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Prediction of Reactor Coolant Pump Performance Under Two-Phase Flow Conditions

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이상유동시 원자로 냉각재 펌프의 성능 예측

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Abstract

A performance of reactor coolant pump in two-phase flow is examined using the pump geometric conditions and the performance of the pump in single-phase flow. Wall friction loss of the reactor coolant pump in single-phase flow is prdicted using the Truckenbrodt boundary layer theory, and the head loss in two-phase flow is predicted with calculated wall friction loss and separation loss coefficients. The analysis results are compared with the Combustion Engineering pump test data. The effect of two-phase multiplier on the peak clad temperature in Loss-of-Coolant Accident is also examined using the RELAP5 and the results indicate the importance of its accuracy.

요 약

이상유동시 원자로 냉각재 펌프의 성능을 펌프의 기하학적 형상 및 단상 유동시의 펌프 성능을 이용하여 예측하였다. 단상 유동시의 원자로 냉각재 펌프의 벽면 마찰손실은 Truckenbrodt의 경계층 이론을 이용하여 예측하였으며, 계산된 벽면 마찰 손실 및 분리 손실을 사용하여 이상유동시의 수두손실을 예측하였다. 해석결과는 Combustion Engineering 사의 펌프 실험 데이터와 비교하였다. 또한 냉각재 상실사고시 이상유동배수가 첨두 피복재 온도에 미치는 영향을 RELAP5 를 사용하여 평가하였으며, 분석결과는 이상유동배수의 정확성이 중요한 영향을 미치는 것으로 나타났다.

1. Introduction

Reactor Coolant Pump (RCP) should provide adequate cooling for the reactor core in both normal operation and transient or accident conditions. Understanding and predicting RCP performance under two-phase flow condition are important in predicting core coolability and hence the loop flow rate in safety analysis. Analyses of a postulated Loss-of-Coolant Accident (LOCA) of Pressurized Water Reactor (PWR) involve the prediction of core flow and broken loop pump overspeed, both of which are dependent on the performance characteristics of RCP. The pump in the broken leg pipe directly affects the rate of system depressurization by changing blowdown flow, whereas the remaining pumps affect the flow rates and distribution throughout the system. During LOCA, RCPs may undergo two-phase flow conditions, and the pump performance characteristics will be changed drastically from that for single-phase flow. Analyzing pump performance characteristics in two-phase flow is complicated by many parameters both characterizing the pump design (pump geometry, scale, and pump type) and thermalhydraulic conditions (void fraction, slip velocity between the phases, condensation effect, etc.) [1]. Until recently, RCP two-phase performance has not been accurately predicted due to such a compexity.

During the past decade, progress in thermalhydraulics of RCPs was made through various test programs which provided large and impressive data base for empirical correlations [2−6]. Recently, as part of EPRI's comprehensive pump testing and analysis program, a mechanistic model that is based on rational analysis was developed [7, 8]. The major features of the model were a predictive capability as well as an insightful understanding of the physical mechanisms involved in pump head degradation in two-phase operation, an independency on specific empiricism rather than those of other models, an applicability to any type of pump, and a consideration of the effects of geometry, condensation, local void fraction, relative velocity between the phases, and compressibility. However, the model has a limitation in predicting an overall thermalhydraulic behavior of reactor coolant system (RCS) during LOCAs especially Large Break LOCA, although it has a transient capability.

To identify an effect of RCP two-phase flow on RCS thermalhydraulics during LOCA, it needs to use system thermalhydraulic codes such as RELAP5 [9], TRAC [10], etc. And those codes usually require the data associated with RCP performance characteristics both in single-phase and two-phase conditions. It is, therefore, important to generate RCP two-phase performance data suitable to the system codes and the pump geometry considered. From those points of view, the ERPI's model [7, 8], which emphasized the model accuracy, needs an additional data and model to generate the required data. The practical models, rather less accurate, need to be developed for the application to the well known nuclear system codes.

There are four principal methods which have been developed to describe the behavior of centrifugal pumps in the two-phase flow. These methods have demonstrated effectiveness to certain areas of pump operation but have not proven to be universally applicable. Those models are Semiscale/RELAP pump performance model, B&W pump performance model, NASA performance prediction method, and Westinghouse equivalent density method. The RELAP pump model employs a two-phase head degradation multiplier as a function of average void fraction. The B&W model uses a similar method to the RELAP pump model except using the different definition of the multiplier. As a results, the scattering of the two-phase data indicates that these might be uncertainty in the formulation of the multiplier. The NASA method requires as inputs a large number of details of the fluid physics and the fluid properties. This method is therefore considered not applicable to RCP. In the Westinghouse equivalent density method, the two-phase head characteristic was correlated with the single-phase head characteristic by means of an equivalent density. When applying this method to Semiscale results a considerable scatter was found to exist when plotting equivalent density against inlet void fraction. Asthe method is not based on the fundamental flow physics occurring within a pump it may turn out not to be of general application. Details on these models are described in Ref. [3].

