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SCALING CRITERIA FOR TWO-PHASE FLOW NATURAL AND FORCED CONVECTION LOOP AND THEIR APPLICATION TO CONCEPTUAL 2 × 4 SIMULATION LOOP DESIGN

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ABSTRACT

Scaling criteria for a natural and forced convection circulation loop under single-phase and two-phase flow conditions are derived from the fluid balance equations, boundary conditions, and solid energy equations. For a single-phase flow case the continuity, integral momentum, and energy equations in one-dimensional area-averaged forms are used. For a two-phase flow case the one-dimensional drift-flux model obtained from the short time temporal averaging and the sectional area averaging is used.

The scaling criteria are applied to a conceptual design of a 2 x 4 loop facility for simulating the Babcock and Wilcox 177 NSSS design. Feasibility of practical solutions are studied and some conclusions on this proposed facility in terms of the proper scaling are obtained.

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A2026	Phenomenological Modeling of Two-phase Flow in Water Reactor Safety Research

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NOMENCLATURE

NOMENCLATURE

A	Non-dimensional area
a	Flow area
a_d	Peripheral area at the outlet of pump impeller
a_s	Wall cross sectional area
B	Biot number
C_p	Fluid heat capacity
C_{ps}	Solid heat capacity
C_o	Distribution parameter
d	Hydraulic diameter
F	Friction number
Fr	Modified Froude number (stratified flow Froude number)
f	Friction factor
$f(x)$	Void-quality relation
g	Gravity
Gr	Grashof number
G	Mass velocity
H_ℓ	Liquid level
h	Heat transfer coefficient
j	Total volumetric flux
k	Thermal conductivity of liquid
k_s	Thermal conductivity of solid
K	Orifice coefficient
ℓ	Axial length
ℓ_e	Equivalent lengt for distributed losses
ℓ_h	Length of hot fluid section
L	Non-dimensional length
ℓ_{nb}	Non-boiling length (heated section)
Nu	Nusselt number
N_{pch}	Phase change number
N_{sub}	Subcooling number
N_{Fr}	Froude number
N_d	Drift flux number
N_ρ	Density ratio group
N_f	Friction number (two-phase)
N_o	Orifice
N_q	Critica
n	Number
Pr	Prandtl
$\dot{P}$	Thermal
p	Pressu
$\dot{q}_s'''$	Heat ge
Q_s	Heat sc
q''	Heat fl
q_c''	Critica
R	Richar
Re	Reynol
r_d	Radius
St	Stanto
t	Time
T	Fluid
T_s	Solid
T_{sat}	Satura
T^*	Charac
u	Veloci
u_r	Repres
u_f	Liquid
u_g	Vapor
u_m	Mixtur
U	Non-di
V_{gj}	Drift
V	Volume
X_{tt}	Turbul
x	Qualit
x_e	Exit q
y	Transv
Y	Non-di
z	Axial
Z	Non-di

NOMENCLATURE (Cont'd)

number (two-phase)
heat flux number
f rods
number
power

eration density in solid
rce number
x
heat flux
on number
number
f pump impeller
number

mperature
mperature
on temperature
ristic time ratio
(liquid)
tative velocity
elocity
locity
velocity
nsional velocity
locity

t flow Martinelli parameter

lity
se distance
nsional transverse distance
stance
nsional axial distance

NOMENCLATURE (continued)

NOMENCLATURE (Cont'd)

Greek Symbols

α	Void fraction
α_e	Exit void fraction
α_s	Solid thermal diffusivity
β	Thermal expansion coefficient
β_d	Exit blade angle of pump impeller
δ	Conduction thickness
ΔH_d	Pump head on impeller
ΔH_{fg}	Latent heat
ΔH_{sub}	Subcooling
Δp	Pressure drop
$\Delta \rho$	Density difference
$\Delta \mu$	Viscosity difference
η	Mechanical efficiency of pump
θ	Non-dimensional temperature
μ	Viscosity of liquid
ξ	Wetted (heated) perimeter
ρ	Density of liquid
ρ_s	Density of solid
σ	Surface tension
τ	Non-dimensional time
ω	Rotational speed of pump

Subscripts

d	Pump
i	ith section
o	Reference constant (heated section)
r	Representative variable
s	Solid
h	Hot
c	Cold
R	Model to prototype ratio
m	mixture
g	Vapor
$(\ )_m$	Model
$(\ )_p$	Prototype

I. INTRODUCTION

Nuclear reactor safety regulations have required extensive thermal-hydraulic testing of simulated fuel cores and other reactor system components. In view of the inherent difficulties associated with full-scale testing, scale models of prototype systems have been extensively used to predict the behavior of nuclear reactor systems during normal and abnormal operation as well as under accident conditions. The severity of the accident that occurred at the Three Mile Island Unit-2 plant has increased interest in this area. In particular, re-examination of the scaling laws and the scaling difficulties introduced by the scaling criteria in experimental facilities is an area currently receiving significant attention. The major source of the scaling difficulties is due to the existing uncertainties in the basic formulation of the two-phase flow systems, two-phase flow correlations, and flow regime transition criteria.

Several studies have been performed to establish similarity relations between the prototype and the scale model. In a study of heat transfer problems associated with the loss-of-fluid test (LOFT) program, Rose [1] employed a one-dimensional form of the conservation of mass, momentum, and energy equations to devise similarity relations between the LOFT geometry and its scale model. Carbiner and Chudnik [2] have examined scaling requirements for modeling blowdown loads imposed on a reactor core during a loss-of-coolant accident (LOCA). Basically, two scaling laws were developed using the conservation equations and presented two scaling laws, one time-reducing and one time-preserving. Ybarrando et al. [3] investigated thermal-hydraulic similarity laws with respect to the LOFT experimental system and assessed the effect of scaling compromises on the LOFT program. Recently, Nahavandi et al. [4] examined the scaling laws for modeling nuclear reactor systems. In this work, three types of scaling procedures are considered: time-reducing, time-preserving volumetric, and time-preserving idealized model/prototype. The necessary relations between the model and prototype are developed for each scaling type. It was shown that scaling procedures can lead to distortion in certain areas. The development in each of these investigations is limited to the power to volume scaling and is based on single-phase flow conservation equations that cannot reflect the phenomena associated with two-phase flow systems.

The available methods to develop similarity criteria for two-phase flow systems has been reviewed by Ishii and Jones [5], and the similarity analysis for a two-phase flow system has been carried out by Ishii and Zuber [6], and Zuber [7]. The results based on the local conservation equations and ones based on the perturbation method were utilized by Ishii and Kataoka [8]. The extension of the similarity analysis to a natural circulation system was achieved by considering the scaling criteria from a small perturbation method and the steady state solution. For this purpose, the relatively well established drift-flux model and constitutive relations [9,10] were used.

In the present analysis, the scaling criteria developed by Ishii and Kataoka [8] is used to obtain the preliminary conceptual design parameters for the 2 x 4 forced and natural circulation loop system under single and/or two-phase flow conditions. The 2 x 4 loop scaled system contains representative components of all thermal-hydraulic systems considered important in performing tests to obtain data representative of the response of the prototype plant. This system contains an electrically powered reactor vessel simulator, and two loops with representative hot leg, once through steam generator, and two active cold legs with pumps. For the purpose of this study the TMI Unit-2 plant was used as the prototype B&W 177 NSSS plant. Some preliminary conclusions on this facility in terms of the proper scaling are obtained.

II. SINGLE PHASE SIMILARITY LAWS

A. Similarity Parameters

The similarity parameters for the forced and natural convection circulation loop systems can be obtained from the integral effects of the local balance equations along the entire loop. A typical system under consideration is illustrated in Fig. 1. This system consists of a thermal energy source, energy sink, connecting piping system between components, and a circulation pump. For a single-phase flow case, a method similar to that used by Heisler and Singer [11] and Heisler [12] for a liquid metal system is applied here to develop similarity criteria.

The dimensionless variables and parameters used in the similarity study are obtained from the following dimensionless balance equations:

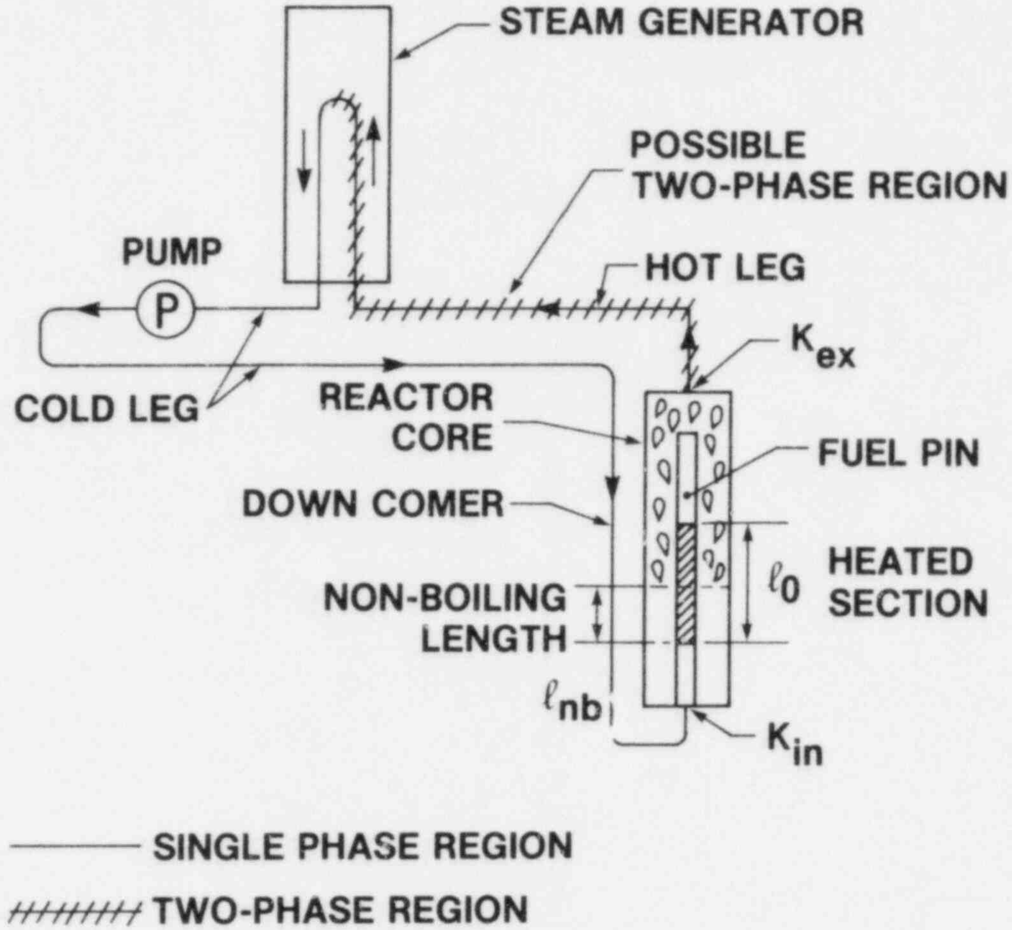

Fig. 1. Schematic System Under Consideration

Fluid Continuity Equation for ith Section:

$$U_i = U_r/A_i \tag{1}$$

Fluid Momentum Equation for Loop:

$$\sum_i (L_i/A_i) \frac{dU_r}{dt} = (\theta_h - \theta_c)\, R\, L_h - \sum_i (F_i/A_i^2) \frac{U_r^2}{2} + F_d \tag{2}$$

Fluid Energy Equation for ith Section

$$\frac{\partial \theta_i}{\partial \tau} + (U_r/A_i) \frac{\partial \theta_i}{\partial Z_i} = (\theta_{si} - \theta_i)\, St_i \tag{3}$$

Solid Structure Energy Equation for ith Section:

$$\frac{\partial \theta_{si}}{\partial \tau} + T_i^\star\, \nabla^{\star 2}\, \theta_{si} = Q_{si} \tag{4}$$

and

Fluid-Solid Structure Boundary Condition for ith Section:

$$\frac{\partial \theta_{si}}{\partial Y_i} = B_i(\theta_{si} - \theta_i) \tag{5}$$

(See Nomenclature for the definitions of various parameters.)

In these equations, the fluid properties are assumed to be constant except for the buoyancy term, where the Boussinesq approximation is used. The significant dimensionless similarity parameters of the system can be expressed in terms of the Richardson, Stanton, and Biot numbers, a dimensionless friction number, F_i, a dimensionless pump characteristic number, F_d, a dimensionless time ratio number, $T_i^\star$, and a dimensionless heat source number, Q_{si}, as follows:

$$R \equiv g\, \beta\, \Delta T_o \ell_o / u_o^2 \tag{6}$$

$$St_i \equiv 4\ h_i \ell_o/\rho\ C_p\ u_o\ d_i \tag{7}$$

$$B_i \equiv h_i\ \delta_{si}/k_{si} \tag{8}$$

$$F_i \equiv f_i\left(\frac{\ell_i}{d_i} + \frac{\ell_{ei}}{d_i}\right) + K_i \tag{9}$$

$$F_d \equiv g\ \Delta H_d/u_o^2 \tag{10}$$

$$T_i^\star = (\alpha_{si}/\delta_{si}^2)(\ell_o/u_o) \tag{11}$$

and

$$Q_{si} = \frac{\dot{q}_{si}'''\ \ell_o}{\rho_{si}\ C_{psi}\ u_o \Delta T_o} \tag{12}$$

where u_o, ΔT_o, and ℓ_o are a prescribed reference velocity, temperature difference, and equivalent length, respectively, for the system. ℓ_{ei} is the equivalent length for minor losses distributed over the ith section, i.e., bends, elbows, etc., and K_i is the singular loss coefficient defined at the inlet and outlet of the ith section, i.e., expansion and contraction coefficient. Finally, ΔH_d is the pump head on the impeller.

It is to be noted that for a natural circulation loop without an operating pump on the loop $\Delta H_d = 0$ ($F_d=0$) and the buoyancy force is the only driving force. On the other hand, for a forced flow circulation loop $\Delta H_d \neq 0$ ($F_d \neq 0$), and the effect of buoyancy can be neglected, i.e., $R \approx 0$.

In addition to the above defined physical similarity groups, several geometric similarity groups are obtained. These are:

$$\text{Axial Length Scale:}\quad L_i \equiv \ell_i/\ell_o\ (L_h \equiv \ell_h/\ell_o) \tag{13}$$

$$\text{Flow Area Scale:}\quad A_i \equiv a_i/a_o \tag{14}$$

It is noted here that the hydraulic diameter and the conduction depth δ_{si} are defined by

$$d_i \equiv 4\, a_i/\xi_i \tag{15}$$

and

$$\delta_{si} \equiv a_{si}/\xi_i \tag{16}$$

where a_i, a_{si}, and ξ_i are the flow cross sectional area, solid structure cross sectional area, and wetted perimeter of ith section, respectively. Hence, d_i and δ_{si} are related by

$$d_i = 4\,(a_i/a_{si})\,\delta_{si} \tag{17}$$

The reference velocity u_o, and temperature difference ΔT_o can be obtained by using the steady-state solution. By taking the heated section as a representative section, one can solve the fluid energy equation for temperature rise. Thus,

$$\Delta T_o = (\dot{q}_o''' \ell_o/\rho\, C_p\, u_o)(a_{so}/a_o) \tag{18}$$

where subscript o denotes the heated section. Substituting the above expression into the steady-state momentum integral equation, the solution for the velocity becomes

$$u_o = \left[\frac{2\,\beta\, g\,(\dot{q}_o''' \ell_o/\rho\, C_p)\,\ell_h\,(a_{so}/a_o)}{\sum_i (F_i/A_i^2)}\right]^{1/3} \tag{19}$$

for the natural circulation loop or

$$u_o = \frac{2\ g\ \Delta H_d}{\sum_i (F_i/A_i^2)} \tag{20}$$

for the forced circulation loop.

B. Similarity Laws

Equations (6) through (14) represent relationships among the dimensionless coefficients and the generalized variables of the differential equations. From the dimensionless form of differential equations it is evident that if the similarity is to be achieved between processes observed in a model and in the prototype, it is necessary to satisfy the following identities:

$$A_{iR} = \frac{(a_i/a_o)_m}{(a_i/a_o)_p} = 1 \tag{21}$$

$$\sum_i (L_i/A_i)_R = \frac{\sum_i (\ell_i/\ell_o)_m/(a_i/a_o)_m}{\sum_i (\ell_i/\ell_o)_p/(a_i/a_o)_p} = 1 \tag{22}$$

$$L_{hR} = \frac{(\ell_h/\ell_o)_m}{(\ell_h/\ell_o)_p} = 1 \tag{23}$$

$$\sum_i (F_i/A_i^2)_R = \frac{\left\{\sum_i \left[f_i\left(\frac{\ell_i}{d_i} + \frac{\ell_{ei}}{d_i}\right) + K_i\right]\left(\frac{a_o}{a_i}\right)^2\right\}_m}{\left\{\sum_i \left[f_i\left(\frac{\ell_i}{d_i} + \frac{\ell_{ei}}{d_i}\right) + K_i\right]\left(\frac{a_o}{a_i}\right)^2\right\}_p} \tag{24}$$

$$R_R = \frac{(\beta g \Delta T_o \ell_o/u_o^2)_m}{(\beta g \Delta T_o \ell_o/u_o^2)_p} = 1 \text{ or } F_{dR} = \frac{(g\Delta H_d/u_o^2)_m}{(g\Delta H_d/u_o^2)_p} = 1 \tag{25}$$

$$St_{iR} = \frac{(4h\ell_o/\rho C_p du_o)_{im}}{(4h\ell_o/\rho C_p du_o)_{ip}} = 1 \tag{26}$$

$$T^*_{iR} = \frac{(\alpha_s \ell_o/\delta^2 u_o)_{im}}{(\alpha_s \ell_o/\delta^2 u_o)_{ip}} = 1 \tag{27}$$

$$B_{iR} = \frac{(h\delta/k_s)_{im}}{(h\delta/k_s)_{ip}} = 1 \tag{28}$$

$$Q_{siR} = \frac{(\dot{q}_s''' \ell_o/\rho_s C_{ps} u_o \Delta T_o)_{im}}{(\dot{q}_s''' \ell_o/\rho_s C_{ps} u_o \Delta T_o)_{ip}} = 1 \tag{29}$$

It is apparent from the above set of equations that the complete transverse area similarity is required as expressed by Eq. (21). On the other hand, axial length similarity is required for the hot fluid section, Eq. (23). The complete loop axial length similarity expressed by Eq. (22) is somewhat weaker restriction than the complete axial geometrical similarity required by

$$L_{iR} = \frac{(\ell_i/\ell_o)_m}{(\ell_i/\ell_o)_p} = 1 \tag{30}$$

However, for simplicity both the transverse area and axial length similarities are assumed where the energy transfer is important. These are given by Eqs. (21) and (30).

