

NUMERICAL MODELLING OF TRANSIENT FLOW IN LONG OIL PIPE LINE SYSTEM

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Abstract

The problem of unsteady flow are frequently encountered in long oil pipelines without the provision of surge tank due to sudden closing or opening of valve and pump trip. Transient analysis of the pipe flow is often more important than the analysis of the steady-state operating conditions ,that engineers normally use to with stand these additional loads resulting from rapid flow change situations.

The oil-hammer are analyzed and predicted by using the Eulerian approach (MOC)and the Lagrangian approach (WCM). The investigation was carried out for sufficiently long time to demonstrate changing of pressure head with time. Transient conditions arising in long oil pipeline with pumping station, valve and branches are studied in details for KRK pipeline by these two approaches. A Computer programs for all these components were developed.

The effect of valve closure, line branch junction and pump shut-down are studied taking into account the effect of line friction on pressure wave. A numerical model " UNSTEADY_ FRIC_WH " using MOC and Barr's explicit friction factor has been presented for solution of the transient flow situation of water hammer. Assessment of friction factor at any section in this unsteady transient flow conditions clearly indicates the effectiveness of using variable friction factor in contrast to the steady state friction as in the available numerical models.

Introduction

Cross-Country pipelines transporting crude oil or refined product are usually several hundred Kilometers long and have several pumping stations located along their length.

The analysis of transient in oil pipeline, some time called oil-hammer analysis or surge analysis, is rather complex because of pipe line friction losses are large compared to the instantaneous pressure change caused by sudden variation of flow velocity. Prior to availability of high –speed digital computer these analysis were approximate and a large factor of safety had to be used to allow for uncertainties in the computed results.

The basic unsteady flow equations along pipe due to transient effect (closing valve, pump start up-or shutdown,...etc.) are non linear and hence analytical solution are not possible. Allevi and Bergan developed classical solution by both analytical and graphical methods. Graphical solution mentioned above had some practical application in pipe design before advent of Computers. Streeter1969 developed a numerical model by using a constant value of turbulent friction factor. Wiggert and Sundquist 1977 solved the pipe line transient fixed grids projecting the characteristic form outside the fundamental grid size. Their analysis shows the effects of interpolation, spacing, and grid size on numerical attenuation and dispersion . Watt 1980 have solved for rise of pressure by Method Of Characteristics (MOC) for only 1.2 seconds and the transient friction values have not been considered. Shimadg and Okashimo 1984 solved the second order

equation of water hammer by series of solution method and a Newton-Raphson method. They calculate only maximum surge pressure with constant friction factor. The solution was not carried out for sufficiently long time to demonstrate damping of pressure head with increase time. Chaudhry and Hussaini 1985 solved the water hammer equations by McCormack technique. Samani and khayatzadeh 2004 also used this method to solve the system of equations related to the effect opening or closing a valve located in a pipeline . They solved the non-linear terms by McCormack predictor corrector, and the friction losses are estimated by applying a coefficient of pressure drop.

Many works on the analysis of transient flow focused mainly on specific type of flows such as liquid flows in pipelines(as opposed to flow through none- pipe components such as pumps and valves). Although these methods are suitable for the types of the flow for which they were developed, they usually suffer limitations when applied to other types of flow. An example is the popular method of characteristics(MOC) which was initially developed to analysis fast transient in oil pipeline (Wylie and Streeter,1993). The equations of water hammer , as derived by this method are applied to oil pipe system. It is shown that oil pipeline transient may be calculated directly by these methods; experimental configuration is given by comparison with field tests. In the MOC the partial differential equation first converts into ordinary differential equation which then can be solved by an explicit finite –difference technique. For pipe line (closed conduit) applications, the MOC widely used (Boules et al 2004).The MOC is considered the most accurate of Eulerian methods in its representation of governing equations but require numerous steps or calculation to solve a typical transient pipe flow problem. As the pipe system becomes more complex , the number of required calculation increase. This method has been summarized by other researchers (Boules et al 2004,Larock et al 1999, Chaudhry 1987, Watters 1984,Streeter and Wylie 1967) and implemented in various computer programs for pipe system transient analysis (Axworthy et al,1999; Karney and McInnis, 1990) . Transient condition arising in long oil pipelines can be adequately analyzed and predicted by use of the MOC; attenuation, line packing ,pyramiding and rarefaction can be completely taken into account . As compared with shorter piping systems, oil pipelines with several pumping stations, having units in series or in parallel , and with special speed controls or valving action, have somewhat different boundary conditions. Since the pumping heads are almost completely used to overcome fluid friction , careful attention to fluid properties is essential ; also the complications arising from various batches of oil traversing the system at any one time must be examined (Rainer 2004).

Another method used to solve transient flow equations in the event- oriented system simulation environment is the LaGrange approach (Wood 2005).In this method, the pressure wave propagation process is driven by distribution system activities . The wave characteristics method (WCM)is an example of such approach(Wood et al 2005,Boules et al 2004) and was first described in the literature as the wave plan method (Wood et al 1966). The method tracks the movement and transformation of pressure wave as they propagate through- out the system and compute new conditions either at first time intervals or at times which a change actually occurs (variable time internal).The effect of line friction on

pressure wave using a nonlinear characteristics relationship describing the corresponding pressure head changes as a function of the lines flow rate.

PART I: THE METHOD OF CHARACTERISTIC

Governing Equations

The dynamics equations in transients-state flow in closed conduit are the equations for the conservation of mass and the momentum equation. These equations are a set of hyperbolic partial differential equations. Using the Reynolds transport theorem and assuming one-dimensional flow of elastic conduit with slightly compressible fluid, which stretch or contract with respect to time, these equations can be written as follow (Chaudhry 1987).

Continuity equation

$$\frac{\partial p}{\partial x} + V \frac{\partial \rho}{\partial x} + \rho a^2 \frac{\partial V}{\partial x} = 0 \quad \dots\dots\dots(1.1)$$

Where p; pressure V; velocity a; wave speed, ρ; fluid density; t; time, x; coordinate axis along conduit length.

For thin walled conduit Robert (1999), proposed following expression for the wave speed;

$a = \sqrt{\frac{K}{(1 + \psi D/e K/E)}}$, where K; compression modules for fluid, E; Young modules of pipe material, D; pipe diameter, e; pipe wall thickness, $\psi = (1 - \Phi^2)$, in which Φ is the Poisson ratio of pipe material.

Momentum equation;

$$\frac{\partial V}{\partial t} + V \frac{\partial V}{\partial x} + \frac{1}{\rho} \frac{\partial p}{\partial x} + g \sin\theta + f \frac{V|V|}{2D} = 0 \quad \dots\dots(1.2)$$

where D; pipe diameter, g; acceleration due to gravity, θ; pipe angle w.r.t the horizontal, f ; Darcy-Weisbetch friction factor.

In most engineering applications the convective acceleration term, $V (\partial V / \partial x)$ and $(V \partial P / \partial x)$ are very small compared to the other term and can be neglected. Expressing pressure in term of piezometric head, above a specified datum, and using the discharge $Q = A V$, instead of the flow velocity, equations 1 and 2 becomes;

$$\frac{\partial H}{\partial t} + \left(\frac{a^2}{g A} \right) \frac{\partial Q}{\partial x} = 0 \quad \dots\dots\dots(1.3)$$

Equation is modified as

$$\frac{\partial Q}{\partial t} + g.A.\frac{\partial H}{\partial x} + \left(\frac{f}{2DA} \right) Q|Q| = 0 \quad \dots\dots\dots(1.4)$$

where Q: flow discharge, A: pipe cross-sectional area and H: piezometric head.

Characteristic Equation

In time domain methodology, Equ. 3 and 4, are normally solved using the method of characteristics(see for example, Streeter 1993). This involves finding moving coordinate system in which the equation may be written an ordinary rather than partial differential equations. Consider the relation that results when we multiplying Eq. 4 by and add to Eq.3 yields;

$$(\partial Q / \partial t + \lambda a^2 \partial Q / \partial x) + \lambda g A (\partial H / \partial t + \partial H / \partial x) + f Q |Q| / 2DA = 0 \quad \dots\dots\dots(1.5)$$

If the coefficient of $\partial Q / \partial x$ and $\partial H / \partial x$ inside the bracket were identical, in other word if $\lambda = \pm a$, then the expressions in bracket could be written as;

$$\partial Q / \partial t = \pm a \partial Q / \partial x ; \text{ and } \partial H / \partial t = \pm a \partial H / \partial x \quad \dots\dots\dots(1.6)$$

and these are the derivative dV/dt and dH/dt on $dx/dt = \pm a$. These lines $dx/dt = \pm a$, are the characteristics lines, and the solution of these equations is accomplished by the MOC, yielding in difference notations. Use these expression Eq. 4 becomes,

$$C+; Q_p - Q_R + (g A / a) (H_p - H_R) + (f \Delta t / 2DA) Q_R |Q_R| \quad \dots\dots\dots(1.7a)$$

$$C-; Q_p - Q_S - (g A / a) (H_p - H_S) + (f \Delta t / 2DA) Q_S |Q_S| \quad \dots\dots\dots(1.7b).$$

The C+ equation is valid along a C+ characteristics line in the x – t plane;(Fig.1) given by $dx/dt=a$. The C- equation is valid only along C- characteristics line $dx/dt = -a$.

Equation 7a can be written as

$$Q_p = C_p - C_a H_p \quad \dots\dots\dots(1.8a)$$

and equ.(1. 7b) as;

$$Q_p = C_n + C_a H_p \quad \dots\dots\dots (1.8b)$$

Equations(1.8a) and(1.8b) gives;

$$Q_p = 0.5 (C_p + C_n) \quad \dots\dots\dots(1.8c)$$

In which

$$\left. \begin{aligned} C_p &= Q_R + (g A / a) H_R - (f \Delta t / 2DA) Q_R |Q_R| \\ C_n &= Q_S - (g A / a) H_S - (f \Delta t / 2DA) Q_S |Q_S| \\ C_a &= g A / a \end{aligned} \right\} \quad \dots\dots\dots(1.9)$$

Calculation of unknown values of H (x, t) and Q (x, t) are performed for a limited number of points (nodes) on the x-t plane, called the grid.

The Method Of Characteristics

In order to calculate value of function H (x, t), Q (x, t) in internal nodes of the grid, finite difference equation are as follow:

The Method Of Characteristic (MOC) takes advantage from the fact that all variation in pressure head and flow rate move with the same velocity as the wave a, and that all these variation in principle are connected according to the Joukowsky equation, which is extended incorporate friction. As shown in figure 1 the calculation take place along characteristics in x-t plane.

The solution to a problem of liquid transients in pipeline usually begins with steady state conditions at time zero, so that H and Q along the pipe are known initial values at each commutation section at (t=0) (Estuti and Streeter1993).

