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APPLICATIONS OF GEOMETRIC MODELS
FOR THE FFTF HYDRAULIC CORE MOCKUP

D. S. Trent

November 1967

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FOR THE FFTF HYDRAULIC CORE MOCKUP

By

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November 1967

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PACIFIC NORTHWEST LABORATORY
RICHLAND, WASHINGTON

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TABLE OF CONTENTS

LIST OF FIGURES	v
NOMENCLATURE	viii
SIMILARITY PARAMETERS	xi
INTRODUCTION	1
SUMMARY	3
CONCLUSIONS	6
DISCUSSION	10
Modeling Requirements	10
Modeling Criteria and Similarity Equations	12
Hydraulic Modeling	14
Model Fluid	16
Ideal Modeling	18
Nonideal Modeling	20
Dissimilar and Distorted Geometries	22
Distorted Dynamics	24
Model Scale	27
Hydraulic Abnormalities	37
Cavitation	38
Natural Convection	39
Gas Entrainment	40
Abnormal Transients	40
Mechanical and Mechanics Modeling	42
Model Fluid	42
Model Scale	44
Vibrations	45
Modeling Vibrations With Gas Model Fluid	46
Modeling Vibrations With Water Model Fluid	47
Vibrations Modeling Feasibility	49
ACKNOWLEDGEMENTS	53
REFERENCES	54
BIBLIOGRAPHY	55

TABLE OF CONTENTS (contd)

APPENDIX A	
FLUID FLOW MODEL THEORY	A-1
APPENDIX B	
NONIDEAL STEADY-FLOW MODELING	B-1
APPENDIX C	
THE EFFECTS OF DIMENSIONAL TOLERANCES ON MODEL ACCURACY	C-1
APPENDIX D	
MODELING WITH GAS MODEL FLUID	D-1
APPENDIX E	
FLUID PROPERTIES	E-1
APPENDIX F	
DISTRIBUTION	F-1

LIST OF FIGURES

1	Pressure Drop Ratio Versus Model Scale Ratio (Ideal model)	19
2	Reynolds Number Ratio Versus Scale Ratio	25
3	Froude Number Ratio Versus Scale Ratio	26
4	Reciprocal Friction Factor Ratio (Smooth surfaces) Versus Reynolds Number Ratio	28
5	Approximate HCM Water Flow Versus Model Scale	30
6	Model Theoretical Required Pumping Horsepower Versus Model Scale	31
7	Temperature Difference Ratio Versus Core Inlet Model Water Temperature (Full scale model)	41
8	Pressure Drop Ratio, Reynolds Number Ratio, and Linear Damping Ratio Versus Model Water Temperature	50
9	Velocity Ratio and Reynolds Number Ratio Versus Model Water Temperature	51
A-1	Required Scale Ratio Versus Model Water Temperature for Ideal Modeling Pressure Ratio = 1.0	A-9
A-2	Required Scale Ratio Versus Model Water Temperature for Ideal Modeling Pressure Ratio = 1.5	A-10
A-3	Required Scale Ratio Versus Model Water Temperature for Ideal Modeling Pressure Ratio = 2.0	A-11
A-4	Pressure Ratio, ΔP_r , Versus Model Water Temperature for Ideal Modeling at Full Scale	A-12
B-1	Q_{rv}/Q_{ri} Versus Re_r	B-5
B-2	Friction Factor Versus Reynolds Number	B-6
B-3	Reynolds Number Ratio Versus Scale Ratio Sodium Temperature = 600 °F	B-9

B-4	Reynolds Number Ratio Versus Scale Ratio Sodium Temperature = 700 °F	B-10
B-5	Reynolds Number Ratio Versus Scale Ratio Sodium Temperature = 800 °F	B-11
B-6	Reynolds Number Ratio Versus Scale Ratio Sodium Temperature = 900 °F	B-12
B-7	Reynolds Number Ratio Versus Scale Ratio Sodium Temperature = 1000 °F	B-13
B-8	Reynolds Number Ratio Versus Scale Ratio Sodium Temperature = 1100 °F	B-14
B-9	Reynolds Number Ratio Versus Scale Ratio Sodium Temperature = 1200 °F	B-15
B-10	Reynolds Number Ratio, Re_r , and Volumetric Flow Ratio, Q_r , Versus Model Water Temperature; Scale Ratio = 0.5	B-16
B-11	Reynolds Number Ratio, Re_r , and Volumetric Flow Ratio, Q_r , Versus Model Water Temperature; Scale Ratio = 0.6	B-17
B-12	Reynolds Number Ratio, Re_r , and Volumetric Flow Ratio, Q_r , Versus Model Water Temperature; Scale Ratio = 0.7	B-18
B-13	Reynolds Number Ratio, Re_r , and Volumetric Flow Ratio, Q_r , Versus Model Water Temperature; Scale Ratio = 0.8	B-19
B-14	Reynolds Number Ratio, Re_r , and Volumetric Flow Ratio, Q_r , Versus Model Water Temperature; Scale Ratio = 0.9	B-20
B-15	Reynolds Number Ratio, Re_r , and Volumetric Flow Ratio, Q_r , Versus Model Water Temperature; Scale Ratio = 1.0	B-21
B-16	Pressure Drop Ratio Versus Model Water Temperature for Constant Velocity	B-23
C-1	Estimated Model Flow Error Versus Variation of Characteristic Length for Overall Hydraulic System or Component	C-4

D-1	Reynolds Number Ratio Versus Pressure Drop Ratio; Air at 100 °F, 14.7 psia	D-2
D-2	Flow Ratio Versus Pressure Drop Ratio; Air at 100 °F, 14.7 psia	D-3
D-3	Reynolds Number Ratio Versus Pressure Drop Ratio; Air at 150 °F, 20 psia	D-4
D-4	Flow Ratio Versus Pressure Drop Ratio; Air at 150 °F, 20 psia	D-5
D-5	Reynolds Number Ratio Versus Pressure Drop Ratio; Air at 100 °F, 100 psia	D-6
D-6	Flow Ratio Versus Pressure Drop Ratio; Air at 100 °F, 100 psia	D-7
D-7	Reynolds Number Ratio Versus Pressure Drop Ratio; CO ₂ at 175 °F, 20 psia	D-8
D-8	Flow Ratio Versus Pressure Drop Ratio; CO ₂ at 175 °F, 20 psia	D-9
D-9	Reynolds Number Ratio Versus Pressure Drop Ratio; CO ₂ at 175 °F, 100 psia	D-10
D-10	Flow Ratio Versus Pressure Drop Ratio; CO ₂ at 175 °F, 100 psia	D-11
D-11	Reynolds Number Ratio Versus Pressure Drop Ratio; CO ₂ at 175 °F, 800 psia	D-12
D-12	Flow Ratio Versus Pressure Drop Ratio; CO ₂ at 175 °F, 800 psia	D-13
D-13	Reynolds Number Ratio Versus Pressure Drop Ratio; CO ₂ at 175 °F, 1200 psia	D-14
D-14	Flow Ratio Versus Pressure Drop Ratio; CO ₂ at 175 °F, 1200 psia	D-15
E-1	Density of Liquid Sodium	E-2
E-2	Kinematic Viscosity ($\frac{\mu}{\rho}$) of Liquid Sodium	E-3
E-3	Vapor Pressure of Sodium	E-4
E-4	Density of Water	E-5
E-5	Kinematic Viscosity of Water	E-6
E-6	Vapor Pressure of Water	E-7

NOMENCLATUREBASIC QUANTITIES

F	Force
θ	Time
L	Length
T	Temperature
A	Area L^2
a	Acceleration, L/θ^2
a'	Hydraulic hammer wave velocity, L/θ
B	Bulk modulus of elasticity (fluid) F/L^2
c	Specific heat $L^2/\theta^2 T$
C	Constant
d	Diameter, L
D	Diameter, L
De	Equivalent diameter, L
E	Modulus of elasticity, F/L^2
g	Gravitational constant
h	Convective film coefficient, $F/LT\theta$
I	Moment of inertia, L^4
k	Spring constant, F/L
K	Thermal conductivity $F/LT\theta$
ℓ	Length, L
L	Characteristic length, L
M	Moment (or torque) FL
P	Pressure, F/L^2
ΔP	Pressure drop, F/L^2
P_v	Vapor pressure, F/L^2
P_s	Static fluid pressure, F/L^2
q	Distributed load, F/L

Q	Volumetric flow rate, L^3/θ
R	Distributed load, F/L^2
S	Tolerance, L
t	Thickness, L
T_o	Reference temperature, T
ΔT	Temperature drop, T
U	Free stream velocity, L/θ
V	Velocity, L/θ
V_o	Reference velocity, L/θ
V'	Volume, L^3
w	Width, L
W	Concentrated load, F
Y	Vibrational amplitude, L

GREEK

α	Thermal expansion coefficients (solids) $1/T$
β	Temperature coefficient of volume expansion $1/T$
β_m	Bulk modulus of fluid elasticity, F/L^2
δ	Deflection, L
ϵ	Surface roughness, L
η	Damping coefficient, $F\theta/L$
γ	Specific weight, F/L^3
μ	Absolute viscosity, $F\theta/L^2$
ν	Kinematic viscosity, L^2/θ
π	Dimensionless parameter
ρ	Density, F/L^3 or $F\theta^2/L^4$
ρ_{eff}	Effective mass density, $F\theta^2/L^4$
ρ_{virt}	Virtual mass density, $F\theta^2/L^4$
σ	Surface tension, F/L
τ	Period, θ

Φ	Functional
Ψ	Functional
ω	Frequency, $1/\theta$

SUBSCRIPTS

m	Model
p	Prototype
r	Ratio, Model/Prototype
v	Viscous
i	Inertial or Index
T	Total

SIMILARITY PARAMETERSTITLED DIMENSIONLESS GROUPS:

Cn	$\frac{P_s - P_v}{\rho V^2}$	Cavitation Number
Eu	$\frac{\rho V^2}{\Delta p}$	Euler Number
Fr	$V/\sqrt{Lg_c}$	Froude Number
f	$2\Delta PL/\ell \rho V^2$	Darcy Friction Factor
Gr	$g_c \rho^2 \beta \Delta T L^3 / \mu^2$	Grashof Number
M	$V/\sqrt{E/\rho}$	Mach Number
Nu	$h L/K$	Nusselt Number
Pn	$a' V_o \rho / 2P_s$	Hydraulic Hammer
Pr	$C_p \mu / K$	Prandtl Number
Re	VL/ν	Reynold Number
Sl	$\omega L/V$	Strouhal Number
Wn	$\rho LV^2/\sigma$	Weber Number

OTHER DIMENSIONLESS GROUPS:

π_1	ϵ/L	Relative Roughness
π_2	ℓ/L	Length/Equivalent Length Ratio
π_3	δ/L	Normalized Deflection
π_4	q/EL^2	Normalized Distributed Load
π_5	W/EL	Normalized Point Load
π_6	M/LL^3	Normalized Moment (or torque)
π_7	R/E	Normalized Distributed Load
π_8	L/E	Normalized Specific Weight

π_9	S/L	Tolerance Ratio
π_{10}	$\alpha \Delta T$	Thermal Expansion
π_{11}	T/T_o	Temperature Distribution
π_{12}	F/kL	Amplitude Ratio
π_{13}	$\omega L \sqrt{\rho/E}$	Frequency Ratio
π_{14}	$\eta/L^2 \sqrt{\rho E}$	Damping Ratio
π_{15}	Y/L	Amplitude Ratio
π_{16}	$\sqrt{EI/\rho_{eff}} \frac{\tau}{L^3}$	Frequency Parameter
π_{17}	ρ_{virt}/ρ_{eff}	Virtual Mass Ratio
π_{18}	FL^2/EI	Fluid Induced Forces
π_{19}	$\sqrt{\rho_{virt}/EI} VL^2$	Normalized Parallel Velocity

APPLICATIONS OF GEOMETRIC MODELS
FOR THE FFTF HYDRAULIC CORE MOCKUP

D. S. Trent

INTRODUCTION

To carry out the desired modeling experiments in support of the FFTF core design and design verification efforts, the Hydraulic Core Mockup facility (HCM) has been proposed.⁽¹⁾ Model objectives for the HCM facility require both hydraulic and structural testing. The first question that must be answered is--what hydraulic, structural, and mechanical problems need to be resolved? Once the HCM problems or problem areas have been defined, one must then establish which similarity parameters dominate the physical processes to be modeled. Then it must be decided how test requirements and similarity requirements can best be satisfied.

There are four major variables or questions that must be answered before detailed model design of the HCM can be initiated, i.e.,

- What is the most feasible model fluid?
- What is the most appropriate model scale?
- Is a full (360°) core model needed or will a radial segment model suffice?
- Can all model test requirements be carried out in a single multipurpose model or are single purpose models required for meeting testing and similarity requirements?

In this study, these questions are answered in view of similarity and testing requirements and modeling practicability; however, model flexibility, economics and timing of data acquisition must also be considered in the final decisions.

In the design of the Fast Flux Test Facility (FFTF) reactor core hydraulic system, it is imperative that a means be available for adequately describing and predicting the sodium

coolant flow and core mechanical behavior. The fluid flow in the hydraulic core is complex and not readily amenable to analytic techniques. In addition, the coolant flow causes various mechanics problems which are further complicated by thermal effects. To provide adequate information for both the reactor core design effort and design verification, it is necessary to provide reasonably accurate prediction of the core flow distribution, pressure distribution, coolant mixing, flow induced vibrations, flow and thermal induced stresses and deflections, and mechanical fit-up. Both steady and unsteady conditions should be investigated along with various hydraulic abnormalities which might occur during reactor operation such as cavitation, gas entrainment, hydraulic hammer, vessel and closed loop rupture, and pumping failure.

It is virtually impossible to construct a sufficiently accurate analytic model to obtain this information because of the system complexity. For this reason the most expeditious way to obtain the required reactor core design development information is through techniques of physical modeling.

Physical modeling is an analog technique for obtaining comprehensive experimental data to aid and expedite the design and analysis of an engineering system. The ideas and applications of scale modeling are by no means new; Froude used modeling ideas to aid in ship design as early as the 1870's. Since that time modeling has found widespread use with a great deal of success in such applications as dam construction, ship building, aerodynamics, architecture, bridge building, chemical plant design and more recently, design of nuclear reactor systems.⁽²⁻¹¹⁾ Rarely today, is any engineering system built without first having undergone an intensive series of model tests to aid in the design, redesign, and diagnosis of the system.

