HEFAT2010
7th International Conference on Heat Transfer, Fluid Mechanics and Thermodynamics
19-21 July 2010
Antalya, Turkey

TEHRAN RESEARCH REACTOR PRIMARY LOOP PUMP TRANSIENT PERFORMANE DURING STOPPING

kazem Farhadi
Engineering Science Research Group, Nuclear Science Research School, Nuclear Science and Technology Research Institute, AEOI, P. O. Box: 11365-3486, Tehran, Iran.
E-mail: kfarhadi@aeoi.org.ir

ABSTRACT

In this work an attempt is made to achieve characteristics curve needed for Tehran Research Reactor (TRR) during stopping. The TRR pump speed during the event is time dependent. Use is made of pump speed variation and rate of pump speed which during stopping time is not constant to obtain velocity component, flow rate, and head as a function of time. This is done for the TRR first quadrant operating characteristics. The obtained results indicate that the time needed for the coolant flow rate to reach zero is greater than the time necessary for pump to come into complete stoppage. This feature is attributed to coolant inertia in the TRR primary piping.

INTRODUCTION

The characteristics of a pump under unsteady operational conditions have usually been thought to follow along its steady-state characteristics curves [1]. This assumption, called hereafter quasi-steady change, is obviously acceptable in the situation where the change in operational condition is slow. When the rate of change of the operating point exceeds a certain limit, however, the centrifugal pump cannot respond quickly enough to traverse its steady-state characteristics curves, thus resulting in a significant change of the actual characteristics curves. There has been, therefore, a need for understanding the dynamics of the centrifugal pump characteristics. Another aspect of unsteady characteristics involves the transient ones with a large shift of the operating point in cases such as starting or stopping, and the quick closure or opening of the discharge valve. From the viewpoint of the theoretical treatment, these cases offer far more difficulties than the previous ones; this is because the linearization of the problem is fundamentally impossible. Pressure response during quick opening or closure of the discharge valve was studied by [2]; while transient characteristics during quick startup were reported by [3], and [4]. An analytical model was developed in that an effective energy ratio replaces centrifugal pump characteristics curve [5]. The authors determined the retarding torque which replaces motor torque during coastdown transient. In a separate study, the effect of system inertia on the acceleration head was investigated [6]. However, a few studies on the transient characteristics during the stopping period have been reported to date. Consequently it is interesting to have some knowledge of how the dynamic characteristics deviate from the quasi-steady ones as the rotational speed decreases. The aim of this study is to try to explain the nature of pumping action during stopping transient. In order to find a stopping characteristic curve for the TRR beyond which the assumption of the quasi-steady change becomes unacceptable, retarding torque for the TRR induction motor were measured. During stopping pump the retarding torque acts as an effective torque. Another aim is to establish the method by which the transient characteristics of a centrifugal pump during the stopping period can be predicted theoretically.

NOMENCLATURE

A	[m^2]	Fluid flow cross-section
B	[m]	Baffle spacing of heat exchanger
C	[m]	Clearance of tubes of heat exchanger
d	[m]	Orifice diameter
d_0	[m]	Inlet diameter of tube
D	[m]	Diameter
E	[volts]	Input voltage to motor
f	[-]	Friction factor
g	[m/s^2]	Gravity acceleration
h	[m]	Pump head
I_p	[kg.m^2]	Pump moment of inertia
I_f	[m^{-1}]	Fluid inertia
K	[-]	Loss coefficient
k_{mech}	[-]	Experimental constant of windage torque loss
L	[m]	Length of flow path
n	[rad/s]	Pump rotational speed
N_b	[-]	Number of baffle in heat exchanger
P_t	[m]	Pitch of tubes of heat exchanger
P_{in}	[W]	Power loss in stator
q	[kg/sec]	Flow rate
r_e	[Ω]	Effective resistance
r_s	[Ω]	Stator resistance
r_m	[Ω]	Rotor resistance
t	[sec]	Time
T	[kg.m]	Torque
u	[m/sec]	Fluid velocity
V	[m^3]	Volume

