conditions has been investigated by employing a radial-flow, open-impeller pump. Single- and two-phase pump models were developed based on the experimental data.

The predicted results for single-phase pressure distributions were calculated by dividing the pump flow area into three different domains. The first domain was the space between the pump suction flange and impeller inlet. The second domain was the impeller vane area. The third domain was the pump volute. The summation of these measured values was then compared with the total pressure rise of the pump. The agreement between these data points showed less than 3% percentage of deviation.

High-speed photography was applied to determine the basic mechanism that causes this remarkable pump performance degradation. Three observation windows were installed on the impeller area. Two observation windows were installed on the volute periphery in order to monitor the flow regimes in the volute flow channel. Through the two-phase flow experiment we observed bubble flow regime with variable bubble sizes.

The mechanism of the present model is based on the fact that the air injected into the liquid stream under a certain threshold void fraction fills up the impeller flow.
channels. At low suction void fraction, the air bubbles in the impeller were distributed uniformly and moved without accumulation, causing no pressure degradation. The flow regime between the vanes is bubbly over the full length of the vanes. When a percentage of the injected air is increased beyond the threshold value, a separated flow regime develops at small distances along the vane but remains bubbly for the rest of the length along the vane. At this point, the impeller differential pressure starts to decrease. If the flow rate of the air is gradually increased, the separated regime extends over the full length of the vane. At this point, a large air-filled space extends from the inlet of the pump to the discharge of the impeller. In this stage, the impeller loses almost all of its pressure-generating ability, and the differential pressure across the pump is reduced to under 10% of its original value.

Ongoing research should be completed in order to more effectively determine the phenomena of two-phase flow pump pressure degradation. A well-designed, comprehensive set of pump two-phase flow tests should be performed to build up a range of correlating curves in order to make the present analytical model more accurate. To accomplish this, one needs to test a representative range of radial pumps of different specific speeds, sizes, number of blades, blade angles, open and closed impellers, with impellers with and without shroud, and condensable gas phase [21].

REFERENCES


RELEVANT PUBLICATIONS