The reason model testing is necessary in the design of a complex hydraulic system is because of the difficulty or impossibility of constructing a mathematical model in which we have sufficient confidence. Mathematical models are inherently an aggregation of established theory, past experiment and assumptions based on engineering judgement. In many cases, the time and expense required for model construction and testing may not be merited because we have sufficient confidence in engineering analysis and judgement, and/or the consequences of failure are not of paramount importance. In other cases, which certainly includes the FFTF Core, the consequences of a system failure may be extremely expensive, endanger human life, cause a drastic loss of time, or result in an unacceptable or poor system performance. Thus, we are not willing to depend strictly on analytical models and must base the design on a combination of analytical and experimental techniques which eliminate a great many of the uncertainties. We must keep in mind, however, that models are approximations and that model design also demands analysis and judgement to insure that reliable data may be obtained.

SUMMARY

The objective of this study was to perform a model analysis of the Hydraulic Core Mockup which would provide analytical criteria for model design and experiment. This objective was carried out by defining requirements for attaining adequate physical similarity of the FFTF hydraulic core concerning hydraulics, mechanics and certain thermal aspects.

It has been recommended that 1) the HCM facility should include a thermal mechanical studies model, and 2) a hydraulic studies model. The study objectives were formulated in terms of the following questions.

- What is the most feasible model fluid in each of these cases?

- Is scaling down model lengths feasible, and if so, what is the minimum practicable scale in view of obtaining reliable, comprehensive model data?

On the basis of model requirements, analysis, and practicability it is recommended that

- The Hydraulic studies model should be a full core with a minimum model scale of 0.2 to 0.5
- The thermal mechanical model should be full scale and accommodate either gas or water as a model fluid. A full core model is not required, only a radial segment of mechanical symmetry.

The detailed model analysis was carried out as follows:

A. Hydraulic Modeling

- The hydraulic similarity criteria was formulated from conditions required for modeling flow distribution and pressure distribution throughout the core, and bypass and plenum mixing. Euler number, Reynolds number, Froude number, were the dynamic similarity parameters considered.
- Both gases and water were considered as possible HCM model fluids. Water was considered the most feasible fluid for modeling hydraulics and all subsequent analysis was based on water.
- The conditions and requirements for ideal or true modeling were established. It was found that ideal modeling is possible, but might be impracticable for a desired HCM hydraulic model scale. (Also, a true model requires true geometric modeling).
- Since an ideal or true model may not be practicable, methods of inexact modeling were investigated. The analysis considered distorting fluid dynamics, kinematics and model geometry. It was determined that Euler

number is the dominant similarity parameter since viscous energy losses are a relatively weak function of Reynolds number over the model flow ranges and surface condition of interest. Results are presented in terms of distorted Reynolds number and Froude number as a function of model temperature and model scale, where the Euler number is assumed to be the same in model and prototype. The effects of these distortions are also discussed.

B. Mechanical and Mechanics Modeling

- The model criteria was formulated from conditions required for simulating mechanical fit, both cold and during heatup, and hydraulically and thermally induced loads, stresses, and distortions.
- Choosing the most feasible model fluid was the first problem investigated. It was determined that thermally induced stresses and distortions should be modeled at prototypic temperatures. This would require heater rods and an inert gas coolant to simulate the nuclear heat generation and to establish the temperature distribution. However, it was also determined that the magnitude of hydraulic loads could not be modeled with gas. To establish stresses and distortions at prototypic temperatures, the flow induced loads would have to be modeled mechanically. Gas is not a good fluid to model flow induced vibrations; water is the most feasible in this case. Hence, the thermalmechanical model should accommodate both gas and water model fluids.
- Determination of the most feasible model scale was based on the problem of scaling prototypically small tolerances, measuring scaled deflection, and adequate satisfaction of several other similarity parameters. Modeling stresses and distortions was found to be the most feasible at full scale.

C. Modeling Hydraulic Abnormalities

Modeling of several hydraulic abnormalities which might occur in the FFTF include:

- Cavitation
- Natural circulation
- Gas entrainment
- Abnormal transients (pump failure, hydraulic hammer, vessel and gas loop ruptures).

CONCLUSIONS

Modeling the FFTF hydraulic core to establish hydraulic and structural behavior is entirely feasible and is essential for obtaining reliable, comprehensive engineering data to both guide and confirm the reactor core design. From the standpoint of modeling similitude it is concluded that

- One, multifluid, full scale multipurpose model is feasible for accomplishing required hydraulic and structural test programs. However; it may be desirable, because of timing requirements, distinct objectives of hydraulic and structural test programs, and the earlier delivery possibility for a reduced scale hydraulic model, to consider two models (i.e., a full scale structural and hydraulic model and a reduced scale hydraulic model).
- The full scale structural model capability should encompass the required thermal mechanical studies and some hydraulic studies, but the reduced scale hydraulic studies model need only meet hydraulic test requirements. This would not only aid in expediting model testing, but would also relax or eliminate certain model similitude requirements in each case, and lead to more practicable and flexible modeling.
- Various single purpose facilities should be constructed and tested as required by the reactor design effort.

These models would be simple to construct, using any appropriate scale depending on purpose, and would be very useful in expediting design test information. Such tests might include subassembly vibrations and hydraulic tests, fuel rod spacer hydraulic tests, orifice and nozzle hydraulic tests, etc.

Specific conclusions are based on consideration of two major models: 1) hydraulic studies model, 2) thermal mechanical model; directed specifically to the needs and requirements of adequate modeling in each case.

1. Hydraulic Studies Model

- Water is the most feasible model fluid.
- It is feasible to use a scale model, but the scale should not be less than about 0.2 to 0.5 (for steady flow a model scale of about 0.2 is feasible; however, for unsteady testing the scale should be closer to 0.5). Determination of an absolute minimum scale for adequate modeling would require more specific details concerning the prototypic design than is available at the present time and would require a more detailed analysis than is presented in this study.
- The model should consist of a full (360°) core as opposed to a core radial segment.
- In the case of unsteady flow simulation the model should be geometrically similar in all respects which includes scaled tolerances where prototypic characteristic lengths are small. It is sufficient to make all model surfaces smooth.
- In the case of steady flow modeling at or near a predetermined reactor flow, the fuel rod bundles may be simulated with orifice plates. The orifice plates serve a dual purpose in that they can be used as flow metering

devices, and at the same time eliminate the cost, time, and difficulty of fabricating redundant scaled fuel pins which are prototypically small. Orifice plate simulated fuel rod bundles are not valid for unsteady flow modeling.

- Dynamic similarity is dictated by Euler number and Reynolds number. If in the final core design there is significant dynamic interaction between the sodium pool and the core hydraulics, Froude number similarity must be considered.
- Ideal or true modeling is unnecessary in obtaining adequate model results; i.e., it is not necessary to insist that $Re_r = Eu_r = 1.0$. In view of the relatively weak dependence of viscous energy losses on Reynolds number (i.e., $f \sim Re^{-0.2}$), it is appropriate to assume that Euler number similarity insures sufficient dynamic similarity for adequate modeling, except where Reynolds number ratios are small while the model flow is significantly viscous dependent.

Model tests or investigations which are feasible using the full core steady flow hydraulic studies model.

- Plenum flow distribution
- Plenum pressure distribution
- Plenum mixing
- Bypass mixing or core effluent attemperation
- Gas entrainment (possibly only qualitatively)
- Cavitation

Model tests or investigations which are feasible only with a geometrically similar, transient flow, hydraulic studies model.

- Hydraulic hammer (possibly only qualitatively)
- Local flow stoppages
- Partial or complete pumping failure
- Vessel rupture
- Closed loop rupture

2. Thermal mechanical Model

- The facility should accommodate both a heated water and a heated inert gas model fluid.
- The facility should be full scale and use prototypic components if possible.
- A full (360°) core is not necessary, only a convenient segment of mechanical symmetry (nominally 90° appears adequate with present reference concepts).
- The model should be geometrically similar to the prototype (except for segment) in every respect including tolerances.
- Thermal stresses and distortions should be modeled using the predicted prototypic temperature distribution. This model would require heat generation simulation with heater rods, and inert gas flow to establish the proper temperature magnitude and distribution, and superimposed loads to simulate fluid dynamic, and pressure forces. (It is impracticable to quantitatively model combined hydraulically and thermally induced stresses and distortions using gas or water model fluid to simulate hydraulic forces).
- Water is the most feasible fluid for modeling flow induced vibrations, (it may not be possible to obtain all of the required quantitative information with a gas model fluid).

Model tests or investigations which are feasible using the segmented thermal mechanical model.

Gas Model Fluid

- Mechanical fit-up displacement and operational tasks, cold and hot.
- Thermal stresses and distortion
- System and component vibrations (possibly qualitative only)
- Hydraulics (excluding plenum and pool effects)

Water Model Fluid

- Natural circulation (with heater rods)
- System and component vibrations
- Hydraulics (excluding plenum and pool effects)

DISCUSSION

MODELING REQUIREMENTS

The general purpose of the Hydraulic Core Mockup is both to guide and to diagnose the reactor core design through experimentation. In this capacity, the facility must first meet the following general criteria.

- The model facility must be flexible with regard to anticipated design changes resulting from efforts to improve undesirable reactor hydraulic and structural behavior.
- The model testing program must be compatible with the FFTF design schedule. That is, the required model testing must be carried out at a time which is the most beneficial to the reactor design effort.

The above criteria imply that the model must not only provide reliable, comprehensive engineering data, but also it must provide the information in the most expeditious way possible.

The HCM testing requirements⁽¹⁾ are formulated in terms of several individual key hydraulic and/or structural design problems and uncertainties. Certain of these problems have a major influence on the reactor core and reactor vessel configuration, and need to be investigated and evaluated experimentally early in the reactor design phase. These problem areas and associated modeling requirements are summarized as follows.

- Core Inlet Flow: characterization of the inlet and lower plenum flow to determine flow and pressure distribution. The facility must accommodate design changes in the inlet nozzle, baffle, tubesheet, and subassembly inlet arrangements.

- Core Outlet Flow: characterization of core outlet plenum to determine flow and pressure distribution and mixing efficiency. The facility must accommodate design changes in the plenum geometry, plenum outlet nozzles, and sub-assembly design.
- Core Reflector Flow: characterization of flow and pressure distribution through the reflector and core regions. Model design must accommodate changes in reflector and core design.
- Coolant Leakage Flow: characterization of the coolant leakage flow with respect to the mechanical fit of the core subassemblies and core pressure drop.
- Core Bypass Flow: characterization of the bypass mixing efficiency in the outlet plenum. The model must accommodate variable bypass flow and design changes in the mixing region.
- Mechanical Fit: characterization of component and system mechanical compatibility to investigate and evaluate mechanical fit, both cold and at prototypic operating temperatures.
- Component and System Stresses: characterization of thermally and hydraulically induced component and system loads and moments to investigate thermally and hydraulically induced stresses and mechanical distortions.
- Core Vibrations: characterization of mechanical response from dynamic loading to investigate the possible occurrence of adverse vibrational resonances within the prototypic system, and evaluate methods for minimizing these conditions should they occur.

In the early design stage, the above problem areas will be investigated under simulated reactor conditions at or near full design flow. Subsequent testing would involve the same

problem areas and requirements, but at various nonstandard reactor operating conditions---provided these tests are feasible. Some of these nonstandard conditions might include

- Flow transients, including hydraulic hammer
- Thermal or power transient
- Flow stoppages which might occur at various points in the system
- Pump failure (partial loss or complete loss of coolant flow)
- Vessel rupture
- Closed loop rupture
- Cavitation
- Gas entrainment
- Emergency cooling (natural circulation).

MODELING CRITERIA AND SIMILARITY EQUATIONS

Once the Hydraulic Core Mockup requirements have been defined, it is then a question of designing and constructing a model or models which will serve the purpose for obtaining comprehensive and reliable results with acceptable accuracy. The model similarity criteria must include requirements for both flow and structural similarity. Ideally, the requirements for flow similitude are satisfied if model and prototype are geometrically, kinematically, and dynamically similar. Structural similarity is somewhat more complicated and difficult to achieve in a reactor model because not only must hydraulically induced loads and moments be simulated, but, external static loads and moments, structural materials properties, temperature distributions, and structural dynamic responses must be similar.

Similarity functionals and parameters which will govern or influence the HCM design are given below. A more detailed discussion of the similarity parameter development and implications is found in Appendix A.

HYDRAULICS:

$$\psi_1 \left[\frac{\varepsilon}{L}, \frac{\ell}{L}, Re, Fr, Eu \right] = 0, \quad 1$$

STATIC DEFLECTION:

$$\psi_2 \left[\frac{\delta}{L}, \frac{q}{EL}, \frac{P}{EL}, \frac{M}{EL^3}, \frac{R}{E}, \frac{\gamma L}{E}, \frac{S}{E}, \alpha \Delta T \right] = 0, \quad 2$$

HEAT TRANSFER (FORCED CONVECTION):

$$\psi_3 \left[\frac{\varepsilon}{L}, \frac{\ell}{L}, \frac{T}{T_o}, Re, Pr, Nu \right] = 0, \quad 3$$

VIBRATIONS:

In general

$$\psi_4 \left[\frac{F}{KL}, WL\sqrt{\frac{\rho}{E}}, \frac{\eta}{L^2\sqrt{\rho E}} \right] = 0, \quad 4$$

For structural elements in parallel fluid flow,

$$\psi_5 \left[\frac{Y}{L}, \sqrt{\frac{EI}{\rho_{eff}}}, \frac{\tau}{L^3}, \frac{\rho_{virt}}{\rho_{eff}}, \frac{FL^2}{EI}, \sqrt{\frac{\rho_{virt}}{EI}}, VL^2 \right] = 0. \quad 5$$

In addition, other similarity parameters that have importance for modeling certain reactor operating conditions are

- Cavitation Number: $(P_s - P_v)/P_v^2$, for investigation of vapor pocket formation.
- Hydraulic Hammer (pipe line): $\frac{a' V_o \rho}{2P_s}$, where a' is the pressure wave celerity. For

$$1. \text{ Rigid Pipes } a' = \sqrt{\frac{B}{\rho}},$$

$$2. \text{ Nonrigid Pipes } a' = \sqrt{\frac{B}{\rho \left[1 + \left(\frac{B}{E} \right) \left(\frac{L}{t} \right) \right]}}.$$

- Strouhal Number: $\omega L/V$, to investigate the effects of vortex shedding.
- Grashof Number: $\rho^2 g_c \beta \Delta T L^3 / \mu^2$, to investigate natural circulation.

In the final analysis some of the above similarity parameters will undoubtedly be negligible and some may not apply.