Special character

β	[-]	Ratio of orifice diameter to pipe diameter
ΔP	[kg/m^2]	Pressure drop
ΔP_p	[kg/m^2]	Developed pump pressure head
η	[-]	Efficiency
Φ_s	[-]	μ/μ_s
ρ	[kg/m^3]	Density

Subscripts

0	Conditions at time 0
bv	Butterfly valve
cv	Check valve
e	Equivalent
elec	Electrical
en	Entry
ex	Exit
f	Fuel
gv	Gate valve
hex	Heat exchanger
ht	Hold-up tank
hyd	Hydraulic
inst	Instantaneous
L	Loss
mech	Mechanical
p	Pump
R	Reactor
ss	Steady-state
s	Shell
w	Waste

PUMP STOPPING MODELING

The system, Tehran Research Reactor, to be analyzed in the present work is shown in **Figure 1**.

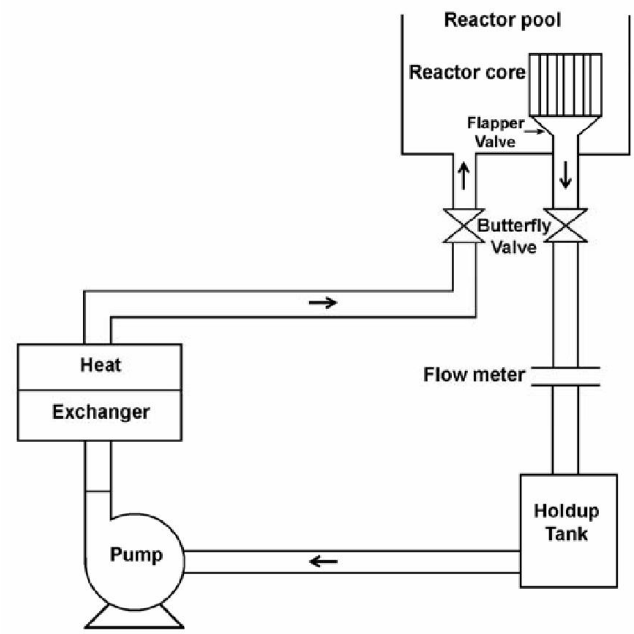

Figure 1 Primary cooling system of Tehran Research Reactor

All components shown in **Figure 1** offer resistance change in the primary cooling system and must be considered for a transient analysis such as pump stopping. The basic flow equation for a system involving i loop associated with a reactor which acts as a common flow path is given by

$$\frac{1}{g}\left(\sum\frac{L}{A}\right)_j \frac{dq_j}{dt} + \frac{1}{g}\left(\sum\frac{L}{A}\right)_R \frac{d}{dt}(q_1+q_2+...+q_i) = \Delta P_{Pj} - \Delta P_R - \Delta P_{wj} \quad (1)$$

This general equation could be further simplified when considering the primary cooling loop of the TRR

$$\frac{1}{g}\left(\sum\frac{L_j}{A_j}\right)\frac{dq}{dt} + \Delta P_L = \Delta P_P \quad (2)$$

where $\Delta P_L = \Delta P_R + \Delta P_w$

The first term on the left hand side of Eq. (2) represents pressure drop caused by inertial of the fluid in the primary cooling system while the second term accounts for the entire pressure drop arising in the system. Head losses due to change in pipe diameter, pipe bends, and valves…etc. may also be included in the second term. The centrifugal pump must provide enough head to compensate for the entire head and inertial losses in the system. Technical specifications of the TRR primary pump and piping are given in **Table 1**.

Physical quantities	Arker and Lewis (1956)	Yokomura (1969)	Grover and Koranne (1981)	Tehran Research Reactor (1986)
NR (RPM)	3580	1470	Unknown	1450
HR (m)	14.49	41.9	57.79	30.480
QR (m^3/s)	0.436	a 0.1259 b 0.0290 c 0.1256 d 0.0279	0.01572	0.1387
η (%)	86	75.6	83	83
I_P (kg.m^2)	0.8	3.7	a 0.2 b 0.5 c 1.0	0.2
$\sum\frac{L}{A}$ (m^{-1})	38.56	959	d 6916 e 13832 f 27664	1717
ε	0.342	0.1131	a(e) 0.3250 b(e) 0.1300 c(e) 0.0650 d(a) 0.0161 e(a) 0.3250 f(a) 0.6506	0.947

