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# DEVELOPMENT OF ANC-TYPE EMPIRICAL TWO-PHASE PUMP MODEL FOR FULL SIZE CANDU PRIMARY HEAT TRANSPORT PUMP

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## ABSTRACT

The development of an ANC-type empirical two-phase pump model for CANDU (Canadian Deuterium) reactor primary heat transport pumps is described in the present paper. The model was developed based on Ontario Hydro Technologies' full scale Darlington pump first quadrant test data. The functional form of the ANC model which is widely used was chosen to facilitate the implementation of the model into existing computer codes.

The work is part of a bigger test program with the aims: (1) to produce high quality pump performance data under off-normal operating conditions using both full-size and model scale pumps; (2) to advance our basic understanding of the dominant mechanisms affecting pump performance based on more detailed local measurements; and (3) to develop a "best-estimate" or improved pump model for use in reactor licensing and safety analyses.

## 1.0 INTRODUCTION

The operation procedures of CANDU reactors allow the main coolant pumps to remain running for a finite length of time under two-phase flow conditions resulted from loop depressurization. The two-phase performance of the main coolant pumps, however, has not been properly characterized. A test program was thus initiated in Ontario Hydro to investigate systematically the performance of a full size CANDU reactor pump under two-phase flow conditions.

The Darlington Pump Test Loop in Ontario Hydro Technologies (OHT) was used in the test program. A schematic of the pump test loop is shown in Figure 1. Specifications of the Darlington pump are given in Table 1.

The test loop was extensively instrumented. A full complement of transmitters were used to cover the process conditions in the flow loop. This included motor parameters, pressures and temperatures at the pump suction and discharge, pressure rise across the pump and venturis for single-phase liquid flow measurements (Figure 1). For two-phase testing, special instrumentation were installed to measure the suction and dishcarge void fraction as well as the suction mass fluxes. They are described below (Figure 1):

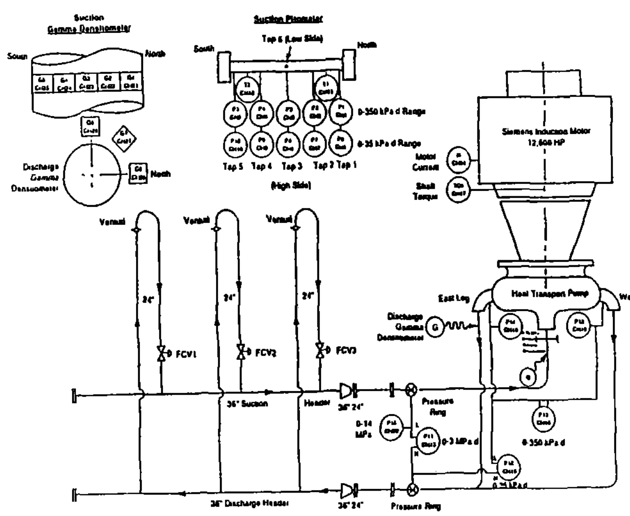


FIGURE 1:
SCHEMATICS OF DARLINGTON NGS PUMP TEST LOOP WITH INSTRUMENTATION

. A pump suction gamma densitometer which divides the cross-section of the suction pipe into five parallel segments and measures the local and average void fractions at the pump suction;

. A rake of five pitot tubes across the diameter of the pipe at the pump suction to measure the local velocity heads. Combined with the local void fraction measurements, the two-phase mass fluxes or two-phase volumetric flow rates at the pump suction can be deduced [1];

. A pump discharge gamma densitometer which provides three chordal void fraction measurements at one of the two discharge pipes (east leg). Since the discharge gamma densitometer does not cover the whole cross-sectional area of the pipe, no average void fraction at the discharge pipe was obtained. The device was used only as a void indicator.

