TRANSIENT FLOW PERFORMANCE
IN A MULTI-LOOP NUCLEAR REACTOR SYSTEM

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ABSTRACT

The method presented can be used to determine fluid flow and coolant pump speed transients in a nuclear reactor system. Included are transients due to power failure, starting pumps in idle loops, and the opening of an active pump's discharge valve. Parallel pumps in the system may be analysed independently of each other. Typical cases are presented for the N. S. Savannah (NMSR) and the Consolidated Edison Thorium Reactor (CETR).

Partial failure of pumping power results in an unbalanced change in primary flow, i.e., flow in the active loops accelerates while flow in the failed loops decelerates rapidly. Maximum deceleration occurs when all pumps are operating and only one fails. The maximum rate of cold water is pumped into the reactor when the pumps in an idle loop are started while the discharge valve is open.

I. INTRODUCTION

A rigorous analysis to predict system behavior during transients in a pressurized water reactor (PWR) was undertaken by The Babcock and Wilcox Company. Consequently, a mathematical simulation has been written whereby system coolant flow and pump speed can be determined for various transient conditions.

The manner in which primary system flow responds to a disturbance is important since flow through the reactor decreases rapidly when pumping power is lost; thus the fuel elements could become overheated and produce burnout. This phenomenon — known as coastdown — is also of interest in analyzing water hammer in the primary piping. Another important consideration is this: If a large quantity of cold water from an out-of-service loop were suddenly inserted into a reactor having a negative temperature coefficient, there would be an undesired increase in reactor power output.

The reference method — based on the work of Arker and Lewis¹ — takes into account the inertia of the system fluid and rotating equipment, the system pressure drop, and the characteristics of the pump and pump motor. Also considered are the variation of coupling between the pump and the system fluid and the stop valve operator's characteristics. Although Arker and Lewis regard all major effects applicable to coastdown, they do not consider idle loops or the operation of pumps at various speeds. Furthermore, the data required for predicting the electrical braking effect during coastdown were not available in the form indicated.

As shown in Figure 1, all components that offer resistance change in the system must be considered. The pressurizer, purification system, and smaller components, not subject to full primary flow, are not regarded.

NOMENCLATURE

A	= cross-sectional area of flow path, ft^2
C	= a constant
e	= generated voltage
g	= acceleration due to gravity, 32.2 ft/sec^2
I_p	= moment of inertia of rotating parts, lb-ft^2
j	= number of sub-loops operating in loop N
K	= resistivity coefficient
K_W	= constant for windage and friction torque
KE	= kinetic energy, ft-lb
k	= a constant
l	= length of flow path taken along center lines, ft
L	= electrical inductance of rotor, henrys
n	= constant exponent
P_c	= stator can and core losses when operating as a motor
R	= electrical resistance of rotor, ohms
r	= radius of gyration, ft
T_E	= electrical torque, ft-lb
T_H	= hydraulic torque, ft-lb
T_{KE}	= torque due to fluid having kinetic energy, ft-lb
T_W	= windage and friction torque, ft-lb
W	= weight, lb
w	= fluid flow rate, lb/sec

$\overline{w}$	= flow at point of discontinuity, lb/sec
ΔP	= pressure drop, lb/ft ²
ΔP_c	= pressure drop, check valve, lb/ft ²
ΔP_{KE}	= pressure head, fluid kinetic energy, lb/ft ²
ΔP_L	= pressure drop external to reactor, lb/ft ²
ΔP_P	= developed pump pressure head, lb/ft ²
ΔP_v	= pressure drop, stop valve, lb/ft ²
ϵ	= a constant, 2.718
η	= hydraulic efficiency, %
ρ	= fluid density, lb/ft ³
τ	= time, sec
ϕ	= magnetic flux
ω	= angular speed of rotating parts, rad/sec

Subscripts

i	= instantaneous condition
j	= number of sub-loops operating in loop N
jM	= total number of sub-loops operating
N	= loop N or large piping
O	= full flow conditions
o	= conditions at time o (electrical equations only)
R	= reactor
s	= small piping

II. DISCUSSION AND ANALYSIS

Since this analysis is based on the law of conservation of energy, the net pressure drop around a closed channel like a pipe must equal zero. When pumping power is lost, system losses dissipate the kinetic energy stored in the pumps and the fluid. At steady state conditions, the developed pump head is equal to the sum of the pressure drops through the system; however, during a transient period, the fluid's kinetic energy is:

$$KE = \left(w^2 / 2g\rho \right) \sum l/A, \quad (1)$$

after summing up each part of the loop and applying the continuity equation. Differentiating equation (1) with respect to time gives the power produced by the kinetic energy in the fluid. Dividing the power by the volumetric flow gives the pressure head developed by that kinetic energy.