Table 1 represents design data of some of stopping transient

The centrifugal pump head will vary according to the product of the square of the impeller speed and a function depending on the ratio of flow to impeller speed so that

$$\Delta P_p = n^2 f_1\left(\frac{q}{n}\right) \tag{3}$$

The primary cooling loop pressure drop as a function of flow must be determined for the specific geometry involved. Therefore

$$\begin{aligned}\Delta P_L &= \Delta P_{cv} + \Delta P_{gv1} + \Delta P_{hex} + \Delta P_{gv2} + \Delta P_{bv1} + \Delta P_{pool} \\ &+ \Delta P_{fuel} + \Delta P_{bv2} + \Delta P_{orifice} + \Delta P_{ht} + \Delta P_{piping} + \Delta P_{bends}\end{aligned} \tag{4}$$

The total pressure drop in the primary loop is sum of the pressure drop of the individual components which offer resistance change in the fluid system. Pressure drop due to check valve is calculated by

$$\Delta P_{cv} = K_{cv}\frac{1}{2\rho g}\left(\frac{q}{A_{cv}}\right)^2 \tag{5}$$

This is a general equation and can be equally used for orifice, gate valve, check valve…etc.

Pressure drop caused by the gate valves before and after the heat exchanger is given by

$$\Delta P_{gv} = K_{gv}\frac{1}{2\rho g}\left(\frac{q}{A_{gv}}\right)^2 \tag{6}$$

Pressure drop due to heat exchanger is given by

$$\Delta P_{hex} = \frac{f\,(N_b+1)D_s}{2\rho g D_e \phi_s}\left(\frac{qP_t}{D_s CB}\right)^2 \tag{7}$$

For triangular tube arrangement

$$D_e = \frac{8\left(\sqrt{3}\frac{P_t^2}{4} - \frac{\pi d_\circ^{\,2}}{8}\right)}{\pi d_\circ}$$

$$\phi_s = \left(\frac{\mu}{\mu_s}\right)^{0.14}$$

$$f = \exp(0.576 - 0.19\ln(\mathrm{Re}))$$

The butterfly valve before the pool is fully opened and consequently offers least resistance.

Nevertheless its pressure drop is given by

$$\Delta P_{bv1} = K_{bv1}\frac{1}{2\rho g}\left(\frac{q}{A_{bv1}}\right)^2 \tag{8}$$

The pool pressure drop is modelled as

$$\begin{aligned}\Delta P_{pool} &= \frac{fL_p}{D_p 2\rho g}\frac{q^2}{A^2} = \frac{K_{pool}}{2\rho g}\frac{q^2}{A^2} \\ f &= 0.004(\mathrm{Re})^{-0.16}\end{aligned} \tag{9}$$

For the analysis presented in this paper, the total pressure drop across a fuel element consists of losses in the upper and lower end boxes of the element and the pressure drop across the fuel plates. The pressure drop across the fuel plates includes the pressure loss at the entrance to the fuel element, the friction loss in the fuel element, and the pressure loss at the exit of the fuel element, respectively. Alternatively

$$\Delta P_{fuel} = \Delta P_{en} + \Delta P_f + \Delta P_{ex}$$

The entrance, friction, and exit pressure losses for single-phase flow can be calculated from the following standard formulae (McAdams, 1954)

$$\Delta P_{en} = \frac{K_{en}\rho u^2}{2g} \tag{i}$$

$$\Delta P_f = \frac{4fL_c\rho u^2}{2D_e g} \tag{ii}$$

$$\Delta P_{ex} = \frac{\rho(u-u_\circ)^2}{2g} \tag{iii}$$

Assuming constant coolant density, it is possible to write

$$\frac{u_\circ}{u} = \frac{A_c}{A_\circ} \tag{iv}$$

Where A_c the total cooling channel cross-sectional area in the fuel element and $A_\circ$ is the cross-sectional area of the end box immediately beyond the channel exit. Combining relations (i) through (iv) results in an expression for the pressure losses across the fuel plates (IAEA-TECDOC-233, 1980)

$$\Delta P_{fuel} = \frac{\rho u^2}{2g}\left\{K_{en} + \frac{4fL_c}{D_e} + \left(1 - \frac{A_c}{A_\circ}\right)^2\right\} \tag{v}$$