A total of twenty-three (23) first quadrant two-phase flow tests were performed covering a range of operating conditions. This included five loop temperatures (140° C to 265° C nominal) and three initial flow settings (80%, 100% and 120% rated). The following procedure was used in the test program: (1) pressurize the loop and heat up the loop using pump motor heat; (2) adjust the settings of the flow control valves (FCV1 to FCV3) to establish the desired initial single-phase flow conditions; (3) depressurize the loop to the corresponding saturation pressure by bleeding the loop after the desired loop temperature is achieved; (4) produce two-phase flow conditions in the loop by bleeding water from the loop slowly and continuously; (5) log all sensor signals as the loop void increases; and (6) terminate the test when it is evident that the pump is free spinning or the loop vibrations become excessive.

The first quadrant test data were processed. The head degradation characteristics of the pump under various two-phase flow conditions were obtained. It was found that the head degradation characteristics are strongly influenced by the loop operating conditions (i.e., temperature and initial flow rates). The pump head tends to stay higher and longer for higher temperature operations (e.g. at 265 deg C).

It should be noted that the set of test data obtained in the test program is quite unique in the sense that: (1) a real-size reactor pump was used and (2) void fraction and two-phase mass fluxes at the pump suction were measured. The test data thus provide a more reliable data base for two-phase pump model development for reactor primary heat transport pumps. The development of an "improved" empirical pump model is given in the present paper. It is hoped that by adopting the improved and more realistic pump model into the reactor licensing codes, our prediction capabilities in reactor thermal hydraulic transient calculations under various accident scenarios can be enhanced.

## 2.0 EXISTING PUMP MODELS

Because of the importance of heat transport pumps in nuclear reactor operations, a large amount of work has been done on pump performance under both single- and two-phase flow conditions. Some of the more important pump test work are given in [2-11]. A brief review of the major empirical two-phase pump models was provided by Kim [12]. Analytical models are described in [13-16]. In general, the analytical models are still under development and are more difficult to apply.

Among all the models, the empirical ANC model [17] is the most widely-used in reactor safety analyses. It was, therefore, decided that the present model should have the same form as the ANC model so that it can be readily implemented into the existing computer codes.

## 3.0 OHT MODEL

### 3.1 Approach

The same basic formulation as in the ANC model [17] is used:

$$h_{2\phi}(v,\alpha) = h_{1\phi}(v) - M(\alpha) * [ h_{1\phi}(v) - h_{fd}(v) ] \qquad (1)$$

where $h_{2\phi}(v,\alpha)$ is the normalized two-phase head
$h_{1\phi}(v)$ is the normalized single-phase head
$h_{fd}(v)$ is the normalized fully-degraded head
$M(\alpha)$ is the head degradation multiplier

Equation (1) states that for a given pump with a known single-phase head-flow curve, the two-phase pump head can be calculated if the head degradation multiplier and the fully-degraded head-flow curve are known.

Since pump heads (single-phase, two-phase and fully-degraded) are, in principle, measurable in a test program, the model is thus reduced to determining the head degradation multiplier as a function of suction void:

$$M(\alpha) = [ h_{1\phi} - h_{2\phi} ] / [ h_{1\phi} - h_{fd} ] \qquad (2)$$

### 3.2 Pump Head Degradation Characteristics

The two-phase head degradation characteristics of the Darlington Pump were investigated systematically in the present test program. The effects of the loop temperature and initial flow settings were studied.

Five loop temperatures (140, 170, 200, 230 and 265° C) were attempted. However, for the two lower temperature tests (140 and 170° C), the pump went into cavitation as soon as the loop was depressurized. The tests were thus terminated without any measurable suction voids. No two-phase pump performance data were obtained for these two test conditions.