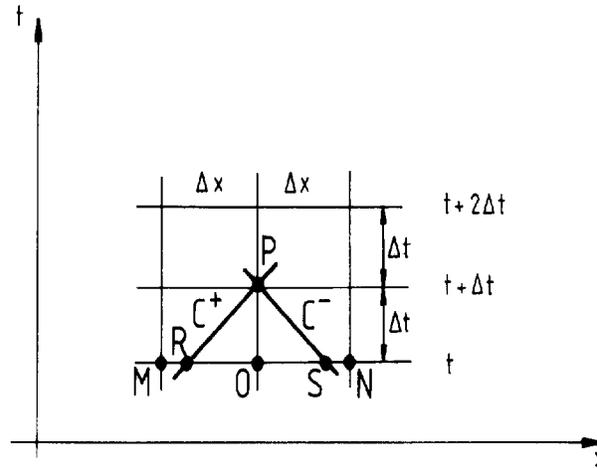


Figure 1 Basic element of the grid for an internal node .

The criteria for convergence indicate that the most accurate and stability criteria for finite difference equations solution are obtained if $\Delta t/\Delta x \leq 1/a$ is satisfied. This is called Courant's stability condition. If the time interval, Δt is such that the segment length for any condition in the system is not equal to $(a \Delta t)$, then Δx must be greater than $(a \Delta t)$.

The x-t grid is selected so that the characteristics through the point P pass through the points R and S , and not through the points M and N. According to Evangelisti 1969, a time interval of 1/16 to 1/24 of the transient time, should give sufficiently accurate results. Streeter and Lai in their pioneer 1983 and Streeter and Wylie 1993 proposed an interpolation procedure for computing R and S for the known conditions M,O and N. Repeating this for all points at time $t+\Delta t$ allow one to march forward in time.

$$\left. \begin{aligned} Q_R &= Q_O + G.a (Q_M - Q_O) \\ Q_S &= Q_O + G.a (Q_N - Q_O) \\ H_R &= H_O + G.a (H_M - H_O) \\ H_S &= H_O + G.a (H_N - H_O) \end{aligned} \right\} \dots\dots\dots(1.10)$$

where index M,N,O,P,R,S are related to nodal points of the grid, and should obtain values interpolated from function Q and H; and $G = \Delta t/\Delta x$. With quantity given in Eq.10 determined, by using Eqs. (1.7a,b) the unknown conditions at point P, which is the next time step, can be found.

The two characteristics equations for the relationships between change in head can be found.

$$H_P = 0.5 (H_R + H_S) + (a / g A) (Q_R - Q_S) - (a / g A^2) (f \Delta t / 2D) (Q_R |Q_R| - Q_S |Q_S|) \dots\dots\dots (1.11)$$

$$Q_p = 0.5(Q_R - Q_S) + (g A/a) (H_R - H_S) - (f \Delta t / 2DA) (Q_R |Q_R| - Q_S |Q_S|) \dots\dots(1.12)$$

The governing equations are solved with the assumption of friction factor at every time step by Barr's explicit friction factor which is a suitable combination of the Poiseuille equation and the Colebrook-white function to cover the full range of flow conditions from laminar to turbulent(Mimi Das Saikia 2006) . The friction factor in the above equations is replaced by the following Barr's explicit approximation.

$$1/\sqrt{f} = -2 \log_{10} [5.02 \log_{10} \{ Re / 4.518 \log_{10}(Re/7) \} / \{ 1+(Re)^{0.52}/29 (D+\epsilon)^{0.7} \} + 1/3.7(D/\epsilon)] \dots\dots\dots (1.13)$$

Where, D; diameter of conduit (pipe), ϵ ; pipe roughness size, Re; Reynolds number = (VD/ν) , in which ν is the kinematic viscosity of the fluid. Equations (1.7, 1.11 and 1.12) are nucleus in most computer programs for transient, although the equation is more advanced program can be slightly more complicated (see for example Streeter 1983 and Casey 1992). Equations (1.7a) and (1.7b) are used together with boundary conditions. Whereas equations (1.11) and (1.12) concern the internal points [Fig.1].

Boundary Conditions

When analysis a hydraulic system to determine pressure and flow rate variations its necessary to break the system down into the component parts. One can consider the various pipes having a uniform properties and separated from each other and the surrounding by reservoir, pump, valve junctionetc. These terminal points are known as boundaries and the variation of head and flow rate at the points are known as boundary conditions. Knowledge of boundary conditions and the way in which they vary with the time is important. Since boundary conditions from system to another ,although the basic equations remain the same , this solution differ from system to another Chaudhry (1987).

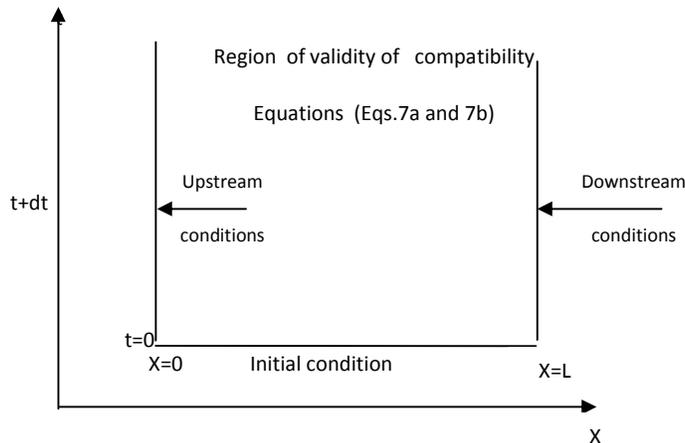


Figure 2 Region of Validity of a single pipe.

For the boundary conditions at the upstream end of a C pipe, the C+ characteristics, equation (1.7b) is valid and provides one equation in two unknowns. Hence, only one external condition is needed, which may be any relation containing one or both of the unknowns. For the downstream boundary condition equation 7a is valid. The same value of Δt is used through the piping system, as this permits solution of boundary condition at common junction of the various pipes (Streeter 1993). Cause of transients when two or (more) pumping stations are on the same pipeline, a slight differential in piping rate can either cause a rise or fall in the Hydraulic grade Line (HGL) between the stations. This transient is somewhat self-adjusting with centrifugal pumps as the consequence, say, of a greater flow rate at downstream station to increase its head and to decrease the head on upstream station, thereby bringing the flow in balance. The cutting off power to station produces a large negative surge wave downstream and positive wave upstream the station which is reflected to as potential surge ($\Delta H = a V_0/2g$), which V_0 is the steady state velocity.

Valves may also be closed, or opened rapidly, under adverse condition which create unwanted transient. The initial head rise at the valve is correctly computed from the relation ($a V/2g$), however because of line packing, the head continues to rise at valve with increasing time. As a wave moves upstream, V change and become smaller and not reduced to zero. Thus at $t+\Delta t$ the head rise is ($\Delta H = a \Delta V/2g$) where $\Delta V < V_0$. At time $L/a \leq t \leq 2L/a$, negative wave reflected travels towards the valve, such that the pressure behind the wave (upstream side) is H_0 , and at time $t = 2L/a$ the reflected wave reaches the valve. In long oil pipeline the reflected wave may be small as to be determined (Robert Bea 1999). Other service conditions would include breakage of a pipe or the trapping of air in the system before starting the pumps.

As preceding the special boundary conditions are required to determine the conditions at the boundaries. These are developed by solving Eqs. 7a and/or 7b, and the conditions imposed by equ.(1.7a) is used for the downstream boundaries and (1.7b) for the upstream boundaries.

CASE STUDY :

A: Valve Positioned Within A pipe

For steady-state condition the flow rate $Q_0 = Q_{\text{initial}}$, and $H_0 = H_{\text{initial}} = H_{\text{res}}$.

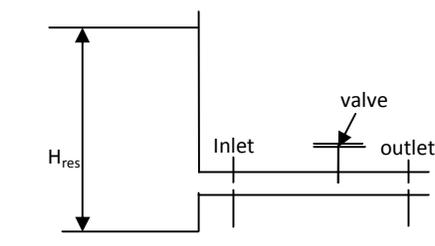


FIGURE (3)

For transient condition, the discharge (Q_p) at valve at time n can be calculated as follow(Jerry, Lescovich 1998).

For given gate valve, $C_v = 12800$ for Q in gpm and ΔP in lb/in^2 , for S.I Unit use $K_v = 0.865 C_v$ for Q in m^3/hr and ΔP in bar, or $K_v = 14.42 C_v$ for in l/min and ΔP in bar. For water,

$$Q = C_v \sqrt{\Delta P} / s \quad \dots\dots\dots (1.14)$$

For another liquids expressed ΔP as follows

$$\Delta P_1 / \Delta P_2 = (\mu_1 \rho_2 / \mu_2 \rho_1)^{1/4} \cdot \rho_1 / \rho_2 \quad \dots\dots\dots (1.15)$$

where μ is the viscosity and subscript 1 refer to liquid and 2 for water.

The flow rate at each opening ratio (Y/D) or area ratio AR is;

$$Q = e^{x/\tau} \cdot Q_0 \quad \dots\dots\dots (1.16)$$

$$x = \ln(\tau) \cdot Y/D \quad \dots\dots\dots (1.17)$$

and,

$$Q/Q_0 = \tau \sqrt{\Delta H} / \Delta \quad \dots\dots\dots (1.18)$$

where $\tau = C_v / C_{v0}$ and Y/D is the percent caudal obtained from a typical valve performance curve provided by the manufacturer.

Using these equations and eq. (1.7a) and (1.7b) to solve for Q_p and H_p .

