Journal of Engineering and Development, Vol. 12, No. 3, September (2008) **ISSN 1813-7822**

Comparison between Oil and Water System Due to Power Failure in Pumping Station

Asst. Lect. Muhammad Asmail Eleiwi
Mechanical Engineering Department, College of Engineering
Al-Mustansiriya University, Baghdad, Iraq

Abstract

In this paper, the power failure in a pumping system has been studied for different flow (Water and Oil). The transient flow rate and head of water and oil had been calculated at different section and effect of fluid properties had been studied.

A computer program (Oil Hammer) was developed to solve the mathematical model using predictor-corrector scheme of characteristic methods.

الخـلاصـــــــة

في هذا البحث، سيتم دراسة فشل الطاقة في محطات الضخ لسوائل مختلفة (الماء والنفط).كم تم حساب معدل الجريان المضطرب و الضغط المضطرب لمقاطع مختلفة وأيضا تم دراسة تأثير خصائص السائل.

برنامج الحاسبة (Oil Hammer) طور لحل النموذج الرياضي باستخدام أسلوب (Predictor Corrector Scheme) التابع لأسلوب Characteristic.

1. Introduction

In pipelines, transporting crude oil or refined products have pumping stations. The pumping head of these stations is mainly used to overcome the friction losses in pipelines.

The analysis of transients flow in oil pipelines, sometimes called oil-hammer analysis or surge analysis, is rather complex because the pipeline friction losses are large compared to the instantaneous pressure changes caused by a sudden variation of the flow velocity [1].

The dynamics of turbulent flow is quite difficult to describe compared with the laminar flow. Many simulation models have been developed for simulating pipe flow with high viscosity liquids such as oil, presented by Leino and Linjama [2].

When the dynamic properties of fluid power systems are to be described, the dynamics of the pipelines are often of great importance. In fluid power system the propagation of pressure pulsations from pumps studied by Petter and Kenneth [3]. The effect of line dynamics in control systems to simulation of the hydraulic system used for the operation of sub sea oil/gas line valve gate.

Current knowledge on the practice of earthquake engineering for oil and gas pipeline systems is presented by Nyman [4].

The governing equations for transient flow in pressurized pipes are solved directly in the frequency domain by means of the impulse response method. The analytical expression of the piezometric head spectrum at the downstream end section of a single pipe system during transients is then derived by Marco and Burno [5].

The aim of this work is to obtain a mathematical model to find out the flow rate and pressure of flow variation along an oil pipelines and to find out the flow rate and pressure of oil pump station. A predictor-Corrector scheme of characteristic methods is used to analyze and calculate these variables.

2. Theoretical Formulation

The following assumptions are made in the derivation of the equation [1]:

1. Flow in the conduit is one-dimensional and the velocity distribution is uniform over the cross section of the conduit.
2. The conduit walls and the fluid are linearly elastic, i.e., stress is proportional to strain. This is true for most conduits such as metal, concrete and wooden pipes, and lined or unlined rock tunnel.
3. Formulas for computing the steady-state friction losses in conduits are valid during the transient state.

2-1 Dynamic Equation

The following notation: Distance, x, discharge, Q, and flow velocity, v, are considered positive in the downstream direction see **Fig.(1)**, and it is the piezometric head at the centerline of the conduit above the specific datum.

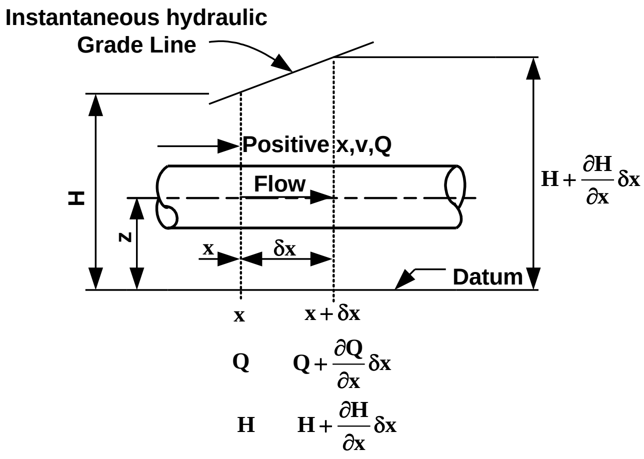


Figure (1) Notation for dynamic equation

Dynamic equation can be written as [1]:

$$\frac{\partial Q}{\partial t}+gA\frac{\partial H}{\partial x}+\frac{f}{2DA}Q|Q|=0 \qquad (1)$$

2-2 Continuity Equation

The pressure change, δp, during time interval δt is $\left(\partial p/\partial t\right)$ δt. This pressure change causes the conduit walls to expand or contact readily and causes the length of the fluid element to decrease or increase due to fluid compressibility see **Fig.(2)**.