The degradation of the pump head as a function of the measured average suction void fraction is shown in Figures 2 to 4 for nominal loop temperatures of 265, 230 and 200° C respectively. The pump head was calculated using the pressure rise measurement across the pump and the measured suction mixture density:

$$H = \Delta P/(\rho_m * g) \qquad (3)$$

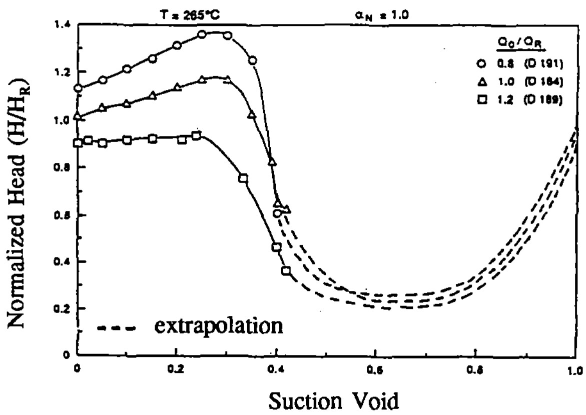


FIGURE 2:
PUMP HEAD DEGRADATION CHARACTERISTICS
(T = 265 deg C)

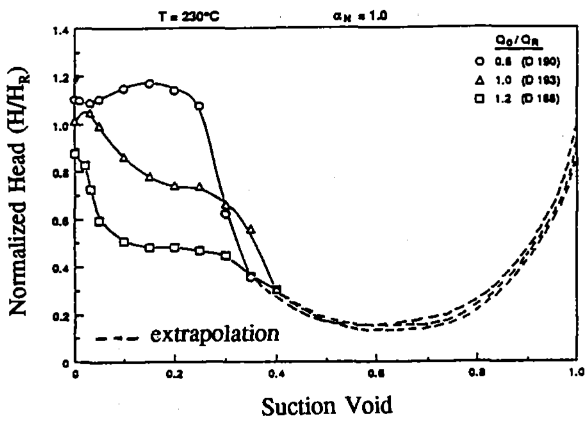


FIGURE 3:
PUMP HEAD DEGRADATION CHARACTERISTICS
(T = 230 deg C)

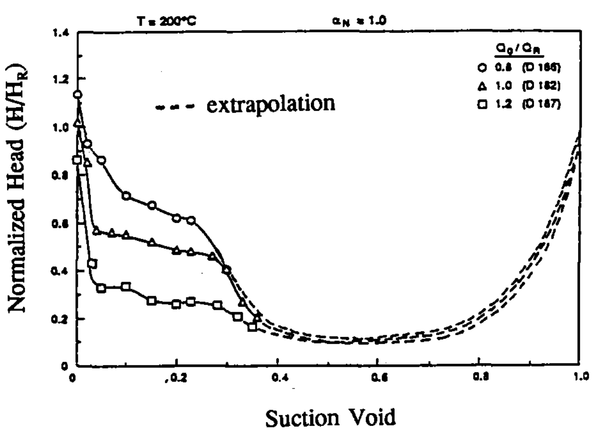


FIGURE 4:
PUMP HEAD DEGRADATION CHARACTERISTICS
(T = 200 deg C)

It should be noted that pump head data were only obtained for average suction void fractions of up to about 40%. At about 40% suction void, the pump head and thus the flow became so low that damage to the pump bearings became a real concern. The tests were therefore terminated. The head curves at higher suction voids shown in the Figures were extrapolated based on theoretical considerations that the pump head degrades as suction void increases and achieves a minimum at high suction voids. The pump head as defined by Eqn (3), then begins to recover as the single-phase steam flow condition is approached and achieves its rated single-phase (steam) value when the suction void becomes unity. The behaviour of the pump head as a function of suction void has also been observed experimentally [6,7].

From Figures 2 to 4, two observations can be made:

. The head degradation characteristics are strongly affected by the loop temperature. At higher loop temperatures, the pump tends to maintain its head for higher suction voids. At lower loop temperatures, the head degrades more readily in response to an increase in void in the suction pipe.

. The head degradation also depends on the initial flow rates. The two-phase pump behaviour for different initial flow conditions are quite different at low suction voids. However, at higher suction voids (0.4 or higher), the pump head curves for a given loop temperature tend to converge. This suggests that the loop resistance was dominated more by the two-phase flow than by the control valves at high voids.