B: Branching Junction

For the branching junction shown in figure , the following equations can be written:

Continuity equation

$$Q_{p\ i,n+1} = Q_{p\ i+1,1} + Q_{p\ i+2,1} \quad \dots\dots\dots (1.19)$$

2.Characteristic equations

$$Q_{p\ i,n+1} = C_{p\ i} - C_{a\ i} H_{p\ i,n+1} \quad \dots\dots\dots (1.20)$$

$$Q_{p\ i+1,1} = C_{n\ i+1} + C_{a\ i+1} H_{p\ i+1,1} \quad \dots\dots\dots (1.21)$$

$$Q_{p\ i+2,1} = C_{n\ i+2} + C_{a\ i+2} H_{p\ i+2,1} \quad \dots\dots\dots (1.22)$$

3.Equation for total head

$$H_{p\ i,n+1} = H_{p\ i+1,1} = H_{p\ i+2,1} \quad \dots\dots\dots (1.23)$$

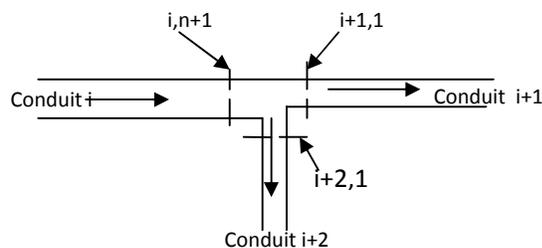


FIGURE4 Branching Junction

In equation the head losses at the junction are neglected, and it is assumed that the velocity heads in all conduits are equal. Simultaneous solution of Eqs.(1.19) Through (1.23) yields;

$$H_{P_{i,n+1}} = (C_{P_i} - C_{n_{i+1}} - C_{n_{i+2}}) / (C_{a_i} + C_{a_{i+1}} + C_{a_{i+2}}) \quad \dots\dots\dots(1.24)$$

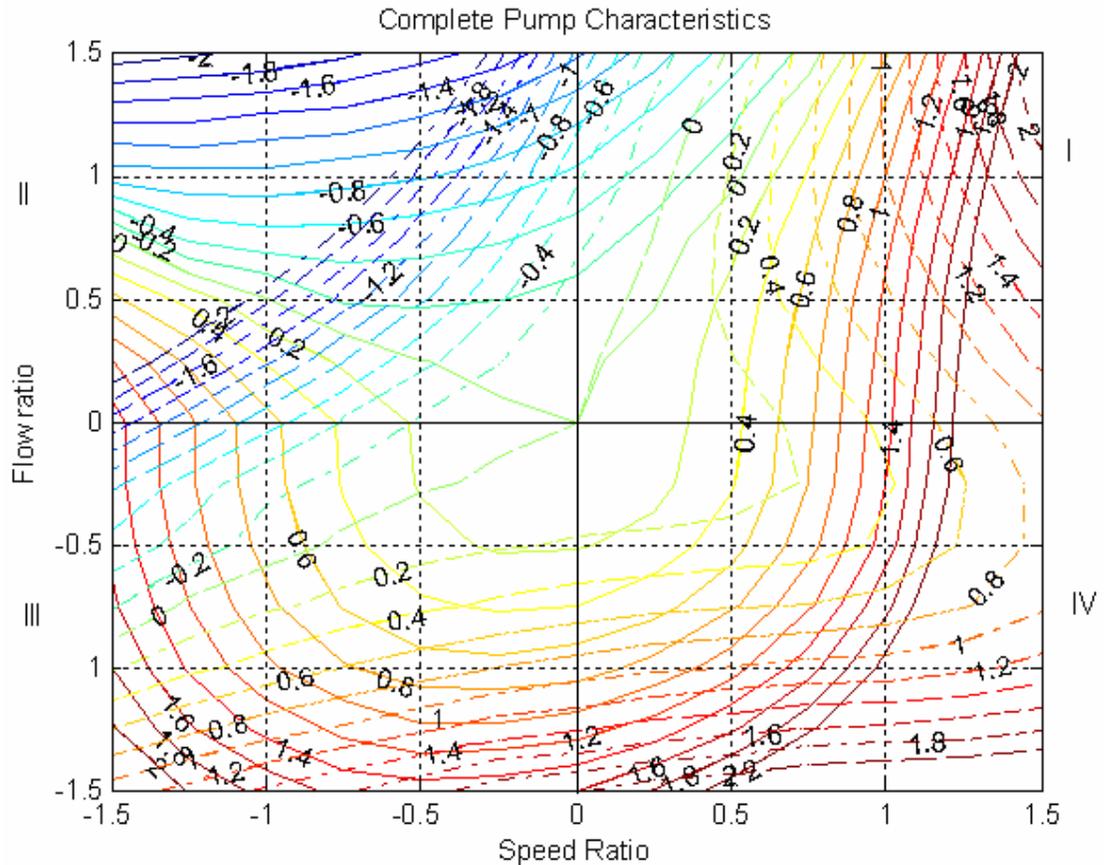
Now $H_{P_{i+1,1}}$ and $H_{P_{i+2,1}}$ can be determined from Eq.(1.24) and $Q_{P_{i,n+1}}$, $Q_{P_{i+1,1}}$ and $Q_{P_{i+2,1}}$ from equ..(1.20) through equ.(1.22).

C: Centrifugal Pump At Upstream End

Complete Pump Characteristics

If a sudden power failure happens the electric motor no longer provides any torque to the pump, and the torque on the pump is solely from the flow in the system. Although the power supply to the pump stops suddenly, the impeller does not stop rotating immediately. The inertia of the rotating parts will keep the impeller rotating for some time. Without the electrical power to keep the pump running at full speed, the pump is not able to keep the same flow rate as before the power failure. The higher head on the upstream side of the pump would cause the flow to decrease in the system and this change in flow causes a high pressure wave on the upstream side of the pump and negative pressure wave on the downstream side of the pump. The flow continues to decrease and eventually will reverse direction and flow in its natural direction. At this point the pump is still rotating in the positive direction but the flow is moving opposite to the rotation of the pump. The negative total torque on the impeller continues the negative rotational acceleration and at some time the impeller will stop rotating in the positive direction and begin rotating in the negative direction. The pump acts as a turbine and continues to accelerate in the reverse direction until the impeller is rotating at such a speed that the fluid can not apply any more torque on the impeller. At this point, equilibrium is reached and the condition is called a runaway turbine.

The pump characteristics are needed to provide the relationships between flow, head, speed and torque to solve the pump operation problem. The efficiency is a function of the other variables so it can be determined if the other values are known. Pump characteristics can normally be obtained from the manufacturer for the normal conditions. However, during transient conditions, pumps act abnormally and the characteristics of abnormal operation are needed. The first investigation of pumps operated under abnormal conditions was done by Kittredge and Thoma in 1931 where they ran a small pump under conditions of negative head, flow, and speed. A few years later, Knapp,1937 continued this investigation with a larger, more efficient pump that represented a modern installation. He used two pumps in a loop to simulate the full range of operated conditions. The results were a set of characteristic curves for all types of operation.



Figure(5). Karman – Knapp Complete Characteristics Diagram

A normal pump curve provided by a manufacturer would consist of a small part of positive-rotation head-discharge curve. Von Karman suggested that a comprehensive diagram be presented including all characteristics in a single figure (Knapp 1937). This figure, known as the Karman-Knapp circle diagram, is arranged with speed on the x-axis and flow on the y-axis dividing the figure into four quadrants, and plotting lines of constant head and constant torque.. The four quadrants are arranged in a counter-clockwise fashion. Quadrant I, has positive flow and positive rotation. Quadrant II has positive flow and negative rotation. Quadrant III has negative flow and negative rotation, and Quadrant IV has negative flow and positive rotation. In addition to combining four sets of curves into one figure, the Karman-Knapp circle diagram is useful in visualizing and understanding the physical picture as well. Many critical operation points can be obtained directly from the diagram.

Pump curves for a selected pump for normal operation can usually be found but complete characteristics are rare. The parameter which used to compare different pump models for different applications is a specific speed. So an expression for Specific speed (NS) is needed;

$$N_s = N_R \cdot (Q_R)^{0.5} / (H_R)^{0.75} \dots\dots\dots(1.25)$$

Where, in US units N_R is in Rpm, Q_R is in gpm, and H_R is in feet. For specific speed in S.I units, the input units for the equation are rad/s, m^3/s , and m.

One S.I unit of specific speed equals 2733 U.S units. Specific speed is important because it is said that pumps with the same specific speed tend to be geometrically similar and pumps of the approximately same specific speed can use

the same set of complete pump characteristics Chaudhry1987. Martin (1983) provides a summary table of pumps that have their complete characteristics documented and their references for a wide range of specific speeds Work by Benjamin Donsky (1961) examined three different pumps with different specific speeds. The pumps examined were radial flow, mixed flow, and axial flow. Radial flow pumps develop their head by centripetal force and are generally used for large heads. Axial flow pumps develop their head by the lifting of water on the vanes of the pump, and are used for lower heads. Mixed flow pumps displace water in both the radial and axial directions pumps are used for intermediate heads. Radial flow pumps correspond with lower specific speeds an axial flow pumps correspond with higher specific speeds. Donsky (1961) concluded that radial flow pumps, the lowest specific speed, cause a greater change in head and surge during a power failure event than the mixed and axial flow machines, and for design purposes choosing a lower specific speed will produce a more conservative design

Transient Pump Operation Mathematics

Transient pump starts when the start-up lasts for a shorter period than the time acceleration of fluid column in the pipe. A rough estimate for the time acceleration is the reflection time $T_f = 2L/a$. General expression is that the pipeline attains its steady state flow in less one or two time T_f . A short pipe is a pipe where the period of acceleration is considerably longer than the start-up of the pump. By definition short pipe do not cause water hammer problems. Figure(6) shows the progress of typical pump start-up at a point just downstream the pump. Point 1 is the situation just before pump begins to rotate, with non-return valve being closed and the pressure head corresponding to the geometric head. Point 2 denotes the situation just after pump has come to its full speed. This is the intersecting points between the pump characteristic and the characteristic for transient. The characteristic for transient is a straight line through point 1 and with a slope α given by Joukowsky equation ($\Delta H = \alpha \Delta V = [a/g A] \Delta Q$).

Due to friction in the pipe the head will increase slightly after pump start-up until the time marked 3, when the initial wave return from the upstream end of the pipe. The system has reached steady state and the duty point already after one or two reflection. As illustrated in figure the maximum pressure can easily be found graphically with a reasonable accuracy without use of computer programs.

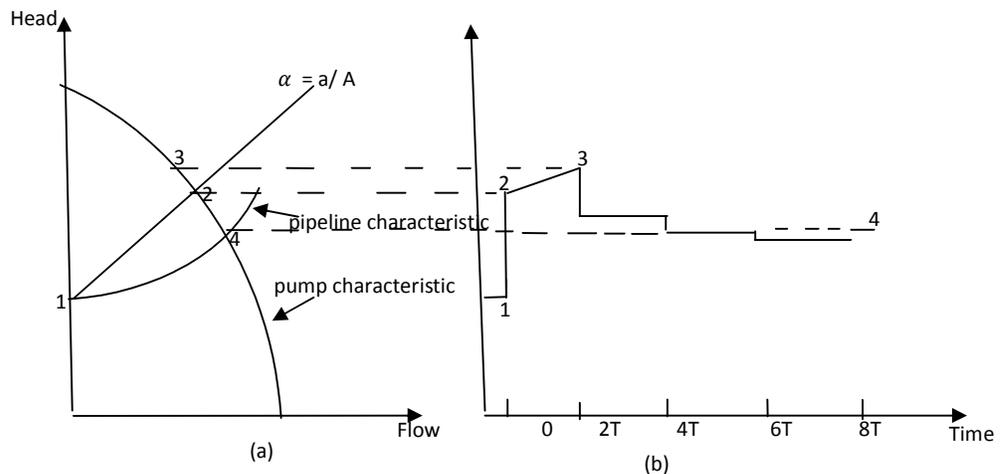


Figure 6 Water hammer at pump run-up

Transient in pump shut down are normally much more complicated and critical than at pump start-up. Most cases of pipe burst occurs at this case. For large system through investigation including computer simulation are needed. When pump shut-down the rotation speed reduce rapidly, but it is important to understand that the a stopped pump does not imply that the flow also stops. The most important is the ratio between duration of shut-down and the reflection time of pipeline (M. Niclacny).

