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— PUMP —

Analog-Hybrid Reactor Coolant
Hydraulic Transient Model

by

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Reactor Coolant System

ABSTRACT

The PUMP hybrid computer code simulates flow and pressure distribution; it is used to determine real time response to starting and tripping all combinations of reactor coolant pumps in a closed, pressurized, four-pump, two-loop primary system. The simulation includes the description of flow, pressure, speed, and torque relationships derived through pump affinity laws and from vendor-supplied pump zone maps to describe pump dynamic characteristics. The program affords great flexibility in the type of transients that can be simulated.

CONTENTS

	Page
1. INTRODUCTION	1
2. MATHEMATICAL MODEL	3
2.1. Pump Dynamics	3
2.2. Fluid Dynamics	4
2.2.1. Pressure Drop Equations	5
2.2.2. Flow Equations	6
3. PROGRAM LIMITATIONS AND FEATURES	8
4. DISCUSSION	9
APPENDIX — Nomenclature	A-1

List of Figures

Figure

1. Reactor Coolant System, Schematic Diagram.	10
2. Typical Centrifugal Pump Characteristics	11
3. Complete Pump Homologous Characteristics	12
4. Pump Motor Starting Performance.	13
5. Pump Speed Dead Space Logic.	14
6. Four-Pump Reactor Coolant Flow Coastdown	15
7. Average Reactor Coolant Pump Speed Coastdown	16

1. INTRODUCTION

The loss of power to one or more pumps in a pressurized water reactor system may be harmful to the system because the flow to the reactor is not sufficient to prevent departure from nucleate boiling (DNB) due to excessive power-to-flow ratio. The accurate prediction of flow and pressure as a function of time during the flow transient will permit the optimum design of components for the system and plant and of the reactor protection system and will extract the maximum power capabilities from the reactor.

The pump analog-hybrid computer program predicts the steady-state and transient flow and pressure (subject to user-specified conditions and program limitations) of a closed, pressurized, four-pump, two-loop system.

Sufficient system equations are developed to adequately describe the chosen plant. The digital computer generates the characteristics of each pump and the appropriate friction factors, and the analog computer simulates the remainder of the reactor coolant system. The reactor coolant system configuration chosen for analysis comprises two loops with one steam generator in each loop. In each loop, the flow path branches into two parallel paths, each of which contains a pump. Figure 1 depicts such a system schematically and indicates the state variables.

The performance of each pump may be represented by the "pump affinity laws."* Performance data for a centrifugal pump indicate that a family of curves can be plotted, relating flow and rotor speed to hydraulic torque and dynamic pressure. Figure 2 shows these performance data as obtained from the manufacturer, and is referred to as a "zone map." Two curves are made from the family of curves in Figure 2 relating torque

* - - - -
A. J. Stepanoff, Centrifugal and Axial Flow Pumps, 2nd Ed., John Wiley & Sons, Inc., New York (1957).

as a function of flow and speed, and pressure as a function of flow and speed. Such curves are shown in Figure 3, where the ratio of percent torque and pressure to percent speed squared is plotted as a function of the ratio of percent flow to percent speed. These curves are the homologous pump relationships where, in this instance, the fluid density is assumed constant.

2. MATHEMATICAL MODEL

2.1. Pump Dynamics

To determine the pump rotor speed during a flow transient, Newton's second law is applied to its rotating mass and takes the form

$$J\dot{\omega} = \sum_{i=1}^N T_i \quad (1)$$

where the right-hand side is the sum of all the torques on the rotor and the torques on the rotating parts include windage and bearing torque, the torque on the rotor induced by imparting a pressure to the fluid, and motor torque. The magnetic torque is assumed to be negligible. Since the hydraulic torque of Figure 3 includes these torques, equation 1 can be rewritten

$$J\dot{\omega} = T_e - T. \quad (2)$$

Referring to Figure 3, T is the hydraulic torque; the arrow indicates the direction of movement during a coastdown transient. Since motor torque is a function of impeller speed, Figure 4 shows this relationship at 100% motor voltage as given by the motor manufacturer.