$$\Delta P_{KE} = \left(1/g \right) \sum (l/A) dw/d\tau \quad (2)$$

During a transient the kinetic energy in the fluid is added algebraically to the other resistive forces in opposing the pump.

$$\Delta P_{KE} + \Delta P_L + \Delta P_R = \Delta P_P. \quad (3)$$

Since the systems analyzed had pumps in parallel, the pressure head developed from the inertia of the fluid is divided into three sections: the reactor, the large pipe, and the small pipe containing the pump. The sum of these terms gives the total pressure head developed by the inertia of the fluid, and equation (3) may be written as

$$\begin{aligned} & \left(1/g \right) \left(\sum l/A \right)_s \frac{dw}{d\tau} + \left(1/g \right) \left(\sum l/A \right)_N (d/d\tau)(w_1 + w_2 + \dots + w_j) \\ & + \left(1/g \right) \left(\sum l/A \right)_R (d/d\tau)(w_1 + w_2 + \dots + w_{jM}) = \Delta P_P - \Delta P_R - \Delta P_L \end{aligned} \quad (4)$$

If the quantities that describe the operation of a centrifugal pump are known, according to a theorem of dimensional analysis, a complete equation can be written to describe the relation among the different quantities measured with fundamental units.² The head, capacity, and brake horsepower vary with speed according to definite rules (affinity laws). After the pump is determined, the head-capacity curves for any size or speed — constant or variable — can be determined from constants that the affinity laws follow.

For a given pump the capacity varies directly as the speed, and the head varies directly as the square of the speed. The developed head may be determined from the pump's head-capacity curve. If the speed is known, the head for a centrifugal pump may be equated according to the affinity laws as

$$\Delta P_P = \omega^2 f_1 \left(\frac{w}{j\omega} \right) . \quad (5)$$

The reactor pressure drop and the pressure drop in sections of the loop vary as a parabolic function of flow. If full-flow conditions are known for a specific reactor, equation (6) may be written:

$$\Delta P_R = \left(\frac{\Delta P_{RO}}{w_{RO}^n} \right) (w_1 + w_2 + \dots + w_{jM})^n . \quad (6)$$

Since the loop is divided into additional sections — large pipe, small pipe, check valve and gate valves —

$$\Delta P_L = \Delta P_N + \Delta P_s + \Delta P_c + \Delta P_v . \quad (7)$$

The large pipe pressure drop (ΔP_N) also includes the entrance and exit losses of the reactor nozzles, large pipe fittings, and the heat exchanger. The small pipe pressure drop (ΔP_s) — found in the parallel channel that contains the pumps — includes the small pipe fittings. Each of these terms may be written similarly,

$$\Delta P_L = \left(\frac{\Delta P_{LO}}{w_O^n} \right) w^n . \quad (8)$$

If a loop is idle while one or more parallel loops are active, there is a pressure differential across the seated check valve — provided that all other valves are open. This differential will be equal to the reactor pressure drop resulting from the flow of the active loop or loops. If the

idle loop were cold and the pumps were accidentally started under these conditions, a reactor cold water accident could result. This flow equation will not balance at the steady state existing before such a transient begins. Since only the pressure changes due to fluid flow around the loop are considered, the static pressure differential across the check valves of idle pumps causes an unbalanced condition in the equation. Therefore, to start such a transient, a steady state condition must be developed, i.e., the pump discharge pressure equal the reactor inlet pressure, or the pump developed head at zero flow equal the reactor inlet pressure drop. The speed of the pumps can be determined then.

The resistance that the check valve imposes on the system is complicated by the variable disc position. A check valve's steady state flow versus pressure drop curve usually consists of two sections: One applies while the disc is in the fully open position, the other while the disc is partially open. In addition, during some of the faster transients, the disc may not always be in the steady state position corresponding to a given flow. Since a complicated analysis is required to determine the exact disc position³ the steady state data are assumed to be applicable. If $w \leq \bar{w}$, the disc is partially open, and the pressure drop may be closely approximated by a polynomial equation:

$$\Delta P_c \approx k_1 (w/j)^4 + k_2 (w/j)^3 + k_3 (w/j)^2 + k_4 (w/j) + k_5 . \quad (9)$$

If $w > \bar{w}$, the flow is large enough to hold the disc wide open, and

$$\Delta P_c \approx k_6 (w/j)^n . \quad (10)$$

Equations (9) and (10) cannot be used at the same time. A typical tilting disc check valve curve is presented in Figure 2.