AS shown in figure (7) situation 1 is immediately before pump shut-down , and situation 2 is just after the rotation has stopped. Again the characteristic of transient is a line through point 1 and with a slop according to Joukowsky equation.

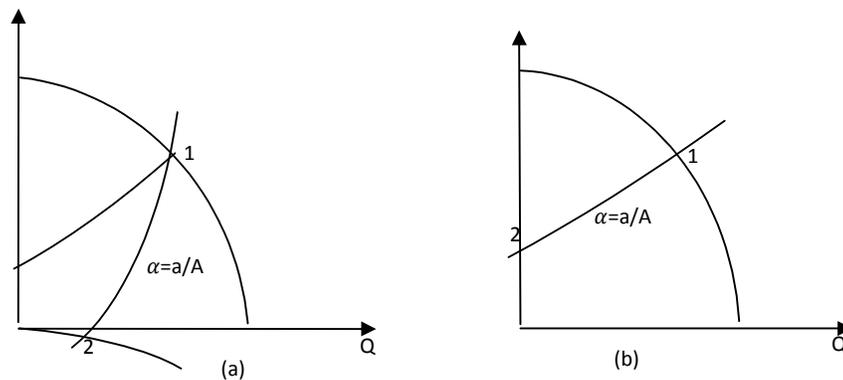


Figure 7 Transient after pump shut-down a) flow continues through pump and non return valve

The characteristic of the stopped pump is a parabola equivalent to head-loss equation for an orifice. If the intersection between the characteristic of transient and the ordinate axis shows a positive head (Fig7.b), the non -return valve needs to close to avoid backwards flow. An intersection with a negative head (Fig7.a) , Indicate that the non-return valve is still open and the flow continues through the pump.

Calculation of flow rate Q_p and pressure head H_p in a boundary node placed at the beginning of the conduit figure 8 (Marian Nielacny)fed by the pump, are performed based on equation (1.7b).

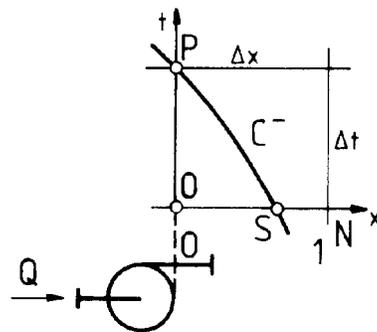


Figure 8 Grid element of boundary node next to the pump

Calculated based on the following equation

$$Q(t+\Delta t) = Q(t) \cdot N(t+\Delta t)/N(t) \quad \dots\dots\dots(1.26)$$

being similarity law of pump characteristic at rotational speed change. Decreases rotational speed of pump units for short time interval Δt is described by an exponential curve (T. Larsen 2006) in the following form;

$$N(t+\Delta t) = N(t) \cdot e^{-Gr/N \cdot \Delta t} \quad \dots\dots\dots(1.27)$$

$N(t+\Delta t)$: rotational speed at the end of time interval Δt

$N(t)$: rotational speed at the beginning of time interval t ,

$$Gr \approx dN/dt = 900 \cdot P/\pi^2 \cdot I \cdot N \quad \dots\dots\dots (1.28)$$

where P :pump power consumption(hydraulic power imparted to fluid) expressing as ;

$$P = \rho g Q H_t / \xi \quad \dots\dots\dots(1.29)$$

H_t :change in total head,

ξ :pump efficiency

I :inertia moment of pump at rotational speed, its value supplied by manufacturer. Thorley(1999) presented an empirical equations, for centrifugal pump and motor if information is not available:

$$\text{Centrifugal pumps} \quad I_p = 0.03768 [P/N]^{0.9556} \quad \dots\dots\dots(1.30)$$

$$\text{Motors} \quad I_M = 0.0043 [P/N]^{1.48} \quad \dots\dots\dots(1.31)$$

$$\text{Then,} \quad (I = I_p + I_M)$$

A rough estimate of duration of shut-down period can be found from

$$\Delta t = K.E / P = 0.5 (I\omega\xi/2\rho gQH) \quad \dots\dots\dots (1.32)$$

where $K.E=1/2I\omega^2$, is the kinetic energy from rotating of pump wheel, shaft, clutch and motor, and $(\omega=2\pi N/60)$, is the angular velocity of rotating with N in revolution per second.

Now, on the basic of present equations(1.7b),and characteristics grid it is possible to determine change in flow rate Q_p and pressure head H_p behind pump during stopping action. Chaudhry (1987) and Streeter and Wylie (1978) presented methods for solving the pump equations, however, they use an explicit scheme instead of an implicit scheme as shown here.

PART II: THE WAVE CHARACTERISTIC METHOD

The wave characteristics method(WCM) is based on the physically accurate concept that the transient pipe flow results from the generation and propagation of pressure wave that occur as a result of a disturbance in the pipe system (valve closure, pump trip,...etc). A pressure wave ,which represent a rapid pressure and associated flow changes, travel at sonic velocity for the liquid – pipe medium, and the wave is partially transmitted and reflected at all discontinuities in the pipe system

Illustrating The Wave Characteristic Method

The(WCM)consist of essentially tracking with time the propagation and attenuation of pressure wave, and calculation of their effects on the pressure and flow (in both time and space),trough out the piping system. It represents the pipe

system as connected by frictionless pipe segments, which serve only to transmit pressure waves. A pressure wave can be modified by pipe wall resistance (Wood 1966 and 2005). Temporal variations of pressures and flow rates are then computed at each point in the pipe system with time contribution of impinging pressure wave. The simulation model is therefore, reduced to a rule of an itellegement shoulder that creates, parameterize, identifies, and arranges transient event in each a chronological order during the life time of a given transient simulation. The effect of line friction (or viscous resistance) influence by pressure wave propagation and are simulated by using " orifice analogy ". They are modeled as a square-law friction orifice with properly chosen orifice coefficient. Time histories of flow rate, pressure, and pressure waves leaving each discontinuity are determined explicitly in terms of the magnitude of impinging wave action. New information at each discontinuity, and conditions prior to wave action. New information at each discontinuity is available only at discrete time intervals because all of the disturbing functions are approximately by a series of small time step change occurring at specified times in the simulation.

The following definitions are used:

Definition 1. A transient event is an internal event that take place when a pressure wave reaches a discontinuity such as valve node, pump, or any element of piping system that will cause the wave to be transmitted and reflected.

Definition 2. A transient time step (Δt) takes place between two consecutive transient events and is a time period during which pressure waves remain unchanged structurally in the simulation.

The transient time step (Δt) defines the time interval successive computations. The value of (Δt) must be very small to accurately represent all disturbing function, and all modifications of pressure, flow and pressure waves occur at an integer number of time intervals.

Component Analysis

Wave characteristics method is basically a simulation system that uses the next – transient event scheduling approach that is the simulation (clock)time is advanced to the time of the next transient event when a discontinuity is reached(in incremented time step).

The next – event approach requires the transient event have a known time at which they are to occur. When this occur, transient calculation is needed to compute the effect of pressure wave, which are generated by this action and transmitted down each of the connecting pipe.

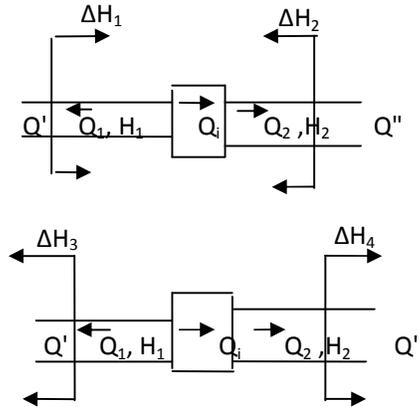
Figure(9) shows a general situation of component (discontinuity)where pressure wave ΔH_1 and ΔH_2 are impinging. At the same time, the characteristic of the component may be changed. The relation between the head and flow rate, satisfy a second order characteristic –head flow rate equation for the component having the general form

$$\Delta H = A(t) + B(t) |Q| + C(t) Q |Q| \dots\dots\dots(2.1)$$

where ΔH ; pressure head change across the component, Q ; flow rate through component, A, B and C constants for characteristic equation (may be time

dependent or can be determined at all time).The absolute value of Q are employed to make the resistance term dependent on flow direction.

This representation applies to both passive resistance elements such as valves, orifices, fittings and friction elements and active elements such as pumps.



Figure(9) Condition at component before and after wave action.

In figure (9) the subscripts 1 and 2 conditions left and right hand side of component before impinging wave arrive. The subscripts 3 and 4 denote these conditions after the pressure wave action. \$Q_1', Q_2'\$ are the pipe flow rates left of pressure wave \$\Delta H_1\$ for pipe 1 and \$\Delta H_2\$ for pipe 2, respectively.

$$\Delta H_3 = \Delta H_1 + F_1(Q_3 - Q_1) \dots\dots\dots(2.2)$$

$$\Delta H_4 = \Delta H_2 + F_2(Q_4 - Q_2) \dots\dots\dots(2.3)$$

where, \$F_1 = a_1 / g A_1\$, \$F_2 = a_2 / g A_2\$

The head \$H\$, after wave action is;

$$H_3 = H_1 + \Delta H_1 + \Delta H_3 \dots\dots\dots(2.4)$$

$$H_4 = H_2 + \Delta H_2 + \Delta H_4 \dots\dots\dots(2.5)$$

After action the pressure and rate characteristic through component is;

$$H_4 - H_3 = A(t) + B(t)|Q_0| + C(t)Q_0|Q_0| \dots\dots\dots(2.6)$$

The coefficient of characteristic equation \$A(t), B(t)\$ and \$C(t)\$ may vary with time and these represent the values at time of wave action. Equations(2-2)to(2-6) can be solved to obtain quadratic relationship for \$Q_0\$. This is;

$$C(t)Q_0|Q_0| + B(t)|Q_0| - (F_1 + F_2) + b = 0 \dots\dots\dots(2.7)$$

where,

$$b = [A(t) + H_1 + 2\Delta H_1 - H_2 - 2\Delta H_2 + (F_1 + F_2)Q_i] \dots\dots\dots(2.8)$$

Equations 2-7 may be solved by using the quadratic formula or using Newton – Raphson iterative procedure starting with initial condition (\$Q_0 = Q_i\$). The solution of equation (2-7) for the flow rate after the action, \$Q_0\$, is easily obtained

. If there is no vapor cavity present, the flows in the pipe are numerically equal to the flow through the component and,

$$Q_3 = - Q_0 \text{ and } Q_4 = Q_0 \text{ (2.9)}$$

Equations 2-2 and 2-3 are solved to compute the magnitude of pressure wave produced by the action and the equations 2-4 and 2-5 gives the pressure head after action take place

Pipe Junction Analysis

The effect of pipe junction on a pressure wave are easily handled if continuity of flow and continuity of pressure at junction are employed. This approach essentially neglects small energy losses at the junction that cause only minor effects. Figure (10) shows the condition considered.