The rotor speed equations programmed for each of the pumps are listed below.

$$\dot{\omega}_1 = 12(T_{e1} - T_1)/J \quad (3)$$

$$\dot{\omega}_2 = 12(T_{e2} - T_2)/J \quad (4)$$

$$\dot{\omega}_3 = 12(T_{e3} - T_3)/J \quad (5)$$

$$\dot{\omega}_4 = 12(T_{e4} - T_4)/J \quad (6)$$

where

J = mass moment of inertia, in.-lbf-s²

= J_W/g where J_W = weight moment of inertia and

$$J_W = K^2 W,$$

where K = radius of rotating part gyration,

W = weight of rotating part, lbf,

g = gravitational constant, in./s².

2.2. Fluid Dynamics

Newton's second law also applies to the flow of fluid. Applying this law to a flow path and assuming complete turbulence and uniform flow produces

$$(L/A)\dot{w} = (\delta P + \Delta VP + \delta H - \Delta P)g \quad (7)$$

where

δP = differential pressure across flow path along direction of flow,

ΔVP = net recoverable pressure across flow path,

δH = net pressure due to elev difference across flow path,

ΔP = pressure drop across flow path due to fluid friction and form loss.

The pressure drop due to form loss is assumed to be proportional to the square of the flow. The pressure drop due to friction is assumed to be proportional to the square of the flow and a function of the Reynolds number. This function of Reynolds number is called the friction factor (f) and is calculated from the regression equation:

$$f = 1/\left\{ 2 \log_{10} [F - (5.028/Re) \log_{10} (F + 16.76/Re)] \right\}^2 \quad (8)$$

and

$$F = \epsilon/3.715(D). \quad (9)$$

After referring to Figure 1, unscaled reactor coolant system flow and pressure drop equations are derived and discussed below.

2.2.1. Pressure Drop Equations

Reactor Coolant Pump Outlet to Core Inlet

$$\Delta P_{N1} = (L/A)_N \dot{w}_1 + K_N w_1^2 + V_{HTI} w_T^2 - V_{H1} w_1^2 - \Delta H_{p-c} \quad (10)$$

$$\Delta P_{N2} = (L/A)_N \dot{w}_2 + K_N w_2^2 + V_{HTI} w_T^2 - V_{H1} w_2^2 - \Delta H_{p-c} \quad (11)$$

$$\Delta P_{N3} = (L/A)_N \dot{w}_3 + K_N w_3^2 + V_{HTI} w_T^2 - V_{H1} w_3^2 - \Delta H_{p-c} \quad (12)$$

$$\Delta P_{N4} = (L/A)_N \dot{w}_4 + K_N w_4^2 + V_{HTI} w_T^2 - V_{H1} w_3^2 - \Delta H_{p-c} \quad (13)$$

where

$$(L/A)_N = (\text{length-to-area ratio})/g_c,$$

$$K_N = f_L/D\rho A^2 2g = \text{pipe loss coeff, } (-K_N \text{ for negative flow coeff}),$$

$$V_{HTI} = 1/\rho A^2 2g = \text{velocity pressure coeff at core inlet } (-V_{HTI} \text{ for negative flow coeff}),$$

$$V_{H1} = 1/\rho A^2 2g = \text{velocity pressure coeff at pump outlet } (-V_{H1} \text{ for negative flow coeff}),$$

$$\Delta H_{p-c} = 12\rho[(\text{elev at pump outlet, ft}) - (\text{elev at core inlet, ft})], \text{ lb/in.}^2$$

Steam Generator Outlet to Core Inlet

$$\Delta P_{L1} = \Delta P_{p1} - \Delta P_{N1} - (L/A)_L \dot{w}_1 - K_L w_1^2 - F_L w_1^2 + V_{HS} w_B^2 - V_{H1} w_1^2 - \Delta H_{s-c} \quad (14)$$

$$\Delta P_{L3} = \Delta P_{p1} - \Delta P_{N1} - (L/A)_L \dot{w}_2 - K_L w_2^2 - F_L w_2^2 + V_{HS} w_A^2 - V_{H1} w_3^2 - \Delta H_{s-c} \quad (15)$$