The pressure drop resulting from the stop valves is written separately so that flow can be determined as the valves are opened or closed. These valves here are gate valves equipped with either automatic or remote manual operators. If the discharge valve is closed and the pumps are operating at constant speed, again the equation will not balance in a steady state, because the pump head at no-flow is greater than the differential pressure across the closed valve. Therefore, a steady state flow is calculated to provide enough pressure drop across the valve, now slightly open; added to the reactor pressure drop and loop losses,

if any, this pressure drop will equal the pump head. Flow through a valve causes a reduction in static head which may be expressed in terms of velocity head. Applying the continuity equation, the pressure drop is given by

$$\Delta P_v = K \frac{w^2}{2g\rho A^2} \quad (11)$$

The resistivity coefficient, K , varies with valve size and with time as the valve opens and closes. Data⁴ for various valve openings and sizes were found to fit a logarithmic curve when plotted with a constant valve stem speed. The pressure drop through the stop valve then follows the equation

$$\Delta P_v = \frac{(w_1 + w_2 + \dots + w_j)^2 K}{2g\rho A^2} \quad (12)$$

Substituting into equation (4), the flow equation becomes

$$\begin{aligned} & (1/g) \sum (l/A)_s \frac{dw_s}{d\tau} + (1/g) \sum (l/A)_N (d/d\tau)(w_1 + w_2 + \dots + w_j) + \\ & (1/g) \sum (l/A)_R (d/d\tau)(w_1 + w_2 + \dots + w_{jM}) = \\ & \omega^2 f_1 (w/j\omega) - \frac{\Delta P_{RO}}{w_{RO}^{n_1}} (w_1 + w_2 + \dots + w_{jM})^{n_1} - \\ & \left(\frac{\Delta P_O}{w_O^{n_2}} \right)_N (w_1 + w_2 + \dots + w_j)^{n_2} - \left(\frac{\Delta P_O}{w_O^{n_3}} \right)_S w_s^{n_3} - \\ & [k_1 (w/j)^4 + k_2 (w/j)^3 + k_3 (w/j)^2 + k_4 (w/j) + k_5] - \\ & k_6 (w/j)^{n_4} - \frac{(w_1 + w_2 + \dots + w_j)^2 K_1}{2g\rho_1 A_1^2} - \\ & \frac{(w_1 + w_2 + \dots + w_j)^2 K_2}{2g\rho_2 A_2^2} \end{aligned} \quad (13)$$

To solve equation (13) when the pump itself is in a transient, the pump speed must be found at each instant. This is accomplished by simultaneously solving equation (13) and one that predicts speed. The kinetic energy of the pump's rotating parts,

$$KE = \frac{W}{2g} \omega^2 r^2 = \frac{1}{2g} I_p \omega^2 \quad (14)$$

is dissipated into several losses, and the power developed by the inertia is given by differentiating equation (14). Dividing by the speed gives the total torque developed by depletion of the kinetic energy.

$$T_{KE} = \frac{1}{g} I_p \frac{d\omega}{d\tau} \quad (15)$$

In order for the pump to operate, the electrical torque in the pump motor must equal the sum of the other torques.

$$T_E = T_{KE} + T_H + T_W \quad (16)$$

The electrical torque — produced by the electromagnetic circuits in the motor — is not the actual shaft output torque, but bears the relation:

"shaft output torque = electrical torque — friction and windage torque".

Electrical torque as a function of speed is required for all analyses of startup and coastdown. For startup a speed-torque curve, usually available from the motor manufacturer, may be used directly if the power source to the motor is not subject to voltage droop during starting. If a high impedance source is used, however, speed-torque curve corrections must be made to adjust for the reduced terminal voltage. It is programmed as a function of speed in equation (24). The electrical torque associated with coastdown requires a more complex method of evaluating, being an unfamiliar quantity to most equipment suppliers. For evaluation here, the three-phase induction motor may be considered as a synchronous generator. Magnetic flux, comparable to a generator's field flux, is trapped in the rotor when the electrical power is turned off.⁵ As the rotor continues to turn, the rotating flux generates a voltage in all stationary members, and currents flow in closed circuits. These closed circuit currents (eddy currents) exist in the magnetic circuit and motor can materials, and along with hysteresis losses resulting from the changing flux in the stator, produce a negative or retardation torque on the rotor. This torque would normally be quite small, with modern magnetic materials and laminated cores, but in canned motor pumps the stator can is subject to high eddy current losses. Since the eddy current losses constitute such a large portion of the total loss, the retardation torque can be evaluated accurately and more simply by considering it a result of eddy current losses alone.