HYDRAULIC MODELING

The premise on which modeling is based is the theory of similitude. Similitude means that all physical quantities that govern a certain phenomena in a prototype bear the same relative influence in the model. Through dimensional analysis one may group the basic quantities governing a physical phenomena into dimensionless groups called similarity parameters. A physical process may then be described by a functional

$$\psi(\pi_1, \pi_2, \pi_3, \dots, \pi_k) = 0,$$

where the π 's are the dimensionless groups or similarity parameters. As long as the π 's and the relationship between the π 's are the same, the functional will have the same value for both the prototype and model, and the conditions for physical similitude will be satisfied.

The first task in model design is to establish what quantities or phenomena the model is to simulate (e.g., pressure drop, flow distribution, vibration, etc). Once this has been decided, one must then establish which similarity parameters must be considered, the approximate influence of each on the desired result, and the probable accuracy of results. The relative influence of each similarity parameter on the model behavior is important because it is usually impossible to ideally model more than two or perhaps three parameters simultaneously; thus, those adjudged to have minor importance must be neglected in the model design. Even when modeling two parameters, severe restrictions may be imposed on the modeling fluid and model scale to meet the criteria for exact modeling. As it turns out, exact or ideal modeling in a complex system is usually impractical, if not impossible. But, more often, it is unnecessary to insist that a model be ideal or simulate the prototypic behavior in every respect. It is not the purpose of the model to simulate the prototype in every respect, but to provide an

adequate description of the prototypic behavior and yield data which may be related quantitatively, when possible, to the prototypic system within a predictable degree of accuracy.

Information gained from a model may either be quantitative or qualitative. Quantitative information is that which establishes numbers and references for such basic quantities as force, time and distance. On the other hand, qualitative information can only identify or distinguish certain phenomena and system characteristics.

Modeling fluid flow in a closed reactor coolant system, such as a BWR or PWR, requires similarity of the functional

$$\Psi(\ell/L, \epsilon/L, Re, Eu) = 0 \quad . \quad (6)$$

For a system that includes a dynamically interacting coolant pool, the surface configuration may require simulation, thus involving Froude number similarity. In this case the functional becomes

$$\Psi(\ell/L, \epsilon/L, Re, Eu, Fr) = 0 \quad . \quad (7)$$

If the model has complete geometric similarity (i.e., a unique and uniform scale which insists that essentially every detail be of true scale including surface roughness), the similarity functionals become

$$\Psi(Re, Eu) = 0 \quad (8)$$

or

$$\Psi(Re, Eu, Fr) = 0 \quad . \quad (9)$$

From Equation (6) above, the requirement for ideal dynamic similarity is

$$Re_r = Eu_r = 1.0 \quad , \quad (10)$$

$$\left(\frac{VL}{\nu}\right)_r = \left(\frac{\rho V^2}{\Delta p}\right)_r = 1.0 \quad . \quad (11)$$

From Equation (7), ideal dynamic similarity requires that

$$\text{Re}_r = \text{Eu}_r = \text{Fr}_r = 1.0 \quad , \quad (12)$$

$$\left(\frac{VL}{\nu}\right)_r = \left(\frac{\rho V^2}{\Delta P}\right)_r = \left(\frac{V}{\sqrt{L}}\right)_r = 1.0 \quad . \quad (13)$$

Generally, unless a model fluid is used with density and viscosity similar to that of the prototypic coolant, it is difficult to satisfy Equation (6) exactly and more difficult to satisfy (7). Even in this case, exact simulation may not be practical. In practice it is not expected to satisfy Equations (6) or (7) exactly, but to obtain the best approximation that is practical.

Model Fluid

The two most common model fluids used in reactor core flow modeling are gases and water. Of the gases, air, nitrogen, and high pressure carbon dioxide are probably the most widely used. Low pressure gases are quite useful for modeling flow distributions and mixing in highly turbulent closed system flow. In these applications, the model can be fabricated from inexpensive materials and model modifications can be made with minimum effort and expense.

Dynamic similarity is difficult to achieve in gas model systems where both viscous and momentum effects are to be modeled simultaneously; especially if unsteady flow is to be simulated or prototypic flow is not highly turbulent. The reason for this is that typically pressure drops are only a small fraction of prototypic and Reynolds numbers may be only 1/30 to 1/100 of prototypical (see curves in Appendix D). This is no serious disadvantage if prototypic flow is turbulent enough to permit fully turbulent flow in the model; but if the prototypic flow is not highly turbulent, a completely different flow regime will be present in the model.

In the FFTF core plenums, the Reynolds numbers will probably be in the range of 1×10^5 to 5×10^6 . A Reynolds number reduction by a factor of 50 or so, could put model Reynolds numbers as low as 20,000 or less. In addition, if the pressure ratio is not increased simulation of partial pumping failure could reduce the model flow into the critical or laminar flow regime.

If a gas such as carbon dioxide is used for a model fluid, dynamic similarity is much easier to achieve at high pressure than at low pressure (See curves in Appendix D). This is mainly a result of the increased gas density; hence, kinematic viscosity. In the case of high pressure CO_2 , ideal hydraulic modeling for a closed system (i.e., $\text{Re}_r = \text{Eu}_r = 1$), especially at small scales, is entirely feasible. In modeling practice, this does not mean that the Reynolds number would be exactly simulated, but that Re_m would not be grossly different than Re_p ; hence, reasonably accurate account could be given the frictional energy losses if Eu number is used as the modeling criterion.

If a gas model fluid were used in the HCM, the flexibility of the facility would be limited, even under high pressure conditions. The sodium pool surface configuration could not be modeled (Froude number similarity) to investigate jetting through possible apertures in the reactor thermal shield. Possible gas entrainment at the pool surface and cavitation could not be investigated.

On the other hand, water adds flexibility to the model facility in that it is a liquid, like sodium, and has a density and viscosity quite similar to sodium over the temperature ranges of interest. Hence, the sodium pool can be modeled if need be; gas entrainment and cavitation can be observed visually; and hydraulic similarity parameters can be more readily and accurately modeled; thus, lending a greater degree of accuracy and confidence to the experimental data. For

these reasons it is suggested that water be used for hydraulic model fluid. All of the following hydraulic considerations are based on water modeling sodium.

Ideal Modeling

An ideal or true model requires that geometric, kinematic, and dynamic similitude be unconditionally satisfied throughout the model. If the FFTF sodium pool is not modeled, flow dynamics similarity requires that

$$Re_r = \frac{V_r L_r}{\nu_r} = 1.0 \quad , \quad (14)$$

and

$$Eu_r = \frac{\rho_r V_r^2}{\Delta P_r} = 1.0 \quad . \quad (15)$$

These equations can be combined in several ways, but in terms of the scale ratio and fluid properties, the pressure drop ratio may be stated as

$$\Delta P_r = \frac{\rho_r \nu_r^2}{L_r^2} \quad . \quad (16)$$

Hence,

$$L_r = \frac{\nu_r \sqrt{\rho_r}}{\sqrt{\Delta P_r}} \quad , \quad (17)$$

which defines the required scale for an ideal model. Either Equation (14) or (15) may be used to define the velocity ratio:

$$V_r = \frac{\nu_r}{L_r} \quad , \quad (18)$$

or

$$V_r = \sqrt{\frac{\Delta P_r}{\rho_r}} \quad . \quad (19)$$

Figure 1 illustrates the required pressure ratio for water ideally modeling sodium at 1000 °F (thermodynamic properties of water and

FIGURE 1. Pressure Drop Ratio Versus Model Scale Ratio (Ideal Model)

sodium are given in Appendix E) as a function of model scale and for various model temperatures. This figure reveals that the required model water temperature might be impracticably high for ideally modeling the FFTF core hydraulics on a scaled down basis. For instance, when $\Delta P_r = L_r = 1.0$, the required model temperature is about 235 °F. If $\Delta P_r = 1.0$ and $L_r = 0.5$, the model temperature would have to be about 475 °F; whereas in this case, if $\Delta P_r = 2.35$ the required model water temperature would need to be about 300 °F.

If the sodium pool were also to be ideally modeled at the same time, Froude number similarity must be satisfied, or

$$Fr_r = \frac{V_r}{\sqrt{L_r}} = 1.0 \quad . \quad (20)$$

The similarity equation is

$$L_r = v_r^{2/3} \quad , \quad (21)$$

or equivalently

$$\rho_r L_r = \Delta P_r = \rho_r v_r^{2/3} \quad . \quad (22)$$

By satisfying an additional similarity parameter, a degree of freedom has been lost. Requirements for ideal dynamic similarity which includes the Froude number are also shown in Figure 1. A significant illustration in this plot is that Euler and Reynolds number similarity is satisfied by a family of curves, whereas Froude number similarity is satisfied only along the curve A-B. For example, if $\Delta P_m = \Delta P_p$ for a model simulating 1000 °F, then the model water temperature and model scale is fixed at 260 °F, and 0.89, respectively.

Nonideal Modeling

According to Figure 1, if the similarity criteria is based on the three parameters Eu, Re, and Fr, ideal simulation of the

FFTF core hydraulics is extremely restrictive of the model scale, pressure drop ratio, and water temperature. In fact, for water modeling sodium at a given temperature, there is only one degree of freedom among the remaining variables. Thus, by setting the pressure drop ratio, the required model scale and water temperatures are fixed. As pointed out previously, for $\Delta P_r = 1.0$, the model scale is fixed at 0.89 and the water temperatures fixed at 260 °F (sodium at 1000 °F). Ideal modeling at 0.5 scale is impracticable because water temperature in the critical region would be needed.

If Froude number similarity is neglected there is one additional degree of freedom among the modeling parameters, but ideal simulation may still be difficult at smaller scales (e.g., 0.5 scale) because of the necessary high water temperatures or required increase in the pressure drop ratio. It is desirable to keep the water temperature as low as practical to mitigate heating problems and reduce the required system pressurization. Single component testing at full scale would be rather easily accomplished because at full scale the required water temperature is about 237 °F. In view of the above examples and implications illustrated by Figure 1, and from the standpoint of required model accuracy, it is probably more practicable to use the techniques of inexact or nonideal modeling (Appendix B).

Nonideal modeling, as defined in this study, includes all modeling techniques that are designed to yield acceptable experimental results without insisting that all conditions for geometric, kinematic, and dynamic similarity are satisfied. These methods may include distorted or dissimilar geometry, and distorted similarity parameters. Exactly what distortions can be made and the extent of dissimilarity that is acceptable really depends on the model purpose and desired accuracy of results.

Dissimilar and Distorted Geometrics

Using dissimilar and distorted model geometry is a technique for simplifying models and/or correcting for nonideal flow dynamics in steady flow applications. The discussion in this section applies to designed geometric distortions and dissimilarities, and should not be confused with unintended or unavoidable distortions such as incorrect scaling of tolerances and surface roughness.

Steady flow models are designed for testing at, or very near, a predetermined flow rate. In this case the more complicated and redundant model components (e.g., FFTF driver fuel subassemblies) can be replaced with orifice plates, nozzles, or other dissimilar devices which are calibrated to pass the correct flow and effect the correct pressure drop. The calibration would be carried out by using the data from full scale, single-component testing done in a separate facility.

Some of the advantages of modeling the FFTF driver fuel bundles with orifices plates are given below.

- The time, expense and difficulty of fabricating scaled down components that are redundant and have small prototypic characteristic lengths, is avoided. These items are especially important in view of testing expediency and anticipated changes in fuel designs.
- The model scale is not limited by inability to model fuel pins and flow passages accurately.
- Orifice plates are dual purpose devices in that they can also be used for flow measurement.

Use of orifice plates is not without disadvantages, some of which are listed below.

- Calibration is only valid at or near a predetermined flow; hence, only steady flow testing is possible.

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- It is difficult to size the orifice plates correctly because the effective pressure drop through a subassembly (prototype or model), in a single component test device (orifice plate calibration apparatus) may not be the same as it would be in the assembled core. This is because plenum effects such as neighboring subassemblies, frictional losses, and increases in velocity head, would not be simulated in a calibration device. The result of incorrect subassembly calibration might lead to appreciable errors in the model plenum flow and pressure distribution unless corrected through analytical and/or experimental methods similar to that discussed in Reference 11.
- The model is only useful for mixing, pressure and flow distribution evaluation (in steady flow).

The use of scaled, exact replica fuel bundles would eliminate the orifice plate disadvantages listed above, but as noted previously, exact replication of all fuel bundles would be expensive, time consuming, and difficult. Also, the scale ratio cannot be made too small because tolerances cannot be scaled down past a certain point without introducing unintended and unavoidable geometric distortions which would lead to erroneous test results.

One example of model length distortion, which may find application in the HCM, is distorting the flow path length to correct nonideal viscous energy losses. If the Reynold number ratio is less than 1.0, then viscous energy losses in the prototype will be dissimilar in the model. In fact, the pressure loss in the model will be too large. A correction can be made by decreasing the flow path length---i.e., decreasing the ℓ/L ratio. The only apparent disadvantage in this technique is that the distortion is strictly valid only for a predetermined steady flow.

Distorted Dynamics

Nonideal simulation of the Reynolds, Euler and Froude number is reflected in inaccurate flow dynamics. A dissimilar Reynolds number leads to inaccurate viscous energy losses, whereas an incorrect Euler number leads to poor simulation of form or shock losses. If the Froude number must be considered, an incorrect model would yield misleading data for the sodium pool simulation and its interaction on the rest of the system. On the otherhand, it will probably be impractical (although possible, Figure 1) to model Re, Fr and Eu ideally in the FFTF. Modeling Re and Eu ideally with small model scales may also be unattractive because of the required model water temperature or pressure ratio.

The approach to nonideal dynamical modeling in this study is to base the analysis on Euler number similarity, and let both Reynolds number and Froude number vary accordingly. Hence, the following set of equations becomes the basis for the model analysis.

$$Eu_r = \frac{\rho_r V_r^2}{\Delta P_r} = 1.0 \quad , \quad (23)$$

$$V_r = \sqrt{\Delta P_r / \rho_r} \quad , \quad (24)$$

$$Q_r = \sqrt{\Delta P_r / \rho_r} L_r^2 \quad , \quad (25)$$

$$Re_r = \sqrt{\frac{\Delta P_r}{\rho_r}} \left(\frac{L_r}{\nu_r} \right) \quad , \quad (26)$$

$$Fr_r = \sqrt{\frac{\Delta P_r}{\rho_r L_r}} \quad . \quad (27)$$

Figures 2 and 3 illustrate the effect of model water temperature, pressure drop ratio, and model scale on the Reynolds number ratio and Froude number ratio, respectively, $Eu = 1.0$, prototypic

FIGURE 2. Reynolds Number Ratio Versus Scale Ratio

FIGURE 3. Froude Number Ratio Versus Scale Ratio

sodium at 1060 °F. Euler number similarity was chosen as the base criterion because pressure drop and flow distribution is a relatively weak function of Reynolds number according to

$$f_r = Re_r^{-0.2} \quad .$$

Only for flow in the model
(28)

Figure 4 demonstrates this relationship. The conclusion that can qualitatively be drawn from Equation (28) and Figure 4 is that viscous energy losses will be reasonably well modeled using Euler number similarity, except in cases where Re_r is small or a different flow regime occurs in the model (e.g., laminar or critical flow in the model, turbulent flow in prototype).