A pressure wave of magnitude ΔH impinging in one of junction legs, it is transmitted to each of the adjoining legs. The magnitude of the transmitted wave $\Delta H_T = T_i \Delta H$ where the transmission coefficient, T_i , is given by

$$T_i = (2/F_i) / \sum (1/F_j) \text{(2-10)}$$

Where the summation refer to all legs (pipes) connecting at the junction. A reflection back in leg i occurs magnitudes $R_i \Delta H$ where (R_i is the reflection coefficient = $T_i - 1$) (Wood et al 2005).

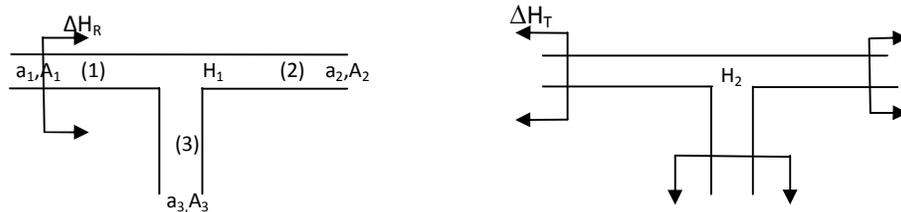


FIGURE (10) Effect of pipe junction on pressure wave

Consider a pressure wave of magnitude ΔH approaching a three pipe junction with initial flow rates $Q_1, Q_2,$ and Q_3 and an initial pressure head of H_1 (Fig. 10). The wave action at the junction result in a reflected wave of magnitude $R_i \Delta H$ in the pipe 1 and a transmitted wave of magnitude $T_i \Delta H$ down each of the remaining two pipes 2 and 3 .the new flow rates as a resultant of the wave action are Q'_1, Q'_2 and Q'_3 and the resultant pressure head at the junction is H_2 . Based on the basic transient flow relationship an expression for computing the magnitude of the transmitted and reflected wave can be written ;

$$\Delta H (1-R_1) = F_1 (Q'_1 - Q_1) \text{(2.11)}$$

$$\Delta H.T_1 = F_2 (Q'_2 - Q_2) \text{(2.12)}$$

$$\Delta H.T_1 = F_3 (Q'_3 - Q_3) \text{(2.13)}$$

$$Q_1 = Q_2 + Q_3 \text{(2.14)}$$

$$Q'_1 = Q'_2 + Q'_3 \text{(2.15)}$$

$$(Q'_1 - Q_1) = (Q'_2 - Q_2) + (Q'_3 - Q_3) \text{(2.16)}$$

Component Characteristics

The coefficient of the component characteristic equation(2-6) are determined using head / flow operating data for the component. Some components such as pump will utilize all three coefficient to represent the head / flow variations. In some cases, the characteristic equation will be based on data, which represents the head / flow relationship for a relatively small range of operation. For these applications, the coefficients used for the component analysis will be based on data valid for the operation in the vicinity of the operation point and will be recalculated as the operation point changes. This is true for the analysis of variable speed pumps for pumps using data representing a wide range of operating conditions, including abnormal situation such as flow reversal.

Many components ,such as valves can be modeled using only C coefficient. These are referred to a resistive term. For this application, the coefficient C(t) is defined as the component resistance. The term resistive is used to characterize a resistive component, and is defined as the head drop divided by the flow squared ($\Delta H/Q^2$). The component resistance is directly related to other resistive parameters such as minor loss (K_M), valve flow coefficient (C_V),and others, which characterize the head / flow characteristic of a resistive component.

Application To Resistive System

Resistive system as valve (for these element, only C coefficient represented the effect of irreversible loss is not zero).This coefficient($C=H/Q^2$). For hydraulic considerations, this type of square – law is usually appropriate, The sign of pressure head change is dependent on the direction of flow through the component. This necessitates to use absolute value of flow rate as in equations.

A – Internal Element

A-1- Pipe line leading From a Reservoir To Closing Valve:

To determine and calculate surge generated by a slow valve closer. When the valve closure initiated , the general pressure wave travel toward the reservoir, get reflected on the reservoir and reach a closure valve in $2L/a$ times. If the valve closed time (T_C), is greater than the $T = 2L / a$ seconds, this may be classified a slow valve closure. Once the pressure wave is generated, the condition at the valve remains the same until the pressure wave reflected of the reservoir reach the valve. This case, uses a computational time step of ($2L / a$) sec to demonstrate the pressure change as the valve close linearly in $T_C \gg 2L/a$. Based on initial flow rate, steady state heads across the valve, the valve initial resistance C(t) can be calculated. At the end of computational time period ($\Delta t = 2L/a$), the valve area ratio (AR) changes from 1 to 0. Then the corresponding value of resistance may be computed by using,

$$C(t = 2L / a, \dots T_C) = C(t=0)/(AR)^2 \quad \dots\dots\dots(2.17)$$

The decreased area ratio will reduce the flow rate across the valve simultaneously generating a pressure wave that travels towards the reservoir and gets reflected. The new flow rate Q across valve depends on the new head on the upstream side of valve, which in turn depends on the change in flow rate

across the valve. Equations (2.7) and (2.8) account for combined effect and, therefore, can be used to solve for the new flow rate through the valve at end of one computational time increment. Equation (2.2) then used to calculate the magnitude of resulting pressure wave. Equation (2.4) gives the new head upstream side valve. The above procedure may be repeated after updating the $C(t)$ to reflect a new area ratio (AR).

A-2 : Valve Positioned With A Pipe

Figure (2-3) shows a valve positioned with a pipe (rather than at the end). The relationship between flow rate and pressure head is known for various element;

$$\Delta H = -1/(C_d)^2 (A_0)^2 \cdot 2g Q^2 \quad \dots\dots\dots(2.18)$$

Where A_0 is the open area of the element and C_d is a discharge coefficient that is depend on the type of element. In this case, the characteristic coefficient for equation(2-1) is ,

$$C(t) = -1/(C_d)^2 (A_0)^2 \cdot 2g \quad \dots\dots\dots(2.19)$$

For element that stay in a set position, $C(t)$ remains constant but this term varies for an element such as valve that is opening and closing and depends on the open area of the valve. In each case of area ratio (AR), the data given before the action plus the values of the coefficient of the characteristic equation after the action are used to solve equation (2.7) for flow through the element after action. These are used to determine the pipe flow, pipe pressure head, and pressure wave magnitude that are presented.

A-3 : Terminal resistive Element

A general analysis applies to a situation where the resistive element such as valve is located at the end of the pipe adjacent to pressure reservoir. For this situation the values of F and ΔH for this side of the component representing the reservoir are taken as zero and the analysis is carried out in the normal fashion. In this manner, conditions on the reservoir side of the component will be not affected by the wave action that is proper influence of pressure reservoir. (The pressure head in the reservoir remains constant; $H_4 = H_2$).

B:Line Friction Element

The effect of line friction on a pressure wave is easily account for by employing an element in the analysis to represent the pressure drop due to line friction in length of pipe L . This element will attenuate a pressure wave in a manner similar to total attenuation that occurs as the wave travels the length L in the pipe. Then a pressure head change across the element corresponds to the loss in the pipe is ;

$$\Delta H = -f L Q^2 / 2g D A_L^2 \quad \dots\dots\dots(2.20)$$

where f is the friction factor. In this case the characteristic coefficient for equation (2.1) for friction element is ;

$$C(t) = -f L / 2g D A^2 \quad \dots\dots\dots(2.21)$$

Since variation in friction factor f occurs due to flow change, this coefficient can change Equ.(1.13). Normally the friction factor will be adequately determined by the flow rate through the element prior to the wave action. In most simulations, it is sufficient to treat this coefficient as a constant.

C : Active Element (Pumps)

Element that alter the fluid energy due to a mechanical means are referred as active element. The general form of the characteristic equation 2-1 was chosen to be able to represent pump operation. The characteristic $A(t)$, $B(t)$ and $C(t)$ may depend on various conditions including pump speed, pump head and other operating conditions and are explicitly determined before calculation for the effect of wave action are made. Various techniques have been suggested to represent pump characteristics. A simple expression that is useful for pump startup or shutdown is :

$$\Delta H = A_R a^2 + B_R a |Q| + C_R Q |Q| \quad \dots\dots\dots(2.22)$$

where A_R , B_R and C_R are the coefficients of a quadratic curve that represents normal pump operating at full (rated) speed and a is the speed ratio ($a = N/N_R$). For variable speed operation during a startup or shutdown, N/N_R and represent the ratio of rotational speed at any time during the transitional period, N , to the rated speed of rotation, N_R . In this manner, it is assumed that the first two terms, which represent the gross pump head developed, are function of the rotational speed. In addition, these terms are not dependent on flow direction but rather the rotational direction as determined the pump installation. The third term can be considered to represent the resistance and account for the energy loss of the pump and is taken as proportional to the square of the flow pump.

Transient pump operation is complex phenomenon that is affected by many factors. The above description does account the principle (pump speed) and offers a basic for evaluating unsteady flow in piping systems that continues pump operating in a normal manner at both constant and variable speed

RESULTS AND DISCUSSION:

The computational accuracy and performance of hydraulic transient modeling approach have not been comprehensively compared using object testing applied to system. Certainly, well-known numerical challenges in solving the transient flow equations exist, for example avoiding numerical dispersion and attenuation and eliminating unnecessary distortion of either the physical pipe system or its boundaries.

both Eulerian and Lagrangian solution schemes are commonly used to approximate the solution of governing equations. The MOC requires numerous steps or calculations to solve a typical transient-pipe flow problem. Lagrangian methods, update the hydraulic state of the system at fixed or variable time interval at times when a change actually occurs. Each approach assume that a steady-state hydraulic equilibrium solution is available that gives initial flow and pressure distribution throughout the system.

Both the MOC and the WCM obtain solutions at interval time of Δt at all junctions and components. However the MOC also requires solution at all interior points for each time step. This requirement basically handles the effect of pipe wall frictions. The WCM handles these effects by using the pressure wave characteristics. Results for pressure head and flow variation are calculated for each time step at all components and junctions in the pipe system.