The present study aims to investigate the two-phase flow effect of the RCP during LOCA calculation under the standpoint of conservatism and to develop the pump two-phase performance degradation data to be applied to RELAP5 code using the main body of Wilson model [3]. For the additionally required model to develope a two-phase performance data, as mentioned above, two kinds of model were considered; a case using a simple Moody friction factor [11] and a case with Truckenbrodt boundary layer theory [12]. And the prediction results were compared with Combustion Engineering (CE) pump test data.

2. Two-phase Flow Effect of RCP on LOCA

During LOCA in a PWR, the RCPs are usually assumed to be unpowered, which leads to a rapid decrease in differential pressure over the pump. This sudden initial decrease in differential pressure is caused by the acceleration of fluid through the pump to the break. As a result of flow acceleration, a choking at the break will occur and thus a short-term recovery of differetial pressure in the intact loop pump will be found. And then differential pressure decrease as the inlet void fraction of the pump increases into the transient. The timing and period of recovery vary with the geometry, fluid conditions, etc. According to the Semiscale experiment [13], a significant change in pump differential pressure, hence the developed pump head was observed for a small change in pump inlet void fraction, which also caused a significant difference core flow rate and fuel cladding temperature during blowdown phase of LOCA.

Such an two-phase pump head effect should be considered in the postulated LOCA calculation using a best-estimate thermal-hydraulic code such as RELAP5. The RELAP5 has a pump model which is based on the model developed by Idaho National Engineering Laboratory (INEL) from the Semiscale test data. The RELAP5 pump model play a role of adding the pump developed head to a source term of the fluid momentum equation during the solution algorithm. The value of the developed head by pump operation is calculated by interpolation over the range of pump data specified by input. The imput data include single-phase homologous curves, two-phase fully degraded homologous curves and degradation multipliers as a function of void fraction. These data should be prepared for both the head and torque. The actual two-phase pump head can be calculated by the following equation:

$$H_{2\phi} = H_{1\phi} + M(\alpha)\,(H_{DEGRAD} - H_{1\phi}) \quad (1)$$

where $H_{2\phi}$, $H_{1\phi}$, HDEGRAD, $M(\alpha)$ are two phase head, single phase head, fully degraded head and two-phase multiplier, respectively. The last three items in this list have to be supplied through input. The $M(\alpha)$ function is an interploating function for interpolating between the single-phase head curve and the fully degraded or lowest two-phase head curve.

The effect of two-phase multiplier can be found in the postulated LOCA calculation for a PWR. Figure 1 shows a comparison of cladding temperature transient for LBLOCA in YGN Units 3/4 from the author's RELAP5 sensitivity study on the pump two-phase multiplier. The "original" in legend of the figure 1 means the case using the two-phase multiplier data provided in PSAR of YGN Units 3/4, while the "modified" means the case using the proprietary data which was proposed by CE pump vendor. The latter can be regarded as more realistic than the former since it was based on the test result. The comparison result shows that the differences of the peak

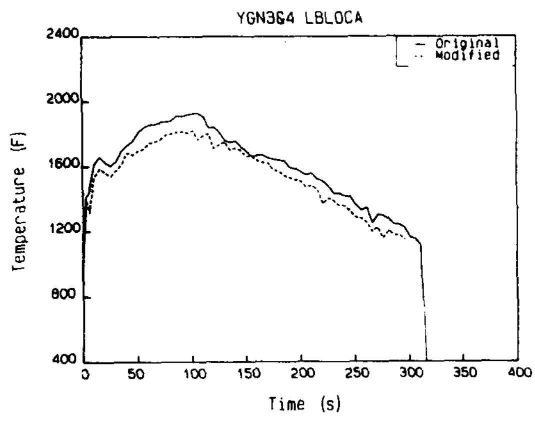

Fig. 1. Peak Cladding Temperature vs.

cladding temperature (PCT) are about 70°F during blowdown phase and 100°F during reflood phase. The difference in reflood PCT was larger than that of blowdown PCT, which can be regarded as a combination of the blowdown difference due to pump two-phase performance plus the propagation of the blowdown difference. From those comparison, the conservatism of the pump two-phase multiplier provided by YGN Units 3/4 PSAR may be identified with a viewpoint of RELAP5-based evaluation ignoring the uncertainty of the proprietary CE pump test data. The calculation was performed by RELAP5/MOD2, however, such an effect of two-phase multiplier on PCT can be expected at the further version of RELAP5 unless the pump model will be changed.