In view of the complete transverse area similarity, the dynamic similarity condition, Eq. (24), reduces to

$$\sum_i (F_i)_R = 1 \tag{31}$$

This expresses that pipe friction loss and the minor losses associated with the loss coefficient $f_i\left(\frac{\ell_i}{di}+\frac{\ell_{ei}}{di}\right)$ and K_i can be interchanged without changing the overall value of the pressure loss term. By adding or removing bends or by providing additional flow restriction in the form of orifices, it should be possible to simulate a wide range of scaling conditions. In expectation that the pressure loss term can be satisfied independently of the remaining scaling requirements, it appears reasonable not to consider it further in the remaining part of the scaling discussions.

In view of the reference temperature difference, Eq. (18) and the reference velocity, Eq. (20) or (21) the Richardson number and/or the pump characteristic number are automatically satisfied provided that the model and prototype pumps are homologous.

In general, the solid materials as well as the working fluid need not be the same between the model and the prototype. However, for simplicity the use of the same material for structures and of the same working fluid with the same operating conditions, i.e., with the same system pressure, are assumed in the present analysis. Hence, the remaining similarity requirements expressed by Eqs. (26) through (29) reduce to the following scaling ratios:

Wall Conduction Depth Ratio:

$$\delta_{iR} = \delta_R = \sqrt{\ell_R/u_R} \tag{32}$$

Hydraulic Diameter Ratio:

$$d_{iR} = d_R = \sqrt{\ell_R/u_R} \tag{33}$$

Convective Heat Transfer Coefficient Ratio:

$$h_{iR} = h_R = \sqrt{u_R/\ell_R} \tag{34}$$

Heat Generation Ratio:

$$\dot{q}'''_{oR} = u_R\ \Delta T_R/\ell_R \tag{35}$$

where the reference velocity ratio is given as

$$u_R = [\dot{q}_{oR}''' \ell_R^2 (\delta_R/d_R)]^{1/3} \tag{36a}$$

for the natural circulation loop, and

$$u_R = \langle \Delta H_d \rangle_R \tag{36b}$$

for the forced flow circulation loop.

It can be seen that the above equations, i.e., Eqs. (32) through (36), comprise a set of five equations in the following seven unknowns δ_R, d_R, ℓ_R, u_R, h_R, $\dot{q}_{sR}'''$, and ΔT_R for the case of the natural circulation loop. In the case of the forced flow circulation, however, one additional unknown ΔH_{dR} has to be included resulting in the total of eight unknowns. Therefore, two of the unknowns in the former case and three of the unknowns in the latter case must be specified independently before a solution of the set can be obtained. For both cases, two of the obvious choices are the system axial length scaling ratio ℓ_R and the power scaling factor $\dot{q}_{oR}'''$ because the first one is limited by the size of the model whereas the second is limited by the available power. For the forced convection case an additional independent parameter to be specified is ΔH_{dR}. In view of these choices, it is important to note that the time scale will be distorted by the factor

$$t_R = \ell_R/u_R \tag{37}$$

Therefore, for a real time simulation an additional constraint of

$$t_R = 1 \text{ or } u_R = \ell_R \tag{38}$$

should be imposed. In this case, the power level and the pump head can be uniquely determined by

$$\dot{q}_{oR}''' = \ell_R \tag{39}$$

and

$$\Delta H_{dR} = \ell_R \tag{40}$$

Then all the other scaling ratios can be expressed in terms of the single geometric system scaling ratio, ℓ_R. The results of a simultaneous solution are given by the following:

$$\delta_R = d_R = 1 \quad (41)$$

$$h_R = 1 \quad (42)$$

and

$$\Delta T_R = \ell_R \quad (43)$$

In contrast to the design parameters such as δ_i and d_i, the heat transfer coefficient cannot be determined independent of a flow field and fluid properties. Thus, Eq. (42) imposes an additional constraint on the flow field.

In general, the heat transfer coefficient depends on the fluid properties and flow conditions. It is customary to represent a correlation for h in terms of the Nusselt number defined by

$$Nu \equiv \frac{hd}{k} \quad (44)$$

where d and k are the hydraulic diameter and thermal conductivity of fluid. There are a number of correlations for Nu for a flow in a relatively long tube, however, the following three correlations typically represent the three important groups

Laminar Flow (q" given) $Nu = 4.36$

Turbulent Flow ($1 < Pr < 20$) $Nu = 0.0155\ Re^{0.83}\ Pr^{0.5}$ (45)

Liquid Metals ($Pr < 0.1$) $Nu = 4.82 + 0.0185\ (RePr)^{0.83}$

This shows that the Nusselt number depends on the Reynolds number and fluid properties.

On the other hand, in a free convection range, Nu depends on the length of a heated surface and the Grashof number defined by

$$Gr \equiv \frac{\beta\, g\, (T_s - T)\, \ell^3}{(\mu/\rho)^2} \tag{46}$$

Then for a wide range of Gr, the Nusselt number may be correlated by

$$Nu = 0.3\, (GrPr)^{0.3}\, (d/\ell) \tag{47}$$

Those different constitutive relations for h indicate that the similarity requirements from the Stanton and Biot numbers are not easy to meet over a wide range of operating conditions, particularly in the case of a fluid to fluid simulation. The correlation for forced convection flow given by Eq. (45) shows that it is desirable to use a fluid with a similar Prandtl number.

With the assumption of the same fluid used in the model and prototype operations, the heat transfer correlation yields

$$h_R = 1/d_R \tag{48}$$

for a laminar flow range and

$$h_R = u_R^{0.83}/d_R^{0.17} \tag{49}$$

for a turbulent flow with ordinary fluids.

It is evident that Eqs. (48) and (49) impose different constraints on operational and design parameters. In vie of the design parameter given by Eq. (41), Eq. (48) reduces to Eq. (42), and, therefore, the similarity condition based on the Biot and Stanton number is satisfied in the laminar flow regime. In the case of the turbulent flow, Eq. (49) reduces to

$$h_R = u_R^{0.83} \tag{50}$$

which indicates that the similarity condition based on the Biot and Stanton number is not satisfied in a real time simulation since $u_R = \ell_R$ in this case. Because of this, the similarity condition based on the Biot and Stanton number should be carefully evaluated.

Another operational constraint is set by Eq. (40) for a forced flow circulation loop. However, this condition can be achieved by adjusting the speed of the circulation pump. It is to be noted that the pump head on the impeller can be expressed as

$$\Delta H_d = \eta \frac{(r_d \omega)^2}{g} \left(1 + \frac{a_o u_o}{a_d r_d \omega}\right)\left(\text{Cot } \beta_d\right) \tag{51}$$

where r_d, ω, a_d, and β_d are the radius, the rotational speed, the peripheral area at the outlet, and the exit blade angle of the impeller, respectively, whereas η is the mechanical efficiency of the pump. For homologous pumps the flow coefficient, the term in the parenthesis of Eq. (51), is duplicated between the model and the prototype. Hence, in view of Eq. (51), Eq. (40) reduces to

$$\frac{(r_d \omega)_m}{(r_d \omega)_p} = \ell_R \tag{52}$$

which can be met by adjusting the rotational speed of the pump.

To summarize, the effects of each of the terms appearing in the conservation equations are preserved in the model and prototype without any distortion, if one satisfies the requirements of:

a) Equal Pressure and Properties:

$$\frac{P_m}{P_p} = \frac{\rho_m}{\rho_p} = \frac{\mu_m}{\mu_p} = \frac{C_{pm}}{C_{pp}} = 1 \tag{53}$$

b) Scaling Relations:

$$\frac{u_m}{u_p} = \frac{\dot{q}_m'''}{\dot{q}_p'''} = \ell_R \tag{54}$$

which imply

$$\frac{t_m}{t_p} = 1 \tag{55}$$

c) Equal Area Ratios:

$$\frac{(a_i/a_o)_m}{(a_i/a_o)_p} = 1 \tag{56}$$

d) Equal Friction Numbers:

$$\frac{\sum_i F_i/(a_i/a_o)_m^2}{\sum_i F_i/(a_i/a_o)_p^2} = 1 \tag{57}$$

If some of these requirements are not satisfied, then the effects of some of the processes observed in the model and prototype will be distorted.

C. Scaling Law Results

In this section, we shall determine power and other core characteristic parameters that satisfy the scaling laws developed in the preceeding section.

Power: $\dot{P}$

The power is obtained from overall energy generation rate, that is, from

$$\dot{P} = a_{so}\ \ell_o\ \dot{q}_o''' \tag{58}$$

Hence, model to prototype ratio is given by

$$\dot{P}_R = a_R \ell_R \dot{q}'''_{oR} = a_R \ell_R^2 \quad (59)$$

Number of Rods: n

With the rod diameter of δ_s, the number of rods is determined as

$$n = 4 a_{so}/\pi \delta_s^2 \quad (60)$$

Then the number of rods ratio can be expressed by

$$n_R = a_R/\delta_{sR}^2 \quad (61)$$

In view of Eqs. (32) and (41), the ratio becomes

$$n_R = a_R/\sqrt{\ell_R/u_R} = a_R \quad (62)$$

Similarly, it can be shown that the number-of-tube ratio in the steam generator can be expressed by the same equation as Eq. (62).

Heat Flux: q"

Energy balance yields

$$q'' = \dot{P}/(\pi \delta_s \ell_o n) \quad (63)$$

and the heat flux ratio is expressed as

$$q''_R = \dot{P}_R/(\delta_{sR} \ell_R n_R) \quad (64)$$

Using Eqs. (41), (59), and (62) in the above equation, one obtains

$$q''_R = \dot{q}'''_R = \ell_R \quad (65)$$

The single-phase scaling laws are summarized in Table I.

Table I. Single-Phase Flow Time-Preserving Scaling Laws

Scaled Quantity		Model/Prototype Ratio
Length		ℓ_R
Hydraulic Diameter	For Core and	$\sqrt{\ell_R/u_R} = 1$
Rod Diameter	Steam Generator	$\sqrt{\ell_R/u_R} = 1$
Number of Rods (or Pipes)	Only	a_R
Flow Area		a_R
Volume		$a_R \ \ell_R$
Velocity		ℓ_R
Time		$\ell_R/u_R = 1$
System Pressure (Fluid Properties)		1
Solid Structure Properties		1
Heat Generation/Unite Volume/Unit Time		ℓ_R
Power		$a_R \ \ell_R^2$
Heat Flux (Steam Generator)		ℓ_R

III. TWO-PHASE FLOW SIMILARITY

A. Similarity Parameters

The similarity parameters for a closed loop system under a two-phase flow condition can be obtained from the integral effects of the local two-phase flow balance equations along the entire loop. Under a natural circulation condition, the majority of transients are expected to be relatively slow. Furthermore, for developing system similarity laws, the response of the whole mixture is important rather than the detailed responses of each phase and phase interactions. Therefore, the drift-flux model formulation is more appropriate for the derivation of system similarity parameters under a natural circulation condition [5,6,13]. This is because the drift-flux model can properly describe the two-phase mixture-structure interactions over a wide range of flow conditions.

The similarity criteria based on the drift-flux model have been developed by Ishii and Zuber [6] and Ishii and Jones [5]. Two different methods have been used. The first method is based on the one-dimensional drift-flux model by choosing proper scales for various parameters. Since it is obtained from the differential equations, it has the characteristics of local scales. The second method is based on the small perturbation technique and consideration of the whole system responses. The local responses of main variables are obtained by solving the differential equations, then the integral effects are found. The resulting transfer functions are nondimensionalized. From these the governing similarity parameters are obtained.

The first method based on the balance equations of the drift-flux model is useful in evaluating the relative importance of various physical effects and mechanisms existing in the system. However, there are certain problems encountered in this method when it is applied to a system similarity analysis. In developing the similarity criteria, the most important aspect is to choose proper scales for various variables. However, this may not always be simple or easy, because in a natural circulation system the variables change over wide ranges. Therefore, the scaling parameters obtained from this method are more locally oriented than system oriented.

The second method requires that the field and constitutive equations are firmly established and that the solutions to the small perturbations can be obtained analytically for the system under consideration. When these conditions are met, it gives quite useful similarity laws.

Recently, the combination of the results obtained from the above two methods has been used by Ishii and Kataoka [8] to develop practical similarity criteria for a natural circulation system under two-phase flow conditions. Rather than duplicating this work, the essential parts of the similarity criteria are summarized below.

The important diemensionless groups which characterize the kinematic, dynamic, and energetic flow fields are obtained by nondimensionalizing the conservation equations based on the drift-flux model formulation. They are given as follows:

Kinematic Similarity Parameters:

$$\text{Phase Change Number, } N_{pch} \equiv \left(\frac{4\ \delta\ \dot{q}_o'''\ \ \ell_o}{d\ u_o\ \Delta H_{fg}\ \rho}\right)\left(\frac{\Delta\rho}{\rho_g}\right) \tag{66}$$

$$\text{Drift-Flux Number, } N_d \equiv \frac{V_{gj}}{u_o} \text{ (or Void-Quality Relation)}$$

$$\text{Density Ratio Group, } N\rho \equiv \frac{\rho_g}{\rho} \tag{67}$$

Dynamic Similarity Parameters:

$$\text{Froude Number, } N_{Fr} \equiv \frac{u_o^2}{g\ \ell_o\ \langle u_o \rangle}\ \frac{\Delta\rho}{\rho} \tag{68}$$

$$\text{Friction Number, } N_f \equiv \frac{f(\ell + \ell_e)}{d}\left\{\frac{1 + x\ (\Delta\rho/\rho_g)}{[1 + x\ (\Delta\mu/\mu_g)]^{0.25}}\right\}\left(\frac{a_o}{a_i}\right)^2 \tag{69}$$

$$\text{Orifice Number, } N_o \equiv K\ [1 + x^{3/2}\ (\Delta\rho/\rho_g)]\left(\frac{a_o}{a_i}\right)^2 \tag{70}$$

Energetic Similarity Parameters:

$$\text{Subcooling Number, } N_{sub} \equiv \left(\frac{\Delta H_{sub}}{\Delta H_{fg}}\right)\left(\frac{\Delta\rho}{\rho_g}\right) \quad (71)$$

$$\text{CHF (Critical Heat Flux) Number, } N_q \equiv \frac{q_c''}{\delta\, \dot{q}_{so}'''} \quad (72)$$

$$\text{Time Ratio Group, } T_i^* \equiv \left(\frac{a_s}{\delta^2}\frac{\ell_o}{u_o}\right)_i \quad (73)$$

$$\text{Heat Source Number, } Q_{si} \equiv \frac{\dot{q}_{si}'''\, \ell_o\, C_p}{\rho_{si}\, C_{psi}\, u_o\, \Delta H_{sub}} \quad (74)$$

The Froude, friction, orifice, and heat source numbers together with the time ratio group have their standard significance. On the other hand, the subcooling, phase-change, drift, density, and CHF numbers are associated with the two-phase flow systems.

The subcooling number takes into account the time-lag effects in the single-phase liquid region due to the subcooling of the fluid entering the heated section. It is one of the important parameters for the similarity of the systems.

The phase change number corresponds to Damköhler's Group I in chemical kinetics [14], and it scales the change of phase due to the heat transfer to the system. Since N_{pch} is the inverse of the nondimensional inlet velocity, it can be seen that N_{pch} is one of the decisive parameters for the kinematic similarity. Both N_{pch} and N_{sub} are significant not only for dynamics of the system but also for the description of the steady-state operational conditions. It can be shown from the steady-state energy balance over the heated section that N_{pch} and N_{sub} are related by

$$\left(\frac{\Delta\rho}{\rho_g}\right) x_e = N_{pch} - N_{sub} \quad (75)$$

where x_e is the vapor quality at the exit of the heated section.

The drift number takes into account the drift effects due to the relative motion of the fluids and thus plays a role in two-phase flow similar to that of Damköhler's Group II in diffusion processes. Since the vapor drift velocity V_{gj} depends on the flow regime [15], this group characterizes the flow pattern.

The density ratio group actually scales the dynamic effect of the system pressure, which is taken into account by the term $(\Delta\rho/\rho_g)$ also appearing in N_{sub}, N_{pch}, N_f, and N_o.

The CHF number replaces the groups related to the thermal boundary layer in single phase flow, i.e., the Stanton and Biot numbers. In two-phase flow systems with heating the boiling heat transfer is rather efficient, and the value of two-phase flow convective heat transfer coefficient is generally very high. In normal conditions, the wall superheat, $T_s - T_{sat}$ is relatively small. However, the occurrence of the critical heat flux is significant, because the heat transfer coefficient is drastically reduced at CHF. Therefore, in two-phase flow the simulation of the CHF condition is more important than that of the thermal boundary layer. The occurrence of CHF can be considered as a flow regime transition due to a change in heat transfer mechanisms.

It should be noted that the constitutive relations for the relative motion between two phases, hence for the drift velocity, V_{gj}, and the critical heat flux, q''_c, should be specified in the above similarity groups.