A scientific Computer program (WHOP)(Water Hammer Oil Pipeline) was designed by Quick Basic Language and applied on Pentium Four with(Ram 512 MB).The(WHOP) calculated the friction factor, head H, and flow rate Q variation with variable time.

The Method Of Characteristic(MOC)

The following information are available;

$K_{oil} = 1.5 \text{ GPa}$; $E_m = 210 \text{ GPa}$; $\Psi = 1 - \Phi = 0.91$; $\Phi = 0.3$; $D = 0.672 \text{ m}$; $A = 0.456 \text{ m}^2$;
 $e = 0.0111125 \text{ m}$; $e/D = 0.01458335$; $D/e = 68.57134$; ϵ (roughness) $= 10^{-4} \text{ m}$;

$\rho_{oil} = 836.6 \text{ Kg/m}^3$; $\vartheta = 5.1 \cdot 10^6 \text{ m}^2/\text{s}^2$; $L = 19312 \text{ m}$; $Q = 0.34 \text{ m}^3/\text{s}$; $H_{suc.} = 153.3 \text{ m}$;
 $H_{Disc.} = 413 \text{ m}$; a (wave speed) $= 1130 \text{ m/s}$ [calculated].

The boundary conditions may be very complicated, but the use of the conditional statement in a computer programming make it is relatively easy to consider many possible optional boundary relations.

Calculations were carried out using the data, for a complete pump shut –down occurring over a time of 35 seconds. The most interesting object of this simulated system, by modeled the pipeline into N (node =13)equal reaches, each $\Delta x = 1500 \text{ m}$ length. A computational time period of $\Delta t = 1.458 \text{ s}$ is necessary. The N/N_R is the ratio of effective speed of pump to the rated speed. The solution to problem in oil transients are began with steady-state condition at $t = 0 \Delta t$ (table 1),so that H and Q along the pipe are known initial values of each communication section. At any time interior grid intersection point, the two compatibility equation are solved and integrated along C+ line and C- line characteristics simultaneously for the unknown Q&H.

TABLE 1.(steady-state condition $t = 0 \Delta t$)

Node:	0	1	2	3	4	5	6	7	8	9	10	11	12	13
X m:	0	1500	3000	4500	6000	7500	9000	10500	12000	13500	15000	16500	18000	19500
Q m ³ /s:	.34	.3329	.326	.3187	.3116	.3045	.2974	.2903	.2832	.276	.269	.2619	.2549	.2477
H m:	413	410.1	407.2	404.32	401.42;	398.5;	395.63;	392.74;	389.84;	386.95 ;	384 ;	381.16;	378.3;	375.4

The computations then proceeded using results obtained from the previous calculations. Results for pump trip are shown in table 2 and fig(29).

TABLE 2.(transient values just downstream the pump.)

Δt :	0	Δt	$2\Delta t$	$3\Delta t$	$4\Delta t$	$5\Delta t$	$6\Delta t$	$7\Delta t$	$8\Delta t$	$9\Delta t$	$10\Delta t$	$11\Delta t$	$12\Delta t$	$13\Delta t$	$14\Delta t$
N/N _R :	1.0		0.919		.845		0.7765		0.7137		0.656		0.603		0.554
QP:	.34	.326	.3125	.2966	.2873	.2754	.264	.253	.243	.233	.223	.214	.205	.1966	.188
Δt :	$15\Delta t$	$16\Delta t$	$17\Delta t$	$18\Delta t$	$19\Delta t$	$20\Delta t$	$21\Delta t$	$22\Delta t$	$23\Delta t$	$23.7\Delta t$	$24\Delta t$ / over				
N/N _R :		0.509		0.468		.43		0.3955		0.379					
QP:	.181	.173	.166	.159	.153	.1463	.140	.134	.129	0.00	reverse flow				

The figures from(11--14),indicate the variation of transient heads and flow rates at pump shutdown at each node along pipeline, for variable time intervals. These figures show that there is a dramatically change transient pressure and rate, depending on time and location. Also shows that H&Q just at downstream of pump are less affected by transient flow.

The variation of transient head and flow rate for valve closure are shown in figures (15—18).These figs. indicated that H&Q upstream valve varies along pipeline with time interval at all grid points. Only at nodes near to the pump the values does not affected by valve closing.

In fig.(19) the fluctuation of transient head along pipeline for pump shutdown and valve closure are drawn, indicates that the pressure fluctuating is directly proportional to the pump speed ratio(N/N_R) and valve closing ratio(AR).

The effect of pipe friction on transient H&Q along pipeline are shown in figs.(20) and(23),at pump trip and valve closure. It is clearly indicated the highly variations of results if applying the frictionless solution of characteristic equations.

Fig (24), shows the head variation for transient flow at suction and discharge side of pump after the pressure wave act on it(case C3A).The transient head up and downstream valve after wave action(case B1&B2) are shown in figs.(25) &(26)respectively. It is clear from the figs. that the maximal increase of pressure head have been serious restricted to the pump trip and valve admissible for the pipeline.

Figures(27)&(28) compare the transient results obtained using the MOC and the WCM solution approach at the pump and valve, respectively. The two methods produced results that are virtually indistinguishable(identical).

The Wave Characteristic Method(WCM)

The following cases are studied in this approach;

A. pipeline friction

Case A1:Wave act from left to right.

Case A2:Wave act from right to left.

B. Gate Valve

CaseB1:Wave act from left to right side after valve closing at variable area ratio(the wave not reached the valve.

CaseB2:Wave reflected and transmitted at the valve at variable area ratio.

C. The pump

Case C1 :Wave act from suction side only.
Case C2 : Wave act from discharge side only.
Case C3A: Change of pump speed without wave action,
Case C3B : Change of pump speed with wave acts from suction side
Case C4 :Waves act from both sides with changing of pump speed.
The results of all these cases are illustrated in table 3 bellow;

Table 3(WCM results).

Before Action						After Action				
Case	Q i	ΔH1	H1	ΔH2	H2	Qo	ΔH3	H3	ΔH4	H4
A1	.34	37.3	413	0	375.7	.436	12.5	463	24.76	400.5
A2	.34	0	413	37.3	375.7	.231	27.544	440.5	9.756	422.776
B1/AR										
1.0	.09944	0	375.6	0	376.88	.09861	1.278	376.87	- 1.278	375.6
0.9	----	---	-----	-----	-----	.0986	1.293	376.89	- 1.293	375.55
0.80984	1.6	377.1	- 1.6	375.25
0.70982	1.884	377.5	- 1.884	374.96
0.60979	2.37	378	-2.37	374.4
0.50975	3.05	378.7	-3.05	373.8
0.40966	4.33	379.93	-4.33	372.5
0.20960	5.25	381	- 5.25	371.6
0.10753	37.115	412.7	- 37.115	339.8
B2/AR										
1.0	.09944	37.3	375.6	0	376.88	.1225				
0.91225	1.28	414.7	35.5	412.3
0.81223	2.11	415	35.2	412
0.71221	2.57	415.5	34.73	411.3
0.61215	3.34	416.24	33.96	410.7
0.51208	4.42	417.3	32.88	409.63
0.41195	6.42	419.3	30.88	407.63
0.21106	20.12	433	17.18	393.94
0.108988	52.0	465	- 14.72	362.08
C1	.34	51.00	153.3	0	413	.494	10.1	214.4	40.84	453.84
C2	.34	0	153.3	37.3	413	.204	37.09	190.37	- 22.0	450.2
C3A/AR										
0.8	.34	0	153.3	0	413	.162	45.23	198.53	- 81	322
0.7087	63.933	217.33	- 63.9	349
0.60194	81	234.5	-81	332
0.4	-.0074	130.85	284.15	- 130.85	282.15
0.2	-.1778	165.9	319.2	- 165.9	297.1
C3B/AR										
0.8	.34	51.00	153.3	0	413	.3366	51.86	256.16	-0.86	412.12
0.7	.34268	69.2	273.5	- 18.2	394.8
0.6207	84.556	288.856	- 33.56	379.44
0.4115	107.77	312.07	- 56.77	356.23
0.2097	112.31	316.61	-61.31	51.64
C3 C/AR										
0.8	.34	0	153.3	37.3	413	.0468	74.09	227.4	- 36.79	413.5
0.7	-.113	114.173	267.7	- 77.17	73.13
0.6	-.125	117.5	270.8	-80.2	370.10
0.4	-.245	198.9	302.44	-111.54	338.76

Table 4 WCM results for branching System

	Before Action				After Action				
	ΔH1	H1	T1	R1	H2	ΔH4	ΔH5	ΔH6	
D1A:	375.5	375.5	0.93	.076	722.5	346.96	346.96	- 28.5	
D1B:	-----	-----	-----	-----	-----	-----	-----	-----	
D1c	-----	-----	.1397	-.8603	428	52.46	52.46	-323.14	
D2:	-----	-----	2	1	1126.5	375.5	375.5	375.5	
D3:// Δt:	0	1	2	3	4	5	6	8	9
H2 :	375.5	376	390	401.5	409	413.5	416.53	420	422.25

The ability of the WCM to accurately model pipe friction in system using just one calculation was substantiated by the virtually identical results obtained for cases studied. This accuracy held true even though pipe friction has a significant effect on the solution.

The excellent agreement between the MOC and WCM solutions for this (figs.27&28) confirms that the computed effect of wall friction is similar for the two methods.

The MOC presented in this paper, applied to estimate unsteady flow, enables consideration of different boundary conditions and analysis of transient states in hydraulic system.

CONCLUSION

The special problem arising in the analysis of transients in oil pipelines are readily handled on the digital computer by use the method of characteristic solution of the basic equations of water hammer. Since the equation go back to fundamentals, and include fluid friction, such phenomena as line packing , attenuation, pyramiding, and rarefaction are automatically taken into account through proper handling of the boundary conditions.