For a completeness of the discussion above, one requires two areas of investisgation; a scaling problem of the pump two-phase multiplier data and the generation of the data suitable to best-estimate calculation of LOCA. The second concern will be discussed in the next section of this paper.

A scaling problem can be expressed as follows: Whether or not the pump two-phase multiplier data obtained from a small-scale pump can be appled consistently to full-sized NPP calculation? To answer the question the various experimental conditions were investigated as shown in Table 1. And the result was illustrated in figure 2, which shows a compariosn of two-phase pump head degradation data available. As shown in the figure, the two-phase head degradation as a function of void fraction can be deduced. The value of multiplier of all the data become larger as void fraction increases winthin the range from 0.2

Table 1. Comparison of Pump Characteristics

Type	Scale	Volume Flow (gpm)	Total Head (ft)	Rated Speed (rpm)	Specific Speed *	Fluid	Pressure (psia)	Note
Kori 3/4	1/1	103100	278	1150	5423	S/W	15−2250	
WH test	1/3	6210	64.4	1150	5190	S/W& A/W	15−420	N/A
B&W test	1/3 of B&W NPP	11300	390	3580	4317	A/W	20−120	N/A
C−E pump	1/5 of B&J NPP	3500	252	4500	4200	S/W	15−1250	P/A
Creare pump	1/20 of B&J NPP	181	252	18000	4200	A/W& S/W	A/W at 90 S/W at 400	N/A
KWU test	1/5 of RS111 NPP	3148	392.7	8480	6700	S/W	435−1305	N/A
LOFT test	1/48	5000	315	3530	2645	S/W	15−2250	P/A
SEMISCALE		180	192	3560	926	S/W	15−1600	P/A

Note *: Unit of Specific Speed = rpm * gpm ** 0.5/ft ** 0.75
B & J: Byron Jackson
RS111: One of the German NPP
N/A: Not Available
P/A: Partially Available
S/W: Steam-Water
A/W: Air-Water

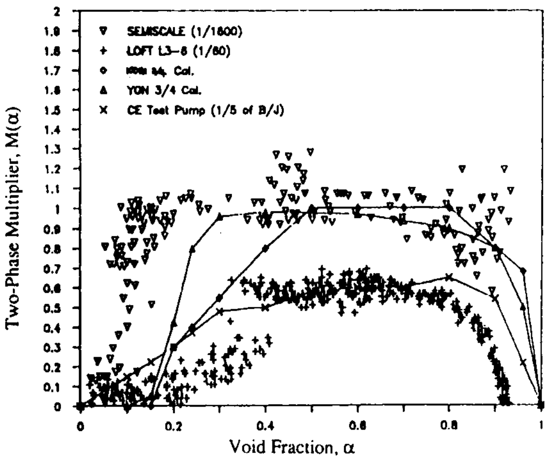

Fig. 2. Pump Two-Phase Multiplier vs. Void Fraction

to 0.9. It is also found that pump degradation of the Semiscale is larger than that of the LOFT. That may mean larger scaled pump has lesser pump degradation. Since more test data are required to verify this, it may be necessary to obtain the data such as BETHSY and LSTF in the near future. For Kori Units 3/4 the two-phase multiplier used is similar to that of the Semiscale except the low void fraction region (0.0−0.2). And the data used in the YGN Units 3/4 calculation almost are close to the Semiscale data. Therefore, the above evaluation results for those two plants may be qualitatively accepatble in terms of conservatism.

3. Prediction of Pump Preformance Under Two-Phase Flow Conditions

In this chapter a semi-empirical method of analysis of two-phase flow behavior in a centrifugal pump is proposed using a simple theory of idealized pump operation and incorporating experimental data for single-and two-phase flow.