The relative motion can be specified by a number of different forms. The representative constitutive equation [20] for the relative motion based on the drift velocity correlation is given by

$$V_{gj} = 0.2\ (1 - \sqrt{\rho_g/\rho})\ j + 1.4 \left(\frac{\sigma g \Delta\rho}{\rho^2}\right)^{1/4} \qquad (76)$$

where the volumetric flux, j, in the heated section is given by

$$j = [1 + x\ (\Delta\rho/\rho_g)]\ u_o \qquad (77)$$

On the other hand, the relative motion based on the classical void-quality correlation may be expressed by

$$\alpha = \alpha\,(x, \rho_g/\rho, \mu_g/\mu_f, \text{etc.}) \qquad (78)$$

Equation (78) is mathematically equivalent to Eq. (76).

In view of Eqs. (76) and (77), the relative motion similarity based on the drift velocity correlation becomes

$$N_d = 0.2\ (1 - \sqrt{\rho_g/\rho})\ [1 + x\ (\Delta\rho/\rho_g)] + \frac{1.4}{u_o}\left(\frac{\sigma g \Delta\rho}{\rho^2}\right)^{1/4} \qquad (79)$$

or it should have the same void quality correlation given by Eq. (78), $\alpha = f(x)$.

The CHF condition at low flow has been reviewed by Leung [16], Katto [17], and Mishima and Ishii [18]. The modified Zuber correlation [19] for low flow rate is given by

$$q_c'' = 0.14\ (1-\alpha)\ \rho_g\ \Delta H_{fg}\left(\frac{\sigma g \Delta\rho}{\rho_g^2}\right)^{1/4} \qquad (80)$$

Based on the limited data on blowdown experiments it is recommended [16] for the mass velocity range of -240 to 100 kg/m^{2}s. This correlation is based on a pool boiling CHF mechanism. Thus it may apply only for transients involving flow reversal.

Katto's correlation [17] for low mass flow rate is given by

$$q_c'' = \frac{1}{4}\ \Delta H_{fg}\ G\ \frac{d_o}{\ell_o}\left[\left(\frac{\sigma g}{G^2 \ell_o}\right)^{0.043} + \frac{\Delta H_{sub}}{\Delta H_{fg}}\right] \qquad (81)$$

which implies that the critical quality is $x_c = (\sigma_g/G^2\rho_o)^{0.043}$. Here G is the mass velocity. The typical value of x_c is 0.5-0.8, thus the underlining mechanism should be the annular flow-film dryout. This correlation can be applied to most slow transient situations at low flow rate range.

However, there is a possibility [18] that the critical heat flux may occur at much lower exit quality than that given above due to a change in two-phase flow regimes. In a natural circulation system with very small flow fluctuations, the occurrences of CHF have been observed at the transition between the churn-turbulent to annular flows. Beyond this transition, the lack of large disturbance waves eliminated the preexisting rewetting of dry patches. This leads to the formation of permanent dry patches and CHF. The criteria developed by Mishima and Ishii [18] for this case is given by

$$q_c'' = \frac{d_o}{4\, \ell_o} \left[\left(\frac{1}{C_o} - 1 \right) \Delta H_{fg} \sqrt{\rho_g \; \Delta\rho g d_o} + G \; \Delta H_{sub} \right] \tag{82}$$

where C_o is the distribution parameter for the drift-flux model [20] and given by

$$C_o = 1.2 - 0.2 \sqrt{\rho_g/\rho} \tag{83}$$

These CHF criteria should be used to develop a similarity criterion for the fluid-solid boundary. This ensures that the critical heat flux occurs under the similar condition in a simulated system.

B. Similarity Laws

Equations (66) through (74) represent relationships among the dimensionless groups and the generalized variables of a two-phase flow system. The dimensionless groups must be equal in the model and prototype if the similarity laws are to be satisfied. In general, scaling of two-phase flow dynamics with complete similarity is practically impossible due to the large number of nondimensional groups. However, a scale model with the same fluid under same system pressure results in significant amount of simplifications. In this case, all the fluid properties can be considered the same between the model and prototype, and the similarity criteria become

$$(N_{pch})_R = \frac{\delta_R \; \dot{q}_R''' \; \ell_R}{d_R \; u_R} = 1 \tag{84}$$

$$(N_{Fr})_R = \frac{u_R^2}{\ell_R \, \langle\alpha\rangle_R} = 1 \tag{85}$$

$$(N_f)_R = \left(\frac{f(\ell + \ell_e)}{d}\right)_R \left(\frac{a_o}{a_i}\right)_R^2 = 1 \tag{86}$$

$$(N_o)_R = K_R \left(\frac{a_o}{a_i}\right)_R^2 = 1 \tag{87}$$

$$(T^*)_R = \frac{\ell_R}{\delta_R^2 \, u_R} = 1 \tag{88}$$

$$(Q_{si})_R = \frac{\dot{q}_R''' \, \ell_R}{u_R \, \Delta H_{subR}} = 1 \tag{89}$$

$$(\Delta H_{sub})_R = 1 \tag{90}$$

$$(N_d)_R = \frac{V_{gj})_R}{u_R} = 1 \text{ or } N_d)_R = \alpha_R = 1 \tag{91}$$

$$(N_q)_R = \frac{q_c''}{\delta \, \dot{q}_{so}'''} = 1 \tag{92}$$

The drift flux number requirement Eq. (91) can be automatically satisfied, if the contribution of the local slip is small in comparison with the slip due to the transverse velocity and void profiles. When the local slip, i.e., the second term on R.H.S. of Eq. (79), is the dominant factor for the relative motion, the similarity requirement is $u_R = 1$ which is a rather severe restriction. From $(N_d)_R = 1$ one obtains,

$$\begin{array}{l} \text{Distribution Controlled Slip} \rightarrow \text{Automatically Satisfied} \\ \text{Local Slip Controlled} \rightarrow u_R = 1 \end{array} \tag{93}$$

For most cases the first condition applies. Even in the second case, the distortion of the velocity will introduce limited changes in the void-quality relation. Therefore, in the following analysis the first condition will be assumed.

In view of requirements expressed by Eqs. (84) and (90) and the quality relation given by Eq. (75), it is evident that basically the similarity in terms of the vapor quality is met. Thus,

$$x_R = 1 \tag{94}$$

And from Eq. (93) this implies that the void similarity also exists,

$$\alpha_R = 1 \tag{95}$$

Furthermore, using the Katto CHF criterion given by Eq. (81) and N_{pch} similarity criterion, it can be shown that the critical heat flux requirement expressed by Eq. (92) becomes

$$(x_c)_R = \left(\frac{1}{u_R^2 \ \ell_R}\right)^{0.043} = 1 \tag{96}$$

It is noted that the exponent of the above expression is very small. Therefore, it may be assumed that Eq. (96) is approximately satisifed if the quality similarity given by Eq. (94) holds.

Physically, the subcooled liquid number is mainly determined by the excess cooling in the condensation section of the loop. The similarity analysis becomes much more complicated when there is not sufficient cooling to condense all of the steam generated in the heated section or the subcooling cannot be well controlled by the condensation section. In such a case, detailed modeling of condensation process and the analysis of the secondary loop may become necessary to determine the exit quality or subcooling at the condensation section. In this analysis it is assumed that the subcooling number is well controlled in the condensation section. In this case, the subcooling number requirement, Eq. (90), can be assumed to be satisfied.

Hence, excluding the friction similarity conditions, together with the requirement imposed by the drift number and CHF number, it is required that

$$\left(\frac{\delta_R}{d_R}\right)\left(\frac{\dot{q}_R''' \ \ell_R}{u_R}\right) = 1 \tag{97}$$

$$\frac{u_R^2}{\ell_R} = 1 \tag{98}$$

$$\frac{\ell_R}{\delta_R^2 \ u_R} = 1 \tag{99}$$

$$\frac{\dot{q}_R''' \ \ell_R}{u_R} = 1 \tag{100}$$

The above set of equations comprise a set of four equations in the following five unknowns δ_R, d_R, u_R, ℓ_R, and $\dot{q}_R'''$. Therefore, one of the unknowns must be specified before a solution of the set can be obtained. The obvious choice is the system length ratio because it is limited by the size of the model. With this choice it becomes necessary that

$$u_R = \sqrt{\ell_R} \tag{101}$$

$$\dot{q}_R''' = \frac{1}{\sqrt{\ell_R}} \tag{102}$$

and

$$\delta_R = d_R = \sqrt{\frac{\ell_R}{u_R}} = (\ell_R)^{1/4} \tag{103}$$

These requirements identically satisfy the single phase scaling criteria on the velocity expressed by Eq. (36) and approximately satisfy the CHF criterion given by Eq. (96) because when Eq. (101) is substituted in Eq. (96) we obtain

$$1 < x_{cR} < 1.22 \text{ for } 1 > \ell_R > 0.1 \tag{104}$$

However, the real time simulation cannot be achieved in case of two-phase flow due to the additional conditions imposed on the system. The time scale is distorted by

$$t_R = \frac{\ell_R}{u_R} = \sqrt{\ell_R} \tag{105}$$

Therefore, the time events will be accelerated in the model for $\ell_R < 1$. Finally, it should be noted that Eq. (102) implies that the power to the system should be increased in the model for $\ell_R < 1$.

To summarize, the effects of each of the terms appearing in the conservation equations are preserved in the model and prototype without any distortion if one satisfy the requirements of:

a) Equal Pressure and Properties:

$$\frac{P_m}{P_\rho} = \frac{\rho_m}{\rho_p} = \frac{\mu_m}{\mu_p} = \frac{C_{pm}}{C_{pp}} = \frac{\rho_{gm}}{\rho_{gp}} = \frac{\mu_{gm}}{\mu_{gp}} = \frac{C_{pgm}}{C_{pgp}} = 1 \tag{106}$$

b) Scaling Relations:

$$\frac{u_m}{u_p} = \sqrt{\frac{\ell_m}{\ell_p}} \tag{107}$$

$$\frac{\dot{q}_m'''}{\dot{q}_p'''} = \sqrt{\frac{\ell_p}{\ell_m}} \tag{108}$$

which imply time distortion

$$\frac{t_m}{t_p} = \sqrt{\frac{\ell_m}{\ell_p}} \qquad (109)$$

c) Equal Area Ratios:

$$\frac{(a_i/a_o)_m}{(a_i/a_o)_p} = 1 \qquad (110)$$

d) Equal Friction Numbers:

$$\frac{\left[\sum_i f_i \left(\frac{\ell_i}{d_i} + \frac{\ell_{ei}}{d_i}\right)\right]_m}{\left[\sum_i f_i \left(\frac{\ell_i}{d_i} + \frac{\ell_{ei}}{d_i}\right)\right]_p} = 1 \qquad (111)$$

$$\frac{K_{im}}{K_{ip}} = 1 \qquad (112)$$

If some of these requirements are not satisfied, then the effects of some of the processes occurring in the model and prototype will be distorted.

C. Scaling Law Results

As it has been done in the single phase case, one can determine power and other loop component characteristic parameters that satisfy the scaling laws presented above.

Power: $\dot{P}$

The power is obtained from overall energy generation rate, that is, from

$$\dot{P} = a_{so} \, \ell_o \, \dot{q}_o''' \qquad (113)$$

In terms of ratios, it can be expressed as

$$P_R = a_R \, \ell_R \, \dot{q}_o''' = a_R \sqrt{\ell_R} \tag{114}$$

Number of Rods: n

With the rod diameter of δ_s, the number of rods can be obtained as follows

$$n = 4 \, a_{so}/\pi\delta_s^2 \tag{115}$$

Then the number-of-rod ratio can be given by

$$n_R = \frac{a_R}{\delta_s^2} \tag{116}$$

It is to be noted that in two-phase similarity, the rod diameter and hydraulic diameter ratio should be such that Eq. (103) should be satisfied. However, from the practical point of view it is desirable that

$$\delta_R = d_R = 1 \tag{117}$$

Since the exponent is very small on the right hand side of Eq. (103), it can be assumed that Eq. (117) is approximately satisfied. With Eq. (103), Eq. (116) becomes,

$$n_R = a_R \tag{118}$$

which is the same as the requirement obtained in the single-phase flow.

Heat Flux: q"

From the energy balance

$$q'' = P/(\pi \, \delta_s \, \ell_o \, n) \tag{119}$$

and the heat flux ratio becomes

$$q_R'' = P_R/(\delta_{SR} \, \ell_R \, n_R) \tag{120}$$

In view of Eqs. (114), (117), and (116), Eq. (120) becomes

$$q_R'' = 1/\sqrt{\ell_R} \tag{121}$$

Note that the heat flux is increased when compared with the single-phase case.

The two-phase scaling laws are summarized in Table II.

D. Additional Considerations of Scaling Laws With Respect to Flow Regime Transition

During a small break, the transition of most concern is that from stratified two-phase flow regime into a slug or to annular-dispersed flow in a horizontal pipe, since the large size of a PWR pipe is expected to promote separated flow during slow transients.

There are several two-phase flow regime maps which can be used to estimate flow regime transitions in horizontal pipes. Zuber [7] has investigated scaling criteria associated with flow regime transition from stratified to slug or to annular flow in a horizontal pipe. The analysis was done using two-phase flow parameters and the Taitel and Dukler [21] flow regime transition criteria.

Taitel and Dukler have shown that this transition boundary can be described in terms of two dimensionless parameters; the turbulent flow Martinelli parameter

$$X_{tt} = \left(\frac{\mu}{\mu_g}\right)^{0.1} \left(\frac{\rho}{\rho_g}\right)^{0.4} \left(\frac{j_f}{j_g}\right)^{0.9} \tag{122}$$

and a modified Froude number given by

$$Fr = \left(\frac{\rho_g \, j_g^2}{g\Delta\rho d}\right)^{1/2} \tag{123}$$

The transition boundary was shown by Taitel-Dukler to occur for values of $X_{tt} > 1.6$. As further addressed by these authors for this value of X_{tt}, the dimensionless liquid depth in a horizontal pipe should be 0.5 and the value of Fr is 0.6. Therefore, in general, the transition occurs for pipe dimensionless

Table II. Two-Phase Flow Time-Distorted Scaling Laws (Satisfies single-phase scaling laws also with time distortion as indicated in the table)

Scaled Quantity		(Model/Prototype) Ratio
Length		ℓ_R
Hydraulic Diameter	For Core and Steam Generator Only	≈ 1
Rod Diameter	For Core and Steam Generator Only	≈ 1
Number of Rods	For Core and Steam Generator Only	$\approx a_R$
Flow Area		a_R
Volume		$a_R \ \ell_R$
Velocity		$\sqrt{\ell_R}$
Time		$\ell_R/u_R = \sqrt{\ell_R}$
System Pressure (Fluid Properties)		1
Solid Structure Properties		1
Heat Generation/Volume/Unit Time		$1/\sqrt{\ell_R}$
Power		$a_R\sqrt{\ell_R}$
Heat Flux (Steam Generator)		$1/\sqrt{\ell_R}$

liquid levels larger than 0.5 or void fractions in the pipe less than or equal to 0.5. As shown in Ref. [21], the dimensionless liquid level (H_ℓ/d) is an explicit function of the Martinelli parameter and since the void fraction and dimensionless liquid level are uniquely related, Zuber indicated that the modified Froude number is an explicit function of void fraction. Consequently, it appears that if transition is to take place at the same void fraction in different pipe sizes, the modified Froude number must be equal or from Eq. (123)

$$Fr_R \equiv \frac{Fr_m}{Fr_p} = \frac{(\rho_g\, j_g^2/g\Delta\rho d)_m^{1/2}}{(\rho_g\, j_g/g\Delta\rho d)_p^{1/2}} = 1 \qquad (124)$$

If the model and the prototype have the same fluid and the processes occur at the same system pressure, Eq. (124) reduces to

$$\frac{j_{gm}}{j_{gp}} = \sqrt{\frac{d_m}{d_p}} = \left(\frac{a_m}{a_p}\right)^{1/4} \qquad (125)$$

The preceding result will be applied to two-phase, time distorted natural convection system scaling. It is shown above that the two-phase, time distorted natural convection scaling leads to

$$u_R = \sqrt{\ell_R} \qquad (126)$$

Rewriting Eq. (124) and substituting

$$\frac{j_{gm}}{j_{gp}} = u_R = \sqrt{\ell_R} \qquad (127)$$

in the resulting equation, one gets

$$Fr_R \equiv \frac{Fr_m}{Fr_p} = \sqrt{\frac{\ell_R}{\sqrt{a_R}}} \tag{128}$$

It can be seen that the modified Froude number scaling when combined with the scaling criteria obtained in the preceding sections, results in a scale distortion given by Eq. (128).

The results are plotted on Fig. 2 for $a_R = 1/300$ with ℓ_R as a parameter. It is apparent from this figure that for the case of equal void which implies that the model and the prototype are geometrically similar, the constant void line intersects the prototype and model curves at different values of Fr. Thus, if conditions in the prototype are such that a flow transition occurs, say from separated to slug flow, then the hot leg in the model will have slug flow.

It can be concluded, therefore, that Froude number scaling when used in conjunction with the general scaling criteria developed in Section III, leads to a distortion so that one cannot satisfy simultaneously geometric similarity (equality of void fractions) and equality of Froude numbers. If one is satisfied, the other is not and vice versa. However, considering the fact that the modified Froude number range lies over several order of magnitude for a minor change in the (H_ℓ/d) ratio, i.e., for a minor change in the void fraction, violation of this requirement does not seem to be severe. Furthermore, it should also be noted that as Eq. (128) indicates, the flow regime transition criterion can be satisfied by changing the dimension of the horizontal section of the hot leg so that

$$\frac{\ell_m}{\ell_p}\left(\frac{d_p}{d_m}\right)^{1/4} = 1 \tag{129}$$

is satisifed. For example, for an area ratio of 1/300, i.e., for a diameter ratio of 1/ 300, the modified Froude number criterion can be satisfied for $\ell_m/\ell_p = 0.24$ in the inverted U-tube section of the hot leg. Similarly, for a length ratio of 0.3, the diameter ratio should be 0.09 instead of 0.06.