The effects of nonideal Froude numbers and the possible corrections are difficult to assess at the present time. To estimate the sodium pool surface hydraulic interactions on the system flow, a reasonably definite description of the upper plenum, thermal shield, and pool design is needed. However, one method of correcting for gravity effects in the pool region might be by distorting the pool depth and perhaps the diameters of the fuel tube assemblies (in the pool region only).

Model Scale

The incentive for scaling down a reactor model is cost. By reducing the model scale, savings are derived through reduced facility and pumping cost. In this study, only pumping or flow requirements will be discussed.

Pumping requirements can be approximated by using Equation (25) or

$$Q_r = \frac{\sqrt{\Delta P_r} L_r^2}{\sqrt{\rho_r}} \quad . \quad (29)$$

FIGURE 4. *Reciprocal Friction Factor Ratio (Smooth Surfaces)
Versus Reynolds Number Ratio*

Figures 5 and 6 show the approximate model water flow and pumping horsepower required as a function of model scale based on a 1000 °F mean temperature prototypic sodium flow of 40,000 gal/min. The model water flow curves shown are solutions to

$$Q_m = 40,000 \sqrt{\Delta P_r} L_r^2 \left(\frac{1}{\sqrt{\rho_r}} \right), \quad (30)$$

where $\rho_r = 1.0$. The appropriate density correction multiplier may be obtained from the inset. For example, at 1/2 scale with $\Delta P_r = 1.0$, the required model flow would be about 9,300 gal/min (200 °F model water). If the model pressure drop were raised to twice prototypic, the model flow would be about 13,100 gal/min. If full scale model water flow is compared to an arbitrary scale model water flow at the same temperature and pressure drop, the ratio of the two flows vary as the scale squared, or

$$\frac{Q \text{ (Model)}}{Q \text{ (Full scale model)}} = L_r^2. \quad (31)$$

Hence, a scaled down model can result in a considerable reduction of model water flow.

It is a reasonably simple task to at least define a proper model scale for the HCM if it is insisted that the model be geometrically, kinematically, and dynamically similar to the prototypic FFTF in every respect. But, as pointed out previously, the required water temperature would be high for small scales (refer to Figure 1) and model scales below about 0.6 are essentially impracticable for ideal modeling. If possible, it would be best to use model water from ambient temperature up to 180 to 190 °F to reduce heating and eliminate pressurization, but on an ideal basis this would require a scale greater than 1.0. Therefore, the practical approach is to arrive at a model scale from a nonideal modeling viewpoint.

FIGURE 5. Approximate HCM Water Flow Versus Model Scale

FIGURE 6. Model Theoretical Required Pumping Horsepower Versus Model Scale

For nonideal modeling, some estimate of inaccuracy introduced by the nonidealization must be made. This, of course, is an impossible task to do exactly because the detailed prototypic flow behavior would have to be known beforehand. However, some rather rough error bounds can be established and as long as they remain within the criterion of acceptable accuracy, the model is assumed to be adequate (Appendix C).

The minimum practicable model scale is strongly dependent on whether the model facility is to be used for steady or unsteady flow testing--or both. As pointed out in the section on nonideal geometrics, unsteady models require a greater degree of geometric, kinematic, and dynamic similarity than do steady flow models, and in fact, the fuel elements would need to be modeled accurately.

First generation hydraulic testing in the HCM will be in support of reactor design at or near full core flow. These tests would involve evaluation of the bypass mixing efficiency and upper and lower plenum mixing, flow and pressure distribution at steady flow. Based on offsite experience⁽⁹⁾ and the work done in the present study, the most expeditious way to carry out these objectives is to use a scaled down geometrically exact model of the full reactor core, except for hydraulically modeled fuel bundles, reflector and seepage flow which would be simulated with orifice plates or nozzles. Individual component testing to establish the correct calibration for the nozzles and orifices would be done in a separate facility. Partial flow (such as caused by pump failure) would be carried out in the model by resizing the orifices and nozzles to accommodate the reduced flow.

In the prototype, at full reactor flow, upper and lower plenum Reynolds numbers will be in the neighborhood of $(1 \times 10^5$ to $5 \times 10^6)$. At these Reynolds numbers, the flow is fully turbulent or Reynolds number independent. Also, form and shock losses will be large in comparison to any frictional energy

losses. Hence, the modeling criterion is that the same type of energy losses should dominate in the model. Euler number similarity assures dynamic similarity in the plenums, and the limitation on the scale, from a similarity viewpoint, is that frictional and Reynolds number dependent losses that are not representative of the prototype should be avoided in the model.

Referring to Figure 2, it is reasonable to expect an excellent degree of dynamic similarity for all scale ratios above about 0.2, with water temperature as low as 120 °F and at the prototypic pressure drop. At a scale of 0.2, water temperature of 120 °F, and $\Delta P_r = 1.0$, the plenum model Reynolds numbers should be above about 10^5 (Reynolds number independent in this case). At this scale only about 4% of the full reactor flow would be required for the model (about 1600 gal/min).

However, there are certain other factors inherent with small scales that detract attractiveness of the small model size and flow requirements. Some of these factors are

- The model is so small that placement of required instrumentation is impossible or measuring equipment affects the flow behavior significantly.
- The model is so small that correct geometric similarity and/or orifice and nozzle sizing is impossible or economically impractical (for steady flow similarity requirements).
- The model scale should not be so small that replica fuel bundles and other components that are typically small at full scale, cannot be fabricated adequately and at reasonable cost (to accommodate possible transient testing).

To assess the above factors quantitatively along with similarity requirements, would require better definition of the reactor system than is possible at this time, and would involve considerably more analysis than covered in this study. Even with complete information at hand, an optimum scale (optimizing

cost against adequate testing) would be difficult to define and probably would not merit the extensive study necessary to define an optimum. For these reasons, choosing a model scale for the HCM requires a good deal of unavoidable qualitative assessment and decisions based on engineering judgement.

Table I illustrates some of the pertinent quantities for modeling at 0.2, 0.5, and full scale, and with model water temperatures at 140 °F and 280 °F. It should be kept in mind that the prototypic values, although typical, are only approximate and by no means fixed. Table II shows a qualitative evaluation of these three model scales.

Scale effect
A 0.2 scale model would probably suffice for modeling mixing and plenum flow and pressure distribution in steady flow. The inability to model frictional effects and scale tolerances is of little consequence in the plenum flow and bypass mixing, because friction contributes little energy loss in these regions and prototypic tolerances are small compared to flow characteristics lengths. The model size and flow requirement is extremely attractive. However, the model would be difficult to instrument and would be useful only for steady flow bypass mixing and plenum modeling. Scaling fuel bundles, or other components with small flow characteristic lengths is necessary for unsteady core flow testing. However, this is impracticable at 0.2 scale; hence, the model facility would be limited in this capacity.

At the opposite extreme, Tables I and II indicate that full scale is superior in every respect except for physical size and pumping requirements. However, it is significant to compare full scale with model water at 140 °F and 0.5 scale at 280 °F. Table I shows that flow dynamics at 0.5 scale (water at 280 °F) are more accurately simulated (except for gravitational effects which may be unimportant) than at full scale with water

TABLE I. Typical Hydraulic Model Geometric and Dynamic Characteristics- $Eu_r = 1.0$
(Prototype values are approximate)

		Prototype 1000 °F Mean Sodium Temp.	Scale		
			2/10	5/10	Full
Mean Active Core Diameter		4.0 ft	0.8 ft	2.0 ft	4.0 ft
Active Core Height		2.8 ft	0.56 ft	1.4 ft	2.8 ft
Fuel Subassembly Diameter		3.0 in.	0.6 in.	1.5 in.	3.0 in.
Fuel Rod Diameter		0.25 in.	0.05 in.	0.125 in.	0.25 in.
Bottom Tube Sheet, Tube Array Diameter		3.50 ft	0.7 ft	1.75 ft	3.5 ft
Reactor Vessel Inside Diameter		15.0 ft	3.0 ft	7.5 ft	15.0 ft
Lower Plenum Diameter		13.0 ft	2.6 ft	6.5 ft	13.0 ft
Reactor Vessel Height		38.0 ft	7.6 ft	19.0 ft	38.0 ft
Model Water Temp.					
Reynolds Number	140		0.105	0.260	0.525
Ratio, Re_r	280		0.240	0.60	1.20
Froude Number	140		2.20	1.30	0.95
Ratio, Fr_r	280				
Upper and Lower	140		0.1 to	0.2 to	0.52 to
			0.5×10^6	1.0×10^6	2.4×10^6
Plenum Re	280	1×10^5 to 5×10^6	0.24 to	0.6 to	1.20 to
			1.2×10^6	3.0×10^6	6.0×10^6
Fuel Bundle Re	140	10^5	0.1×10^5	0.26×10^5	0.52×10^5
	280		0.24×10^5	0.60×10^5	1.20×10^5
Vessel Flow	140	40,000 gal/min	1440	9080	36,700
			gal/min	gal/min	gal/min
	280		1510	9440	37,900
			gal/min	gal/min	gal/min
Friction Factor Ratio	140		0.63	0.76	0.88
f_p/f_m	280		0.75	0.90	1.10

TABLE II. Model Scale Suitability

	Steady Flow Model			Steady Flow or Transient Flow Model		
	Orifice	Simulated Fuel Bundles		(Exact replica)		
	2/10	5/10	Full	2/10	5/10	Full
Scaling Practicability						
Nominal	F+	G	E	P	F	E
Tolerances	F*	G*	E*	P	F	E
Surface Roughness	E*	E*	E*	G	G	E
Pumping Requirements	E	G	P	E	G	P
Instrumenting Requirements	P	G	E	P	G	E
Physical Size	E	F	P	E	F	P
Pressurization Requirements	E	E	E	P	G	E
Dynamic Similarity**						
Flow Distribution	G	E	E	G	E	E
Pressure Distribution	G	E	E	G	E	E
Gravitational Effects	F	F	E	F	F	E
Friction Effects	G	E	E	F	G	E
Mixing-Bypass & Plenum	G	E	E	G	E	E
Outer Reflector & Seepage	F	G	E	F	G	E
Flow Transients	P	P	P	F	G	E

+ E: No undue difficulty, G: Satisfactory, F: Questionable, P: Poor

* Secondary importance

** Assumes that geometric similarity is exact

temperature at 140 °F. The implication is, if similarity is satisfactory at full scale and 140 °F, then it may be more economical to build a 0.5 scale model to accommodate 280 °F water which would require only 25% of full scale flow.

Tables I and II show that 0.5 scale is a reasonable scaling compromise if unsteady flow testing is anticipated. At this scale, physical size and flow requirements are significantly reduced: yet, the size is large enough for practicable fabrication and instrumentation, and appears to satisfy all dynamic similarity requirements adequately. Model flexibility is retained in that the model size does not negate the possibility of scaling small prototypic components such as the fuel bundles. Thus, a 0.5 scale hydraulic model should suffice for all anticipated tests involving mixing, pressure and flow distribution. This is not to imply that 0.5 is an optimum scale; rather, smaller model scales may detract model flexibility and accuracy, whereas larger scales would be adequate from these viewpoints.

If unsteady flow testing is not anticipated, a model scale as small as 0.2 would suffice.

Hydraulic Abnormalities

Hydraulic abnormalities as defined in this study pertains to all hydraulic conditions other than normal flow transients and normal steady flow at or near a design flow level. These conditions may occur under either steady or transient reactor flow and are listed below

- Abnormal Steady Flow
 - A. At Design Flow
 - 1. Cavitation
 - 2. Gas Entrainment
 - B. Total Pumping Failure (immediately after initial flow transient)
 - 1. Natural Circulation

- Abnormal Transients
 - A. Hydraulic Hammer
 - B. Pumping Failure (partial or full)
 - C. Closed Loop Rupture
 - D. Vessel Rupture
 - E. Localized Flow Stoppage

Modeling requirements to study these conditions are discussed briefly below.

Cavitation

Cavitation, or formation of vapor pockets, is a phenomena which occurs in a hydraulic system when the fluid static pressure falls to the fluid vapor pressure. The cavitation number, C_n , is stated symbolically as

$$C_n = \frac{P_s - P_v}{\rho V^2}, \quad (32)$$

where P_v and P_s are the fluid vapor pressure and fluid static pressure, respectively. Since cavitation is a localized phenomena, the flow in model and prototype must be dynamically similar in every respect (at least in the region of interest) to correctly simulate cavitation. This means that the trajectories of and forces on all homologous fluid elements must be similar in model and prototype. Since cavitation is a function of the fluid vapor pressure, total system pressure is of primary importance.

For an ideal model, the Reynolds number ratio is unity and when combined with the cavitation number, the following relationship for the vapor pressure disparity ($P_s - P_v$) ratio may be formed.

$$\left(\frac{P_s - P_v}{P_s - P_v} \right)_m = \Delta P_{v_r} = \frac{v_r^2}{L_r \rho_r} \quad (33)$$

Also, for a model with $Eu_r = 1.0$ cavitation similarity is given by

$$\left(\frac{\Delta P_v}{\rho V^2} \right)_r = \left(\frac{\Delta P}{\rho V^2} \right)_r \quad (34)$$

Hence, $\Delta P_{v_r} = 1.0$ when $\Delta P_r = 1.0$ and in this case

$$P_{sm} - P_{vm} = P_{sp} - P_{vp} \quad ,$$

or

$$P_{sm} - P_{sp} = P_{vm} - P_{vp} \quad (35)$$

Equation (35) states that cavitation will be similar in prototype and model, if the model pressure scale is translated linearly according to the disparity in vapor pressure between the prototype and model fluid. This should not be too difficult to do in the case of water modeling sodium. For example, for 200 °F water modeling ($P_{vm} \approx 11.53$ psia) 1000 °F sodium ($P_{vp} \approx 0.156$ psia), the vapor pressure difference is about 11.37 psi which means that the model total pressure should be increased by this amount. For sodium at 1200 °F, $P_{vp} \approx 9.6$ and in this case the model pressure increase would need to be about 1.9 psi.