3.1. Loss in Two-Phase Flow Regime

Usually, fluid flowing through a pump experienced the resistance which comes from losses due to wall friction, due to sudden expansion and contraction of flow area, or shock in blade upstream, etc. All of the losses also may be important in two-phase flow condition, thus the prediction of the pump performance degradation during two-phase flow should be based on the two-phase loss characteristics. However, the two-phase characteristics of the various losses associated with pump internal flow are not clearly described due to their complexity in thermalhydraulics such as flow regime and turbulence, etc. Therefore, the wall friction loss and a shock loss, which are known to have an almost identical behavior both in single-phase and two-phase flows, are considered as major contributors to the two-phase pump head degradation in the present study. This simplification also based on the fact that the contribution of other loss mechanisms was small compared to the two loss mechanisms.

A simple model was proposed by Lottes and Flinn [11], which enables to calculate the wall friction loss at two-phase flow from the that in single-phase at a given mass flow rate, as follows:

$$\Delta P_{tpf} = R \Delta P_{sf} \qquad (2)$$

where P_{tpf} and P_{sf} are pump head losses in two-phase flow and single-phase flow, respectively. The factor R in Eq. (2) is a friction factor for two-phase flow and can be expressed as following form:

$$R = [\frac{(1-x)}{(1-\alpha)}]^2 \qquad (3)$$

where x and α are steam quality and void fraction, respectively. The wall friction loss in single phase flow, Psf of Eq. (2), can be described as follows:

$$\Delta P_{sf} = f(\frac{L}{d_h})(\rho_1 \frac{C^2}{2g_c}) \qquad (3)$$

where f, L, d_h, 1, C and g_c denote a wall friction factor, a flow length, a hydraulic diameter, an inlet fluid density, a inlet fluid absolute velocity and gravitational accelation, respectively. The absolute fluid velocity, C has calculated from

$$C = Q_{tp} \rho_{tp} / (A \rho_L) \qquad (4)$$

whre A and mean a flow area and fluid density while subscripts tp and L mean "two-phase mixture" and "liquid", repectively. This equation can be incorporated into Eq. (3) to formulate the two-phase pressure drop.

The pressure drop due to separation, which can be assumed to be proportional to square of difference between the actual flow rate and the best efficiency point flow rate, is represented by the following equation.

$$\Delta P_{1s} = K_1 (Q_1 - Q_{BE})^2 \tag{5}$$

where the subscript 1s means a single-phase separation, Q_1 and Q_{BE} denote a single-phase flow rate and a flow rate at the best efficiency point for the given pump design, and K_1 represent a propotional constant. The separation loss for two-phase flow can be described in the similar way as follows:

$$\Delta P_{tps} = K_1 \left[\frac{Q_L}{Q_{tp}} - \frac{Q_{BE}}{Q_{tp}} \right]^2 Q_{tp}^2 \tag{6}$$

The overall head loss considering the wall friction and shock, can be expressed as follows:

$$\Delta P_{tpfs} = R\, f\, G \left(\frac{\rho_{tp}}{\rho_1} \right)^2 + K_1 \left[\frac{1-a}{1-a+as} - \varepsilon \right]^2 Q_{tp}^2 \tag{7}$$

where the subscript tpfs means "two-phase friction and shock", and the following relations were considered.

$$G = L / (d_h A^2) \tag{8}$$

$$a = \frac{\alpha}{(1-\alpha)} \frac{\rho_v}{\rho_L} \tag{9}$$

$$s = C_v / C_L \tag{10}$$

$$\varepsilon = Q_{BE} / Q_{tp} \tag{11}$$

Thus, the head loss ratio between single-phase and two-phase can be deduced as follows:

$$H^* = \frac{\Delta H_{tpth} - \Delta H_{tp}}{\Delta H_{1tp} - \Delta H_1} = \frac{\left(\frac{1+a}{1+sa} \right)^2 + K_3 \left(\frac{1-a}{1-a+as} - \varepsilon \right)^2}{(1-a)(1+a)\left[1 + K_3 (1-\varepsilon)^2 \right]} \tag{12}$$

where the subscript 1 and th denote a single-phase and theoretical, and K_3 is calculated by the following equation:

$$K_3 = \frac{K_1}{f_L\, G\, \frac{\rho_L}{2 g_c}} \tag{13}$$

where constant K_1 may be obtained from single-phase pump performance curve. The calculation of liquid friction factor f_L also can be calculated by a simple model such as Moody diagram. From the equations above, therefore, the two-phase head ratio consequently two-phase head degradation can be calculated with the known thermodynamic state, geometrical data and single-phase performance data.