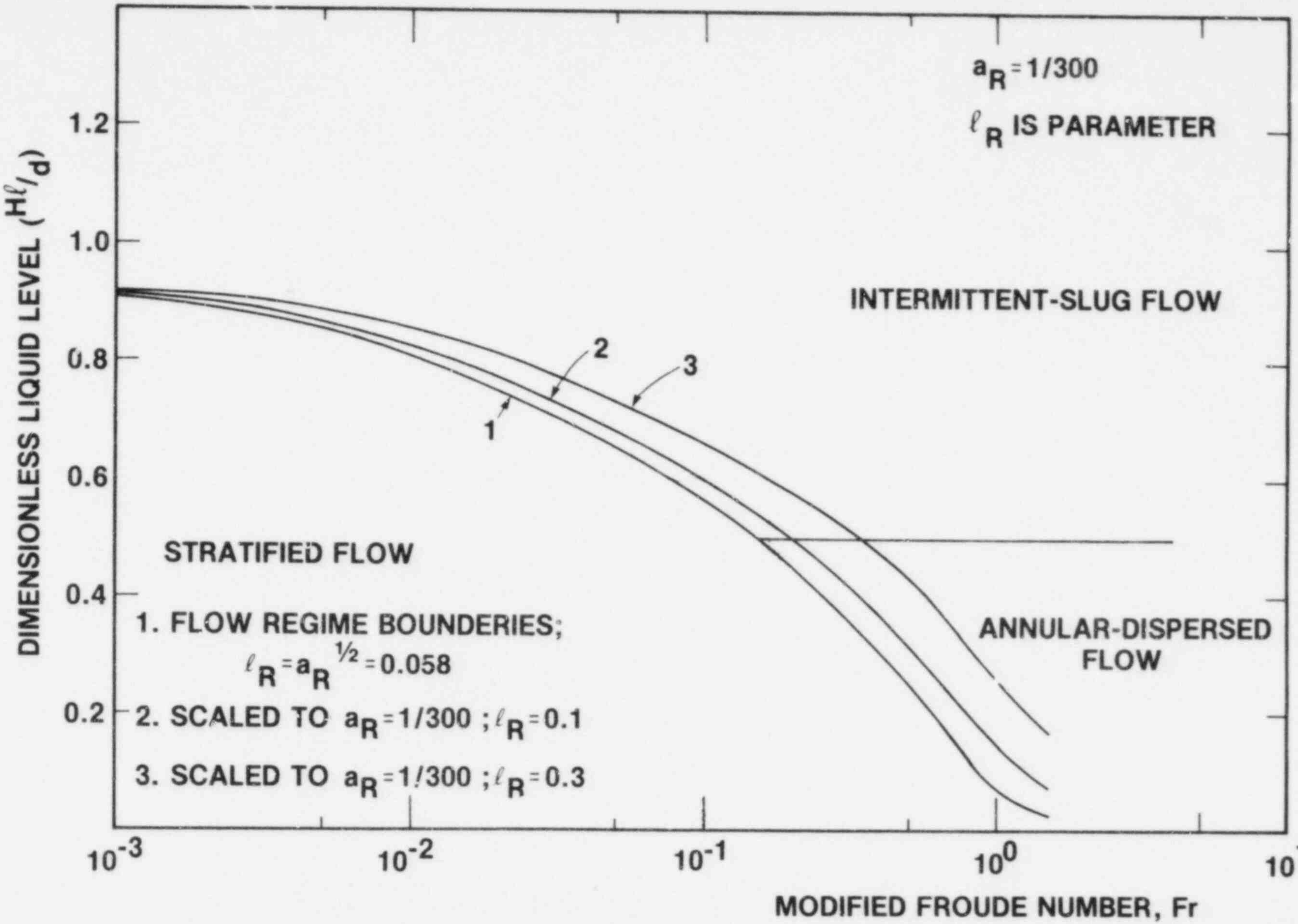

Fig. 2. Dukler-Taitel Flow Regime Map

As a summary, it can be concluded that the distortion introduced by the modified Froude number is not a very severe one in terms of the general requirements of the scaling criteria presented for the two-phase flow case.

IV. APPLICATIONS

A. Choice of Prototype

Since the accident occurred at the Three-Mile Island Unit-2 plant, numerous issues have been raised in regard to 2 x 4 loop NSSS designs [22]. Some of the issues are generic in nature and others are design specific. In particular, concerns have been raised relevant to the Babcock and Wilcox (B&W) NSSSs largely because of some unique components and hardware features contained in the plant design in addition to the 2 x 4 loop geometry. It was suggested that unique design features of the B&W reactor may produce unique and complex thermo-hydraulic behavior during small break loss of coolant accident or during some other abnormal transient that could be misleading to an operator.

Perhaps the most distinguishing feature of the B&W plant design relative to the other PWR vendor designs is the once-through steam generator (OTSG) as illustrated in Fig. 3. In the OTSG system, primary fluid flows downward through the tubes in the steam generator. The secondary feed enters the steam generator secondary riser section from the bottom. The design is such that superheated steam can be produced whereas, the U-tube in shell design, (Fig. 1), used by other vendors can only produce saturated steam. However, use of the OTSG in the B&W NSSS necessitates the use of a vertical hot leg, with a 180 degree U-bend at the top to route the primary fluid from the reactor vessel to the top of the steam generator.

After a thorough study of the technical issues unique to 2 x 4 loop plant design, and more specifically the unique design features of the B&W design, it was suggested by Larson et al. [22] that the best possible execution of a test program to resolve the issues can be affected in a new, well-scaled, 2 x 4 loop simulator. Therefore, the scaling concepts presented in the preceding sections are applied here to obtain a 2 x 4 loop facility conceptual modeling of a B&W 177 NSSS lowered loop plant design, Fig. 4.

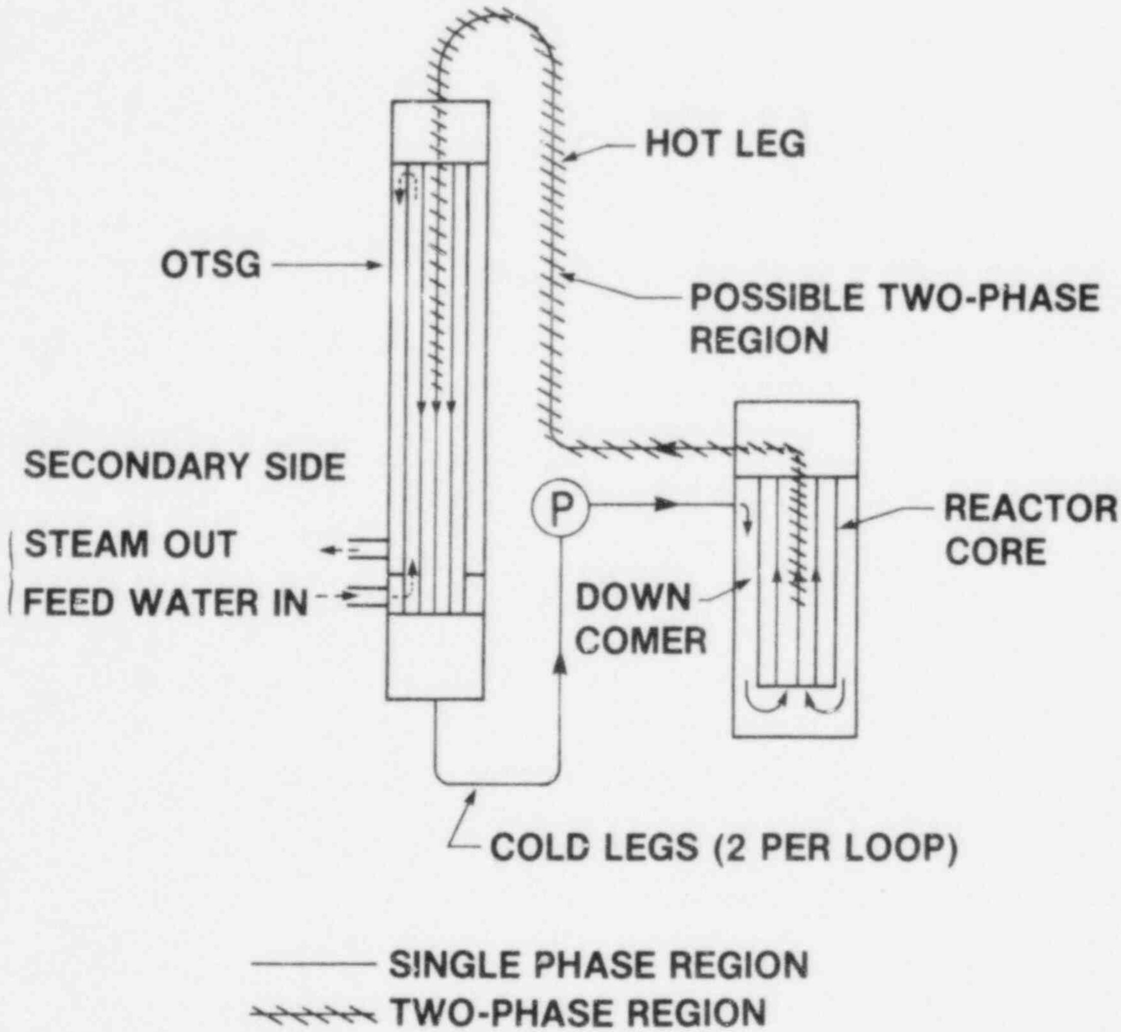

Fig. 3. Typical Loop with Once Through Steam Generator

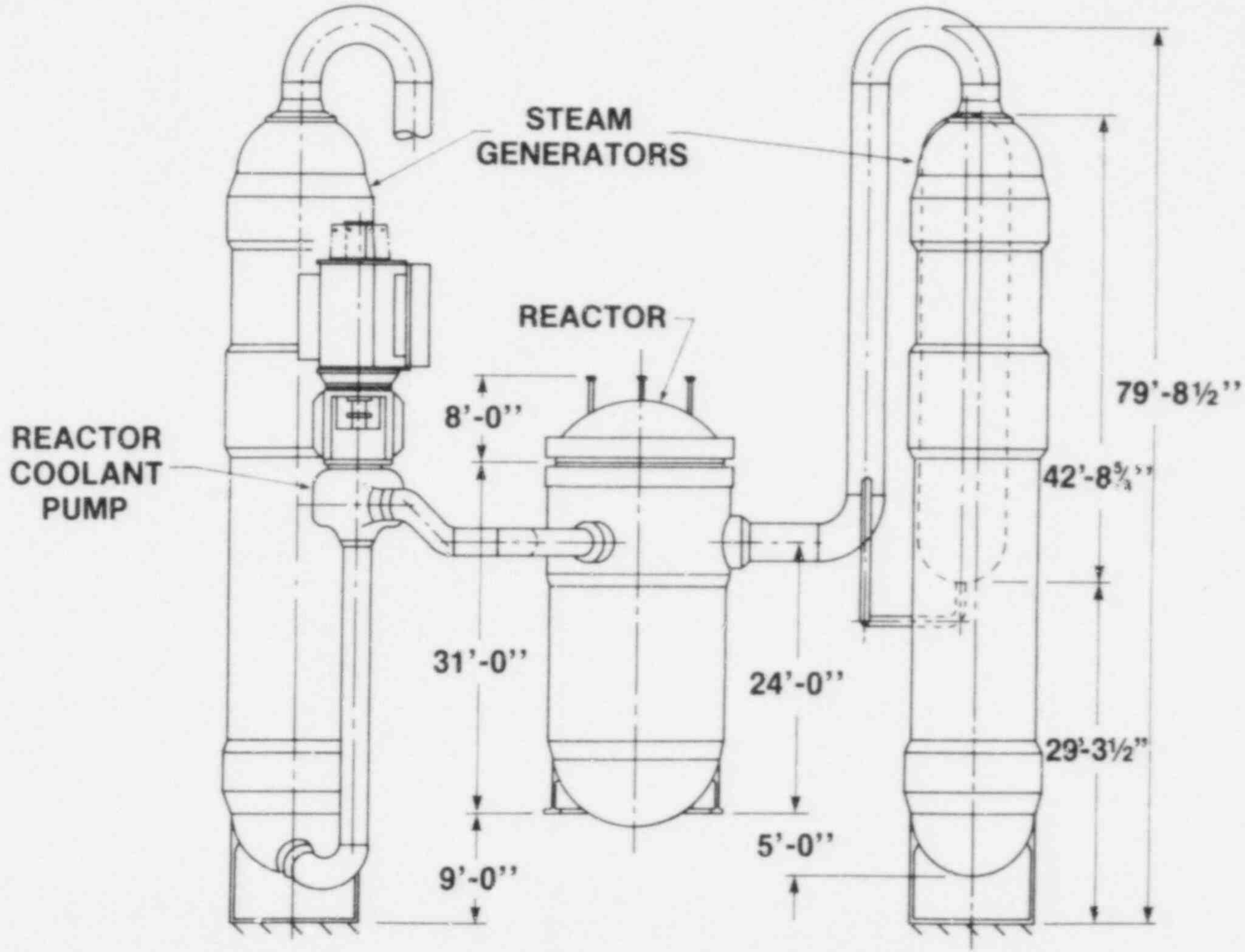

Fig. 4. Babcock & Wilcox Low Loop Plant Design Arrangement [22]

The proposed 2 x 4 loop scaled model as shown schematically in Fig. 5 contains representative components of all thermo-hydraulic systems considered important in performing tests to obtain data representative of the response of the prototye plant, Fig. 4. This system contains an electrically powered reactor vessel simulator and two loops with representative hot leg, once through steam generator, and two active cold legs with pumps. For the purpose of this study the TMI Unit-2 plant is used as the prototype B&W 177 NSSS plant. The prototype system parameters are summarized in Table III.

To obtain the model requirements three separate circumstances are distinguished. They are enumerated as follows:

1. Single-phase forced convection system, time-preserving scaling,
2. Single-phase and two-phase forced convection system, time-distorted scaling,
3. Single-phase and two-phase natural convection system, time-distorted scaling.

The model requirements to meet for each case are described in the following sections.

B. Single-Phase Forced Convection System, Time-Preserving Scaling Model Requirements

From the single-phase flow similarity analysis presented in Section II, it has been concluded that the following conditions should be satisfied;

$$u_R = \ell_R \tag{130}$$

$$\dot{q}_R''' = \ell_R \tag{131}$$

and

$$\delta_R = d_R = 1 \tag{132}$$

Under these conditions, the time preserving scaling is established, thus one has

$$t_R = \frac{t_m}{t_p} = 1 \tag{133}$$

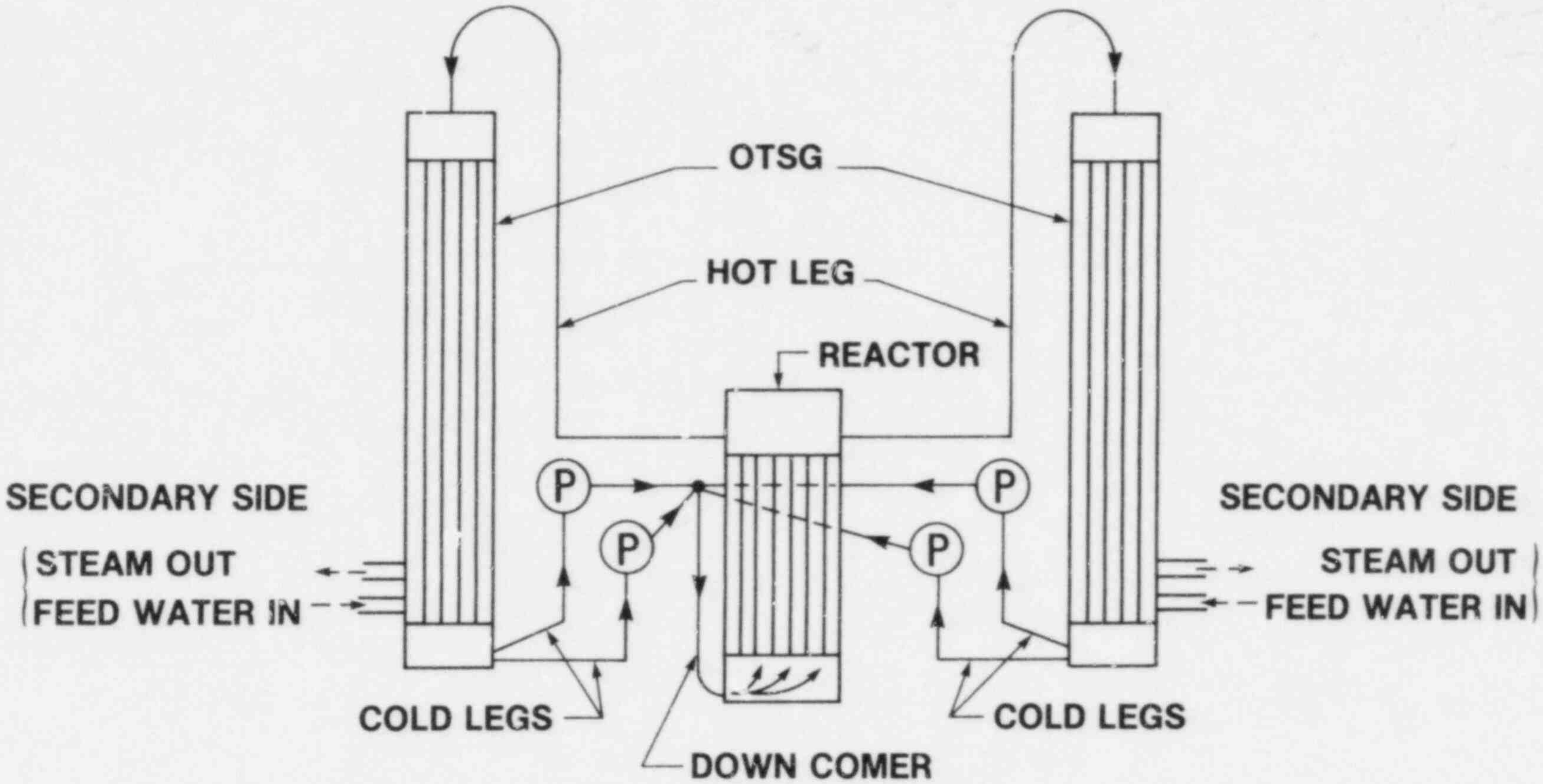

Fig. 5. Schematic 2 x 4 Loop Model (Not to Scale)