where

$$(L/A)_L = (\text{length-to-area ratio})/g,$$

$$K_L = fL/D\rho A^2 2g = (\text{pipe form loss} + \text{steam generator outlet nozzle}) \text{ loss coeff } (-K_L \text{ for negative flow}),$$

$$F_L = fL/D\rho A^2 2g = \text{pipe friction loss coeff } (-F_L \text{ for negative flow}),$$

$$V_{HS} = 1/\rho A^2 2g = \text{velocity pressure coeff inside steam generator } (-V_{HS} \text{ for negative flow}),$$

$$V_{H1} = \text{velocity pressure coeff at outside of steam generator exit } (-V_{H1} \text{ for negative flow}),$$

$$\Delta H_{s-c} = 12\rho[(\text{elev at steam generator outlet, ft}) - (\text{elev at core inlet, ft})], \text{ lb/in.}^2$$

Core Inlet to Core Outlet

$$\Delta P_{\text{core}} = (L/A)_c \dot{w}_T + K_C w_T^2 + \Delta H_{c-o} \quad (16)$$

where

$$(L/A)_c = (\text{length-to-area ratio})/g,$$

$$K_C = \text{core form loss coeff},$$

$$\Delta H_{c-o} = 12\rho[(\text{elev at core inlet, ft}) - (\text{elev at core outlet, ft})], \text{ lb/in.}^2$$

Core Outlet to Steam Generator Outlet

$$\Delta P_{H1} = \Delta P_{L1} - \Delta P_{\text{core}} \quad (17)$$

$$\Delta P_{H2} = \Delta P_{L3} - \Delta P_{\text{core}} \quad (18)$$

2.2.2. Flow Equations

Pump No. 1 Flow

$$\dot{w}_1 = \dot{w}_B - \dot{w}_2 \quad (19)$$

Pump No. 2 Flow

$$\begin{aligned} \dot{w}_2 = [1/(L/A)_L] & \quad (-\Delta P_{L1} + \Delta P_{P2} - \Delta P_{N2} - K_L w_2^2 - F_L w_2^2 \\ & \quad + V H_S w_B^2 - V H_1 w_2^2 - \Delta H_{s-c}) \end{aligned} \quad (20)$$

Pump No. 3 Flow

$$\dot{w}_3 = \dot{w}_A - \dot{w}_4 \quad (21)$$

Pump No. 4 Flow

$$\begin{aligned} \dot{w}_4 = [1/(L/A)_L] & \quad (-\Delta P_{L3} + \Delta P_{P4} - \Delta P_{N4} - K_L w_4^2 - F_L w_4^2 \\ & \quad + V H_S w_A^2 - V H_1 w_4^2 - \Delta H_{s-c}) \end{aligned} \quad (22)$$

Loop A Flow

$$\begin{aligned}\dot{w}_A = [1/(L/A)_H] (\Delta P_{H2} - F_S w_A^2 - K_S w_A^2 + V H_S w_A^2 + \Delta H_{si-so} - K_{OH} w_A^2 \\ - F_H w_A^2 - V H_H w_A^2 + V H_{TO} w_T^2 - \Delta H_{co-si})\end{aligned}\quad (23)$$

where $(L/A)_H$ = (sum of length-to-area ratios of reactor outlet, 36-inch pipe, and steam generator)/g.

Loop B Flow

$$\begin{aligned}\dot{w}_B = [1/(L/A)_H] (\Delta P_{H1} - F_S w_B^2 - K_S w_B^2 + V H_S w_B^2 + \Delta H_{si-so} - K_{OH} w_B^2 \\ - F_H w_B^2 - V H_H w_B^2 + V H_{TO} w_T^2 - \Delta H_{co-si})\end{aligned}\quad (24)$$