Standard generator equations show that the torque that produces the

eddy current losses is proportional to the square of the flux and the angular velocity. Angular velocity is a variable in the final expression for retardation torque, but the flux must be evaluated. Two methods of flux evaluation give relative or proportional values which are used as multipliers with a known value of power loss.

In the first method, the eddy current and hysteresis power loss is measured by standard motor test; i.e., when the machine is running as a motor. The trapped flux is then said to be unity at the given power loss. The trapped flux during coastdown, initially unity, depreciates with time according to the inductive circuit discharge curve, $e^{-R\tau/L}$. The value of $e^{-R\tau/L}$ for the rotor circuit is obtained from the motor designer.

The second method also utilizes motor test to obtain the power loss, again said to be unity. In a second test, the motor power is cut off and the generated voltage is measured on the stator terminals by an oscillograph. Figure 3 shows a typical oscillogram from such a test. This voltage decreases as flux and angular velocity depreciate. Since a correction can be made for angular velocity depreciation, the resultant voltage decrease is due to flux depreciation. A plot of e_i/e_o (instantaneous corrected voltage, e_i , divided by the initial voltage, e_o) should fit the inductive circuit discharge curve. Usually the test curve does not fit the theoretical curve exactly because of non-linearity of the iron in the magnetic circuit.

Electrical torque for coastdown is initially:

$$T_E = k P_c / \omega_o \quad (17)$$

where k is a proportionality constant dependent upon the units used. Substituting ω_i (representing any instantaneous angular velocity) for ω_o , and depreciating P_c as both magnetic flux (ϕ) and angular velocity (ω) decay, the basic equation becomes:

$$\text{by method one, } T_E = k P_c \left(e^{-R\tau/L} \right)^2 \omega_i / \omega_o^2, \quad (18)$$

$$\text{by method two, } T_E = k P_c (e_i^2 \omega_i) / (e_o^2 \omega_o^2) \quad (19)$$

Equation (20) can be used for either method.

$$T_E = k P_c [f(\tau)]^2 \omega_i / \omega_o^2 \quad (20)$$

To determine the torque produced by the fluid (T_H) variation of efficiency with flow and speed must be known. For a given pump, the brake horsepower varies as the cube of the speed, assuming that efficiency remains constant with speed for each point. However, for the canned motor pumps considered here, efficiency drops off as speed drops off. If the design-point efficiency were independent of speed, the hydraulic efficiency could be represented as a constant function of the flow-to-speed ratio.¹ Therefore, if the pump manufacturer supplies a family of efficiency curves at various speeds, an envelope can be developed to show how design-point efficiency varies with speed. The efficiency used here is:

$$\eta = f_4 (w/j\omega) f_5 (\omega) . \quad (21)$$

As stated previously, the hydraulic efficiency must be known. The pump manufacturer can supply the pump efficiency curve. Although hydraulic efficiency involves only hydraulic losses — skin friction, sudden change of area or direction and eddies — they are difficult to separate from mechanical losses and leakage. In addition hydraulic efficiency and gross pump efficiency are probably identical in a canned motor pump. Hydraulic torque is defined as

$$T_H = w \Delta P_P / \omega \eta \rho . \quad (22)$$

The torque due to friction and windage or internal hydraulics (readily available from pump vendors as a lumped quantity) varies with speed according to the relationship

$$T_W = K_W (\omega)^n . \quad (23)$$

Empirical values for "n" vary from 1.7 to 2.4. The factor K_W is generally calculated from test data on pump motor test stands.

Equation (16) may now be written

$$\begin{aligned} (1/g) I_p \, dw/d\tau &= C f_2 (\omega) - P_c [f_3 (\tau)]^2 \omega / \omega_o^2 \\ \frac{w_s \omega f_1 (w/j\omega)}{j f_4 (w/j\omega) f_5 (\omega) \rho_3} &- K_W (\omega)^{n_3} \end{aligned} \quad (24)$$

To consider pumps in parallel channels at different speeds, the piping is divided into large and small pipe sections, as described. Thus, the pressure drop along one flow path can be calculated as well as the pressure head developed by the inertia of the fluid. If the speed of the pumps in parallel is identical during the transient, a shorter program may be used by considering the parallel piping as one channel.

To start a run with the check valve seated, a balanced condition must be found at steady state. Both dw/dt and ΔP_L will be equal to zero, so $\Delta P_P = \Delta P_R$. Since ΔP_R can be determined from the existing flow, ΔP_P will be known; thus the pump speed can be determined if flow is equal to zero and the starting conditions for a transient are known. The time elapsed before the transient begins may be found by integrating equation (24) and solving for τ .