3.2. Wall Friction Loss in Two-Phase Flow Regime

Eq. (12) needs an estimation of constant K_3, and the prediction accuracy of two-phase head degradation will be strongly dependent on the accuracy of K_3. However, the detailed calculation method was not proposed by previous studies. Since a nature of K_3 represents a single-phase wall friction, an improved prediction method rather than conventional Moody diagram method is needed to complete Wilson's model and to increase predictability. The present study employed a model proposed by Balje [12], which was based on Trukenbrodt's integral boundary layer equation, and verified for aerodynamic impeller design.

Trukenbrodt's intergal equation for the momentum thickness of boundary layer flow under adverse pressure gradient is:

$$\theta_2 \left(\frac{\upsilon}{\upsilon_2} \theta_2 \right)^{1/n} \upsilon_2^{\,3+2/n} = C_1 + A \int_0^\infty \upsilon^{\,3+2/n}\, dx \tag{14}$$

where, θ_2 means a momentum boundary thickness defined in Ref. [12] and v is a free stream velocity. n and A are constants depending on the Reynolds number, Re(n=1 and A=0.46 for laminar flow, n=6 and A=0.0076 for turbulent flow, and n=2 and A=0. 46 for transition flow with flow separation). C_i is an initial boundary layer thickness. For an integration of the equation, the velociy distribution on the blade suction surface should be descibed. According to the Balje, a "double peaked suface velocity distribution", and stepwise linear velocity distribution, was recommended for centrifugal impellers. As a result of the integration after introducing the double peaked surface velocity distribution, the momentum thickness can be calculated as follows:

$$\frac{\theta_2}{l} = \frac{\left(\frac{A}{4+2/n}\right)^{\left(\frac{n}{n+1}\right)}}{Re^{\left(\frac{1}{n+1}\right)}} \left[[I_1+I_2+I_3]^{\left(\frac{n}{n+1}\right)} + [I_4+I_5]^{\left(\frac{n}{n+1}\right)} \right] \quad (15)$$

where,

$$I_1 = \frac{\mu\, x_1 \left(\frac{\alpha^*-1}{\mu}\right)^{(4+2/n)}}{\alpha^*-1}$$

$$I_2 = -\frac{(x_2-x_1)\left\{ \beta^{*(4+2/n)} - \left(\frac{\alpha^*}{\mu}\right)^{(4+2/n)} \right\}}{\frac{\alpha^*}{\mu}-\beta^*}$$

$$I_3 = \frac{(1-x_2)\left(1-\beta^{*(4+2/n)}\right)}{1-\beta^*}$$

$$I_4 = \frac{\gamma^* x_3 \left\{ \gamma^{*(4+2/n)} - \left(\frac{1}{\mu}\right)^{(4+2/n)} \right\}}{\gamma^*\mu-1}$$

$$I_5 = \frac{(1-x_2)\left(1-\gamma^{*(4+2/n)}\right)}{1-\gamma^*}$$

where, μ, α^*, β^*, γ^* are a total deceleration ratio (v_2/v_1), a deceleration ratio of the maximum velocity at suction surface ($v_{s\text{-}max}/v_1$), a deceleration ratio of the maximum velocity based on exit velocity ($v_{s\text{-}max}/v_2$), and a deceleration ratio of the minimum velocity at pressure surface ($v_{p\text{-}min}/v_2$), respectively. And x_1, x_2 and x_3 mean inflection points of velocity at suction surface and pressure surface, respectively. In general, these valuse are not known, therefore, the surface velocity distribution of blades is obtained by considering the force balance acting on a particle in a rotating channel and by assuming the blades as circular arc shapes [12]. Then, the loss coefficient at blade surface associated with the momentum thickness is

$$\zeta_0 = 2\left(\frac{\theta_2}{l}\right)\frac{\sigma}{\sin\beta_2^*}\left[1 + \frac{1}{2}\left(\frac{\theta_2}{l}\right)\frac{\sigma H^2}{\sin\beta_2^*}\right] \quad (16)$$

where, ξ_0 is H $2g/c_{ref}^2$, c_{ref} is free stream velocity, σ is a cascade solidity, β^* is a blade exit angle and H is a shape factor defined by δ/θ(δ is a boundary layer displacement thickness). The total friction loss can be deduced using the average outlet velocity as follows:

$$H_{lf} = \frac{\zeta_0}{1-\zeta_0}\,\frac{2L}{d_h}\,\frac{c^2}{2g_c} \quad (17)$$

Eq. (17) can be incorporated into Eq. (13), which is actually used in Eq. (12) to generate a head loss ratio. In Eq. (12), the theoretical head loss both in single-phase and two-phase flow can be calculated by Wilson-derived Euler equation [3] for a given pump geometry and operational condition. The logic diagram of calculating the head coefficients is described in figure 3.