where

$$\begin{aligned}F_S &= f_S L / (D \rho A^2 2g) = \text{steam generator tube friction loss coeff } (-F_S \text{ for negative flow coeff}), \\ K_S &= f_L / D \rho A^2 2g = \text{steam generator form loss coeff } (-K_S \text{ for negative flow coeff}), \\ \Delta H_{si-so} &= 12\rho[(\text{elev at steam generator inlet, ft}) - (\text{elev at steam generator outlet, ft})], \text{ lb/in.}^2, \\ V H_S &= 1/\rho A^2 2g = \text{net velocity pressure coeff inside steam generator, i.e., 36-inch diameter pipe - steam generator } (-V H_S \text{ for negative flow coeff}), \\ K_{OH} &= f_L / D \rho A^2 2g = (\text{form loss + reactor outlet}) \text{ loss coeff } (-K_{OH} \text{ for negative flow coeff}), \\ F_H &= f_H L / (D \rho A^2 2g) H = \text{36-inch-diameter pipe friction loss coeff}, \\ V H_H &= \text{velocity pressure in 36-inch-diameter pipe } (-V H_H \text{ for negative flow coeff}), \\ V H_{TO} &= \text{velocity pressure at core outlet } (-V H_{TO} \text{ for negative flow coeff}), \\ \Delta H_{co-si} &= 12\rho[(\text{elev at core outlet, ft}) - (\text{elev at steam generator inlet, ft})], \text{ lb/in.}^2\end{aligned}$$

Total System Flow

$$\dot{w}_T = \dot{w}_A + \dot{w}_B \quad (25)$$

3. PROGRAM LIMITATIONS AND FEATURES

The program limits the reactor coolant system to two loops and two branches per loop. The initial conditions for the problem are 100% flow and four pumps operating.

As the rotor speed approaches zero, the ordinates of Figure 3 approach infinity. In order to prevent digital computer overload, a dead space around zero rotor speed is required; this dead space logic is outlined in Figure 5. When the analog is operating within this dead space, the digital computer operates on a constant rotor speed, while the time rate of change of speed on the analog continues to be driven by the system dynamics. As a result, transients will be longer during this phase of a transient.

In deriving the system equations, the system is assumed to be isothermal throughout all transients; i.e., the fluid density is not sensitive to changes in heat flow caused by flow changes through the reactor and steam generator. The flow paths through the core are assumed not to change as a function of pump combination and flow dynamics during a transient.

Since the friction factor is calculated as a function of Reynolds number and is formed into arrays from which the appropriate friction factor is interpolated, care must be taken not to exceed the array memory allocation. A flag is available to inform the operator that an array allocation has been exceeded, which requires an increase in array dimensions.

The hybrid nature of the simulation permits unusual flexibility for the transient analyzer. PUMP can simulate such additional transient types as a locked-rotor accident, a pump startup, and transfer function analysis. With minor adjustments to the system model the analyzer can greatly increase the types of transients to be simulated.

4. DISCUSSION

The results of the four-pump flow coastdown predicted by the PUMP simulation compare most favorably with test data recorded at Oconee 1, especially in the first 15 seconds after all pumps are tripped, where pump inertia is high. Thereafter, a small deviation appears, which is well within the computer accuracy. The results are shown in Figures 6 and 7.

The deviation between the speed curves can be attributed to the inaccuracy of the zone map at low speeds and low flows. At approximately 10% speed, the deadband takes effect to force the speed to decrease at a lower rate, resulting in a higher calculated speed relative to what the actual speed would be. However, at low flows and speeds, the influence of speed on the flow is minimal because of the reduced sensitivity of the speed effect on flow; i.e., a large change in speed produces a relatively small change in pressure and torque.