Assuming $u = \frac{\rho q}{A_c}$ represents average velocity throughout the core, relation (v) can be written as

$$\Delta P_{fuel} = \frac{q^2}{2\rho g A_c^2}\left\{K_{en} + \frac{4fL_c}{D_e} + \left(1 - \frac{A_c}{A_\circ}\right)^2\right\} \tag{10}$$

Head losses due to presence of orifice just before the hold-up tank is given by

$$\Delta P_{orifice} = \frac{(1-\beta)^4}{2\rho g} \frac{q^2}{C_d{}^2 A^2} \qquad (11)$$

$$\beta = \frac{d}{D}$$

$$C_d = f(\beta) + 91.71\beta^{2.5}\,\mathrm{Re}^{-0.75} + \frac{0.09\beta^4}{1-\beta^4}F_1 - 0.0337\beta^3 F_2$$

$$f(\beta) = 0.5959 + 0.0312\beta^{2.1} - 0.184\beta^8$$

$$F_1 = \frac{2.54\times10^{-2}}{d} \quad \text{for} \quad 5.08\times10^{-3} \le d \le 5.842\times10^{-2}\,(m),$$

otherwise $F_1 = 0.433$

$$F_2 = \frac{1}{d}$$

Pressure drop in the hold-up tank is given by

$$\Delta P_{ht} = \frac{fL_{ht}}{2\rho g D_{ht}} \frac{q^2}{A^2} = \frac{K_{ht}}{2\rho g} \frac{q^2}{A^2} \qquad (12)$$

$$f = 0.04(\mathrm{Re})^{-0.16}$$

Pressure drop in the piping is given by

$$\Delta P_{piping} = K_{piping} \frac{1}{2\rho g}\left(\frac{q}{A_{piping}}\right)^2 \qquad (13)$$

Also pressure drop caused by bends is given by

$$\Delta P_{bends} = K_{bends} \frac{1}{2\rho g}\left(\frac{q}{A_{bends}}\right)^2 \qquad (14)$$

Substituting relations (3) through (14) into equation (2) results in

$$\frac{1}{g}\left(\sum\frac{L_j}{A_j}\right)\frac{dq}{dt} + K_{cv}\frac{1}{2\rho g}\left(\frac{q}{A_{cv}}\right)^2 + K_{gv1}\frac{1}{2\rho g}\left(\frac{q}{A_{gv}}\right)^2 +$$

$$\frac{f(N_b+1)D_s}{2\rho g D_e \phi_s}\left(\frac{qP_t}{D_s CB}\right)^2 + K_{gv2}\frac{1}{2\rho g}\left(\frac{q}{A_{gv}}\right)^2 + K_{bv1}\frac{1}{2\rho g}\left(\frac{q}{A_{bv1}}\right)^2 +$$

$$\frac{K_{pool}}{2\rho g}\frac{q^2}{A_{pool}^2} + \frac{q^2}{2A_c^2}\left\{\begin{matrix} K + \dfrac{4fL_c}{D_e} + \\ \left(1-\dfrac{A_c}{A_o}\right)^2 \end{matrix}\right\} + K_{bv2}\frac{1}{2\rho g}\left(\frac{q}{A_{bv2}}\right)^2 + \frac{(1-\beta)^4}{2\rho g}\frac{q^2}{C_d{}^2 A_{orifice}^2} +$$

$$\frac{K_{ht}}{2\rho g}\frac{q^2}{A_{ht}^2} + K_{piping}\frac{1}{2\rho g}\left(\frac{q}{A_{piping}}\right)^2 + K_{bends}\frac{1}{2\rho g}\left(\frac{q}{A_{bends}}\right)^2 = n^2 f_1\left(\frac{q}{n}\right)$$

(15)

Time dependent values of q must be known in order to know how flow rate varies with time. Instantaneous values of flow rate depend on speed and torque. Therefore motor and pump speed distributions are needed. For obtaining the imparted speed and torque to the coolant use is made of conservation of energy which results in the following equations. **Figure 2** shows total head versus flow coefficient of the TRR centrifugal pump.