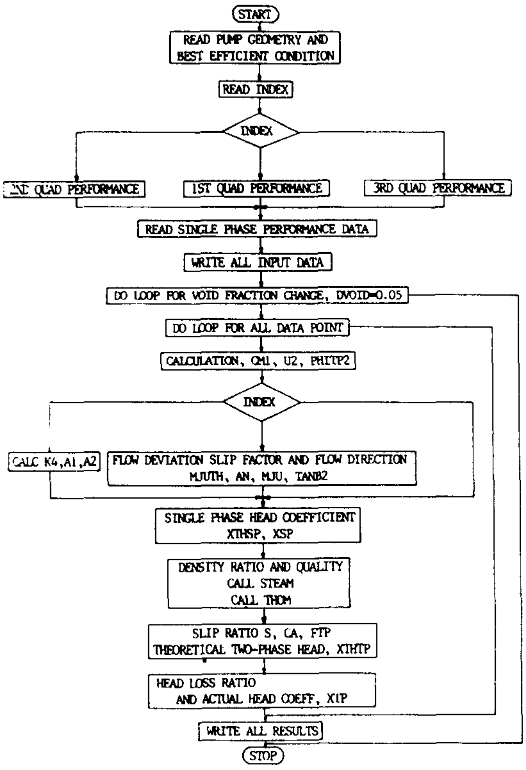

Fig. 3. Logic Diagram of Calculating Head Coefficients

3.3. Analysis Results

To investigate the effectiveness of the model explained above a Combustion Engineering (CE) pump test [5] was used. Tests were performed on a geometrically scaled model of an actual RCP. Both steady-state and transient blowdown tests were performed over sufficiently large ranges of thermal-hydraulic operating conditions. Approximately 1000 steady-state tests were performed. Each test provided about 200 measures and derived parameters. The different conditions for which the scale model pump was tested under steady-state conditions included variations in several parameters. Measurements were after establishing steady-state operation as desired combinations of fluid pressure vessel, void fraction, volumetric flow rate, and impeller speed. These pump performance tests covered forward, zero, and reverse flow, and speeds in various combinations. Fluid conditions upstream of the pump were set to provide a variety of single-phase steam or water and two-phase mixtures of steam and water, ranging from all water to all steam. Also, fluid pressure was set at several different values. Details on the experimental procedures and data are described in Ref. [5].

The major parameters used in this calculations are as follows: inlet blade angle (19.17°), exit blade angle (23.0°), blade thickness (0.004mm), number of the blades (5), volumetric flow rate (0.2m/s), and rotational speed (4500 rpm). The predicted pump heads of the Combustion Engineering in single-phase flow as a function of flow rate using Moody diagram and Truckenbrodt method are shown in figure 4 and figure 5, respectively. Here, the total head loss represents the practical head subtracting from the theoretical head, and is equal to sum of the wall frictional loss and the separational loss. It is shown that the results with the Truckenbrodt method are predicted better than those with Moody diagram. According to the characteristics of the CE pump, the best-efficiency or rated conditions of pump flow rate was at $0.2210\,m^3/s$. At this condition, the separational loss will be closed to zero and the total heas loss is nearly same as the frictional head loss.

In figure 6 the total head losses of CE pump in

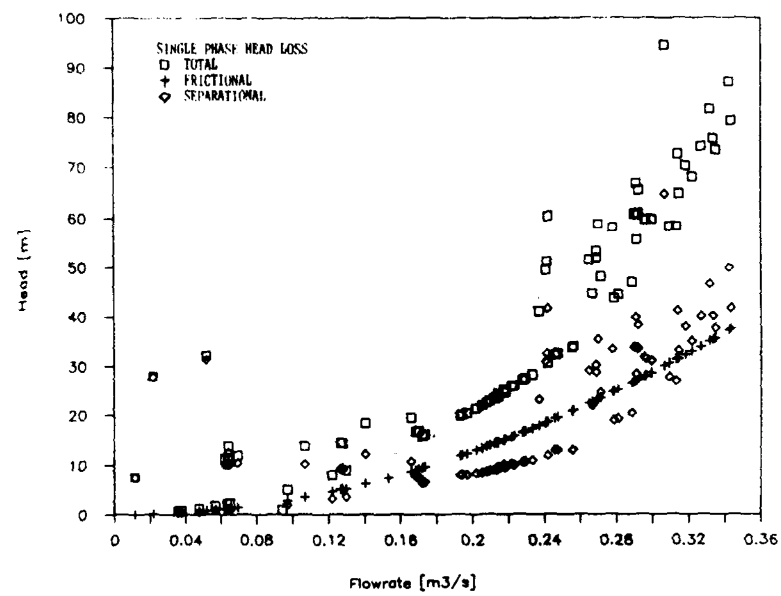

Fig. 4. Head vs. Flowrate Using Moody Diagram (CE Pump)