Table III. Single-Phase Only Forced Convection Prototype and Practically Optimum Scaled Model System Parameters

COMPONENT	PROTOTYPE	MODEL
System Pressure (bar)	148.6	148.6
Area Scale Factor	1/1	1/500
Optimum Length Ratio	1.0	0.194
CORE		
Number of Rods	36,816	73
Rod Diameter (m)	1.092×10^{-2}	1.092×10^{-2}
Rod Pitch (m)	1.443×10^{-2}	1.443×10^{-2}
Hydraulic Diameter	1.326×10^{-2}	1.326×10^{-2}
Flow Area (m^2)	4.572	9.15×10^{-3}
Length (m)	4.206	0.816
Velocity (m/s)	5.125	0.994
Mass Flow Rate (kg/s)	16,682	6.472
Reynolds Number	5.41×10^{5}	1.05×10^{5}
Friction Number	10.604	2.976
Power (MWt)	2772	0.212
Time Ratio	1.0	1.0
HOT LEG (1 OF 2)		
Diameter (m)	0.915	4.09×10^{-2}
Flow Area (m^2)	0.658	1.32×10^{-3}
Length (m)	21.123	4.098
Velocity (m/s)	19.524	3.788
Mass Flow Rate (kg/s)	8820.0	3.422
Reynolds Number	1.45×10^{8}	1.26×10^{6}
Friction Number	1.515	1.515
Time Ratio	1.0	1.0
STEAM GENERATOR (1 OF 2)		
Number of Tubes	15,531	31
Diameter of Tubes (m)	1.42×10^{-2}	1.42×10^{-2}
Flow Area (m^2)	2.441	4.88×10^{-3}
Length (m)	15.880	3.081

Table III (Cont'd)

STEAM GENERATOR (1 OF 2) (Cont'd)	PROTOTYPE	MODEL
Velocity (m/s)	5.075	0.985
Mass Flow Rate (kg/s)	8820.0	3.422
Reynolds Number	5.72×10^5	1.11×10^5
Friction Number	16.686	4.560
Time Ratio	1.0	1.0
COLD LEG (1 OF 4)		
Pump Suction Section		
Diameter (m)	0.711	3.18×10^{-2}
Flow Area (m^2)	0.397	7.94×10^{-4}
Length (m)	11.126	2.158
Velocity (m/s)	15.071	2.924
Mass Flow Rate (kg/s)	4410.0	1.711
Reynolds Number	8.39×10^7	7.28×10^5
Friction Number	1.060	1.019
Time Ratio	1.0	1.0
Pump Discharge Section		
Diameter (m)	0.711	3.18×10^{-2}
Flow Area (m^2)	0.397	7.94×10^{-4}
Length (m)	8.810	1.709
Velocity (m/s)	15.071	2.924
Mass Flow Rate (kg/s)	4410.0	1.711
Reynolds Number	8.39×10^7	7.28×10^5
Friction Number	1.047	0.813
Time Ratio	1.0	1.0
DOWNCOMER		
Hydraulic Diameter (m)	0.505	9.91×10^{-2}
Flow Area (m^2)	3.855	7.71×10^{-3}
Length (m)	6.523	1.265
Velocity (m/s)	5.872	1.139
Mass Flow Rate (kg/s)	16682.0	6.472
Friction Number	0.154	0.153
Time Ratio	1.0	1.0

In addition to the above three constraints, the dynamic similarity and geometric similarity conditions are required. Thus,

$$F_R = \left\{ \left[\sum_i f_i \left(\frac{\ell_i}{d_i} + \frac{\ell_{ei}}{d_i} \right) + K_i \right] \left(\frac{a_o}{a_i} \right)^2 \right\}_R = 1 \tag{134}$$

and

$$\left(\frac{\ell_i}{\ell_o} \right)_R = \left(\frac{a_i}{a_o} \right)_R = 1 \tag{135}$$

Equation (134) requires that the flow resistance similarity be satisfied only for a whole loop, but not for each section separately. However, for a multi-loop system under various transient or accident conditions it is desirable to have similarity conditions for each section separately. Thus,

$$F_{iR} = \left[f_i \left(\frac{\ell_i}{d_i} + \frac{\ell_{ei}}{d_i} \right) + K_i \right]_R = 1 \tag{136}$$

Among the above constraints, Eq. (132) stands for the similarity of the thermal conduction in the solid structure. Generally, it is difficult to satisfy this condition in a scale model. However, it is easily satisfied in the core and steam generator sections where most of the heat transfer takes place.

The most severe condition in terms of the thermo-hydraulic simulation is imposed by Eq. (136), because in a scaled model the hydraulic diameter can be much smaller in piping system. Therefore, for a given value of a_R, calculations are centered to determine ℓ_R to meet $F_R = 1$ for each section as required by Eq. (136). A computer code which is capable of calculating F_i for each section among other system parameters for the prototype and scaled model was developed. (See Appendix for the computer code.) The results gathered from a series of computer calculations indicated that the hot leg simulation imposes the strongest constraint. Once the necessary condition for the hot leg is obtained, the other sections are easily adjusted by increasing the minor loss coefficients. Therefore, in what follows the basic scaling criterion from the hot leg is presented.

The prototype hot leg flow resistence consists of the commercial steel pipe friction and the distributed loss due to elbows. The distributed loss factor, $(f_i\, \ell_{ei}/d_i)$, was calculated to be about 1.2704 for the prototype. For a scale model, lower $(f_i\, \ell_{ei}/d_i)$ value is desirable due to the requirement imposed by Eq. (136). Therefore, the friction factor for a drawn tubing is used. Two different cases are distinguished for (ℓ_{ei}/d_i). One, in which the distributed flow restriction is assumed to be negligibly small, is <u>the ideal case</u>. The other, in which the practically minimum values for the flow restrictions are used, is <u>the practically optimum case</u>. The latter can be achieved by using large radius elbows [23].

The solution to the similarity criteria for both ideal and practically optimum cases is presented in Fig. 6 where a_R is the flow area ratio. The solution can be approximated by

$$\ell_R = 13.18\ (1/a_R)^{-0.64} \tag{137}$$

for the ideal case, and

$$\ell_R = 15.24\ (1/a_R)^{-0.70} \tag{138}$$

for the practically optimum case.

Similarly, volume ratio for both cases are given, respectively, as follows:

$$V_R = 13.18\ (1/a_R)^{-1.64} \tag{139}$$

and

$$V_R = 15.24\ (1/a_R)^{-1.70} \tag{140}$$

Hence, for a sample case of $V_R = 1/815.3$ [22], one obtains the following solutions for the practically optimum modeling:

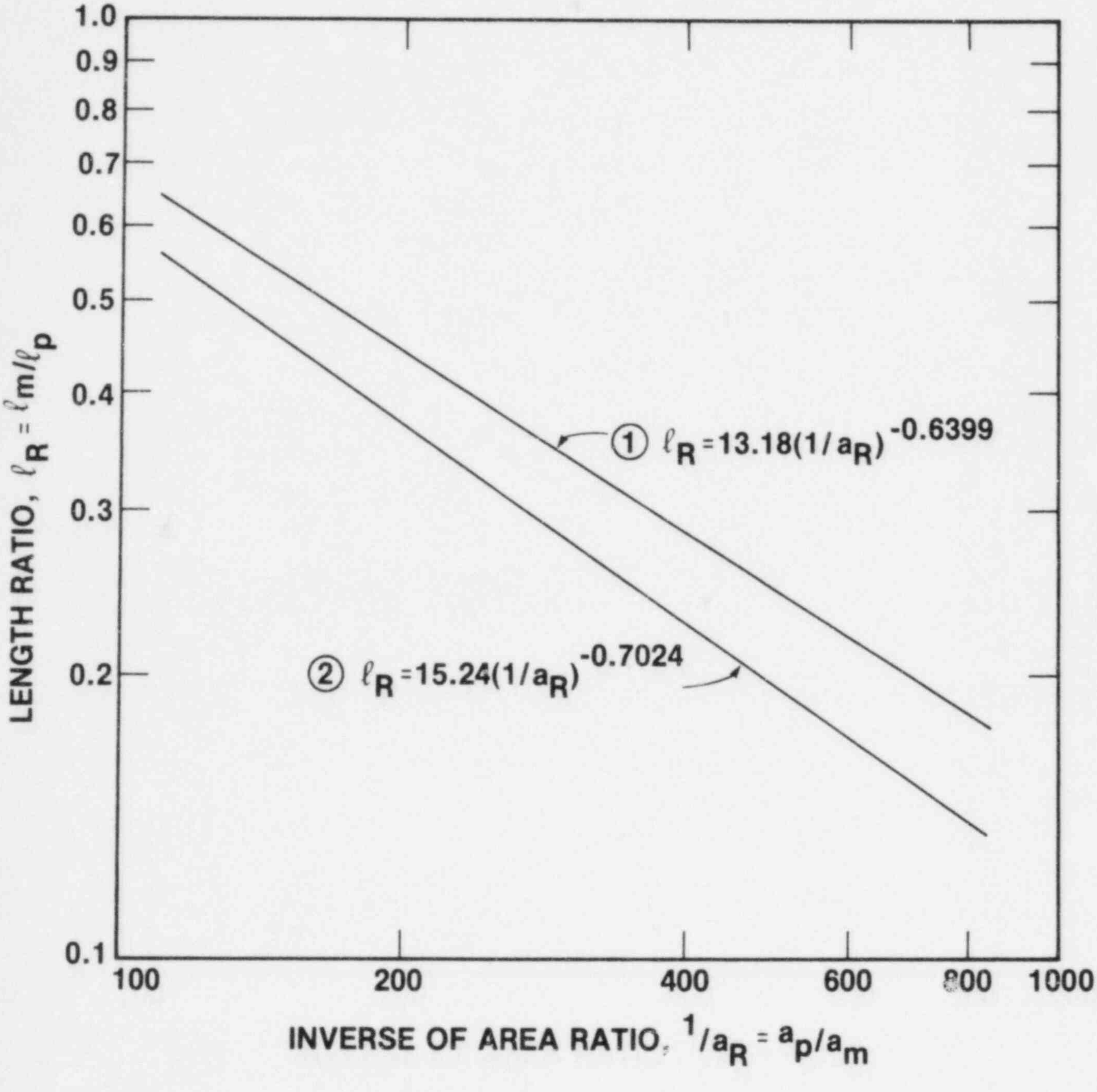

Fig. 6. Single-Phase Forced Convection System, Time Preserving Scaled Model Requirements
1. Ideal Case
2. Practically Optimum Case

$$a_R = 1/254.1$$

$$u_R = \dot{q}_R''' = \ell_R = 0.312 \tag{141}$$

These imply that the flow area should be reduced by 1/254.1 whereas the velocity, the power density, and the length should be reduced by 0.312. The same criteria apply to the other components of the model.

A summary of the important system parameters for the prototype and the practically optimum scaled model of $a_R = 1/500$ case is provided in Table III. It should be noted from the table that in order to meet the condition imposed by Eq. (136), additional flow restrictions must be introduced to those components of the loop where $F_{im} < F_{ip}$. From the practical point of view, this does not introduce additional difficulty in designing the model plant.

Finally, it is to be noted here that, with Eqs. (137) and (138), two phase flow cannot be simulated. For the simultaneous single-phase and two-phase simulations, please see the sections that follow.

C. Single-Phase and Two-Phase Forced Convection System, Time-Distorted Scaling

From the two-phase flow similarity analysis presented in Section III, the following conditions are necessary for two-phase flow simulations:

$$u_R = \sqrt{\ell_R} \tag{142}$$

and

$$\dot{q}_R''' = 1/\sqrt{\ell_R} \tag{143}$$

With these conditions imposed on the modeling, the time is distorted by

$$t_R = \frac{t_m}{t_p} = \sqrt{\ell_R} \tag{144}$$

For $\ell_R < 1.0$, this implies that the time runs faster in the scale model by a factor of $\sqrt{\ell_R}$.

In addition to the above constraints, the dynamic and the geometrical similarity conditions expressed by Eqs. (135) and (136), respectively, are required for a proper scaling.

The solution to the similarity criteria for both ideal and practically optimum scaled models is presented in Fig. 7. It is approximately given by

$$\ell_R = 12.0\ (1/a_R)^{-0.614};\ V_R = 12.0\ (1/a_R)^{-1.614} \tag{145}$$

for the ideal case, and

$$\ell_R = 10.23\ (1/a_R)^{-0.622};\ V_R = 10.23\ (1/a_R)^{-1.622} \tag{146}$$

for the practically optimum modeling case.

Hence, for a sample case of $V_R = 1/815.3$ [22], one obtains the following solutions for the practically optimum modeling:

$$\begin{aligned} a_R &= 261.5 \\ \ell_R &= 0.321 \\ u_R &= 0.566 \\ \dot{q}_R''' &= 1.765 \end{aligned} \tag{147}$$

and

$$t_R = 0.566$$

Equation (147) indicates that the flow area, length, and velocity should be reduced by 1/261.5, 0.321, and 0.566, respectively. On the other hand, the power density should be increased by about 76.5%. The time distortion is given by 0.566, therefore, all the events are expected to occur in shorter time by a factor of 0.566 in the scale model than in the prototype.

A summary of the important system parameters for the prototype plant and the practically possible optimum scaled model is provided in Table IV for the

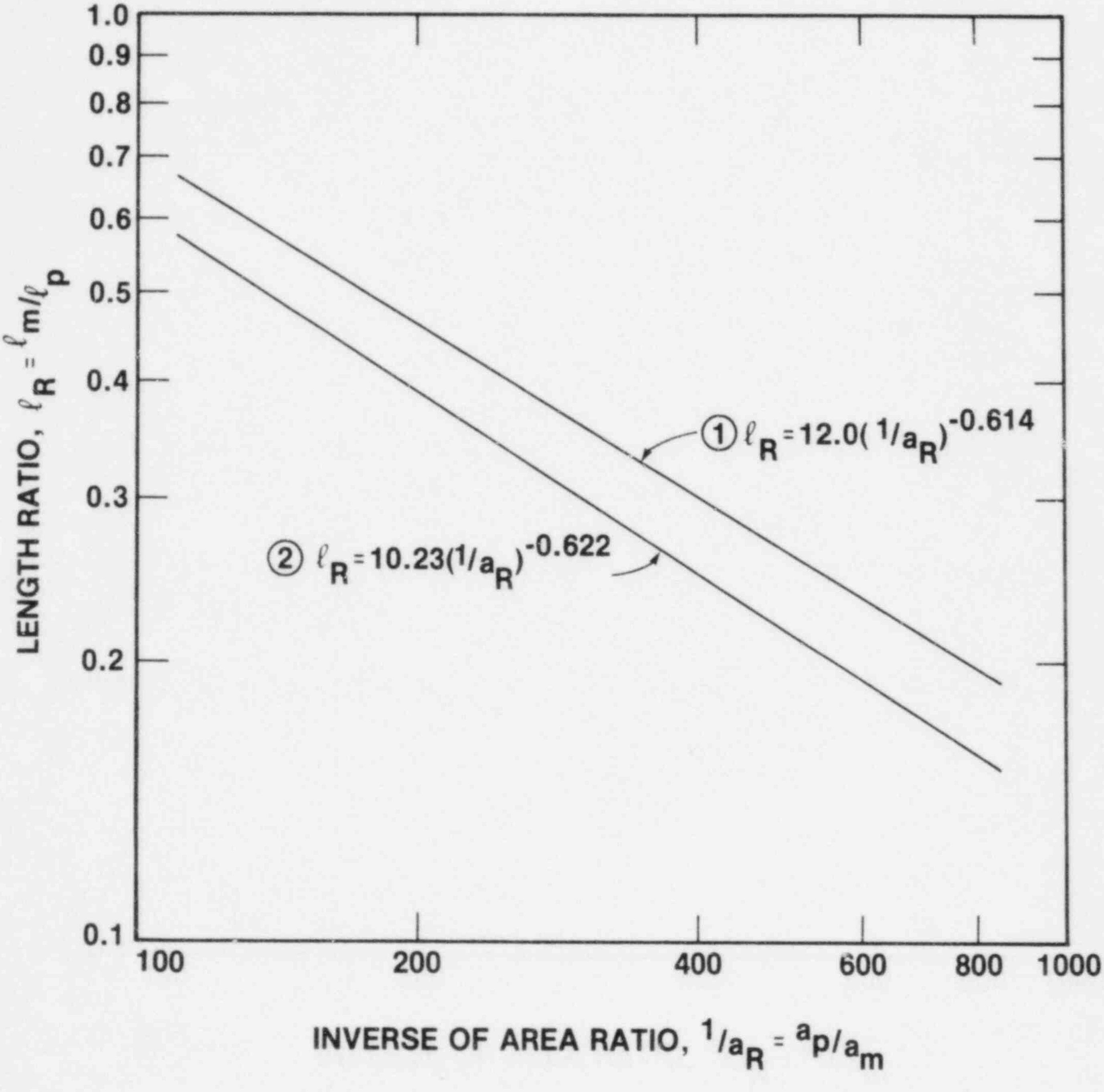

Fig. 7. Single-Phase and Two-Phase Forced Convection System, Time-Distorted Scaled Model Requirements
1. Ideal Case
2. Practically Optimum Case

Table IV. Single-Phase and Two-Phase Forced Convection Prototype and Practically Optimum Scaled Model System Parameters