The equation of motion for the primary pump rotational speed is given by

$$\frac{1}{g} I_P \frac{dn}{dt} = T_{elec.} - T_{hyd.} - T_{mech.} \qquad (16)$$

The authenticity of the basic flow equation as well as pump equation of motion results is depend on the accurate modelling of individual terms which constitute these equations. Therefore like equation of motion for coolant flow in the primary loop, each term in this equation is treated in detail. Torque-speed characteristic of the TRR canned induction motor is not available. Several attempts were made by the author to obtain it from its motor manufacturer. The manufacturer does not support this motor anymore. Since in the present work only pump stopping for the TRR is attempted to be analysed therefore its functional form $Cf_2(n)$ is used. Instead, electrical torque during pump stopping must be analysed in a greater detail.

The electrical torque developed during pump stopping necessitates a cumbersome method for evaluation. The losses or retardation torque must be evaluated as accurately as possible during the pump stopping. This is because during the event this is the only torque available in the system for accelerating mechanical load (circulating the coolant).

In general, skin effect is characterized by a reduction of the effective area of cross-section of the rotor cage. This is caused by the current flows through the least impedance path of the rotor bar surface. During pump stopping, slip frequency increases. This in turn increases the impedance of each motor phase. As a consequence the effective electrical torque decreases. Moreover, at high slips and (currents), reduction of the leakage flux path saturation causes a reduction of stator and rotor leakage inductance. The influence of magnetizing saturation for higher slips is negligible. Nevertheless the effect of decaying magnetic flux and angular velocity is included in Equation (18). Stray losses (eddy current) are already included in the core loss.

When the power is cut-off from the motor the current reduces. This current produces a magnetic flux instantaneously. As a consequence the induced magnetic field in the rotor changes direction with the current frequency. Due to hysteresis in rotor, the iron core lags the new direction which prevents rotor to attain actual speed relative to frequency. The induced current in the rotor opposes the motion of the rotor and consequently reduces the effective rotor torque.

Standard test are conducted on the motor test stands to determine constant and variable losses. Total electrical power losses were measured by two wattmeters (Blondel's theorem). Resistance of each motor phase is determined by applying a direct voltage to terminal of the motor. The effective resistance of each phase is obtained as

$$r_e = 1.25 r_S$$

The total stator copper loss is given by

$$P_{CuS} = \frac{3}{2} r_e I_L^2 \quad \text{(Emanuel, 1985)}$$

In the no-load test slip is very low (rotor low copper loss) and therefore part of input power is consumed by constant losses (stator core loss and total mechanical loss) and some dissipated into heat in the stator. From this test total mechanical loss (friction and windage) can be extracted. In the rated-load test a tachometer is used to monitor the motor speed. A multimeter is employed to measure input current once rated current is attained by the motor. In this test stator copper loss and slip can be calculated. The measured values of the motor parameters during the tests are

$P_{in} \simeq 9kW$; $E = 380V$; $\omega = 153.4 rad/s$; $T_E = 444N.m$; $r_m = 0.1\Omega$

where E is the input voltage of the motor.

During pump stopping the residual magnetic flux diminishes with time in accordance with inductive circuit discharge curve given by

$$\varepsilon^{-\frac{t}{\tau}}$$

where $\tau = \frac{L}{R}$.