Both the MOC and WCM methods are capable of accuracy solving for transient pressure and flows in oil pipe system, including the effect of pipe friction.

any transient analysis is subjected to inaccuracies because of incomplete information regarding the piping system, its components and degree of skeletonization. Properly developed and calibrated models for transient analysis greatly improve the ability of oil utilities to determine adequate surge protection, strengthen

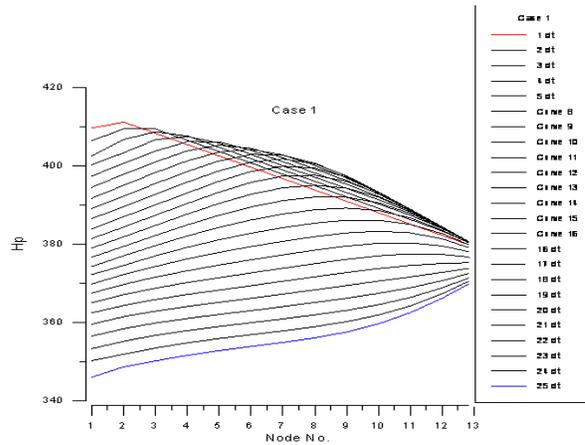


Fig.(11) Variation of head along pipe at pump shut-down (MOC)

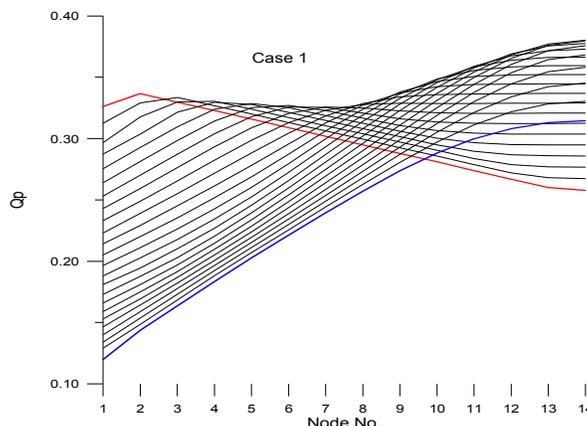


Fig.(12) variation of flow rate along pipe at pump shut-down (MOC).

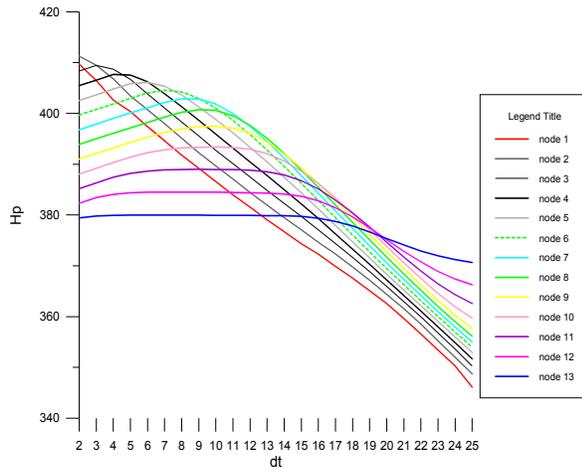


Fig.(13) Variation of head at each node for pump shut down (MOC).

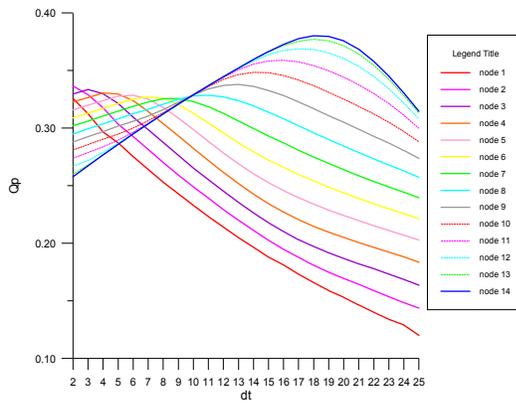


Fig.(14) Variation of flow rate at

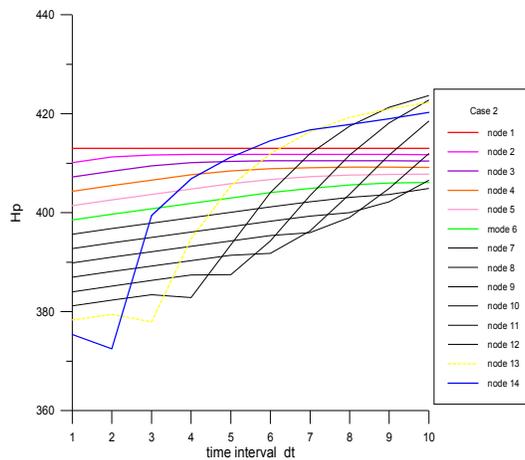


Fig.(15) Variation of head along pipe at valve closing (MOC).

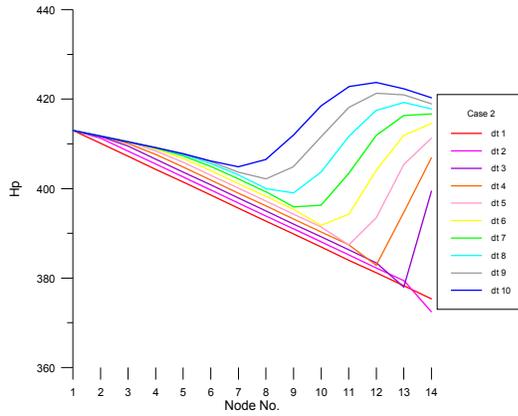


Fig.(16) Variation of head along pipe at valve closing (MOC).

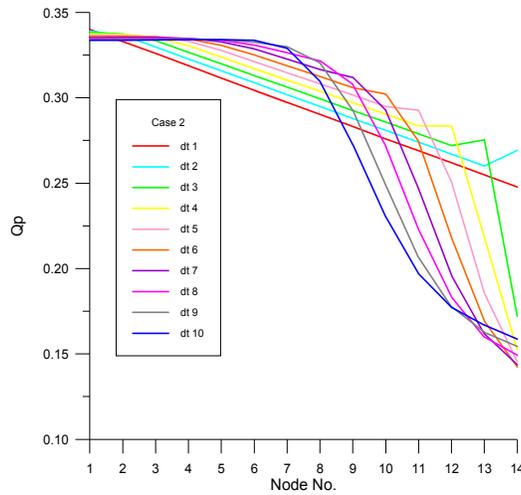


Fig.(17) Variation of flow rate along pipe at valve closing(MOC).

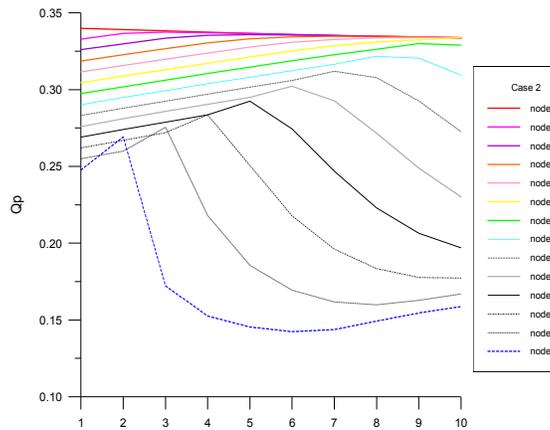


Fig.(18) Variation of flow rate at each node for valve closing(MOC)

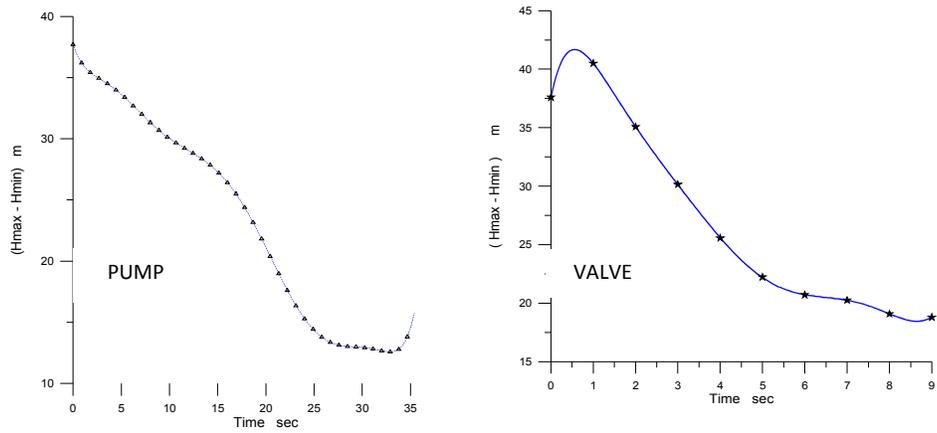


Fig.(19) Head fluctuating at various nodes for pump shut -down and valve closure.

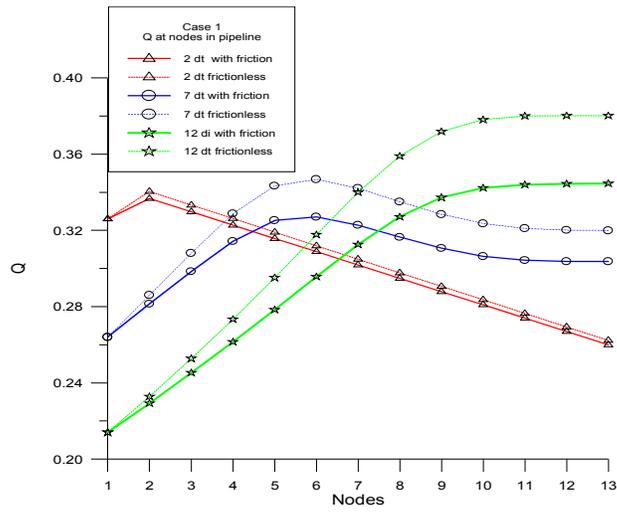


Fig.(20)Effect of pipe friction on transient flow rate (MOC, case 1)

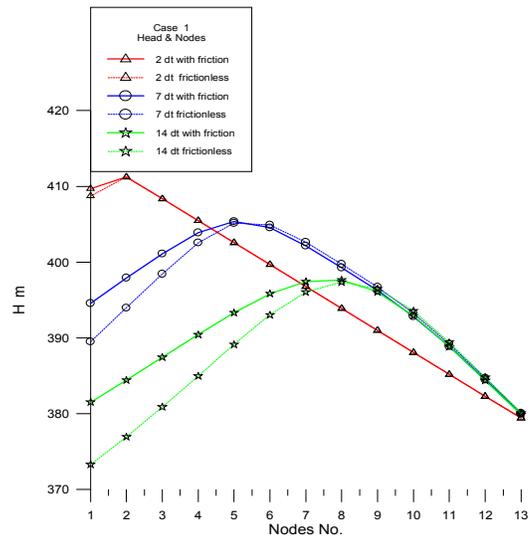


Fig.(21) Effect of pipe friction on pressure transient (MOC,case1)

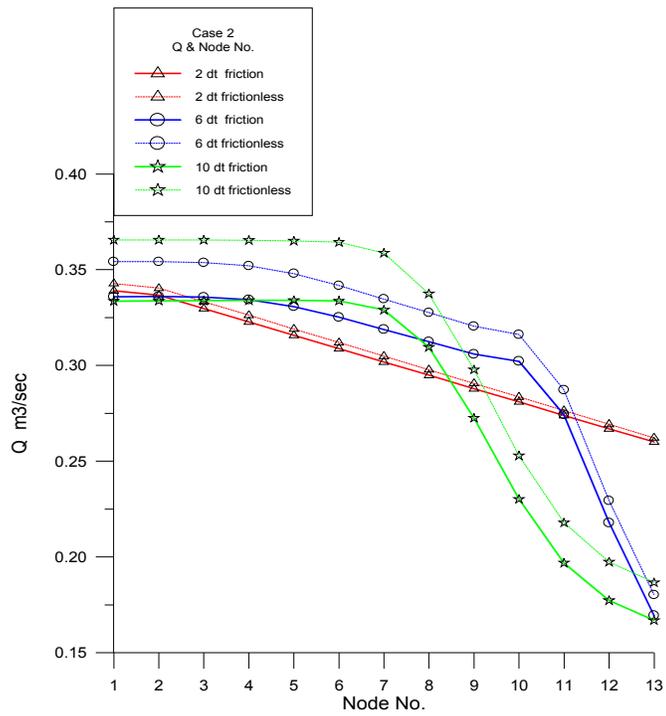


Fig.(22) Effect of pipe friction on transient flow rate (MOC,case2).