COMPONENT	PROTOTYPE	MODEL
System Pressure (bar)	148.6	148.6
Area Scale Factor	1/1	1/500
Optimum Length Ratio	1.0	0.216
CORE		
Number of Rods	36,816	73
Rod Diameter (m)	1.092×15^{2}	1.092×15^{2}
Rod Pitch (m)	1.443×15^{2}	1.443×15^{2}
Hydraulic Diameter (m)	1.326×15^{2}	1.326×15^{2}
Flow Area (m^2)	4.572	9.15×15^{3}
Length (m)	4.206	0.913
Velocity (m/s)	5.125	2.387
Mass Flow Rate (kg/s)	16.682	15.491
Reynolds Number	5.41×10^{5}	2.51×10^{5}
Friction Number	10.604	2.905
Power (MWt)	2772.0	2.573
Time Ratio	1.0	0.466
HOT LEG (1 OF 2)		
Diameter (m)	0.915	4.09×10^{-2}
Flow Area (m^2)	0.658	1.32×10^{-3}
Length (m)	21.123	4.584
Velocity (m/s)	19.524	9.095
Mass Flow Rate (kg/s)	8820.0	8.190
Reynolds Number	1.45×10^{8}	3.01×10^{6}
Friction Number	1.515	1.515
Time Ratio	1.0	0.466
STEAM GENERATOR (1 OF 2)		
Number of Tubes	15,531	31
Diameter of Tubes (m)	1.42×15^{2}	1.42×10^{-2}
Flow Area (m^2)	2.441	4.88×10^{-3}
Length (m)	15.880	3.446

Table IV. (Cont'd)

STEAM GENERATOR (1 OF 2) (Cont'd)	PROTOTYPE	MODEL
Velocity (m/s)	5.075	2.357
Mass Flow Rate (kg/s)	8820.0	8.190
Reynolds Number	5.72×10^5	2.65×10^5
Friction Number	16.686	4.449
Time Ratio	1.0	0.466
COLD LEG (1 OF 4)		
Pump Suction Section		
Diameter (m)	0.711	3.18×10^{-2}
Flow Area (m^2)	0.397	7.94×15^4
Length (m)	11.126	2.414
Velocity (m/s)	15.071	7.021
Mass Flow Rate (kg/s)	4410.0	4.095
Reynolds Number	8.39×10^7	1.74×10^6
Friction Number	0.950	0.945
Time Ratio	1.0	0.466
Pump Discharge Section		
Diameter (m)	0.711	3.18×10^{-2}
Flow Area (m^2)	0.397	7.94×15^4
Length (m)	8.810	1.912
Velocity (m/s)	15.071	7.021
Mass Flow Rate (kg/s)	4410.0	4.095
Reynolds Number	8.39×10^7	1.74×10^6
Friction Number	1.047	0.813
Time Ratio	1.0	0.466
DOWNCOMER		
Hydraulic Diameter (m)	0.505	9.21×10^{-2}
Flow Area (m^2)	3.855	7.71×10^{-3}
Length (m)	6.523	1.415
Velocity (m/s)	5.872	2.735
Mass Flow Rate (kg/s)	16,682.0	15.491
Friction Number	0.154	0.154
Time Ratio	1.0	0.466

area ratio of 1/500. Again, additional flow restrictions should be introduced to components other than the hot leg to insure Eq. (136) is sectionally satisfied.

D. Single-Phase and Two-Phase Natural Convection System, Time-Distorted Scaling

In the case of single-phase natural circulation, the core velocity, u_0, and the temperature rise, ΔT_0, can be determined by Eqs. (19) and (18), respectively. At the system pressure of 148.6 bar and the core inlet temperature of 290°C, the natural circulation flow limit in the single phase region can be obtained by setting the upper limit for the temperature as the saturation temperature (341.4°C). The flow limit obtained in this way is shown in Fig. 8. The maximum single phase core velocity due to natural circulation is about 0.6 m/s. Although it is possible that the flow rate may increase in the two-phase region due to increased driving force, this velocity is taken as a reference velocity for the similarity criteria under a natural circulation condition.

The same similarity criteria as those presented in Section III can be used for this case also. The solution to the similarity criteria for both ideal and practically optimum cases is presented in Fig. 9. It can be given approximately by

$$\ell_R = 12.08\ (1/a_R)^{-0.658} \tag{148}$$

for the ideal case, and

$$\ell_R = 10.0\ (1/a_R)^{-0.672} \tag{149}$$

for the practically optimum case. Similarly, the volume ratio for both cases are given, respectively, as follows:

$$V_R = 12.08\ (1/a_R)^{-1.658} \tag{150}$$

and

$$V_R = 10.0\ (1/a_R)^{-1.672}$$

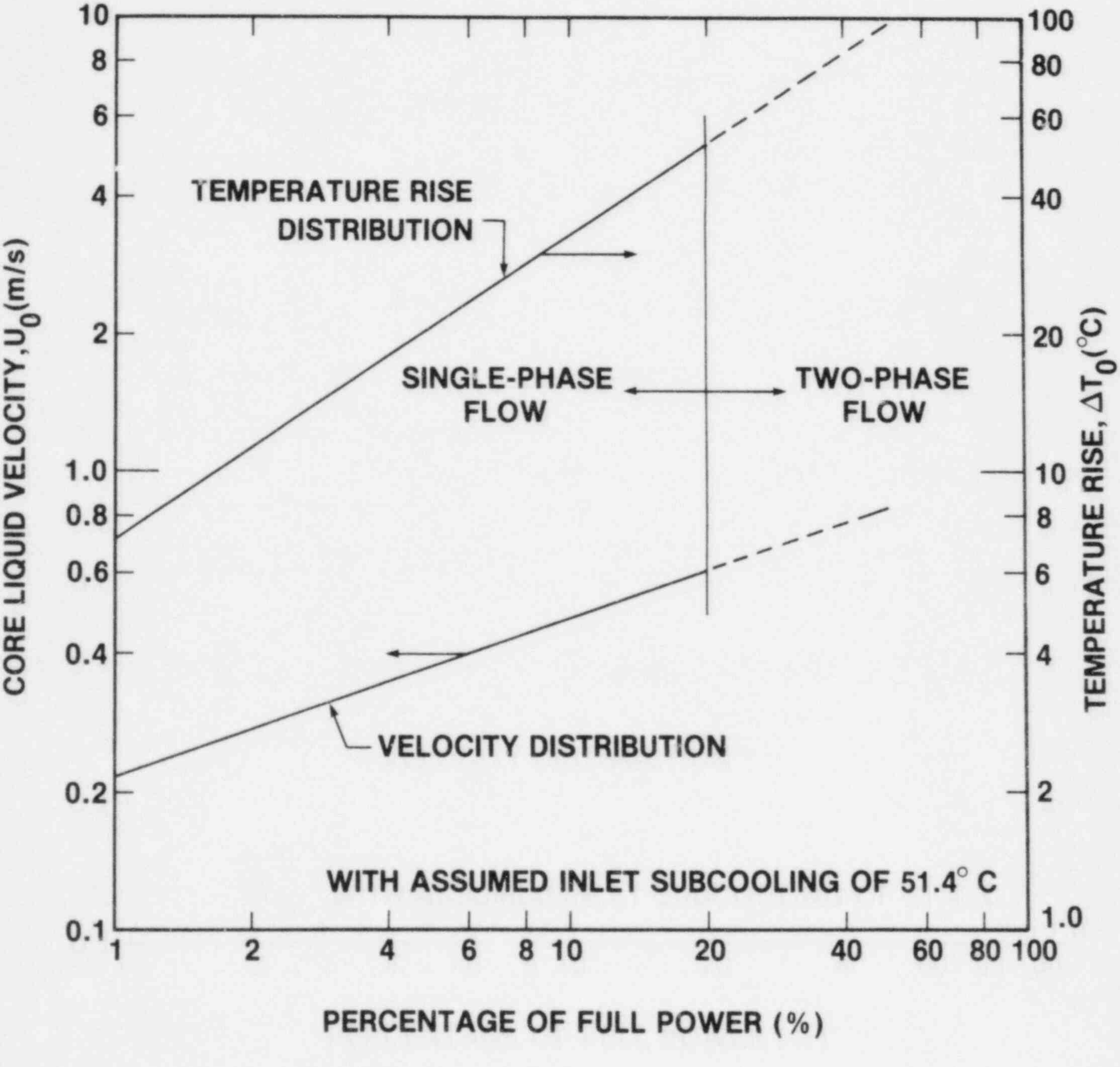

Fig. 8. Steady-State Natural Circulation Operation, Core Velocity (u_o) and Core Temperature Rise (ΔT_o)

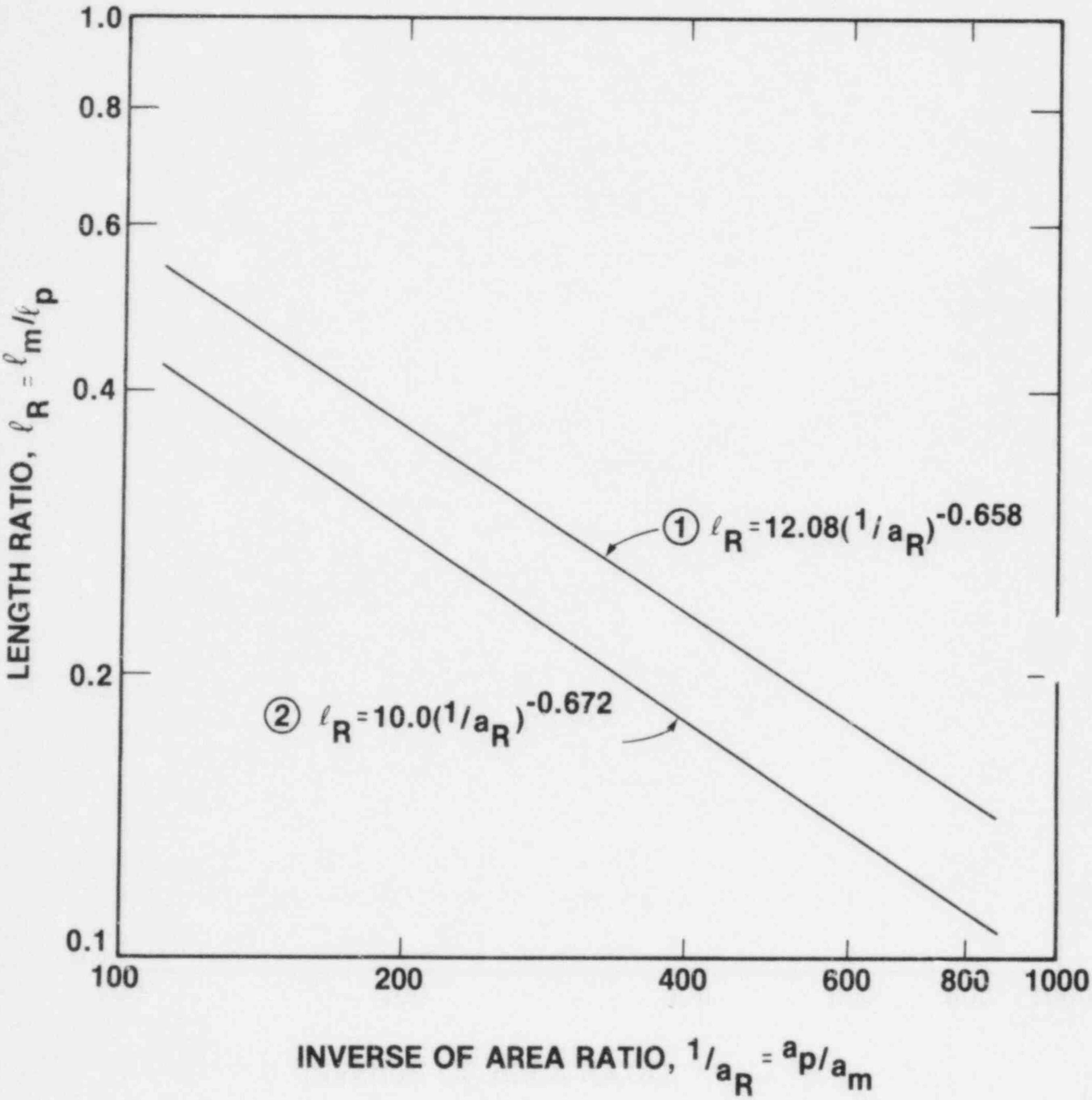

Fig. 9. Single-Phase and Two-Phase Natural Convection System, Time-Distorted Scaled Model Requirements
1. Ideal Case
2. Practically Optimum Case

Hence, for a sample case of $V_R = 1/815.3$, one obtains the following solutions for the practically optimum case:

$$a_R = 1/218.4$$

$$\ell_R = 0.27 \qquad (151)$$

$$\dot{q}_R''' = 1.93$$

In this case, the time distortion can be expressed as

$$t_R = \frac{t_m}{t_p} = 0.52 \qquad (152)$$

The comparison of the forced convection and natural convection, i.e., comparison of Eqs. (146) and (149), indicates that for the volume ratio of 1/815.3 the length ratio suitable for both forced and natural convection simulation is given by

$$0.270 < \ell_R < 0.312 \qquad (153)$$

Overall comparison of the forced and natural convection circulation is given in Fig. 10. The hatched area between two curves indicates the length ratio region where the solution should be sought if the model is to be built for forced and natural circulation simulations. For example, if the model is built for the natural circulation purposes (the lower curve in Fig. 10), then the forced convection friction number will be higher in each section than those corresponding sections in the model. Therefore, for the purpose of making a compromise the model should be built with a length ratio in between two requirements imposed by the natural and forced circulation simulation criteria.

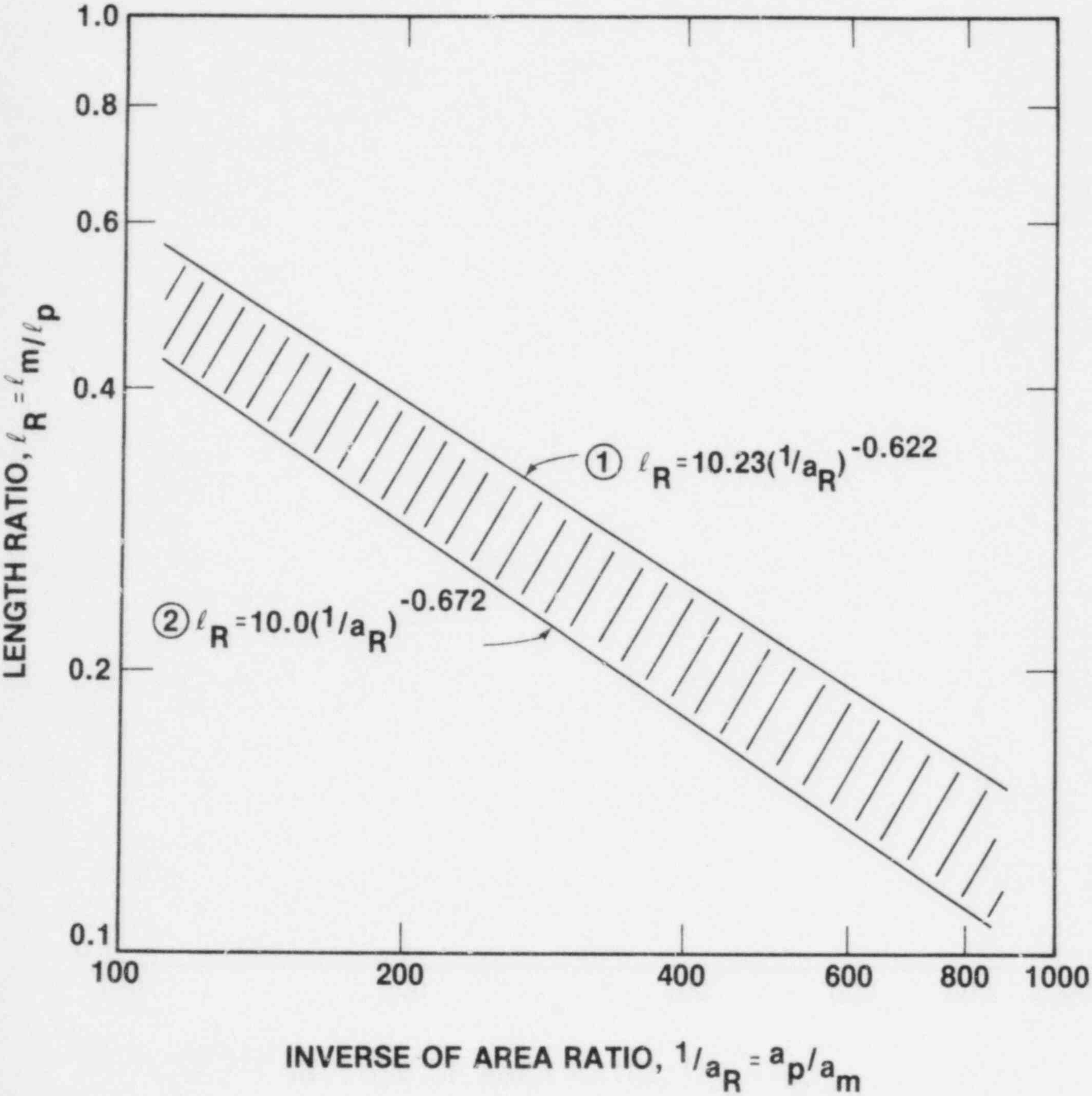

Fig. 10. Comparison of Forced and Natural Circulation Systems, Time-Distorted Scaled Model Requirements
1. Forced Convection
2. Natural Convection

V. SUMMARY AND CONCLUSIONS

Scaling criteria for natural and forced convection circulation loop under single-phase and two-phase flow conditions have been derived from the balance equations. For the single-phase case, the fluid continuity, integral momentum, and energy equations in one-dimensional area-averaged forms have been used together with the appropriate boundary conditions and the solid energy equation. From the non-dimensional form of these equations, important dimensionless groups characterizing geometric, kinematic, dynamic, and energetic similarity between the prototype and a model have been identified. In the case of single-phase natural convection flow, they are the geometric similarity group, friction number, Richardson number, characteristic time constant ratio, Biot number, and the heat source number. In the case of forced convection flow, the Richardson number is replaced by the pump characteristic number. The geometric similarity group consists of the axial length and cross-sectional area ratio of various sections.