This quantity for the rotor circuit is usually available from the motor designer. Initially the electrical torque for the pump stopping is given by

$$T_{coastdown} = \frac{1}{s} \frac{(P_{Core} + P_{CuS})}{n_{initial}} \tag{17}$$

Representing $n_{inst.}$ as any instantaneous angular velocity, the electrical torque can be shown by

$$T_{coastdown} = \frac{1}{s}(P_{Core} + P_{CuS})\left(\varepsilon^{-\frac{t}{\tau}}\right)^2 \frac{n_{inst.}}{n_{initial}^2} \tag{18}$$

Generally, for a given centrifugal pump, the design point efficiency is independent of speed. For this kind of pumps, the impeller efficiency could be shown as a constant function of the ratio of flow to speed for the entire range of first quadrant. However, for the TRR centrifugal pump considered in the analysis, efficiency diminishes as speed decreasing. Therefore, the efficiency used in the analysis is given by

$$\eta = f_3\left(\frac{q}{n}\right) f_4(n) \tag{19}$$

The hydraulic torque is expressed as

$$T_{hyd.} = \frac{q \Delta P}{\rho n \eta} \tag{20}$$

The mechanical torque due to friction and windage varies with speed according to the relationship (Dey et al. 2008)

$$T_{mech.} = k_{mech.}(n)^2 \tag{21}$$

The factor $k_{mech.}$ is generally calculated from test data on induction motor test stands. In the present work the value of $T_{mech.}$ is calculated at the rated rotational speed of the TRR primary centrifugal pump. With these definitions Eq. (16) can be written as

$$\frac{1}{g} I_P \frac{dn}{dt} = Cf_2(n) - \frac{1}{s}(P_{Core} + P_{CuS})\left(\varepsilon^{-\frac{t}{\tau}}\right)^2 \frac{n_{inst.}}{n_{initial}^2} - \frac{qnf_1\left(\frac{q}{n}\right)}{f_3\left(\frac{q}{n}\right) f_4(n)\rho} - k_{mech.}(n)^2 \tag{22}$$

In the case of pump stopping, the second term on the right hand side of Eq. (22) becomes positive.

This equation predicts pump speed during pump stopping transient.

RESULTS AND DISCUSSION

The mathematical model developed in the present work is used to study the influence of retarding torque on the Tehran Research Reactor pump stopping transient.

Figure 3 shows pump stopping resulting from the TRR primary loop pump loss of electric power. Runge-Kutta method is used to solve equation (15) which predicts time dependent flow rate with equation (22). This equation represents primary pump in a transient situation. During the transient, using **Figure 3**, it is possible to determine time dependent variations of flow rate and primary pump speed.

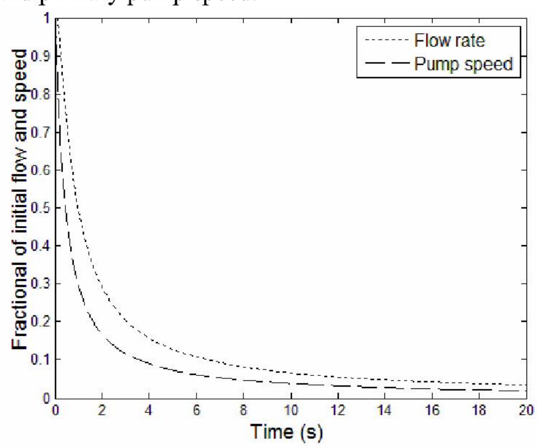

Figure 3 the TRR pump stopping characteristics curve

Figure 4 shows the obtained results from flow coefficient and head coefficient in stopping transient. Flow coefficient begins from an initial value of less than 0.2 and increases. While head coefficient from its initial value at t=0, indicates steady-state, decreases with time. Consequently, in the transient the value of flow coefficient is greater than the entire flow rate obtained at steady-state. In the contrary, the head coefficient has its maximum value at steady-state.