3.3 Fully-Degraded Pump Head

The ANC pump model is based on the assumption that the two-phase pump performance varies between the single-phase liquid and the fully-degraded two-phase head-flow curves (Eqn (1)). The single-phase liquid head-flow curve can be readily measured. However, the fully-degraded head-flow curve is not as well defined. Physically, it can be defined as the lowest possible two-phase head delivered by the pump for a given volumetric flow rate. The fully-degraded head-flow curve can thus be determined experimentally by fitting a curve to the lower bound of all the two-phase performance data for a given pump. The resulting fully-degraded pump head, therefore, depends very much on the size and quality of the test data.

Another important paramter in the ANC-type pump model is the threshold pump suction void fraction ($\alpha_{fd}$) which signifies the beginning of the fully-degraded pump operation. At $\alpha_{fd}$, the two-phase pump head becomes the same as the fully-degraded head and consequently, M becomes unity as given by Equation (2).

Again, the threshold void fraction is not well defined. It was found to be pump dependent and can be correlated by the pump specific speed and the impeller diameter [18]. Its value was estimated to be between 0.0 and 0.2 for the SAWFT pump and 0.4 and 0.6 for the CE pump [18].

3.4 Pump Model

Before an ANC-type model for the Darlington pump can be developed, it is neccesary to determine both the fully-degraded head-flow curve and the threshold void fraction for the Darlington pump.

The fully-degraded head can be estimated by fitting a lower bound curve to the two-phase performance data as described in the previous section. Its normalized value was found to lie between 0.2 and 0.0 for a normalized flow of between 0.2 and 1.0. However, this estimation is not considered reliable due to the small size of the Darlington pump test data.

In the present study, the fully-degraded pump head was obtained from the extrapolated head degradation curves (Figures 2-4). The fully-degraded pump head was taken as the minimum head values in the Figures. This minimum head value depends on the loop temperature only and is independent of the initial flow rates.

They are given in Table 2 for the three loop temperatures used. It should be noted that the fully-degraded pump head so obtained gave only approximate values. The values were acceptable for the present application as the degradation multiplier was found to be rather insensitive to the fully-degraded pump head used. This will be discussed later.

The threshold void fraction for the Darlington pump was also obtained from the head degradation curves (Figures 2 to 4). It was taken as the smallest suction void fraction which corresponds to the minimum pump head value in the extrapolated head degradation curves. Its values are also given in Table 2.

Using the two-phase test data and the fully degraded head values obtained, the head degradation multiplier for the Darlington pump was computed as a function of the suction void (Equation (2)).

The manufacturer provided single-phase liquid head-flow curve was used in the calculations. This curve can be expressed in the form of a polynomial:

$$h_{1\phi}(v) = a_o + a_1*v + a_2*v^2 + a_3*v^3 \qquad (4)$$

where $a_o$ = 1.2158
$a_1$ = 0.1182
$a_2$ = -0.2273
$a_3$ = -0.0930

The single-phase head-flow curve was also obtained experimentally using the Darlington pump loop, it was found to be within 3% of the manufacturer's curve.

The head degradation multiplier for the Darlington pump is shown in Figures 5 to 7 for the three loop temperatures used. The curves can be divided into three regions based on the suction void fraction values:

(1) For $\alpha_s < 0.4$, the curves were calculated directly from test data as described above;

(2) For $0.4 < \alpha_s < \alpha_{fd}$, $M(\alpha)$ was extrapolated using the fully-degraded two-phase pump head and threshold void fraction estimated; and

(3) For $\alpha_s > \alpha_{fd}$, a simple model was used. The pump head was assumed to be fully-degraded up to a suction void of 0.9. After that, the pump head recovered linearly and achieved the single-phase (steam) head at $\alpha_s = 1.0$.

From Figures 5 to 7, it can be seen that the head degradation multiplier is also a function of the loop temperature and initial flow conditions as expected from the head degradation curves shown in Figures 2 to 4. The curves provide detailed pump behaviour under two-phase flow conditions. However, from an application point of view, it may be desirable to reduce the number of dependent variables so that the model can be more easily implemented into the existing computer codes.