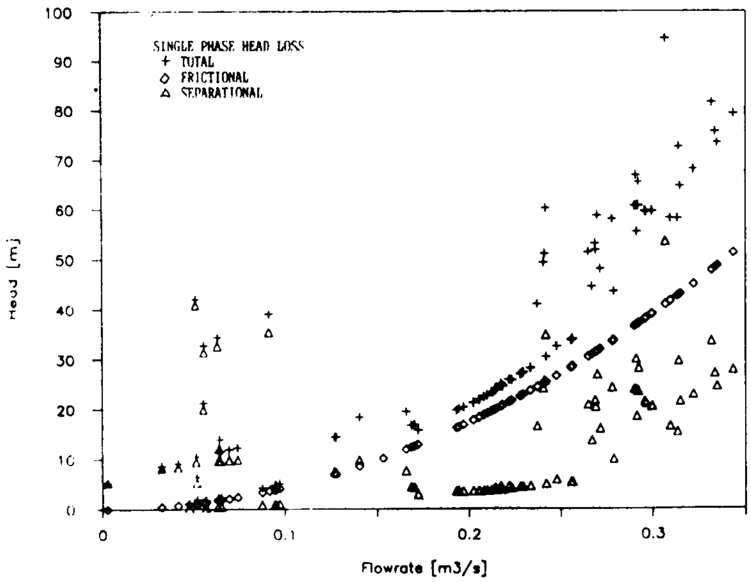

Fig. 5. Head vs. Flowrate Using Truckenbrodt (CEPump)

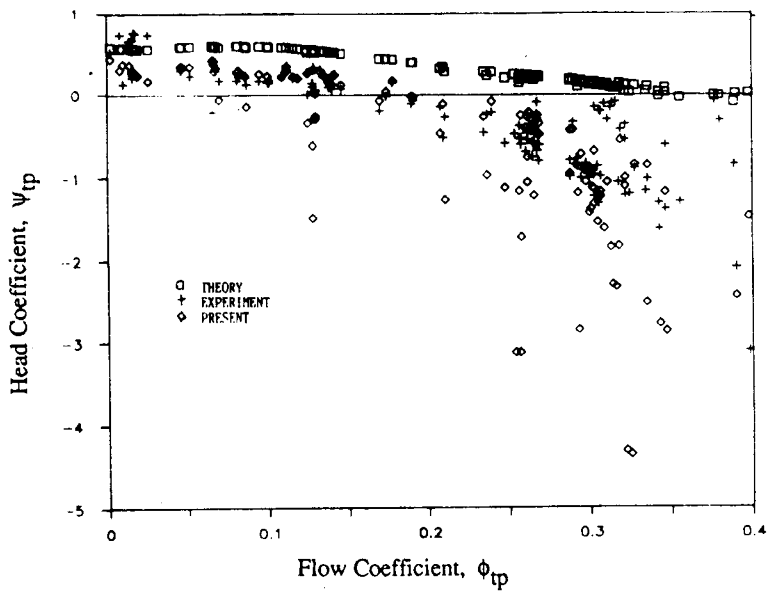

Fig. 6. Head Coefficient vs. Flow Coefficient (CE)

two-phase flow as a function of pump flow rate is shown. Here Ψ_{tp} is a dimensionless head coefficient, defined as g $\triangle H_{tp}/u^2$, and ψ_{tp} is dimensionless flow coefficient, defined as m/ρ_{tp} A u. The variation of head coefficients vs flow coefficients under two-phase flow conditions is compared with the test data and