Simultaneous solution of the equations imposed by the similarity requirements yielded the similarity criteria. These criteria satsify all the similarity requirements except the condition imposed by the Biot number. Since the Biot number involves the heat transfer coefficient, it was shown that it may cause some difficulties in simulating the turbulent flow thermal boundary layer. However, the relaxation of the Biot number similarity condition influences the boundary layer temperature drop only, and, therefore, it may not be very significant in the single phase case.

For a two-phase flow case, the similarity groups were obtained from a perturbation analysis based on the one-dimensional drift-flux model. The dimensional analysis indicated that the phase change number, drift-flux number, friction number, density ratio, and two-phase Froude number are the important similarity groups. The physical significance of these groups were discussed and conditions implemented by them were evaluated to arrive at similarity criteria. The criteria were simplified for the purpose of practical simulation experiments.

In view of the unique design features of the B&W plant design, the scaling concepts presented above were applied to obtain a 2 x 4 loop facility conceptual modeling of a B&W 177 NSSS lowered loop plant design. Numerical calculations performed to meet the similarity requirement indicated that the most severe condition in terms of the thermo-hydraulic simulation is imposed by the friction number requirement over the hot leg section. Therefore, a solution for the similarity criteria based on the hot leg was presented. For three separate circumstances, i.e., single-phase forced convection (time-preserving), single-phase and two-phase forced convection (time-distorted), and single-phase and two-phase natural convection (time-distorted), it was shown that a solution in the form $\ell_R = f(a_R)$ was feasible in each case. It is suggested that once the necessary condition is obtained by the solution, the other sections must be adjusted to meet sectionally the friction number requirement.

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APPENDIX

Here a computer code which is capable of calculating the friction number (F_i) for each section among other system parameters for the prototype and the scaled model is given. For each value of the area scale factor ($a_R = a_m/a_p$), the prototype and model parameters are listed for different values of the length ratio ($\ell_R = \ell_m/\ell_p$). For the purpose of demonstration, a typical output for the case of a_R = 1/500 is given. A series of outputs for different values of a_R is used to determine the solution ℓ_R to meet $F_R = F_m/F_p = 1$ for the hot leg section because the hot leg simulation imposes the strongest constraint.

```
C
      DIMENSION RL(10),AP(6),DP(6),XP(6),WP(6),PP(6),DM(6),E(6),CM(6,10)
     &,UP(6),REP(6),FSP(6),FP(6),CP(6),FTP(6),AM(6),XM(6,10),UM(6,10),
     &WM(6,10),REM(6,10),FM(6,10),FSM(6,10),FTM(6,10),PM(6,10),TRM(6,10)
     &,POWM(10)   ,QM(10),R(6),V(6),PVEP(1),PVEM(10),PSIP(1),PSIM(10),
     &PSEP(1),PSEM(10),PDIP(1),PDIM(10),PPIP(1)  ,PPIM(10),PPEP(1),PPEM(1
     &0),RA1(9)
C
C I:UNITS INDEX NUMBER
C I=1,CORE
C I=2,HOT LEG
C I=3,COLD LEG PUMP SUCTION
C I=4,COLD LEG PUMP DISCHARGE
C I=5,STEAM GENERATOR
C I=6,DOWNCOWMER
C
C SUBSCRIPTS
C       M:MODEL
C       P:PROTOTYPE
C
C NOMENCLATURES:
C       AP ,AM    :FLOW AREA (M2)
C       CP ,CM    :FRICTION NO. (DIST. LOSS ONLY)
C       CVE       :ORIFICE NO. OF VESSEL EXIT
C       CSI       :ORIFICE NO. OF STEAM GENERATOR INLET
C       CSE       :ORIFICE NO. OF STEAM GENERATOR EXIT
C       CDI       :ORIFICE NO. OF DOWNCOMER INLET
C       CPI       :ORIFICE NO. OF PUMP INLET
C       CPE       :ORIFICE NO. OF PUMP EXIT
C       CSP       :FREE-FLOW/FRONTAL-AREA RATIO OF CORE
C       CKP       :CONTRACTION COEFFICIENT OF CORE
C       CEP       :EXPANSION COEFFICIENT OF CORE
C       DLCP,DLCM:DISTRIBUTED LOSS COEFFICIENT
C       DP ,DM    :FLOW DIAMETER (M)
C       DRP,DRM   :ROD DIAMETER (M)
C       E  ,EM    :TUBE ROUGHNESS
C       FP ,FM    :FRICTION FACTOR
C       FSP,FSM   :TOTAL FRICTION NO.
C       FTP,FTM   :FRICTION NO. (FR. LOSS ONLY)
C       HRP,HRM   :ROD PITCH OF CORE (M)
C       NRP,NRM   :NUMBER OF CORE-RODES
C       NSP,NSM   :NUMBER OF TUBES OF STEAM GENERATOR
C       PP ,PM    :PRESSURE DROP (PA)
C       PVEP,PVEM:VESSEL EXIT MINOR LOSSES
C       PSIP,PSIM:STEAM GENERATOR INLET MINOR LOSSES
C       PSEP,PSEM:STEAM GENERATOR EXIT MINOR LOSSES
C       PDIP,PDIM:DOWN COMER INLET MINOR LOSSES
C       POW,POWM:POWER OF CORE (MWT)
C       QP ,QM    :HEAT FLUX OF CORE (KW/M2)
C       R         :DENSITY OF WORKING LIQUID
C       RMA,RAV,RMI:MAXIUM,AVERAGE AND MINIMUM FLUID DENSITIES
C       RA,RA1    :AREA SCALE FACTOR
C       RL        :LENGTH SCALE FACTOR
C       REP,REM   :REYNOLDS NUMBER
C       S         :AREA RATIO OF STEAM GENERATOR
C       SSP       :FREE-FLOW/FRONTAL-AREA RATIO OF STEAM GENERATOR
C       SKP       :CONTRACTION COEFFICIENT OF STEAM GENERATOR
C       SEP       :EXPANSION COEFFICIENT OF STEAM GENERATOR
C       TRM       :TIME SHIFT FACTOR
C       UP ,UM    :VELOCITY (M/S)
C       V         :VISCOSITY OF WORKING LIQUID (KG/S-M)
C       WP ,WM    :MASS FLOW RATE (KG/S)
C       XP ,XM    :LENGTH (M)
C
```

```
      READ (1,*) (RL(J),J=1,10)
      READ (1,*) CVE,CSI,CSE,CDI,CPI,CPE
      READ (1,*) SSP,SKP,SEP,CSP,CKP,CEP
      READ (1,*) DLCP2,DLCP3,DLCP4,DLCM2,DLCM3,DLCM4
      READ (1,*) RMA,RAV,RMI
      READ (1,*) NRP,DRP,HRP,POW
      READ (1,*) DP(1),XP(1),WP(1), R(1), V(1),AP(1)
      READ (1,*) DP(2),XP(2),WP(2), R(2),V(2)
      READ (1,*) DP(3),XP(3),WP(3), R(3),V(3)
      READ (1,*) DP(4),XP(4),WP(4), R(4),V(4)
      READ (1,*) DP(5),XP(5),WP(5),R(5),V(5),AP(5),NSP,S
      READ (1,*) DP(6),XP(6),WP(6), R(6), V(6),AP(6)
      READ (1,*) (RA1(M),M=1,9),MM
 30   READ (1,*) KK
      IF (KK.EQ.0 ) GO TO 20
      K=9
      IF (KK.NE.1) GO TO 96
      WRITE (K,97)
 97   FORMAT (/////,' TWO-PHASE FLOW MODELING',/,1X,23('*'))
      GO TO 99
 96   WRITE (K,98)
 98   FORMAT (/////,' SINGLE-PHASE FLOW MODELING',/,1X,26('*'))
 99   DO 95 M=1,MM
      RA=RA1(M)
      HRM=HRP
      DRM=DRP
      DM(1)=DP(1)
      E(1)=0.0000015
      E(5)=0.0000015
      E(6)=0.000046
      EM=0.0000015
      DO 1 I=2,4
      E(I)=0.000046
      AP(I)=3.1416*DP(I)**2/4.
 1    DM(I)=DP(I)/RA**.5
C CALCULATION OF PROTOTYPE VARIABLES
      DO 2 I=1,6
      UP(I)=WP(I)/(R(I)*AP(I))
      REP(I)=DP(I)*R(I)*UP(I)/V(I)
      FP(I)=1.325/(ALOG(E(I)/(DP(I)*3.7)+5.74/REP(I)**.9))**2
 2    FTP(I)=FP(I)*XP(I)/DP(I)
      FP(1)=2.2*FP(1)
      FTP(1)=2.2*FTP(1)
      CP(2)=3.*FP(2)*DLCP2
      CP(3)=FP(3)*DLCP3
      CP(4)=FP(4)*DLCP4
      C1=CKP+1.-CSP**2
      C2=1.-CSP**2-CEP
      S1=SKP+1.-SSP**2
      S2=1.-SSP**2-SEP
      DO 71 I=2,4
 71   PP(I)=FP(I)*(XP(I)/DP(I)+CP(I)/FP(I))*.5*R(I)*UP(I)**2
      PP(6)=FTP(6)*.5*R(6)*UP(6)**2
      CP(6)=0.
      R11=RAV**2/RMA
      PP(1)=.5*R11*UP(1)**2*( C1 +2.*(RMA/RMI-1.)+FTP(1)*RMA/RAV-C2*
     &RMA/RMI)
      CP(1)=2.*PP(1)/(R(1)*UP(1)**2)-FTP(1)
      R55=RAV**2/RMI
      PP(5)=.5*R55*UP(5)**2*( S1  +2.*(RMI/RMA-1.)+FTP(5)*RMI/RAV-S2
     &*RMI/RMA)
      CP(5)=2.*PP(5)/(R(5)*UP(5)**2)-FTP(5)
      DO 72 I=1,6
 72   FSP(I)=FTP(I)+CP(I)
      QP=POW*1000./(3.1416*DRP*XP(1)*NRP)
      NRM=NRP/RA
      NSM=NSP/RA
      DM(1)=DP(1)
      DM(5)=DP(5)
      DM(6)=(4.*AP(6)/RA/3.1416)**.5
```

```
C CALCULATION OF MODELING VARIABLES
      DO 3 I=1,6
      AM(I)=AP(I)/RA
      DO 4 J=1,10
      XM(I,J)=XP(I)*RL(J)
      IF (KK.NE.1) GO TO 100
      UM(I,J)=RL(J)**.5*UP(I)
      GO TO 101
 100  UM(I,J)=RL(J)*UP(I)
 101  WM(I,J)=AM(I)  *UM(I,J)*R(I)
      REM(I,J)=DM(I)  *UM(I,J)*R(I)/V(I)
      FM(I,J)=1.325/(ALOG(EM/(DM(I)*3.7)+5.74/REM(I,J)**.9))**2
      IF (I.NE.1) GO TO 8
      FM(1,J)=2.2*FM(1,J)
 8    FTM(I,J)=FM(I,J)*XM(I,J)/DM(I)
      IF (KK.NE.1) GO TO 103
      TRM(I,J)=RL(J)**.5
      GO TO 4
 103  TRM(I,J)=1.
 4    CONTINUE
 3    CONTINUE
      DO 73 J=1,10
      CM(2,J)=3.*FM(2,J)*DLCM2
      CM(3,J)=FM(3,J)*DLCM3
 73   CM(4,J)=FM(4,J)*DLCM4
      DO 74 I=2,4
      DO 75 J=1,10
 75   PM(I,J)=FM(I,J)*(XM(I,J)/DM(I)+CM(I,J)/FM(I,J))*.5*R(I)*UM(I,J)**2
 74   CONTINUE
      DO 76 J=1,10
      PM(6,J)=FTM(6,J)*R(6)*UM(6,J)**2*.5
 76   CM(6,J)=0.
      DO 77 J=1,10
      PM(1,J)=.5*R11*UM(1,J)**2*( C1 +2.*(RMA/RMI-1.)+FTM(1,J)*RMA/RAV-
     & C2  *RMA/RMI)
      CM(1,J)=2.*PM(1,J)/(R(1)*UM(1,J)**2)-FTM(1,J)
      PM(5,J)=.5*R55*UM(5,J)**2*( S1  +2.*(RMI/RMA-1.)+FTM(5,J)*RMI/RAV-
     & S2  *RMI/RMA)
      CM(5,J)=2.*PM(5,J)/(R(5)*UM(5,J)**2)-FTM(5,J)
 77   CONTINUE
      DO 78 I=1,6
      DO 79 J=1,10
 79   FSM(I,J)=FTM(I,J)+CM(I,J)
 78   CONTINUE
      PVEP(1)=CVE*RMI*UP(2)**2/2.
      PSIP(1)=CSI*RMI*UP(2)**2/2.
      PSEP(1)=CSE*RMA*UP(3)**2/2.
      PDIP(1)=CDI*RMA*UP(4)**2/2.
      PPIP(1)=CPI*R(3)*UP(3)**2/2.
      PPEP(1)=CPE*R(4)*UP(4)**2/2.
      DO 5 J=1,10
      PVEM(J)=CVE*RMI*UM(2,J)**2/2.
      PSIM(J)=CSI*RMI*UM(2,J)**2/2.
      PSEM(J)=CSE*RMA*UM(3,J)**2/2.
      PDIM(J)=CDI*RMA*UM(4,J)**2/2.
      PPIM(J)=CPI*R(3)*UM(3,J)**2/2.
      PPEM(J)=CPE*R(4)*UM(4,J)**2/2.
      IF (KK.NE.1) GO TO 102
      POWM(J)=RL(J)**.5*POW/RA
      GO TO 5
 102  POWM(J)=RL(J)**2*POW/RA
 5    QM(J)=POWM(J)*1000./(3.1416*DRP*XM(1,J)*NRM)
 OUT PUT THE PROTOTYPE AND MODELING VARIABLES
     'WRITE (K,31)
 31   FORMAT (///,' CORE',/,' ****',/,25X,'PROTOTYPE',43X,'MODEL',/,25X,
     &9('*'),5X,90('*'))
```

```
      WRITE (K,32) RA
   32 FORMAT (' AREA SCALE FACTOR',10X,'1/1',46X,'1/',F4.0)
      WRITE (K,33) NRP,NRM
   33 FORMAT (' NUMBER OF RODS',10X,I9,41X,I8)
      WRITE (K,34) DRP,DRM
   34 FORMAT (' ROD DIAMETER (M)',8X,F9.5,41X,F8.5)
      WRITE (K,35) HRP,HRM
   35 FORMAT (' ROD PITCH (M)',11X,F9.5,41X,F8.5)
      WRITE (K,36) DP(1),DM(1)
   36 FORMAT (' HYDROULIC DIAMETER (M)',2X,F9.5,41X,F8.5)
      WRITE (K,37) AP(1),AM(1)
   37 FORMAT (' FLOW AREA (M2)',10X,F9.4,41X,F8.6)
      WRITE (K,39) (RL(J),J=1,10)
   39 FORMAT (' LENGTH SCALE FACTOR',8X,'1/1',7X,10F9.1,/,1X,128('*'))
      WRITE (K,40) XP(1),(XM(1,J),J=1,10)
   40 FORMAT (' LENGTH (M)',14X,F9.3,4X,10F9.3)
      WRITE (K,41) UP(1),(UM(1,J),J=1,10)
   41 FORMAT (' VELOCITY (M/S)',10X,F9.3,4X,10F9.3)
      WRITE (K,42) WP(1),(WM(1,J),J=1,10)
   42 FORMAT (' MASS FLOW RATE (KG/S)',3X,F9.2,4X,10F9.3)
      WRITE (K,43) REP(1),(REM(1,J),J=1,10)
   43 FORMAT (' REYNOLDS NUMBER',7X,F11.0,4X,10F9.0)
      WRITE (K,44) FP(1),(FM(1,J),J=1,10)
   44 FORMAT (' FRICTION FACTOR',9X,F9.5,4X,10F9.5)
      IF (KK.NE.1) GO TO 110
      WRITE (K,45) FTP(1),(FTM(1,J),J=1,10)
   45 FORMAT(' FR.NO.(FR.LOSS ONLY)',4X,F9.5,4X,10F9.5)
  110 WRITE (K,38) CP(1),(CM(1,J),J=1,10)
      WRITE (K,46) FSP(1),(FSM(1,J),J=1,10)
   46 FORMAT (' TOTAL FR.NO.',      12X,F9.5,4X,10F9.5)
      WRITE (K,47) PP(1),(PM(1,J),J=1,10)
   47 FORMAT (' PRESSURE DROP (PA)',6X,F9.0,4X,10F9.0)
   38 FORMAT (' FR.NO.(DIST.LOSS ONLY)',2X,F9.4,4X,10F9.4)
      IF (KK.NE.1) GO TO 105
      WRITE (K,48) (TRM(1,J),J=1,10)
   48 FORMAT (' TIME SHIFT',17X,'1/1',7X,10F9.3)
      GO TO 106
  105 WRITE (K,107)
  107 FORMAT (' TIME SHIFT',17X,'1/1',7X,10(3X,'1/1',3X))
  106 WRITE (K,49) POW,(POWM(J),J=1,10)
   49 FORMAT (' POWER (MWT)',13X,F9.1,4X,10F9.3)
C     WRITE (K,50) QP,(QM(J),J=1,10)
   50 FORMAT (' HEAT FLUX (KW/M2)' ,7X,F9.1,4X,10F9.2)
      DO 6 I=2,6
      IF (I.EQ.3) GO TO 11
      IF (I.EQ.4) GO TO 12
      IF (I.EQ.5) GO TO 13
      IF (I.EQ.6) GO TO 16
      WRITE (K,51)
   51 FORMAT (///,' HOT LEG',/,' *******')
      GO TO 14
   11 WRITE (K,52)
   52 FORMAT (///,' COLD LEG PUMP SUCTION',/,1X,21('*'))
      GO TO 14
   12 WRITE (K,53)
   53 FORMAT (///,' COLD LEG PUMP DISCHARGE',/,1X,23('*'))
      GO TO 14
   13 WRITE (K,56)
   56 FORMAT (///,' STEAM GENERATOR',/,1X,15('*'))
      WRITE (K,54)
      WRITE (K,32) RA
      WRITE (K,61) NSP,NSM
   61 FORMAT (' NUMBER OF TUBES',9X,I9,41X,I8)
      WRITE (K,57) S,S
   57 FORMAT (' AREA RATIO',14X,F9.4,41X,F8.4)
      GO TO 15
```

```
 16  WRITE (K,58)
 58  FORMAT (///,' DOWNCOMER',/,1X,9('*'))
     WRITE (K,54)
     WRITE (K,32) RA
     WRITE (K,36) DP(6),DM(6)
     GO TO 17
 14  WRITE (K,54)
 54  FORMAT (25X,'PROTOTYPE',43X,'MODEL',/,25X,9('*'),5X,90('*'))
     WRITE (K,32) RA
 15  WRITE (K,55) DP(I),DM(I)
 55  FORMAT (' DIAMETER (M)',12X,F9.5,41X,F8.5)
 17  WRITE (K,37) AP(I),AM(I)
     WRITE (K,39) (RL(J),J=1,10)
     WRITE (K,40) XP(I),(XM(I,J),J=1,10)
     WRITE (K,41) UP(I),(UM(I,J),J=1,10)
     WRITE (K,42) WP(I),(WM(I,J),J=1,10)
     WRITE (K,43) REP(I),(REM(I,J),J=1,10)
     WRITE (K,44) FP(I),(FM(I,J),J=1,10)
     IF (KK.NE.1) GO TO 111
     WRITE (K,45) FTP(I),(FTM(I,J),J=1,10)
111  WRITE (K,38) CP(I),(CM(I,J),J=1,10)
     WRITE (K,46) FSP(I),(FSM(I,J),J=1,10)
     WRITE (K,47) PP(I),(PM(I,J),J=1,10)
     IF (KK.NE.1) GO TO 112
     WRITE (K,48) (TRM(I,J),J=1,10)
     GO TO 6
112  WRITE (K,107)
 6   CONTINUE
     WRITE (K,81)
 81  FORMAT (///,' MINOR LOSSES',/,' ************')
     WRITE (K,82) CVE
 82  FORMAT (/,' VESSEL EXIT: ORIFICE NUMBER=',F4.2)
     WRITE (K,83) PVEP(1),(PVEM(J),J=1,10)
 83  FORMAT (' PRESSURE LOSSES (PA)', 4X,F9.0.4X,10F9.0)
     WRITE (K,84) CSI
 84  FORMAT (/,' STEAM GENERATOR INLET: ORIFICE NUMBER=',F4.2)
     WRITE (K,83) PSIP(1),(PSIM(J),J=1,10)
     WRITE (K,85) CSE
 85  FORMAT (/,' STEAM GENERATOR EXIT: ORIFICE NUMBER=',F4.2)
     WRITE (K,83) PSEP(1),(PSEM(J),J=1,10)
     WRITE (K,86) CDI
 86  FORMAT (/,' DOWNCOMER INLET: ORIFICE NUMBER=',F4.2)
     WRITE (K,83) PDIP(1),(PDIM(J) ,J=1,10)
     WRITE (K,87) CPI
 87  FORMAT (/,' PUMP INLET: ORIFICE NUMBER=',F4.2)
     WRITE (K,83) PPIP(1),(PPIM(J),J=1,10)
     WRITE (K,88) CPE
 88  FORMAT (/,' PUMP EXIT: ORIFICE NUMBER=',F4.2)
     WRITE (K,83) PPEP(1),(PPEM(J),J=1,10)
 95  CONTINUE
     GO TO 30
 20  CALL WINPUT(2)
     STOP
     END
```