Since the dependence of the head degradation multiplier on the initial flow rates is not as strong as the loop temperature, it is thus decided that the effect of the flow parameter be ignored and only the M curves with v = 1.0 be retained. The choice of the v = 1.0 curves is reasonable becasue the v = 1.0 curves provide a good approximation for the other flow conditions, especially for the 265 and 200 deg C cases (Figures 5 and 7).

To facilitate the application of the new two-phase pump model to the existing computer codes, attempts have been made to relate the head degradation multiplier to the pump suction void using various functional forms. However, the outcome of the effort was less than satisfactory. Therefore, it was decided that it will be easier and more accurate to implement the model in tabulated form. This is given in Table 3. Shown in the table are the values of the multiplier at the three loop temperatures for different pump suction void fractions. M values at void fractions between two given points can be linearly interpolated. Interpolation or extrapolation may also be used to obtain M values at temperatures other than those shown in Table 3. However, there are not enough test data to assess the validity or accuracy of this temperature interpolation or extrapolation.

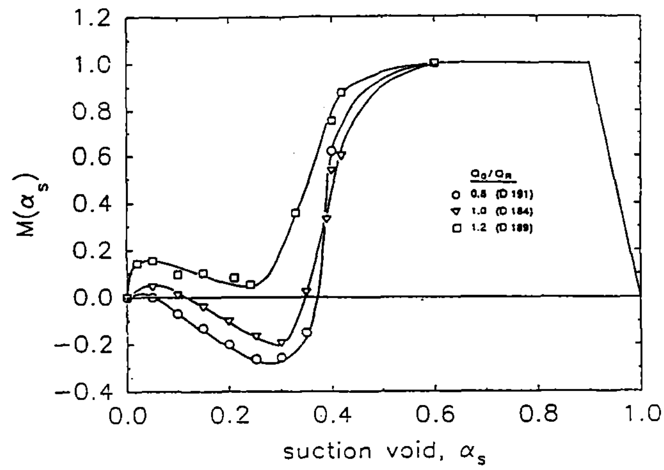


FIGURE 5:
HEAD DEGRADATION MULTIPLIER (T = 265 deg C)

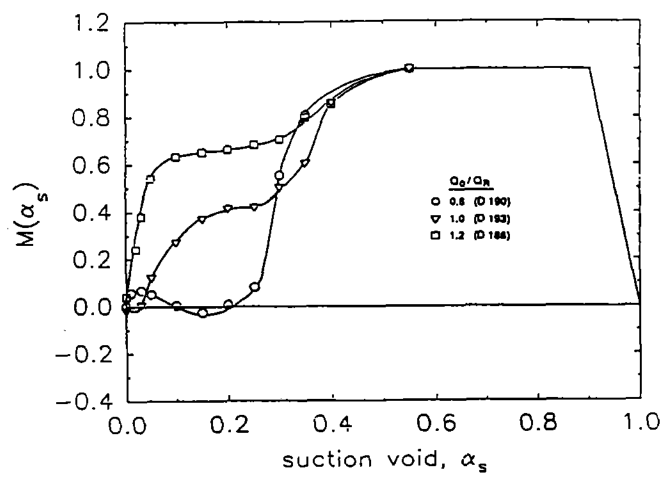


FIGURE 6:
HEAD DEGRADATION MULTIPLIER (T = 230 deg C)

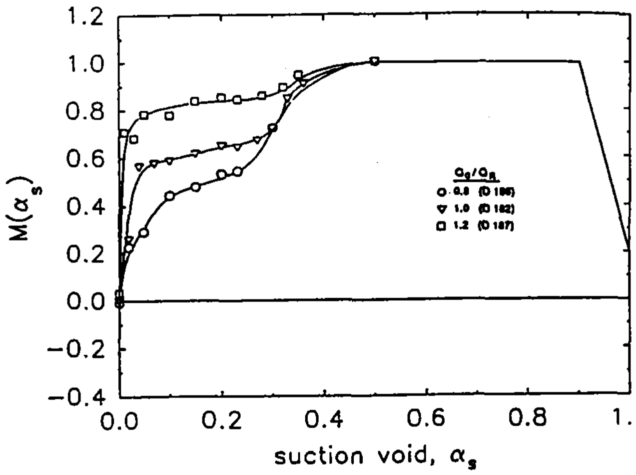


FIGURE 7:
HEAD DEGRADATION MULTIPLIER (T = 200 deg C)