Natural Convection

Although natural convection (or natural circulation) occurs in most all flow systems where thermally induced density variations are present, it will be significant in the FFTF hydraulic core only at very low flows or during a complete pumping failure. The reason for modeling natural circulation in the HCM is to provide experimental data from which to assess the heat removal capability by natural convection during a pumping failure.

Equality of the Grashof number in model and prototype establishes dynamic similarity provided there is also geometric similarity. The Grashof number ratio is given by

$$Gr_r = \frac{g_c \beta_r \Delta T_r L_r^3}{\nu_r^2} \quad (36)$$

In the above equation β is the volumetric expansivity (evaluated at the inlet temperature), and ΔT is the characteristic temperature difference which in the case of a vertical fuel assembly may be taken as

$$\Delta T = T_{\text{wall}} - T_{\text{inlet}} \quad (37)$$

In addition, pressure losses and flow effects resulting from the reactor system piping, etc. should also be modeled. It probably will not be necessary to model the entire core so that these tests could be done in the segmented thermalmechanical model. Fuel assembly geometry and temperature difference distribution needs to be modeled with reasonable accuracy. Figure 7 illustrates for a full scale model the mean temperature difference ratio versus inlet model water temperature for 900 °F inlet sodium, mean prototype $\Delta T = 150$ °F, Grashof number ratio = 1.0.

Gas Entrainment

It is possible that cover gas may become entrained in the FFTF coolant system through vortex formation in the sodium pool. Modeling requirements to permit observation of occurrence of this phenomena involves similarity of system geometry and Froude and Euler numbers, or essentially the same requirements as for modeling surface waves and jetting.

Abnormal Transients

Except for the abnormal transients due to hydraulic hammer, the modeling of pumping failure, rupture, flow stoppages, etc., is essentially the same as the modeling of normal flow transients.

FIGURE 7. Temperature Difference Ratio Versus Core Inlet Model Water Temperature
(Full scale model)

Differences are present, however, because the abnormal transients referred to here will normally cover a greater range of flows, thus requiring a greater degree of dynamic similarity. For instance, modeling with a nonideal Reynolds number (say 0.5) may result in vastly different flow regimes in model and prototype - especially at low prototypic flows. As an example, the Reynolds number at the beginning of the transient may be 500,000 in the model (or dynamically similar); but at the low end of the transient the prototypic Reynolds number may be 4000 and 2000 in the model in which case energy losses are not at all similar.

In addition to flow similarity, there must be similarity of structural materials and pressure wave propagation when modeling hydraulic hammer. The subject of water hammer is covered thoroughly in Reference 12, and will not be discussed further in this document.

MECHANICAL AND MECHANICS MODELING

As pointed out previously, the objective of the mechanics model is to provide an experimental tool to approximate the following for the Prototypic FFTF: mechanical fit, mechanical stresses and deflection, and flow induced vibrations. In view of these requirements, similarity Equations (1) through (5) must be considered. As noted in the model criteria, we cannot expect to model all quantities given by these functionals with a single model; therefore, certain approximations and dissimilarities must be accepted in order to carry out the model objectives.

Model Fluid

To adequately model thermally induced system stresses and distortions, thermal expansion of all system components must be modeled. In this case

$$(\alpha \Delta T)_m = (\alpha \Delta T)_p \quad . \quad (38)$$

Thermal expansion is both a function of materials and temperature distribution. Theoretically at least, we could model with $\Delta T_r < 1.0$ by selecting the correct materials such that

$$\Delta T_r = \frac{1}{\alpha_r} . \quad (39)$$

For components consisting of one or two materials, satisfying Equation (38) would probably not be too difficult, but if more materials were involved this might be impossible.

When modeling with water, practicable model temperatures are anywhere from ambient up to perhaps $300^+ \text{ }^\circ\text{F}$, (but, preferably not over 180-190 $^\circ\text{F}$). Suppose water is used for the HCM model fluid and that the temperature limit is 300 $^\circ\text{F}$. Prototypic sodium is entering the core at 900 $^\circ\text{F}$, exiting at 1200 $^\circ\text{F}$ is to be simulated. In this case $\Delta T_r = 1/4$, the model entrance temperature would be 225 $^\circ\text{F}$ and the temperature rise would be 75 $^\circ\text{F}$. However, since $\alpha_r = 4.0$, it may be impossible to match all materials correctly.

In the discussion above, only one similarity parameter is satisfied for only one similarity functional. To gain any kind of a reliable approximation of thermally and hydraulically induced stresses and distortions, external and internal loads and moments must also be correctly superimposed at the same time. An additional materials property, Young's modulus, must be compatible with the model scale and loads according to Equation (3), which may be impossible in view of the requirements on α_r .

The FFTF core will be a complex hydraulic and mechanical system. It is not reasonable to believe that a reliable model can be constructed to evaluate mechanical fit, distortion and stress without using prototypic materials and temperature distribution. In this case, only modeling of hydraulic and mechanical loads need to be contended with. This is still a difficult job in view of complex system interactions, but it is not impossible.

It is suggested that the thermalmechanical model be designed to accommodate both inert gas and water model fluid. With this capability the prototypic temperature distribution can be effected to accommodate the necessary high temperature testing. Further, tests that are not as sensitive to temperature distribution, such as core vibrations and various hydraulic tests, can be carried out at low temperature using water.

Model Scale

For steady state thermal stress evaluation, a scale model would suffice, but careful scaling of static loads and moments would be required. Scaling the loads and moments that occur from normal assembly, body forces, fluid flow, etc., is certainly possible from a theoretical view; but from a practical standpoint, matching of these quantities might prove difficult or inadequate and result in erroneous data. A full scale model would eliminate a great deal of difficulty and uncertainty in this particular case.

Tolerances and dimensional effects have paramount importance in fast reactor nuclear stability. Thus, it is essential that tolerances, deflections, and thermal expansion be correctly modeled in the thermalmechanical test facility. Again, in principle, it is possible to scale these effects, but by the nature of prototypic requirements, tolerances in general will be "tight"; scaling would be difficult and possibly infeasible. Dimensional changes in the scaled model would be difficult to measure accurately, and lead to an erroneous assessment of these thermal effects.

Since correct assessment of tolerances, static and thermal stresses, deflections and expansions is extremely important to the FFTF core design, use of a model scale less than 1.0 is deemed wholly unacceptable. Also, a full (360°) core mockup is not needed; only a convenient radial segment of mechanical symmetry is necessary.

Vibrations

Modeling fluid-elastic forces, such as flow induced vibrations, is extremely difficult to carry out quantitatively without using a near prototypic model. Any simplification of structural details, idealization of fluid dynamics, changes in wall effects, changes in inertia, changes in damping and mechanical properties will normally alter the dynamic response of the structure. For ideal modeling of vibrations, the following conditions for similarity must be considered.

$$\left(\sqrt{\frac{EI}{\rho_{\text{eff}}}} \frac{\tau}{L^3} \right)_r = 1.0 \quad , \quad (40)$$

$$\left(\frac{\rho_{\text{virt}}}{\rho_{\text{eff}}} \right)_r = 1.0 \quad , \quad (41)$$

$$\left(\frac{FL^2}{EI} \right)_r = 1.0 \quad , \quad (42)$$

$$\left(\sqrt{\frac{\rho_{\text{virt}}}{EI}} VL^2 \right)_r = 1.0 \quad , \quad (43)$$

$$\left(\frac{\omega F}{kL} \right)_r = 1.0 \quad , \quad (44)$$

$$\left(L \sqrt{\frac{\rho}{E}} \right)_r = 1.0 \quad , \quad (45)$$

$$\left(\frac{\eta}{L^2 \sqrt{\rho E}} \right)_r = 1.0 \quad , \quad (46)$$

$$\left(\omega L/V \right)_r = 1.0 \quad . \quad (47)$$

Thus, there may be as many as eight similarity parameters to satisfy in a vibrations model (assuming that fluid flow is kinematically similar). Ideal Modeling is not attainable in this case.

Modeling Vibrations With Gas Model Fluid

The major advantage of modeling vibrations with a gas instead of water is that prototypic temperatures and temperature distributions can be simulated. Therefore, if prototypic structural materials are used, the materials properties, as they enter into Equations (38) through (46), will be ideal.

On the other hand, modeling vibrations with gas has many disadvantages that far outweigh the advantage of ideal materials properties simulation.

- Inertial effects are badly distorted because of the difference between the mass of gas coolant and sodium.
- Fluid dynamic forces, although perhaps ideal with respect to the Euler number, are only a small fraction of prototypic.
- The dynamic viscosity of gases is vastly different from sodium, which tends to distort the fluid damping effect.
- Fluid velocities are usually greatly different in model and prototype.

Frequencies in gas and sodium will be quite different because of the virtual mass effect. It is doubtful that the fluid-elastic operators could be modeled without radically changing the structural stiffness.

From the above considerations, it does not seem reasonable that quantitative vibration data could be gained from gas modeling--even at full scale using prototypic structural components. This does not, however, negate the possibility of obtaining meaningful and useful qualitative information. In a gas cooled thermalmechanical model, gross interactions between system components could be observed. At least, likely trouble areas might be located, thus pointing out where structural support is needed or requires improvement. Also, by modeling the fuel subassemblies in both air and water, references can be established for measuring the effects of fluid viscosity on damping, virtual mass, etc.

Modeling Vibrations With Water Model Fluid

Water has many advantages over gases for the modeling of structural vibrations in sodium flow.

- The densities of sodium and water, over the temperature ranges of interest, are quite similar; hence, virtual and real system masses would be quite similar.
- There are practicable model water temperatures where the linear damping coefficient, η , is close to sodium.
- Fluid dynamic forces can be matched near ideal, if need be.
- Matching model and prototypic velocities is practical.

Although, all similarity requirements cannot be modeled ideally with any combination of sodium and water temperatures, conditions can be chosen such that no parameter will be greatly in error; hence, an adequate quantitative model can be defined.

The only disadvantage in modeling FFTF structural vibrations in a water facility is that prototypic temperatures cannot be achieved, which is reflected in distorted materials properties. Evidently the only temperature dependent properties of importance are Young's modulus, E , and thermal expansion. Corrections for E can be made analytically, but different thermally induced deflections and expansions might lead to different component support conditions.

In the following discussions, it is assumed that suitable structural materials can be used such that real and virtual density ratios and the Young's modulus ratio are suitably close to 1.0. In this case, Equations (40) through (46) become (for true geometric scaling)

$$(\tau/L)_r = 1.0 \quad , \quad (40a)$$

$$(F/L^2)_r = 1.0 \quad , \quad (42a)$$

$$V_r = 1.0 \quad , \quad (43a)$$

$$(F/kL)_r = 1.0 \quad , \quad (44a)$$

$$(\omega L)_r = 1.0 \quad , \quad (45a)$$

$$(\eta/L^2)_r = 1.0 \quad . \quad (46a)$$

Equation (41) is assumed to be satisfied per the previous discussion.

From a standpoint of satisfying similarity parameters, $L_r = 1.0$ is the best choice of model scales. In the case of full scale, the vibrational period is true ($\tau_m = \tau_p$), Equations (42a) and (44a) require fluid dynamic forces to be reasonably prototypic, and Equations (43a) and (47) insist that velocities in model and prototype be reasonably similar. Equation (45a) is automatically satisfied and (46a) then depends primarily on fluid properties (since, $\eta_r = L_r \mu_r$).

In spite of the many restrictions cited above, reasonably accurate vibrations modeling can be carried out, at least in the case of water modeling sodium at full scale. The reason for this is that fluid flow forces and velocities can be modeled with reasonable accuracy in this case. Consider the case where model and prototypic velocities are matched ($V_r = 1.0$). Then for fluid dynamics

$$\Delta P_r = \rho_r \quad , \quad (48)$$

$$Re_r = \frac{1}{\nu_r} \quad . \quad (49)$$

For vibrations,

$$\tau_r = 1.0 \quad , \quad (50)$$

$$\nu_r = 1.0 \quad , \quad (51)$$

$$\eta_r = \mu_r \quad . \quad (52)$$

The resulting ratios of fluid dynamic forces are given by Equation (48).

The results of matching velocities for 1000 °F sodium are illustrated in Figure 8 which shows that a reasonably good job of modeling can be done with water temperatures in the range of 220 to 260 °F (since η_r and Re_r are reasonably close to 1.0). In any case, where $V_r = 1.0$, the fluid dynamic forces are going to be a little high, which, for instance, would tend to distort the vibrational amplitudes.

If, instead of setting $V_r = 1.0$, the pressure ratio is set to 1.0, then for the fluid dynamics

$$V_r = \frac{1}{\sqrt{\rho_r}} \quad , \quad (53)$$

$$Re_r = \frac{1}{\sqrt{\rho_r} v_r} \quad . \quad (54)$$

The velocity and Reynolds number distortion is shown in Figure 9 for water modeling 1000 °F sodium. This figure shows that the velocity ratio in the water temperature range from 220 to 260 °F is distorted by less than 7%. This may be a less dramatic error than the possible 16% error in fluid forces indicated by Figure 9, when velocities are matched.

Modeling the flow induced system structural dynamics becomes decidedly more complicated and difficult when $L_r = 1.0$. No attempt is made in this study to point out specifically what needs to be done to obtain reliable vibrational data with a scaled down HCM model; a more detailed and definite description of the system is needed to accomplish this. However, in general, the first job would be to establish the relative influence of each similarity parameter on the structural dynamic response and to discern the implications and consequences of certain distortions.

Vibrations Modeling Feasibility

Table III summarizes the feasibility of modeling structural vibration in the HCM.

FIGURE 8. Pressure Drop Ratio, Reynolds Number Ratio, and Linear Damping Ratio Versus Model Water Temperature

FIGURE 9. Velocity Ratio and Reynolds Number Ratio Versus Model Water Temperature

TABLE III. *Vibrations Modeling Feasibility*

	<u>Scale</u>	<u>Model Fluid</u>	<u>Quantitative</u>	<u>Qualitative</u>
Single Subassembly	Full	Water	Excellent	Excellent
Full or Partial Core	Full	Water	Excellent	Excellent
Full or Partial Core	Full	Gas	Poor	Good
Full or Partial Core	1/2	Water	Fair	Good
Full or Partial Core	1/5	Water	Poor	Poor

NOTE: *All models are assumed to be true geometric models and feasibility includes fabrication, data acquisition, and vibrational and hydraulic similitude.*

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APPENDIX A
FLUID FLOW MODEL THEORY

APPENDIX A

FLUID FLOW MODEL THEORY

A complete physical equation is a statement of relationship between all physical quantities that govern a particular phenomenon. Symbolically this relationship may be expressed by a functional

$$\Phi (X_1, X_2, X_3 \dots X_k) = 0 \quad , \quad (A-1)$$

where the X's are the pertinent quantities. By the Buckingham Pi theorem, it may be shown that the particular phenomenon may be expressed equivalently by a functional

$$\Psi (\pi_1, \pi_2, \pi_3 \dots \pi_i) = 0 \quad , \quad i < k \quad (A-2)$$

where the π 's are dimensionless quantities formed from the X's. Further, two systems that are described by the same functionals, Ψ or Φ are said to be similar if the corresponding π 's have the same value in both systems, even though the values of the X's do not correspond. The π 's are called similarity parameters. For the similarity parameters to remain unchanged between a model and prototype, certain restrictions concerning geometrics, kinematics, and dynamics are required.