Figure 1. Reactor Coolant System, Schematic Diagram

Figure 2. Typical Centrifugal Pump Characteristics

Figure 3. Complete Pump Homologous Characteristics

Figure 4. Pump Motor Starting Performance

Figure 5. Pump Speed Dead Space Logic

Figure 6. Four-Pump Reactor Coolant Flow Coastdown

Figure 7. Average Reactor Coolant Pump Speed Coastdown

APPENDIX
Nomenclature

Analytic and analog	Symbol	
	Digital	Description
$\{w\}_o$	FLO100	Initial condition, flow rate per pump, lb/s.
$\{\omega\}_o$	RAD100	Initial condition, rotor speed, rad/s.
	PRE100	Initial condition, dynamic pressure per pump.
	TOR100	Initial condition, hydraulic torque per pump, ft-lb.
w_1	W(1), INN(1)	Pump No. 1 flow, lb/s.
w_2	W(2), INN(2)	Pump No. 2 flow, lb/s.
w_3	W(3), INN(3)	Pump No. 3 flow, lb/s.
w_4	W(4), INN(4)	Pump No. 4 flow, lb/s.
ω_1	OMEGA(1), INN(5)	Pump No. 1 rotor speed, rad/s.
ω_2	OMEGA(2), INN(6)	Pump No. 2 rotor speed, rad/s.
ω_3	OMEGA(3), INN(7)	Pump No. 3 rotor speed, rad/s.
ω_4	OMEGA(4), INN(8)	Pump No. 4 rotor speed, rad/s.
w_B	W1(1), IINN(1)	Loop B flow, lb/s.
w_A	W1(2), IINN(2)	Loop A flow, lb/s.
ΔP_{p1}	PFRAC(1), IOUT(1)	Pump No. 1 dynamic pressure, lb/in. ²
ΔP_{p2}	PFRAC(2), IOUT(2)	Pump No. 2 dynamic pressure, lb/in. ²
ΔP_{p3}	PFRAC(3), IOUT(3)	Pump No. 3 dynamic pressure, lb/in. ²
ΔP_{p4}	PFRAC(4), IOUT(4)	Pump No. 4 dynamic pressure, lb/in. ²
T_1	TORFAC(1), IOUT(5)	Pump No. 1 hydraulic torque in pump rotor, ft-lbf.
T_2	TORFAC(2), IOUT(6)	Pump No. 2 hydraulic torque in pump rotor, ft-lbf.
T_3	TORFAC(3), IOUT(7)	Pump No. 3 hydraulic torque in pump rotor, ft-lbf.
T_4	TORFAC(4), IOUT(8)	Pump No. 4 hydraulic torque in pump rotor, ft-lbf.

Symbol		
Analytic and analog	Digital	Description
f_L	F1(1), IIOUT(1)	28-in.-diam-pipe friction factor, location in pump 1 leg.
f_L	F1(2), IIOUT(2)	28-in.-diam-pipe friction factor, location in pump 2 leg.
f_L	F1(3), IIOUT(3)	28-in.-diam-pipe friction factor, location in pump 3 leg.
f_L	F1(4), IIOUT(4)	28-in.-diam-pipe friction factor, location in pump 4 leg.
f_H	F2(1), IIOUT(5)	36-in.-diam-pipe friction factor, loop B.
f_H	F2(2), IIOUT(6)	36-in.-diam-pipe friction factor, loop A.
f_S	FSG(1), IIOUT(7)	Steam generator friction factor, loop B.
f_S	FSG(2), IIOUT(8)	Steam generator friction factor, loop A.
T_{e1}	MTORK(1), LOUT(1)	Pump No. 1 motor torque (torque on water by rotor), ft-lbf.
T_{e2}	MTORK(2), LOUT(2)	Pump No. 2 motor torque (torque on water by rotor), ft-lbf.
T_{e3}	MTORK(3), LOUT(3)	Pump No. 3 motor torque (torque on water by rotor), ft-lbf.
T_{e4}	MTORK(4), LOUT(4)	Pump No. 4 motor torque (torque on water by rotor), ft-lbf.
L		Length of component, in.
A		Cross-sectional area of component, in. ²
g		Gravitational constant, in./s ² .
w		Flow, lb/s.
$\dot{w}$		Time rate of change of flow, lb/s ² .
ω		Rotor speed, 1/s.
$\dot{\omega}$		Time rate of change of rotor speed, 1/s ² .

Symbol		Description
Analytic and analog	Digital	
ρ		Fluid weight density, lbf/in. ³
T		Hydraulic torque, ft-lbf.
T _e		Motor torque, ft-lb.
f		Friction factor.
J		Mass moment of inertia of pump rotor, in.-lbf-s ² .
J _w		Weight moment of inertia of pump rotor, in. ² -lbf.
D		Inside diameter of component, in.
ΔP		Pressure difference, lb/in. ²
ϵ		Roughness factor, rms in.
Re		Reynolds number.