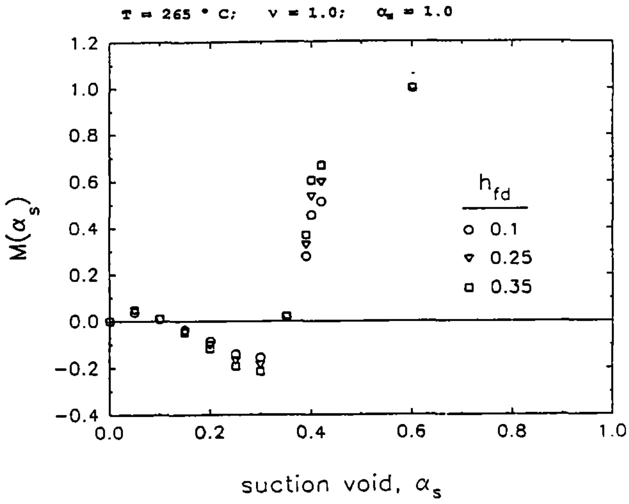


FIGURE 8: SENSITIVITY OF M(α) TO $h_{fd}$ VALUES

## 4.0 DISCUSSIONS

1. The present empirical model was developed using the fully-degraded pump head and threshold void fraction values estimated from the extrapolated head degradation curves. The estimated values appeared to be reasonable and consistent with other studies [18].

2. The fully-degraded pump head, $h_{fd}$, was assumed to be constant for a given loop temperature in the present model. The sensitivity of the model to the fully-degraded pump head used was studied. The results are shown in Figure 8 for the 265° C test at rated flow condition. The fully-degraded pump head was varied between 0.1 and 0.35 which represents a large uncertainty range. It can be seen that the effect of $h_{fd}$ on the degradation multiplier is small.

3. The present model is divided into three regions. For $\alpha < \alpha_{fd}$, M(α) was calculated or extrapolated from the test data. For $\alpha > \alpha_{fd}$, a simple relationship was suggested. Other curves may also be used in this region if they are judged to be more appropriate.

4. The pump heads as predicted using the present empirical model were compared with the test data. The pump heads for the tests with initial flows at rated condition were well predicted as expected. For the range of initial flows used (80% to 120% rated), the uncertainties of the model predictions were around 30%, 45% and 35% for loop temperatures of 265°, 230° and 200° C respectively.

5. The present empirical model is more appropriate to full-size reactor pumps than the ANC model by virtue of the data base used in its development. In comparison, the ANC model was developed using a small pump with internals quite different from typical reactor pumps. The relative size of the pumps which have been used in various two-phase performance tests are compared in Figure 9. Also shown in Figure 9 are typical BWR, PWR and CANDU reactor pumps. It is obvious that the ANC model may not be the best model to used for nuclear reactor applications.