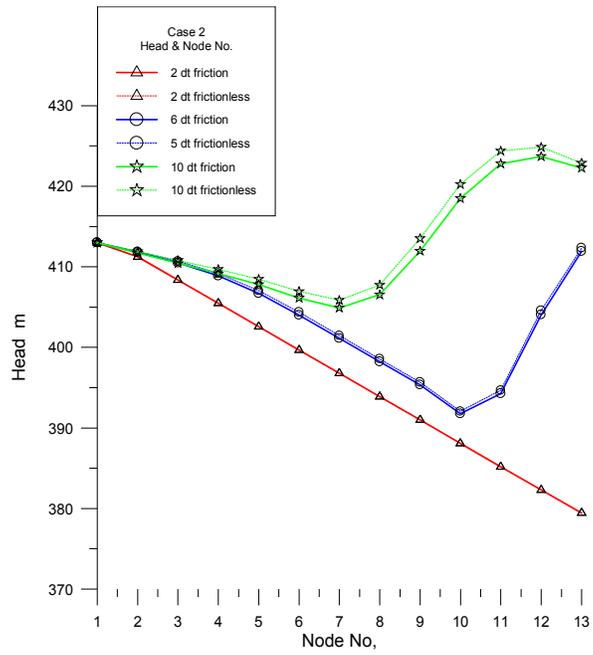


Fig.(23)Effect of pipe friction on pressure transient (MOC,case2)

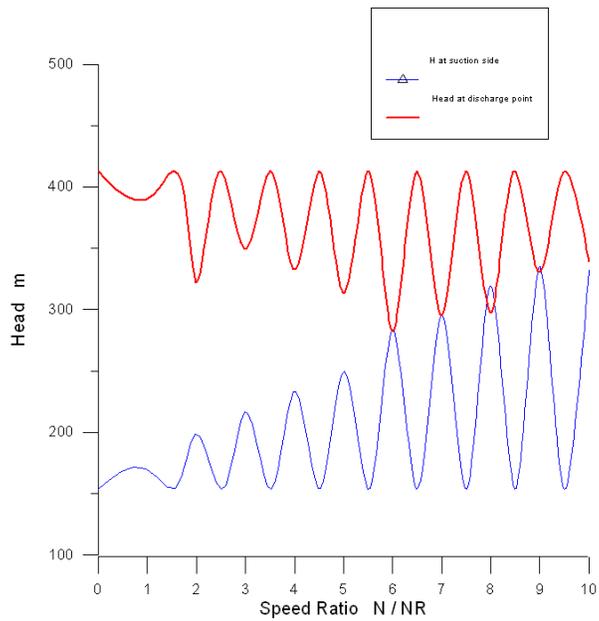


Fig.(24) Pressure head at suction and discharge side of pump after wave action.

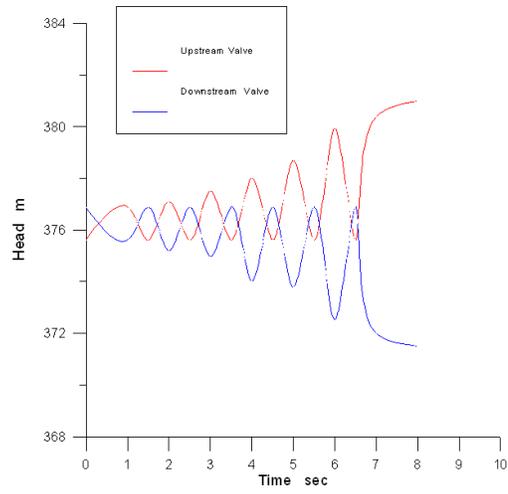


Fig.(25) Pressure variation upstream and downstream of valve after wave action (case B1)

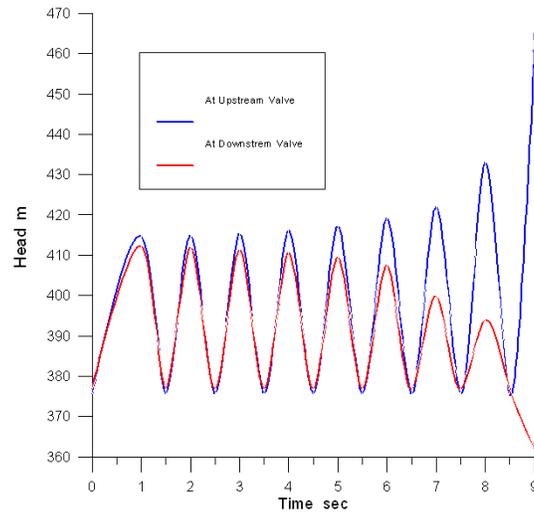


Fig.(26) pressure variation upstream and downstream of valve after wave action(case B2).

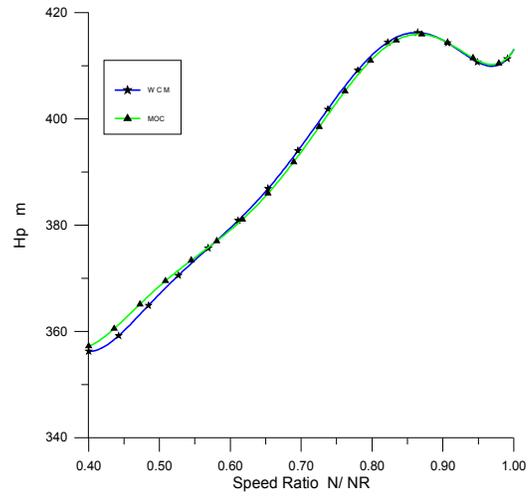


Fig.(27) Comparison of MOC and WCM for pump shut-down.

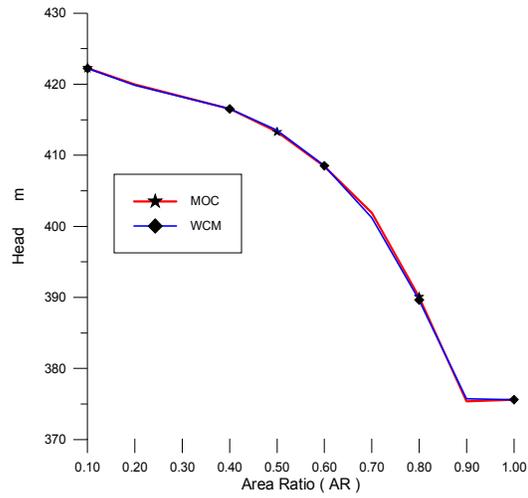


Fig.(28) Comparison of MOC and WCM for valve closure.

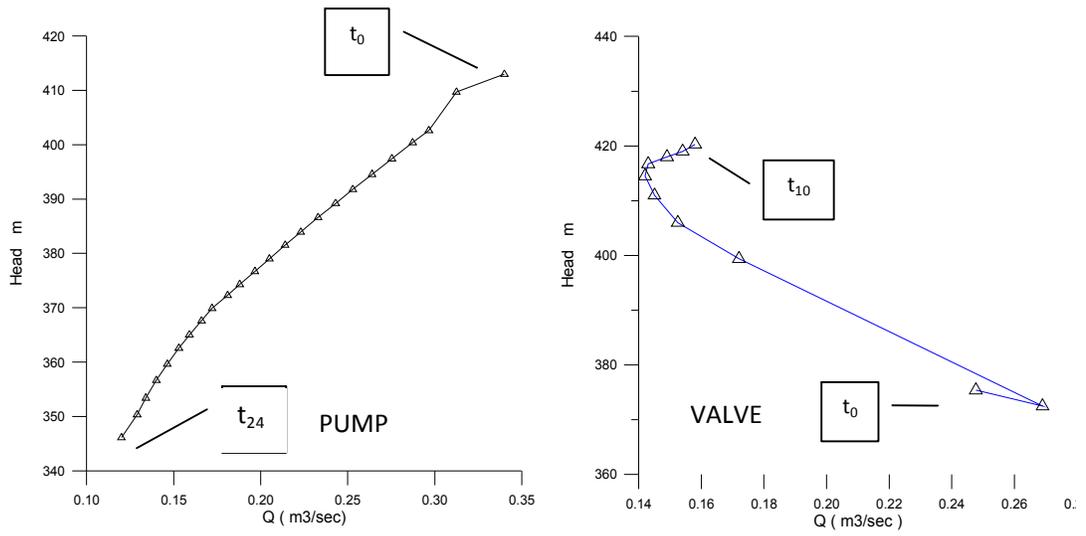


Fig.(29) pressure head characteristics for transient condition.

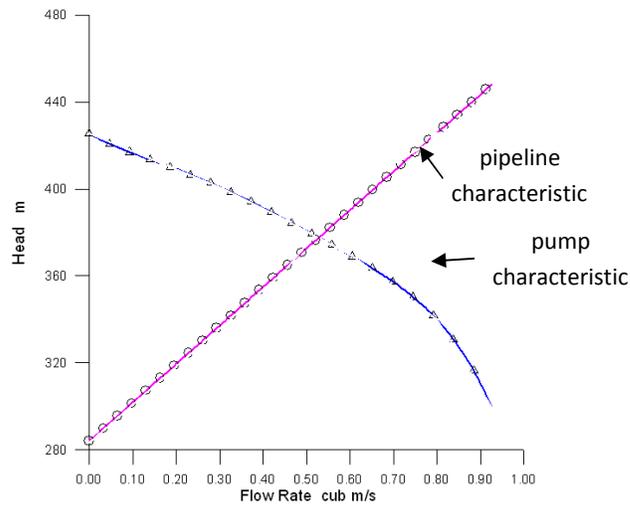


Fig.(30)Head-Discharge curve for a pump

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