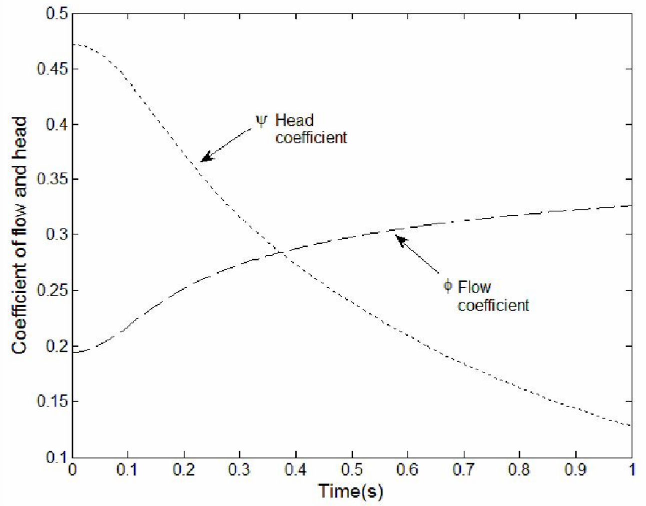

Figure 4 Variations of head and flow coefficient with time

Figure 5 shows variations of head coefficient versus flow coefficient. The degree of variations of head and flow coefficient are seen. These variations appear as a second degree polynomial which is a dynamic characteristics curve from which head and speed can be extracted for different time.

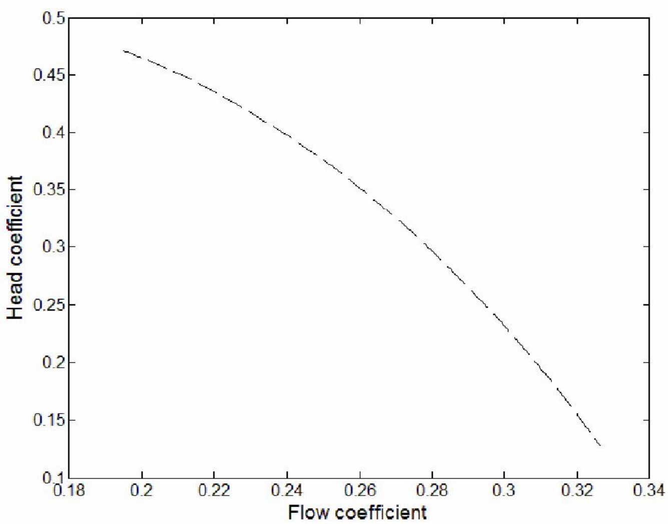

Figure 5 Variation of head and flow coefficient with time

CONCLUSION

The obtained results, based on theoretical method, indicate that flow rate and speed in the stopping transient decreases with time. This decrease in the initial stage soon after stopping pump is faster than the later stage. Also speed reduction in comparison to flow rate reduction is occurring faster and in less time reaching to its minimum value. In addition to this, pump dynamic characteristic curve which was obtained based on theoretical methods. Since most of the obtained results are based on theoretical method and because the theoretical values are not available it was not possible to compare the obtained results from the present study with experimental ones.

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.

REFERENCES

[1] Knapp, R. T., Complete Characteristics of Centrifugal Pumps and Their Use in the Prediction of Transient Behaviour, Trans. ASME, Vol. 59, Nov. 1937, pp. 638-689.

[2] Kaneko, M., and Ohashi, H., Transient Characteristics of a Centrifugal Pump During Quik Change of Flow Rate, Trans. JSME, Ser. B, Vol. 48, No. 426, Feb. 1982, pp. 229-237 (in Japanese).

[3] Tsukamoto, H., and Ohashi, H., Transient Characteristics of a Centrifugal Pump During Starting Period, ASME Journal of Fluids Engineering, Vol. 104, No. 1, Mar. 1982, pp. 6-14.

[4] Saito, S., The Transient Characteristics of a Pump During Start Up, Bulletin of JSME, Vol. 25, No. 201, Mar. 1983, pp. 372-379.

[5] Farhadi, K., and Akbarian Shourkaei, H., Flow transient's performance in an MTR-type research reactor, submitted to Progress in Nuclear Energy, 2010.

[6] Farhadi, K., and Akbarian Shourkaei, H., Transient behaviour of parallel pump in nuclear research reacotors, submitted to Progress in Nuclear Energy, 2010.

[7] [2] Arker, A. J., Lewis, D. G., Rapid flow transients in closed loops. In: Reactor Heat Transfer Conference of 1956. TID-7529 (part 1), Book 1, pp. 33-47.

[8] Yokomura, T. Flow coastdown in centrifugal pump system. Nuclear Engineering and Design 10, pp. 250-258, 1969.

[9] Grover, R. B., Koranne, S. M., Analysis of pump start-up in transients. Nuclear Engineering and Design 6, pp. 137-144, 1981.