To open the gate valve, the run must begin at some steady state point after the valve is cracked open. After selecting a time factor, the flow equations must be balanced for each loop on a trial and error basis. To start an equation for the opening of a valve with a very large resistivity coefficient is difficult — in fact, when $\tau = 0$, resistivity becomes infinite, and the equation obviously will not hold. Therefore, integration must be started at a finite time greater than zero.

III. RESULTS

The equations presented have been programmed and successfully run on a digital computer. These equations are applicable to most any number of loops with an infinite number of pumps arranged in parallel. Both two-loop and four-loop systems with two pumps connected in parallel have been considered here.

This program is compared with another in Figure 4, which shows flow coastdown resulting from the simultaneous failure of all CETR primary pumps. The other method assumes that all pump kinetic energy is converted to useful pumping energy at a constant efficiency — an assumption requiring flow and pump speed to coast down at the same rate. Figure 5 shows the significance of electrical retardation torque compared with other torques in a particular pump. Figure 6 shows coastdown in the NMSR resulting from complete power failure of these pumps. These flow and speed curves are asymptotic (because of the nature of the equations) and may appear to be zero at some finite time. In figures 7 - 11, showing coastdown under various conditions of continuing pumping power, the coasting flow channel very definitely goes to zero. This normally indicates that the check valve in that channel is fully closed and that the dead pump is merely churning water. However, the check valve may not have had time to close because of the rapid transient in flow.

The equations also hold for sequential power failure to the pumps — say power to one loop is cut off, then, before coasting down completely, power to a second loop is cut off. (See Fig. 12.) If the pumps in a cold loop in the N. S. Savannah are accidentally started while the valves in a cold loop are wide open, the worst type of cold water accident results (Fig. 13). This transient actually begins when the pump's discharge pressure equals that of the reactor inlet pressure, and the check valve opens. Without a check valve, there would be negative flow in the cold

loop and an even greater flow in the hot loop, up to this point. In this program, however, the pressure drop in the check valve is a function of flow only since the dynamics have not been considered, thus the results in the low flow region are qualitative.

Figure 14 represents flow in the two loops when the pumps are operating at full speed and the discharge valve in the cold loop is opening. Since the kinetic energy from the acceleration of the fluid is very small, it can be said that the flow through the valve is a function of valve position only. Flow curves for different sized valves may be calculated by hand, using steady state values — if the opening time is not less than one minute.

IV. CONCLUSIONS

1. Flows may be predicted for various combinations of pump failure, for starting pumps in idle loops, and for opening the discharge valve of an active pump.

2. The retardation torque resulting from the loss of power to canned motor pumps is significant.

3. Steady state flow points may be calculated by hand for the case of opening the discharge valve to an isolated cold loop whose pumps are operating — if the opening time is not less than one minute.

4. The worst cold water accident results when pumps in a cold loop are started while the stop valves are wide open.

5. Although no test data were available for comparison, it is believed that all important factors have been considered, and that the results are reasonably accurate.

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FIG. 1: TYPICAL MULTI-LOOP REACTOR AND PRIMARY SYSTEM

FIG. 2: PRESSURE DROP AND FLOW IN A
TILTING DISC CHECK VALVE

FIG. 3: OSCILLOGRAM OF VOLTAGE DECAY
(Measured On Stator Terminals, Power Off)

FIG. 4: CETR COASTDOWN AFTER COMPLETE PUMP
POWER LOSS

FIG. 5: COMPARISON OF TORQUES DURING COASTDOWN

FIG. 6: NMSR COASTDOWN AFTER COMPLETE
POWER LOSS TO THREE PUMPS AT FULL SPEED

FIG. 7: NMSR COASTDOWN (One Pump Fails, All Pumps Full Speed)

FIG. 8: NMSR COASTDOWN IN ONE-PUMP LOOP
(Three Pumps Operating At Full Speed)

FIG. 9: CETR COASTDOWN, ALL PUMPS IN TWO LOOPS FAIL

FIG. 10: CETR COASTDOWN
Two Pumps in Two Loops and One Pump in Two Loops to
Two Pumps in One Loop and One Pump in One Loop

FIG. 11: NMSR COASTDOWN (Three Pumps Fail At Full Speed)

FIG. 12: NMSR, LOOP TWO PUMPING POWER FAILS 0.8 sec AFTER LOOP ONE.

FIG. 13: NMSR COLD LOOP STARTUP (Open Loop)

FIG. 14: NMSR COLD LOOP STARTUP (Discharge Valve Opening,
Pumps At Full Speed)

END