GEOMETRIC SIMILARITY

In fluid flow models, solid boundaries that influence the flow must be geometrically similar to the prototype. Geometric similarity is characterized by a single and uniform scale ratio between prototype and model. Thus,

$$\frac{L_m}{L_p} = L_r \quad , \quad (A-3a)$$

$$\frac{A_m}{A_p} = L_r^2 \quad , \quad (A-3b)$$

$$\frac{V_m}{V_p} = L_r^3 \quad , \quad (A-3c)$$

where L , A , and V refer to length, area, and volume, respectively; the subscripts m and p refer to the model and prototype, respectively.

KINEMATIC SIMILARITY

To establish kinematic similarity not only requires introduction of a time scale, but also requires that paths of all homologous fluid elements are geometrically similar and have the same velocity and acceleration ratios. Thus,

$$\frac{V_m}{V_p} = \frac{L_r}{\tau_r} \quad , \quad (A-4a)$$

$$\frac{a_m}{a_p} = \frac{L_r}{\tau_r^2} \quad , \quad (A-4b)$$

$$\frac{Q_m}{Q_p} = \frac{L_r^3}{\tau_r} \quad , \quad (A-4c)$$

where V , a , Q , and τ are velocity, acceleration, volumetric flow rate, and time, respectively. This simply means that flow patterns are geometrically similar, or velocity and acceleration vectors at each corresponding point in the model and prototypic fluid have similar direction and magnitude.

DYNAMIC SIMILARITY

Dynamic similarity requires, in addition to geometric and kinematic similarity, that all homologous forces have the same ratio between prototype and model. Since forces are vector quantities, they must also be similar in a vectorial sense. From Newton's Second Law:

$$\frac{(\sum \vec{F}_k)_m}{(\sum \vec{F}_k)_p} = \frac{M_m}{M_p} \frac{(d\vec{V}/d\tau)_m}{(d\vec{V}/d\tau)_p} \quad , \quad (A-5)$$

where M is mass, and the $\vec{F}_k$ are the various forces acting on a fluid element. These forces may result from pressure effects,

$\vec{F}_p$, gravitational effects, $\vec{F}_g$, elastic effects, $\vec{F}_e$, viscous effects, $\vec{F}_f$, or surface tension effects, $\vec{F}_\sigma$, to name a few.

DEVELOPMENT OF THE SIMILARITY PARAMETERS FOR STEADY FLUID FLOW

The similarity parameters may be developed directly from the Pi theorem; however, here the development will be presented from a less abstract point of view. From D'Alembert's principle, Newton's Second Law may be considered in a slightly different light

$$\sum \vec{F}_k + \left(-M \frac{d\vec{V}}{d\tau} \right) = 0 \quad , \quad (A-6)$$

where $-M(d\vec{V}/d\tau)$ is treated as a force, namely the inertial force, $\vec{F}_i$, on the fluid element. Thus

$$\sum_k \vec{F}_k - \vec{F}_i = 0 \quad . \quad (A-7)$$

A force polygon may then be drawn for a fluid element as shown below.

The conditions for dynamic similarity are satisfied if a uniform scale of enlargement or contraction is the only difference between the force polygons for the prototype and model, provided the conditions of kinematic and geometric similarity are also satisfied. Therefore, the following ratios may be formed.

$$\frac{(F_i)_m}{(F_i)_p} = \frac{(F_g)_m}{(F_g)_p} = \frac{(F_p)_m}{(F_p)_p} = \frac{(F_\sigma)_m}{(F_\sigma)_p} = \frac{(F_e)_m}{(F_e)_p} = \frac{(F_f)_m}{(F_f)_p} = F_r, \quad (A-8)$$

and

$$\frac{(F_i)_m}{(F_i)_p} = \frac{(\vec{F}_g + \vec{F}_p + F_\sigma + F_e + \vec{F}_f)_m}{(\vec{F}_g + \vec{F}_p + F_\sigma + F_e + \vec{F}_f)_p} \quad (A-9)$$

The above additions are carried out in a vectorial sense. If the inertial force ratio is equated to each of the various other force ratios independently,

$$\left(\frac{F_i}{F_g}\right)_m = \left(\frac{F_i}{F_g}\right)_p \quad \text{inertial-body force ratio,}$$

$$\left(\frac{F_i}{F_p}\right)_m = \left(\frac{F_i}{F_p}\right)_p \quad \text{inertial-pressure force ratio,}$$

$$\left(\frac{F_i}{F_\sigma}\right)_m = \left(\frac{F_i}{F_\sigma}\right)_p \quad \text{inertial-surface tension force ratio,}$$

$$\left(\frac{F_i}{F_e}\right)_m = \left(\frac{F_i}{F_e}\right)_p \quad \text{inertial-elastic force ratio,}$$

$$\left(\frac{F_i}{F_f}\right)_m = \left(\frac{F_i}{F_f}\right)_p \quad \text{inertial-viscous force ratio.}$$

The ratios may be developed further with the following relationships:

$$F_i \sim \rho L^2 V^2, \quad ,$$

$$F_g \sim \gamma L^3, \quad ,$$

$$F_p \sim \Delta P L^2, \quad ,$$

$$F_f \sim \mu V L, \quad ,$$

$$F_{\sigma} \sim \sigma L \quad ,$$

$$F_e \sim EL^2 \quad ,$$

where ρ is density, γ specific weight, μ is viscosity, σ is surface tension, and E is the modulus of elasticity. Therefore,

$$\frac{F_i}{F_g} = \frac{V^2}{Lg} \quad ; \quad Fr = \frac{V}{\sqrt{Lg_c}} \quad , \quad (\text{Froude Number})$$

$$\frac{F_i}{F_p} = \frac{\rho V^2}{\Delta p} = EU \quad , \quad (\text{Euler Number, or pressure coefficient})$$

$$\frac{F_i}{F_{\sigma}} = \frac{\rho LV^2}{\sigma} = W \quad , \quad (\text{Weber Number})$$

$$\frac{F_i}{F_e} = \frac{\rho V^2}{E} \quad ; \quad M = \frac{V}{\sqrt{E/\rho}} \quad , \quad (\text{Mach Number})$$

$$\frac{F_i}{F_f} = \frac{\rho VL}{\mu} = Re \quad . \quad (\text{Reynolds Number})$$

Similarity Parameters

The first step in flow model analysis is to determine which forces dominate or which similarity parameters are necessary to adequately describe the prototypic flow. Generally, steady flow phenomena can be characterized by the similarity parameters derived previously. However, other parameters such as the Grashof and cavitation numbers may be of interest in particular applications.

$$\underline{\text{Froude Number} - V/\sqrt{Lg_c}}$$

The Froude number is usually associated with a free liquid surface. In particular it characterizes gravity waves which associates interactions between gravitational and inertial forces. Application of Froude number similarity will depend, to a certain degree, on the final FFTF concept. For instance, in the split conical core concept, there may be only weak hydraulic

communication between the sodium pool free surface and the reactor core; hence, the pool surface configuration may have negligible effect on the core hydraulics. On the other hand, if a type of hot-cell concept is adapted where the core coolant is discharged directly to a free surface sodium pool, gravity waves and jetting may significantly affect the core hydraulics and thus implicate Froude number similarity in the model design.

Reynolds Number - $\rho VL/\mu$

The Reynolds number associates viscous and inertial forces. In flow through closed passages, it influences the viscous drag or frictional forces. Model design of the FFTF hydraulic core will definitely be influenced by Reynolds number similarity. Although this similarity parameter may not be used per se in every instance, it finds importance in establishing the relative influence of viscous effects on the hydraulic behavior.

Euler Number - $\rho V^2/\Delta p$

The Euler number (or pressure coefficient) relates pressure and inertial forces. It finds importance in modeling momentum energy losses such as shock losses from orificing, sudden enlargements and contractions, and highly turbulent frictional flow.

Grashof Number - $\rho^2 g_c \beta \Delta T L^3 / \mu^2$

In the above equation β is the coefficient of thermal expansion of the fluid and ΔT is a characteristic temperature difference. Gravity not only influences the shape of a free surface, which is described by the Froude number, but it also promotes free circulation when a temperature difference is present, because of density variation. Free convection is present in all flow systems with a temperature gradient, but is usually dominated by other effects. However, when flow ceases (through pump failure, perhaps) natural, or free circulation may become the principal mode of fluid transport.

$$\text{Cavitation} - (P_S - P_V) / \rho V^2$$

Cavitation is the formation of vapor pockets in the flow stream. If the model is designed to reveal the possibility of cavitation, the thermodynamic property vapor pressure, P_V , must be included in the analysis.

Geometric Similarity

For a true model all dimensions have a unique scale. This implies that surface roughness and fabrication tolerances as well as nominal dimensions, must also be similar in model and prototype.

Governing Similarity Parameters

There are various other similarity parameters which may enter in modeling similitude. Some of these are Mach number, which enters in with compressibility considerations; Weber number, which relates the force of surface tension to inertia force; and others which may result from centrifugal, Coriolis and elastic forces. All of these considerations will be of negligible importance in the FFTF Hydraulic Core Mockup.

Pressure-inertial and viscous-inertial forces will undoubtedly dominate the steady flow characteristics in the prototypic FFTF hydraulic core; thus, involving Euler number; (and, perhaps, Froude number). The effects of deviating from geometric similarity by way of surface roughness, fabrication tolerances, and model distortion must also be considered.

IDEAL MODELING IN STEADY FLOW

An ideal or true model demands that the conditions for geometric similarity be unconditionally satisfied. This means that a uniform scale not only applies to nominal dimensions, but surface roughness and fabrication tolerances as well. The flow dynamics require that Euler number and Reynolds number similarity be satisfied simultaneously, or:

$$\frac{\rho_r V_r^2}{\Delta P_r} = 1 = \frac{V_r L_r}{\mu_r} \quad . \quad (A-10)$$

Thus, the following expression can be formulated to relate the necessary conditions for ideal dynamical modeling.

$$\frac{V_r \sqrt{\rho_r}}{\sqrt{\Delta P_r}} = \frac{V_r L_r}{\mu_r} \quad , \quad (A-11a)$$

so that

$$L_r = \frac{\mu_r \sqrt{\rho_r}}{\sqrt{\Delta P_r}} \quad . \quad (A-11b)$$

By the preceding equation, model and prototypic fluid properties, and the pressure ratio dictate the required scale.

Figures A1 through A3 illustrate the required scale ratio for ideal modeling as a function of model water temperature for various sodium temperatures, and for three different pressure ratios, (properties of water and sodium may be found in Appendix E.) Figure A4 illustrates pressure drop ratio as a function of model water temperature at various sodium temperatures for ideal modeling at full scale. These figures illustrate the impossibility of satisfying both similarity parameters without first making a judicious choice of model fluid temperature and pressure ratio.

To avoid excessive model pressure, one might first consider $\Delta P_r = 1$, or less. Mean sodium temperatures in the FFTF under some operating conditions are expected to be above 800 °F. To satisfy both similarity criteria in the case of $\Delta P_r = 1$, the model fluid temperatures must be at least 215 °F for a scale ratio of 1.0 and even hotter for smaller scale ratios. For sodium at 1200 °F, model fluid temperatures of 260 °F and greater would be required for scale ratios of 1.0 and less. If the pressure ratio is increased to 2.0 then the required model water temperature would decrease to about 157 °F and 285 °F for model scales of 1.0 and 0.5, respectively. Modeling with water at high temperature also has the complication in that the total pressure must be high enough to suppress boiling (or cavitation which is not representative of the prototypic flow).

FIGURE A-1. Required Scale Ratio Versus Model Water Temperature
for Ideal Modeling
Pressure Ratio = 1.0

FIGURE A-2. Required Scale Ratio Versus Model Water Temperature for Ideal Modeling
Pressure Ratio = 1.5

FIGURE A-3. Required Scale Ratio vs Model Water Temperature
for Ideal Modeling
Pressure Ratio = 2.0

FIGURE A4. Pressure Ratio, ΔP_r Versus Model Water Temperature for Ideal Modeling at Full Scale

APPENDIX B
NONIDEAL STEADY-FLOW MODELING

APPENDIX B

NONIDEAL STEADY-FLOW MODELING

If it is established that true similarity cannot be effected, or is impractical from a fabrication standpoint, ideas of nonideal modeling must be considered. In all probability, exact similarity will not be necessary since it is really necessary only to provide an adequate description (as opposed to an exact description) of the prototypic flow. Of course, the definition of an adequate model is not always apparent, especially for a complex prototype, and really constitutes the major problem of model design.

Nonideal modeling, as defined in this study, embraces all modeling techniques that are designed to yield acceptable results, without insisting that all conditions for geometric, kinematic and dynamic similarity are satisfied. These methods may include dissimilar Reynolds numbers, pressure coefficients, and distorted or dissimilar geometry.

PRELIMINARY CONSIDERATIONS

A hydraulic system such as the FFTF hydraulic core consists of a number of series and parallel flow channels. Assuming that the flow originates and discharges to constant pressure plenums outside the hydraulic core, then the overall pressure drop through the system is the summation of the series pressure drops, regardless of the continuous path chosen. Along one path the fluid will encounter a variety of flow channel shapes and lengths, and will undergo a variety of momentum and viscous energy losses. Symbolically, pressure loss through the entire system may be expressed as

$$\Delta P_T = \sum_i \Delta P_{ij} \quad , \quad (B-1)$$

where ΔP_{ij} is the i th pressure loss along the j th continuous path. Each of the ΔP_{ij} may be expressed in the following form

$$\Delta P_{ij} = C_{ij} K_{ij}(\ell/L, \epsilon/L, Re) \rho_{ij} \left(\frac{Q}{L^2} \right)_{ij}^2, \quad (B-2)$$

where the K_{ij} 's are functionals that may depend on the flow channel length, ℓ ; characteristic length, L ; roughness, ϵ ; and Reynolds number. Q_{ij} is the volumetric flow at the particular location (i,j) . The C_{ij} 's are appropriate constants. The above equation is equivalent to

$$\frac{\Delta P}{\rho V^2} = C f(\ell/L, \epsilon/L, Re), \quad (B-3)$$

applied to each point in the system. As discussed previously, for an ideal model

$$(\rho V^2 / \Delta P)_Y = 1;$$

hence,

$$\left(\frac{PV^2}{\Delta P} \right)_r = f(\ell/L, \epsilon/L, Re)_r = 1, \quad (B-4)$$

which relieves the restriction on the Reynolds number somewhat, by posing the possibility of distorting ℓ/L , or perhaps ϵ/L .