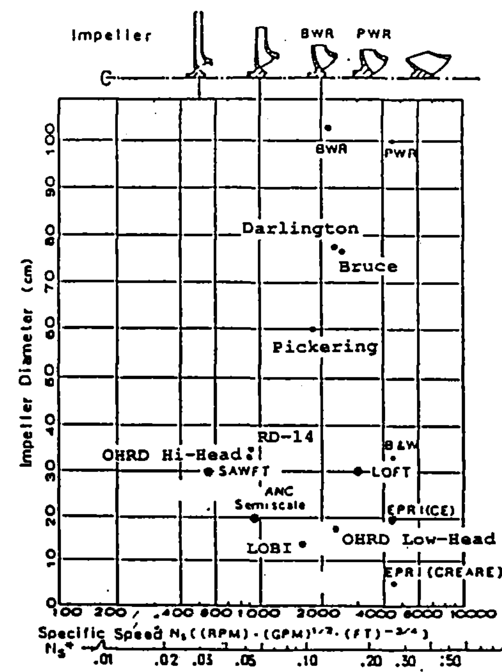


FIGURE 9: SIZE AND SPEED OF PUMPS USED IN TWO-PHASE TESTS

## 5.0 CONCLUSIONS

The development of a new empirical two-phase pump model based on the Ontario Hydro Technologies Darlington pump test data was described. Since a real-size reactor pump was used in the test with direct measurements of void fraction and two-phase mass flux at the pump suction, the new model is believed to be a major improvement over the widely-used ANC model.

Only the first quadrant (forward flow, forward rotation) test data were used in the model development. Therefore, the new model is applicable only to the first quadrant pump operations.

The model was developed based on test data obtained at relativly high temperatures. Application of the model to temperatures below 200° C is not recommended.

ACKNOWLEDGEMENTS
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NOMENCLATURE

| | |
|---|---|
| $h$ | normalized pump head ( $= H/H_R$ ) |
| $H$ | pump head, m |
| $M$ | head degradation multiplier |
| $N$ | pump rotational speed, rpm |
| $Q$ | volumetric flow rate, $m^3/s$ |
| $\alpha$ | void fraction |
| $\alpha_N$ | normalized rotational speed ( $= N/N_R$ ) |
| $\Delta P$ | pressure rise across the pump, KPa |
| $v$ | normalized volumetric flow rate ( $= Q/Q_R$ ) |
| $\rho_m$ | mixture density, $Kg/m^3$ |

Subscripts:

| | |
|---|---|
| $1\phi$ | single-phase liquid |
| $2\phi$ | two-phase |
| fd | fully-degraded |
| o | initial condition |
| R | rated |
| s | pump suction |

TABLE 1
SPECIFICATIONS OF DARLINGTON NGS PUMP

| | |
|---|---|
| Rated Head: | 224 m |
| Rated Flow: | 3.1 $m^3$ / s |
| Speed: | 1800 rpm |
| Specific Speed: | 2828 |
| Impeller Diameter: | 77.0 cm |

TABLE 2
DARLINGTON PUMP HEAD
DEGRADATION CHARACTERISTICS

| Loop Temperature (° C) | Fully-Degraded Head ($h_{fd}$) | Threshold Void Fraction ($\alpha_{fd}$) |
|---|---|---|
| 265 | 0.25 | 0.60 |
| 230 | 0.15 | 0.55 |
| 200 | 0.10 | 0.50 |

TABLE 3
OHRD EMPIRICAL TWO-PHASE PUMP MODEL

| | HEAD DEGRADATION MULTIPLIER ($M(\alpha_S)$) | | |
|---|---|---|---|
| Suction Void ($\alpha_s$) | T = 265 °C | T = 230 °C | T = 200 °C |
| 0.00 | 0.00 | 0.00 | 0.00 |
| 0.02 | 0.022 | 0.023 | 0.258 |
| 0.05 | 0.044 | 0.119 | 0.563 |
| 0.10 | 0.010 | 0.269 | 0.585 |
| 0.15 | - 0.043 | 0.367 | 0.615 |
| 0.20 | - 0.103 | 0.411 | 0.647 |
| 0.25 | - 0.168 | 0.417 | 0.650 |
| 0.30 | - 0.186 | 0.500 | 0.721 |
| 0.35 | 0.020 | 0.601 | 0.886 |
| 0.40 | 0.536 | 0.856 | 0.940 |
| 0.45 | 0.805 | 0.925 | 0.975 |
| 0.50 | 0.890 | 0.973 | 1.00 |
| 0.55 | 0.965 | 1.00 | 1.00 |
| 0.60 | 1.00 | 1.00 | 1.00 |
| 0.90 | 1.00 | 1.00 | 1.00 |
| 0.95 | 0.50 | 0.50 | 0.50 |
| 1.00 | 0.00 | 0.00 | 0.00 |