TWO-PHASE FLOW MODELING
****

CORE
****

	PROTOTYPE	MODEL									
AREA SCALE FACTOR	1/1					1/500.					
NUMBER OF RODS	36816					73					
ROD DIAMETER (M)	0.01092					0.01092					
ROD PITCH (M)	0.01443					0.01443					
HYDROULIC DIAMETER (M)	0.01326					0.01326					
FLOW AREA (M2)	4.5720					0.009144					
LENGTH SCALE FACTOR	1/1	1.0	0.9	0.8	0.7	0.6	0.5	0.4	0.3	0.2	0.1
LENGTH (M)	4.206	4.206	3.785	3.365	2.944	2.524	2.103	1.682	1.262	0.841	0.421
VELOCITY (M/S)	5.125	5.125	4.862	4.584	4.288	3.970	3.624	3.241	2.807	2.292	1.621
MASS FLOW RATE (KG/S)	16682.00	33.364	31.652	29.842	27.914	25.844	23.592	21.101	18.274	14.921	10.551
REYNOLDS NUMBER	541430.	541430.	513645.	484270.	452993.	419390.	382849.	342430.	296553.	242135.	171215.
FRICTION FACTOR	0.03187	0.03187	0.03205	0.03226	0.03251	0.03280	0.03316	0.03363	0.03427	0.03526	0.03716
FR.NO.(FR.LOSS ONLY)	10.11015	10.11015	9.15084	8.18705	7.21801	6.24273	5.25979	4.26712	3.26141	2.23664	1.17879
FR.NO.(DIST.LOSS ONLY)	0.4936	0.4936	0.4936	0.4936	0.4936	0.4936	0.4936	0.4936	0.4936	0.4936	0.4936
TOTAL FR.NO.	10.60378	10.60378	9.64447	8.68068	7.71165	6.73637	5.75342	4.76077	3.75506	2.73028	1.67243
PRESSURE DROP (PA)	99137.	99137.	81151.	64926.	50468.	37788.	26895.	17804.	10532.	5105.	1564.
TIME SHIFT	1/1	1.000	0.949	0.894	0.837	0.775	0.707	0.632	0.548	0.447	0.316
POWER (MWT)	2772.0	5.544	5.259	4.959	4.638	4.294	3.920	3.506	3.037	2.479	1.753

HOT LEG
****

	PROTOTYPE	MODEL									
AREA SCALE FACTOR	1/1					1/500.					
DIAMETER (M)	0.91500					0.04092					
FLOW AREA (M2)	0.6576					0.001315					
LENGTH SCALE FACTOR	1/1	1.0	0.9	0.8	0.7	0.6	0.5	0.4	0.3	0.2	0.1
LENGTH (M)	21.123	21.123	19.011	16.898	14.786	12.674	10.562	8.449	6.337	4.225	2.112
VELOCITY (M/S)	19.524	19.524	18.523	17.463	16.335	15.124	13.806	12.348	10.694	8.732	6.174
MASS FLOW RATE (KG/S)	8820.00	17.640	16.735	15.778	14.759	13.664	12.473	11.157	9.662	7.889	5.578
REYNOLDS NUMBER	145055792.	6487094.	6154196.	5802227.	5427492.	5024882.	4587067.	4102797.	3553127.	29[illegible]1116.	2051400.
FRICTION FACTOR	0.01059	0.01063	0.01066	0.01069	0.01073	0.01077	0.01083	0.01090	0.01101	0.01[illegible]18	0.01151
FR.NO.(FR.LOSS ONLY)	0.24439	5.48629	4.95047	4.41366	3.87567	3.33625	2.79507	2.25161	1.70503	1.15377	0.59406
FR.NO.(DIST.LOSS ONLY)	1.2704	0.2551	0.2557	0.2565	0.2574	0.2585	0.2599	0.2617	0.2642	0.2682	0.2762
TOTAL FR.NO.	1.51477	5.74136	5.20621	4.67016	4.13309	3.59477	3.05497	2.51332	1.96927	1.42198	0.87025
PRESSURE DROP (PA)	198349.	751795.	613540.	489222.	378840.	282428.	200015.	131641.	77359.	37240.	11395.
TIME SHIFT	1/1	1.000	0.949	0.894	0.837	0.775	0.707	0.632	0.548	0.447	0.316

COLD LEG PUMP SUCTION

	PROTOTYPE	MODEL									
AREA SCALE FACTOR	1/1					1/500.					
DIAMETER (M)	0.71100					0.03180					
FLOW AREA (M2)	0.3970					0.000794					
LENGTH SCALE FACTOR	1/1	1.0	0.9	0.8	0.7	0.6	0.5	0.4	0.3	0.2	0.1
LENGTH (M)	11.126	11.126	10.013	8.901	7.788	6.676	5.563	4.450	3.338	2.225	1.113
VELOCITY (M/S)	15.071	15.071	14.298	13.480	12.609	11.674	10.657	9.532	8.255	6.740	4.766
MASS FLOW RATE (KG/S)	4410.00	8.820	8.367	7.889	7.379	6.832	6.237	5.578	4.831	3.944	2.789
REYNOLDS NUMBER	83924416.	3753217.	3560613.	3356980.	3140167.	2907229.	2653925.	2373743.	2055722.	1678490.	1186872.
FRICTION FACTOR	0.01109	0.01129	0.01132	0.01136	0.01141	0.01147	0.01154	0.01164	0.01177	0.01198	0.01239
FR.NO.(FR.LOSS ONLY)	0.17355	3.94928	3.56552	3.18087	2.79516	2.40820	2.01966	1.62911	1.23582	0.83842	0.43378
FR.NO.(DIST.LOSS ONLY)	0.7764	0.0903	0.0906	0.0909	0.0913	0.0918	0.0924	0.0931	0.0942	0.0958	0.0992
TOTAL FR.NO.	0.94991	4.03957	3.65609	3.27177	2.88646	2.49996	2.11201	1.72223	1.33000	0.93427	0.53285
PRESSURE DROP (PA)	79506.	338108.	275410.	219075.	169116.	125546.	88387.	57659.	33396.	15639.	4460.
TIME SHIFT	1/1	1.000	0.949	0.894	0.837	0.775	0.707	0.632	0.548	0.447	0.316

COLD LEG PUMP DISCHARGE

	PROTOTYPE	MODEL									
AREA SCALE FACTOR	1/1					1/500.					
DIAMETER (M)	0.71100					0.03180					
FLOW AREA (M2)	0.3970					0.000794					
LENGTH SCALE FACTOR	1/1	1.0	0.9	0.8	0.7	0.6	0.5	0.4	0.3	0.2	0.1
LENGTH (M)	8.810	8.810	7.929	7.048	6.167	5.286	4.405	3.524	2.643	1.762	0.881
VELOCITY (M/S)	15.071	15.071	14.298	13.480	12.609	11.674	10.657	9.532	8.255	6.740	4.766
MASS FLOW RATE (KG/S)	4410.00	8.820	8.367	7.889	7.379	6.832	6.237	5.578	4.831	3.944	2.789
REYNOLDS NUMBER	83924416.	3753217.	3560613.	3356980.	3140167.	2907229.	2653925.	2373743.	2055722.	1678490.	1186872.
FRICTION FACTOR	0.01109	0.01129	0.01132	0.01136	0.01141	0.01147	0.01154	0.01164	0.01177	0.01198	0.01239
FR.NO.(FR.LOSS ONLY)	0.13743	3.12719	2.82332	2.51873	2.21332	1.90690	1.59925	1.28999	0.97857	0.66390	0.34342
FR.NO.(DIST.LOSS ONLY)	0.9094	0.0903	0.0906	0.0909	0.0913	0.0918	0.0924	0.0931	0.0942	0.0958	0.0992
TOTAL FR.NO.	1.04687	3.21748	2.71389	2.60964	2.30461	1.99867	1.69160	1.38311	1.07275	0.75974	0.44257
PRESSURE DROP (PA)	87622.	269300.	219501.	174739.	135026.	100372.	70792.	46306.	26936.	12718.	3704.
TIME SHIFT	1/1	1.000	0.949	0.894	0.837	0.775	0.707	0.632	0.548	0.447	0.316

STEAM GENERATOR

	PROTOTYPE	MODEL									
AREA SCALE FACTOR	1/1					1/500.					
NUMBER OF TUBES	15531					31					
AREA RATIO	0.3826					0.3826					
DIAMETER (M)	0.01415					0.01415					
FLOW AREA (M2)	2.4410					0.004882					
LENGTH SCALE FACTOR	1/1	1.0	0.9	0.8	0.7	0.6	0.5	0.4	0.3	0.2	0.1
LENGTH (M)	15.880	15.880	14.292	12.704	11.116	9.528	7.940	6.352	4.764	3.176	1.588
VELOCITY (M/S)	5.075	5.075	4.814	4.539	4.246	3.931	3.588	3.210	2.780	2.270	1.605
MASS FLOW RATE (KG/S)	8820.00	17.640	16.735	15.778	14.759	13.664	12.473	11.157	9.662	7.889	5.578
REYNOLDS NUMBER	572155.	572155.	542794.	511751.	478700.	443190.	404575.	361063.	313382.	255876.	180931.
FRICTION FACTOR	0.01432	0.01432	0.01440	0.01449	0.01460	0.01474	0.01490	0.01511	0.01540	0.01584	0.01669
FR.NO.(FR.LOSS ONLY)	16.06958	16.06958	14.54487	13.01300	11.47278	9.92261	8.36024	6.78237	5.18376	3.55481	1.87329
FR.NO.(DIST.LOSS ONLY)	0.6169	0.6169	0.6169	0.6169	0.6169	0.6169	0.6169	0.6169	0.6169	0.6169	0.6169
TOTAL FR.NO.	16.68643	16.68643	15.16174	13.62987	12.08965	10.53948	8.97710	7.39925	5.80064	4.17169	2.49017
PRESSURE DROP (PA)	152987.	152987.	125107.	99971.	77590.	57978.	41153.	27136.	15955.	7650.	2283.
TIME SHIFT	1/1	1.000	0.949	0.894	0.837	0.775	0.707	0.632	0.548	0.447	0.316

DOWNCOMER

	PROTOTYPE	MODEL									
AREA SCALE FACTOR	1/1					1/500.					
HYDROULIC DIAMETER (M)	0.50500					0.09908					
FLOW AREA (M2)	3.8550					0.007710					
LENGTH SCALE FACTOR	1/1	1.0	0.9	0.8	0.7	0.6	0.5	0.4	0.3	0.2	0.1
LENGTH (M)	6.523	6.523	5.871	5.218	4.566	3.914	3.261	2.609	1.957	1.305	0.652
VELOCITY (M/S)	5.872	5.872	5.570	5.252	4.913	4.548	4.152	3.714	3.216	2.626	1.857
MASS FLOW RATE (KG/S)	16682.00	33.364	31.652	29.842	27.914	25.844	23.592	21.101	18.274	14.921	10.551
REYNOLDS NUMBER	23223360.	4556338.	4322521.	4075311.	3812103.	3529323.	3221815.	2881679.	2495609.	2037655.	1440841.
FRICTION FACTOR	0.01188	0.00998	0.01003	0.01009	0.01015	0.01023	0.01033	0.01045	0.01062	0.01088	0.01137
FR.NO.(FR.LOSS ONLY)	0.15351	0.65730	0.59448	0.53139	0.46800	0.40425	0.34006	0.27532	0.20984	0.14328	0.07488
FR.NO.(DIST.LOSS ONLY)	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000
TOTAL FR.NO.	0.15351	0.65730	0.59448	0.53139	0.46800	0.40425	0.34006	0.27532	0.20984	0.14328	0.07488
PRESSURE DROP (PA)	1950.	8351.	6797.	5401.	4162.	3081.	2160.	1399.	800.	364.	95.
TIME SHIFT	1/1	1.000	0.949	0.894	0.837	0.775	0.707	0.632	0.548	0.447	0.316

MINOR LOSSES
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VESSEL EXIT: ORIFICE NUMBER=0.50											
PRESSURE LOSSES (PA)	65472.	65472.	58925.	52377.	45830.	39283.	32736.	26189.	19642.	13094.	6547.
STEAM GENERATOR INLET: ORIFICE NUMBER=1.00											
PRESSURE LOSSES (PA)	130944.	130944.	117849.	104755.	91660.	78566.	65472.	52377.	39283.	26189.	13094.
STEAM GENERATOR EXIT: ORIFICE NUMBER=0.50											
PRESSURE LOSSES (PA)	41849.	41849.	37665.	33480.	29295.	25110.	20925.	16740.	12555.	8370.	4185.
DOWNCOMER INLET: ORIFICE NUMBER=1.00											
PRESSURE LOSSES (PA)	83699.	83699.	75329.	66959.	58589.	50219.	41849.	33480.	25110.	16740.	8370.
PUMP INLET: ORIFICE NUMBER=1.00											
PRESSURE LOSSES (PA)	83699.	83699.	75329.	66959.	58589.	50219.	41849.	33480.	25110.	16740.	8370.
PUMP EXIT: ORIFICE NUMBER=0.50											
PRESSURE LOSSES (PA)	41849.	41849.	37665.	33480.	29295.	25110.	20925.	16740.	12555.	8370.	4185.

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16. ABSTRACT (200 words or less)

Scaling criteria for a natural and forced convection circulation loop under single-phase and two-phase flow conditions are derived from the fluid balance equations, boundary conditions, and solid energy equations. For a single-phase flow case the continuity, integral momentum, and energy equations in one-dimensional area-averaged forms are used. For a two-phase flow case the one-dimensional drift-flux model obtained from the short time temporal averaging and the sectional area averaging is used.

The scaling criteria are applied to a conceptual design of a 2 x 4 loop facility for simulating the Babcock and Wilcox 177 NSSS design. Feasibility of practical solutions are studied and some conclusions on this proposed facility in terms of the proper scaling are obtained.

17. KEY WORDS AND DOCUMENT ANALYSIS	17a. DESCRIPTORS
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17b. IDENTIFIERS/OPEN-ENDED TERMS	

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