In any case, it is not apparent that Equation (B-4) can be satisfied for all points in the HCM. Thus, one may choose to model either the Euler number or Reynolds number, or a compromise between the two. In general, if the pressure coefficient is modeled, then viscous or Reynolds number dependent energy losses will not correspond to those in the prototype. On the other hand, if the Reynolds number is modeled, then the pressure loss and flow rate might be excessive. Energy loss for the entire system, either prototype or model, may in general be expressed by the equation

$$\Delta P = C f(Re) \rho \left(\frac{Q}{L^2} \right)^2, \quad (B-5)$$

or for a ratio between a geometrically similar model and prototype,

$$\Delta P_r = f(Re)_r \rho_r \left(\frac{Q}{L^2} \right)_r^2 \quad (B-6)$$

Here $f(Re)$ is the net effect of all the K_{ij} 's defined in Equation (B-2). Solving for the flow ratio Q_r gives

$$Q_r = \left[\frac{\Delta P_r}{f(Re)_r \rho_r} \right]^{1/2} L_r^2 \quad (B-7)$$

For a case where the flow is Reynolds number independent in both prototype and model, $f(Re)_r = 1$. Then,

$$Q_{ri} = \sqrt{\Delta P_r / \rho_r} L_r^2, \quad (B-8)$$

where Q_{ri} designates the flow ratio for momentum loss dominated systems.

In the case where pressure loss is entirely from effects of turbulent flow in smooth passages,

$$f(Re)_r \simeq Re_r^{-1/5} \quad (B-9)$$

Substitution in the flow equation yields,

$$Q_{rv} = \sqrt{\Delta P_r / \rho_r} Re_r^{1/10} L_r^2, \quad (B-10)$$

where Q_{rv} is the flow ratio in viscous force dominated systems. Another case, but of lesser interest in the FFTF, is complete laminar flow in both model and prototype. In this case

$$f(Re)_r = Re_r^{-1}, \quad (B-11)$$

thus,

$$Q_{rL} = \sqrt{\Delta P_r / \rho_r} Re_r^{1/2} L_r^2 \quad (B-12)$$

It is instructive to compare Equations (B-8) and (B-10), noting that the only difference is the factor $(Re_r)^{1/10}$ in the flow rate equation for complete viscous flow. In the FFTF hydraulic core mockup the flow would probably fall somewhere between the values given by Equations (B-8) and (B-10). Equation (B-10) points out that the flow ratio is a very weak function of the Reynolds number ratio. In fact, it implies that the flow ratio would not deviate a great deal from Equation (B-8) even if Re_r is appreciably different from 1.0. A plot of this deviation is shown in Figure B-1. This argument would, of course, lose its validity if there is a significant difference of flow regime between model and prototype or geometric similarity is not maintained.

By assuming values of Re_r less than, equal to, and greater than 1.0 in Equations (B-8) and (B-10), additional observations are made.

- If $Re_r < 1$, an appropriate upper bound on the maximum flow is given by Q_{ri} .
- If $Re_r = 1$, $Q_{rv} = Q_{ri}$ (ideal model).
- If $Re_r > 1$, an appropriate upper bound on the maximum flow is given by Q_{rv} .

In the above discussion it is pointed out that the system flow response and energy losses are only a weak function of the Reynolds number, at least over the range of validity of Equation (B-9), but that Reynolds number ratios less than one might lead to appreciably dissimilar model behavior because of a significant difference in the flow regimes. By way of explanation, consider the relationship between friction factor and Reynolds number shown in Figure B-2. Suppose the prototypic Reynolds number Re_p has a value corresponding to B in Figure B-2. If the surface roughness is 0.004 or greater, the flow will be independent of increasing Re . Thus, if the value of the model Reynolds number, Re_m , is greater than B, and the relative roughness remains

FIGURE B-1. Q_{rv}/Q_{ri} Versus Re_r

FIGURE B-2. Friction Factor Versus Reynolds Number

unchanged; then, the model flow will, in fact, also be independent of the Reynolds number. On the other hand, if the model Reynolds number is decreased to a value A; then, depending on the relative roughness, the flow may become dependent on Re. In some cases the value of Re might be decreased into the critical region, and results would be highly inaccurate and essentially useless. If the surfaces are smooth, then flow will be Re dependent in all cases. A similar discussion applies to flow through nozzles and orifices.

The above discussion of Reynolds number dependent flow behavior is not to imply that Reynolds number similarity is necessary for adequate modeling but is to point out the difficulties which may be encountered. As a general rule it is safe to conclude that if inertial forces dominate flow characteristics at one Re, they will also dominate at higher Re. If viscous forces dominate, they will also dominate at a lower Re.

MODEL ANALYSIS

The three variables or parameters that might be independently modeled in the HCM are

- Reynolds number, Re
- Velocity, V
- Pressure drop, Δp .

Modeling Δp or velocity would in this case involve Euler number similarity. If the Reynolds number is simulated so that frictional or viscous losses are modeled correctly, the velocity ratio is given by

$$V_r = \frac{v_r}{L_r} \quad . \quad (B-13)$$

For a 1/2 scale model, the model velocities might be more than twice as great as prototypic velocities, depending on model water temperature and prototypic sodium temperature.

The case of primary interest, as far as hydraulic modeling is concerned, is where Euler number similarity is approximated. Thus,

$$V_r = \sqrt{\Delta P_r / \rho_r} ; \quad (B-14)$$

if

$$\Delta P_r = 1 ,$$

then

$$V_r = 1/\sqrt{\rho_r} . \quad (B-15)$$

In this case the Reynolds number ratio is

$$Re_r = L_r / (\sqrt{\rho_r} V_r) . \quad (B-16)$$

If the equation for maximum flow rate when $Re_r < 1$, is substituted into the expression above,

$$Re_r = Q_r / (V_r L_r) . \quad (B-17)$$

Figures B-3 through B-9 illustrate the value of Q_r and Re_r as a function of the model scale for various values of ΔP_r , model water temperature and sodium temperature. Figures B-10 through B-15 show the values of Q_r and Re_r as a function of model water temperature for various values of ΔP_r , L_r and sodium temperature. Generally, Re_r will be appreciably less than 1.0 for small scales and lower water temperatures. For instance, in the case of $\Delta P_r = 1.0$, $T_{Na} = 1200$ °F, $T_W = 200$ °F, and $L_r = 0.5$, the Reynolds number ratio will be about 0.35. At first thought, there might be reason to doubt the validity of these results; but, with reference to Figure B-1, $Q_{rv}/Q_{ri} = 0.9$. If frictional losses contribute only a small percent of model pressure drop, then Q_r will be very close to correct, even though the Re_r is quite different from 1.0 (provided Re_p and Re_m are within the range of validity of Equation (B-9).

For some purposes there may be an advantage in modeling fluid velocities (e.g., when mechanical vibrations and stresses are to be modeled simultaneously with fluid flow). If Euler number

FIGURE B-3. Reynolds Number Ratio Versus Scale Ratio
Sodium Temperature = 600 °F

FIGURE B-4. Reynolds Number Ratio Versus Scale Ratio
Sodium Temperature = 700 °F

FIGURE B-5. Reynolds Number Ratio Versus Scale Ratio
Sodium Temperature = 800 °F

FIGURE B-6. Reynolds Number Ratio Versus Scale Ratio
Sodium Temperature = 900 °F

FIGURE B-7. Reynolds Number Ratio Versus Scale Ratio
Sodium Temperature = 1000 °F

FIGURE B-8. Reynolds Number Ratio Versus Scale Ratio
Sodium Temperature = 1100 °F

FIGURE B-9. Reynolds Number Ratio Versus Scale Ratio
Sodium Temperature = 1200 °F

FIGURE B-10. Reynolds Number Ratio, Re_r , and Volumetric Flow Ratio, Q_r , Versus Model Water Temperature; Scale Ratio = 0.5

FIGURE B-11. Reynolds Number Ratio, Re_r , and Volumetric Flow Ratio, Q_r , Versus Model Water Temperature; Scale Ratio = 0.6

FIGURE B-12. Reynolds Number Ratio, Re_r , and Volumetric Flow Ratio, Q_r , Versus Model Water Temperature; Scale Ratio = 0.7

FIGURE B-13. Reynolds Number Ratio, Re_r , and Volumetric Flow Ratio, Q_r , Versus Model Water Temperature; Scale Ratio = 0.8

FIGURE B-14. Reynolds Number Ratio, Re_r , and Volumetric Flow Ratio, Q_r , Versus Model Water Temperature; Scale Ratio = 0.9

FIGURE B-15. Reynolds Number Ratio, Re_r , and Volumetric Flow Ratio, Q_r , Versus Model Water Temperature; Scale Ratio = 1.0

similarity is maintained, then the pressure drop ratio would be given by

$$\Delta P_r = \rho_r \quad . \quad (B-18)$$

Figure B-16 illustrates the value of the pressure ratio as a function of model water temperature for various prototypic sodium temperatures when $V_r = 1.0$.

GEOMETRIC DISTORTION

As pointed out in the previous sections of this study, if a model is designed such that $Re_r \neq 1.0$, then viscous energy losses are not correctly modeled. If it is adjudged that these losses are a significant contribution to the prototypic pressure loss, a correction may be needed.

One such correction involves geometric distortion of the frictional flow path length. Viscous energy losses in channel flow may be expressed by

$$\Delta P = C \rho \ell / L f(Re) (Q/L^2)^2, \quad (B-19)$$

where ℓ is the channel length. Forming a ratio between prototype and model yields

$$\Delta P_r = \rho_r \ell_r / L_r f(Re)_r (Q/L^2)_r^2, \quad (B-20)$$

solving for the length ratio gives

$$\ell_r = \frac{\Delta P_r L_r^5}{\rho_r Q_r^2 f(Re)_r} \quad . \quad (B-21)$$

For turbulent flow in smooth channels with $\Delta P_r = 1$,

$$\ell_m = \frac{L_r^5 Re_r^{1/5}}{\rho_r Q_r^2} \ell_p \quad . \quad (B-22)$$

In the case of laminar flow

$$\ell_m = \frac{L_r^5 Re_r}{\rho_r Q_r^2} \ell_p \quad . \quad (B-23)$$

FIGURE B-16. Pressure Drop Ratio Versus Model Water Temperature for Constant Velocity

In the above equations ℓ_m and ℓ_p are channel lengths of model and prototype, respectively. The Reynolds number ratio and flow ratio are those predicted by Euler number similarity.

HYDRAULIC DISTORTION - DRIVER FUEL MODELING

Some components of the reactor core are difficult and expensive to model geometrically on a "scaled down" basis. If the interaction between these components and the overall system is not complex, modeling may be carried out on the components individually, exterior to the system on a full or greater scale basis. Once the hydraulic characteristics of the components have been established, a nonideal model may be substituted in the scaled down system model which will give an adequate account of the component with regard to its interaction on the system. Examples are the driver fuel and reflector tubes in the FFTF. Full scale driver fuel pins may be as small as 0.2 in. in diam. True, or ideal, modeling would require, e.g., on a 1/5 scale, model fuel pins of 0.04 in. diam. The problems here are in fabrication tolerances, which should be scaled proportionately; channel fouling should any dirt enter the system; and expense. In the FFTF, the pressure drop contribution from a given fuel bundle will be nearly independent of the rest of the system for a given system flow; thus, any device which will permit essentially the same flow as in the case of an ideal model, and effect the same pressure drop, can be substituted for the ideal model whether it "looks like" a fuel assembly or not. This non-ideal modeling scheme has been carried out successfully by a number of experimenters using orifices which also serve as flow metering devices.^(11,14) Pressure drop through a sharp edged orifice is given by

$$\Delta P = k(d_o/d_1, R_e) \frac{\rho V^2}{2g} \quad , \quad (B-24)$$

where k is an orifice coefficient which depends on Reynolds number, and the orifice diameter to pipe diameter ratio, d_o/d_1 .

Orifice size may be established from single component testing of an exact full scale model of the driver fuel assembly. The orifice must be calibrated to about the expected model flow and should be sized such that the energy loss is independent of Re .

For single component testing at full scale, it may be possible to approach ideal modeling conditions (i.e., $Re_r \simeq 1.0$, $Eu_r = 1.0$). In the case of the fuel assemblies, this would be an advantage because frictional and shock pressure losses are of about equal importance. Referring to Figure B-15, if the model water temperature is 200 °F or greater, the Reynolds number ratio would in all cases be greater than 0.7, when $\Delta P_r = 1.0$. Modeling sodium flow at 800 °F can be done ideally with model water at 215 °F.

If Reynolds numbers are not ideally modeled with $\Delta P_r = 1.0$, then the model might be run at a higher pressure drop to satisfy both similarity parameters. In lieu of operating at $\Delta P_r > 1.0$, one can also plot $\Delta P V_s Q$ for a particular model, and extrapolate, with a good deal of accuracy, to the point $(\Delta P_i, Q_i)$ which satisfies $Eu_r = Re_r = 1.0$.

MODELING SURFACE ROUGHNESS

The friction factor used in computing friction energy losses is not only a function of Reynolds number, but also the flow passage surface roughness. The roughness is usually specified as the dimensionless ratio of a roughness characteristic length to a flow passage characteristic length, or

$$\text{Relative roughness} = \epsilon / De$$

To insure that roughness is similar in model and prototype, the ϵ / De ratio should be kept constant. In fact, if $(\epsilon / De)_r \neq 1.0$, then a geometry distortion is implied which violates the

condition for geometric similitude. However, it should be noted that there are cases where $(\epsilon/De)_r$ can be appreciably different than 1.0 without significantly affecting flow similarity.

It is not exactly clear how the roughness length ϵ , may be determined in terms of measurable quantities, since it is a function of size, height, distribution, shape, etc., of the local protrusions. Values of ϵ for various commercial pipes and tubes have been well established through experiment, but machined surfaces cause some complication in estimating ϵ because the local dimensional variations depend on the cutting tool and direction of cut.