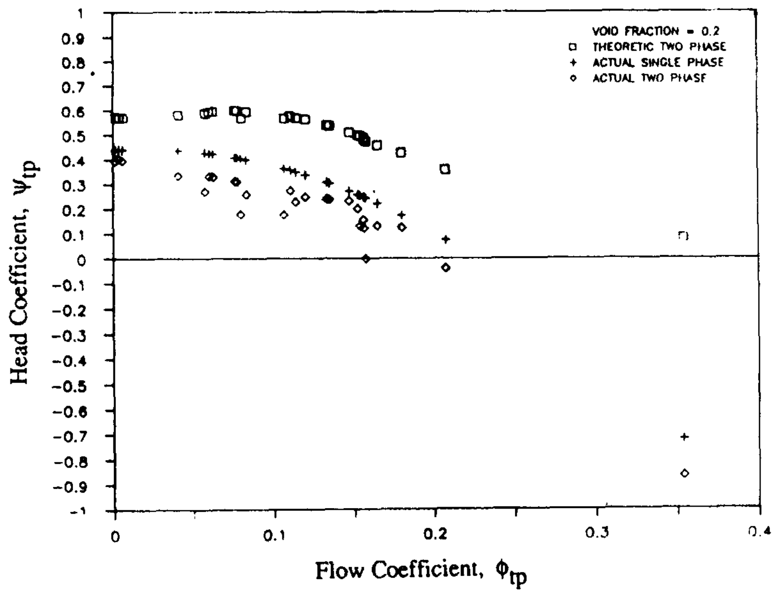

Fig. 7. Head Coefficient vs. Flow Coefficient for α=0.2 (CE)

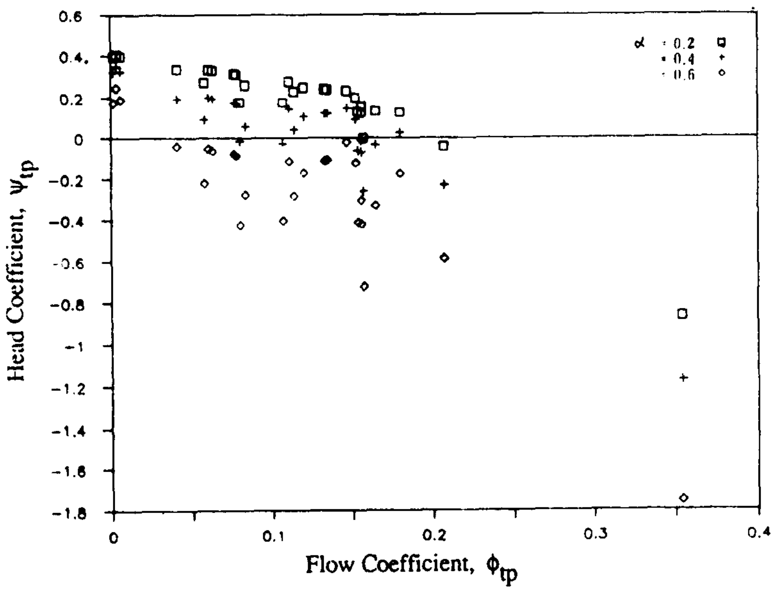

Fig. 8. Head Coefficient vs. Flow Coefficient (CE)

the theoretical values. The comparison shows that the calculational results are well agreed with the test data, however, the deviation from the test data becomes larger as the flow rates increases. One of the reasons for those deviations is considered due to the scattered single-phase head losses at large flowrate in the experiment.

The calculated head coefficient as a function of flow coefficient when the void fraction is 0.2 is also compared with the theoretical two-phase head coefficient and the single-phase head coefficient (Figure 7). Here, "theoretical" represents the conditions without considering the losses due to friction and flow separation. As is to be expected, the calculated results are lower than those of the theoretical and single-phase cases. The head coefficients for various void fractions such as 0.2, 0.4 and 0.6 are shown in figure 8. The head losses become larger and are predicted with large scatter of values as the void increases. That explains the inadequacy of the present model for the larger values of void fraction.

From the aforementioned analysis results, the present model could be used reasonably to predict the effects of pump two-phase degradation and to generate the two-phase multiplier data suitable for system TH code within the accuracy of two-phase pump data used.

4. Conclusions

The performance degradation of the reactor coolant pump under two-phase conditions is investigated in terms of "Head-Loss Ratio" using the pump geometry conditions and performance data under single-phase conditions. For the calculation of the wall friction and separation losses at the rotor blade, the Truckenbrodt boundary layer theory is applied. The results were also compared with Combustion Engineering (CE) test data and were in good agreement with the test data under two-phase conditions. This may explain that the present calculations with considering the wall friction loss and separation loss will be reasonable. The effect of "two-phase multiplier" used as the RELAP5 input on large break LOCA was also examined, and the comparison results showed that its effects resulted in large difference in the peak cladding temperature.

Therefore, the pump degradation under two-phase flow condition could be predicted using the pump geometry and data in single-phase flow and the two-phase multiplier obtained can be used as input to the RELAP5 and to predict the pump performance more accurately during the anticipated accidents.

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