Actually, the problem of modeling roughness is not as difficult as it may seem at first thought. First, we may dismiss the problem for components in laminar flow, because the friction factor is independent of roughness in this flow regime. Second, surface roughness is of minor importance for components where shock pressure losses predominate (e.g., orifices, etc.), or where prototypic friction losses are slight (however, in the latter instance, the model surface should be reasonably smooth to insure that frictional effects do not become significant in the model). Thus, the only instance where surface conditions may require any special consideration is in the case of turbulent viscous flow when frictional energy losses are significant.

Prototypic components which cause appreciable frictional pressure loss will probably be reasonably smooth (e.g., driver fuel assemblies) and the Reynolds numbers will be in the neighborhood of 10^5 or less. Inspection of Figure B-1 reveals that the relative roughness may vary appreciably without seriously affecting the friction factor in this case. If prototypic surfaces can be classified as "smooth" then model surfaces should be "comparably smooth." If prototypic surfaces are other than smooth, then as a general rule, the model surface roughness

should be scaled according to the model scale, $(\epsilon/De)_r \approx 1.0$. It is suggested that the RMS surface roughness may be used as the measurable quantity in establishing an acceptable degree of geometric similarity, provided surface protrusions have the same general orientation.

APPENDIX C
THE EFFECT OF DIMENSIONAL TOLERANCES ON MODEL ACCURACY

APPENDIX C

THE EFFECT OF DIMENSIONAL TOLERANCES ON MODEL ACCURACY

In any hydraulic system, flow and pressure variation are induced by imperfections in component fabrication and assembly. Mechanical tolerances in the prototype are specified throughout the system so that the deviations from a mechanically exact or nominal system are within some known bound. The same ideas apply to scale modeling and, in general, it is necessary only to scale dimensional tolerances in accordance with the model scale to achieve adequate geometric similarity. That is to say, if the model accurately predicts the prototypic performance, then the bound on performance variation caused by dimensional tolerances will be similar in prototype and model.

In many cases tolerances and surface finish need not be scaled exactly, or, for that matter, scaled at all. These cases would include components where the model hydraulic characteristic lengths are large compared to the mechanical deviations and surface roughness effects in the case of small frictional losses or laminar flow.

On the other hand, scaling the surface roughness is extremely important in turbulent flow where prototypic friction losses are high, and scaling mechanical tolerances is a critical factor in model accuracy if characteristic lengths are small. For small scale models the tolerance scaling criterion may be difficult to meet. For instances in a 1/10 scale model, even loose prototypic tolerances may be difficult to scale correctly, and the tighter dimension may be economically infeasible or impossible to scale accurately.

The purpose here is to approximate the error in model performance under these conditions of geometric dissimilarity. Estimates are given for the error in model performance as a function of variation from a nominal model.

By a nominal model is meant a model which has exact dimensions or zero dimensional tolerance. Thus, the error is not presented as an expected error from the modeling process, but is presented as the difference in performance between a theoretical model and the actual model used.

EFFECTS ON THE OVERALL HYDRAULIC SYSTEM

Dimensional variations cause localized flow perturbations which are propagated throughout the hydraulic system in a quite complex manner. To what extent a local dimensional perturbation affects the remainder of the system depends on the hydraulic interaction between system components.

From Equation (B-1)

$$\Delta P_{ij} = C_{ij} K_{ij} (\ell/L, \epsilon/L, Re) \rho_{ij} \left(\frac{Q_{ij}}{L_{ij}^2} \right)^2, \quad (C-1)$$

from which the overall system pressure drop may be written as

$$\Delta P_T = \sum_i \Delta P_i = \sum_i C_{ij} K_{ij} (\ell/L, \epsilon/L, Re) \rho_{ij} \left(\frac{Q_{ij}}{L_{ij}^2} \right)^2, \quad (C-2)$$

where, again, j refers to a particular path through the system. The above relationship results in a set of simultaneous equations which, at least, in principle, can be solved, with a systems analysis approval but would require detailed knowledge of the system for adequate evaluation of the K_{ij} and L_{ij} . Without this knowledge, it is impossible to give an accurate account of what effect localized dimensional variations will have on the entire system.

Nonetheless, a quantitative bound can be established for the maximum flow error by assuming that the inexact model is also a scale model of the prototype. Since tolerances are

not necessarily related to model scale, the inexact model certainly cannot have a uniform scale. However, if the tolerances that have the greatest influence on flow response are used to determine a maximum scaling error factor, then a uniform scale factor can be applied throughout the system. Thus, Equation (B-8) may be used to approximate the flow error. In this case

$$\Delta P_r^* = \frac{Q_r^{*2}}{L_r^{*4}}, \quad (C-3)$$

where the superscript (*) pertains to the ratio between an inexact model and an exact model ($\Delta P^* = \Delta P_{\text{inexact}} / \Delta P_{\text{exact}}$). The density term has been dropped because in this case both model and prototypic fluids are at the same temperature. By Equation (B-10) the above equation implies a maximum flow error when $Re_r < 1.0$. If $\Delta P_r^* = 1$, then the flow error bound is given by

$$Q_r^* = L_r^{*2} = (1 \pm S)^2, \quad (C-4)$$

where $S = \text{tolerance/exact dimension}$. Figure C-1 shows the estimated model flow error as a function of S . As an example of how the above equation can be used, suppose the maximum tolerance on the model critical parts is 0.001 in. If the parts have a nominal characteristic dimension of 0.100 in., then $S = 0.01$. The overall model flow error would be approximately $\pm 3\%$ as a maximum. It should be kept in mind, however, that tolerance tend to be statistical and that Equation C-4 and Figure C-1 are at best a rough approximation.

EFFECT OF DIMENSIONAL VARIATIONS ON INDIVIDUAL COMPONENT

It was pointed out above, that estimating the effects of dimensional variation on the overall system could be, at best, a rough approximation of the maximum flow error. If an

FIGURE C-1. *Estimated Model Flow Error Versus Variation of Characteristic Length for Overall Hydraulic System or Component*

individual component consists of a number of parallel flow channels, such as an FFTF fuel bundle, the same is true because of the statistical nature of the controlling geometric variables. However, using the method mentioned previously, we can arrive at a similar type of quantitative error bound.

For individual components such as orifices, nozzles, and sudden enlargements and contractions, the pressure drop across, or flow through, the component is influenced geometrically by essentially one or two dimensions.

Momentum Losses

In the case of shock loss pressure drops

$$\Delta P_r^* = (Q_r^* / A_r^*)^2, \quad (C-5)$$

where A_r^* = inexact area/theoretical area. If tolerances were proportionate to length, Equation (C-3) could be applied; however, this is not normally the case. Generally speaking, deviation of dimensions will also cause flow coefficients to change unless a uniform scale is maintained. However, for small dimensional variations no appreciable change of flow coefficients will be noticed. For some cases the following approximation is valid.

Equivalent diameter is defined as

$$De = \frac{4A}{P_w}$$

then

$$\frac{De^*}{De} = \left(\frac{A}{A^*} \right) \left(\frac{P_w^*}{P_w} \right),$$

where P_w is the hydraulic perimeter.

- Narrow slits and annuli

In case of a narrow slit or annuli $P_w^* \simeq P_w$; thus,

$$\frac{A^*}{A} \simeq \frac{De^*}{De} \quad , \quad (C-6)$$

and for

$$\Delta P_r^* = 1 \quad , \quad (C-7)$$

$$Q_r^* = 1 + \frac{t}{De} \quad , \quad (C-7)$$

where t is the appropriate tolerance for the equivalent diameter.

In the case of an annulus with $D_o - D_i \ll 1/2 (D_o + D_i)$, then

$De \simeq D_o - D_i$. Therefore,

$$Q_r^* = 1 \pm \frac{t}{D_o - D_i} \quad . \quad (C-8)$$

In the case of a narrow slit $De \simeq 2W$, where W is the width of the slit. If 0.001 in. is the tolerance on the width of the channel, then $t = 0.002$ in. and

$$Q_r^* = 1 \pm \frac{0.001}{W} \quad . \quad (C-9)$$

- Orifices and nozzles

In the cases of orifices and nozzles where (d_i/d_o) for an inexact model does not cause an appreciable change in the flow coefficient;

$$Q_r^* = (1 \pm \frac{t}{d_o})^2 \quad , \quad (C-10)$$

where t is the tolerance on the nozzle or orifice diameter, d_o . The same equation would apply to a square opening. For other configurations, actual areas would have to be calculated.

Frictional Losses

For frictional flow in passageways

$$\Delta p = C f(Re) \ell / De \frac{Q^2}{A^2} \quad . \quad (C-11)$$

Thus, for small dimensional variations and reasonably long passages,

$$\Delta p_r^* = Q_r^{*2} / (De_r^* A_r^{*2}) ; \quad (C-12)$$

again, with a pressure ratio of 1.0,

$$Q_r^* = \sqrt{De_r^*} A_r^* \quad . \quad (C-13)$$

Following are flow deviations for various frictional flow configurations with

$$\Delta P_r^* = 1.0 \quad .$$

- Round pipe

$$Q_r^* = (D_r^*)^{5/2} \quad , \quad (C-14)$$

or

$$Q_r^* = (1 \pm t/D_p)^{5/2} \quad , \quad (C-15)$$

where D_p is the nominal pipe diameter.

- Square passages

$$Q_r^* = (1 \pm t/L)^{5/2} \quad , \quad (C-16)$$

where L is the length of a side.

- Rod Bundle

$$\Delta P_r^* = Q_r^{*2} P_{wr}^* / A_r^{*5/2} \quad , \quad (C-17)$$

$$Q_r^* = A_r^{*5/24} \sqrt{P_{wr}^*}, \quad \Delta P_r^* = 1 \quad (C-18)$$

- Thin annuli and narrow slits

$$Q_r^* = De_r^{5/4} = (1 \pm t/De)^{5/4} \quad (C-19)$$

In all cases above, t , is the tolerance on the quantity dividing it.

APPENDIX D
MODELING WITH AIR AND CO₂

APPENDIX D

MODELING WITH AIR AND CO₂

The figures in this Appendix are for the gases, air and CO₂. They are based on Euler number similarity. ($Eu_r = 1.0$) and neglect variation in energy losses resulting from Reynolds number dissimilarity. In all cases

$$\Delta p_r = \frac{\Delta p_m}{\Delta p_p} = \frac{\rho_m V_m^2}{\rho_p V_p^2} = \frac{\rho_m}{\rho_p} \left(\frac{Q_m}{Q_p} \right)^2 \left(\frac{L_p}{L_m} \right)^4, \quad (D-1)$$

and

$$Re_m = \frac{Re_m}{Re_p} = \frac{V_m L_m \nu_p}{V_p L_p \nu_m} = \frac{Q_m}{Q_p} \frac{\nu_p}{\nu_m} \frac{L_p}{L_m}, \quad (D-2)$$

where $Q_r = \frac{Q_m}{Q_p}$ is given by Equation (D-1).

FIGURE D-1. Reynolds Number Ratio Versus Pressure Drop Ratio;
 Air at 100 °F, 14.7 psia

FIGURE D-2. Flow Ratio Versus Pressure Drop Ratio;
 Air at 100 °F, 14.7 psia

FIGURE D-3. Reynolds Number Ratio Versus Pressure Drop Ratio; Air at 150 °F, 20 psia

FIGURE D-4. Flow Ratio Versus Pressure Drop Ratio;
 Air at 150 °F, 20 psia

FIGURE D-5. Reynolds Number Ratio Versus Pressure
 Drop Ratio; Air at 100 °F, 100 psia

FIGURE D-6. Flow Ratio Versus Pressure Drop Ratio;
Air at 100 °F, 100 psia

FIGURE D-7. Reynolds Number Ratio Versus Pressure
 Drop Ratio; CO_2 at $175^\circ F$, 20 psia

FIGURE D-8. Flow Ratio Versus Pressure Drop Ratio;
 CO_2 at $175^\circ F$, 20 psia

FIGURE D-9. Reynolds Number Ratio Versus Pressure
 Drop Ratio; CO_2 at $175^\circ F$, 100 psia

FIGURE D-10. Flow Ratio Versus Pressure Drop Ratio;
 CO_2 at $175^\circ F$, 100 psia

FIGURE D-11. Reynolds Number Ratio Versus Pressure Drop Ratio; CO_2 at $175^\circ F$, 800 psia

FIGURE D-12. Flow Ratio Versus Pressure Drop Ratio;
CO₂ at 175 °F, 800 psia

FIGURE D-13. Reynolds Number Ratio Versus Pressure Drop Ratio; CO₂ at 175 °F, 1200 psia

FIGURE D-14. Flow Ratio Versus Pressure Drop Ratio;
 CO_2 at $175^\circ F$, 1200 psia

APPENDIX E
FLUID PROPERTIES

APPENDIX E

FLUID PROPERTIESPROPERTIES OF LIQUID SODIUM

Computations presented in the text were done on the Univac-1107 digital computer. The sodium thermodynamic properties were computed from a properties subroutine which included the following expressions for density, dynamic viscosity, and vapor pressure:

- Density (lbs/ft³)

$$\rho = (0.949 - 22.3 \text{ E} - 5 * T - 1.75 \text{ E} - 8 * T * T) * 62.428$$

T in degrees Centigrade

- Dynamic Viscosity (lg/ft-sec)

$$\mu = 0.000671969 * 10. **[- 1.09127 + 687.6/(T + 71.4)]$$

T in degrees Centigrade

- Vapor Pressure (lb/in.²)

$$P_v = \text{Exp} (-7.0909 - 21741.0/T + 3.4061 * \text{ALOG} (T) - 0.00456 * T + 1.1069 \text{ E} - 6 * T * T - 1.2768 \text{ E} - 10 * T * T * T)$$

T in degrees Rankine

Plots of density and vapor pressure are shown in Figures E-1 and E-3 and Figure E-2 shows kinematic viscosity (μ/ρ).

PROPERTIES OF WATER

The thermodynamic properties of water used in the calculations were obtained from a water properties table computer subroutine. Plots of table values for density, kinematic viscosity, and vapor pressure are shown in Figures E-4, E-5, and E-6.

FIGURE E-1. Density of Liquid Sodium

FIGURE E-2. Kinematic Viscosity $\left(\frac{\mu}{\rho}\right)$ of Liquid Sodium

FIGURE E-3. Vapor Pressure of Sodium

FIGURE E-4. Density of Water

FIGURE E-5. Kinematic Viscosity of Water

FIGURE E-6. Vapor Pressure of Water

APPENDIX F
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