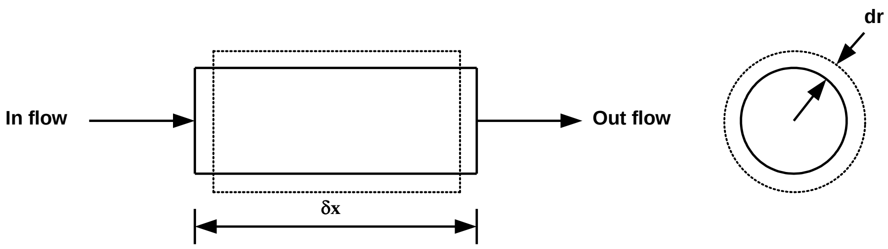


Figure (2) Notation for continuity equation

The continuity equation can be written as [1]:

$$\frac{a^2}{gA}\frac{\partial Q}{\partial x}+\frac{\partial H}{\partial t}=0 \qquad (2)$$

3. Characteristic Equations

Let we rewrite The dynamic and continuity equations Eq.(1) and Eq.(2).

$$L1 = \frac{\partial Q}{\partial t} + gA\frac{\partial H}{\partial x} + \frac{f}{2DA}Q|Q| = 0 \quad \text{.......... (3)}$$

$$L2 = a^2\frac{\partial Q}{\partial x} + gA\frac{\partial H}{\partial t} = 0 \quad \text{.......... (4)}$$

Let us consider a linear combination of Eq. (1) and Eq. (2), i.e,

$$Le = L1 + \lambda L2 \quad \text{.......... (5)}$$

Or:

$$\left(\frac{\partial Q}{\partial t} + \lambda a^2\frac{\partial Q}{\partial x}\right) + \lambda gA\left(\frac{\partial H}{\partial t} + \frac{1}{\lambda}\frac{\partial H}{\partial x}\right) + \frac{f}{2DA}Q|Q| = 0 \quad \text{.......... (6)}$$

If H=H(x,t) and Q=Q(x,t) are solutions of Eq.(3) and Eq.(4), then the total derivatives may be written as:

$$\frac{dQ}{dt} = \frac{\partial Q}{\partial t} + \frac{\partial Q}{\partial x}\frac{dx}{dt} \quad \text{.......... (7)}$$

And:

$$\frac{dH}{dt} = \frac{\partial H}{\partial t} + \frac{\partial H}{\partial x}\frac{dx}{dt} \quad \text{.......... (8)}$$

By defining the unknown multiplier λ as:

$$\frac{1}{\lambda} = \frac{dx}{dt} = \lambda a^2 \quad \text{.......... (9)}$$

And by using Eq.(7) and Eq.(8) , Eq.(6) can be written as:

$$\frac{dQ}{dt} + \frac{gA}{a}\frac{dH}{dt} + \frac{f}{2DA}Q|Q| = 0 \quad \text{.......... (10)}$$

If:

$$\frac{dx}{dt} = a \quad \text{.......... (11)}$$

And:

$$\frac{dQ}{dt} - \frac{gA}{a}\frac{dH}{dt} + \frac{f}{2DA}Q|Q| = 0 \quad \text{......(12)}$$

If:

$$\frac{dx}{dt} = -a \quad \text{......(13)}$$

Note that Eq.(10) is valid if Eq.(11) is satisfied and that Eq.(12) is valid if Eq.(13) is satisfied. In other words, by imposing the relations given by Eq.(11) and Eq.(13), we have converted the partial differential equations (Eq.(3) and Eq.(4)) into ordinary differential equations in the independent variable t.

In the x-t plane, Eq.(11) and Eq.(13) represent two straight lines having slopes ± 1/a. these are called characteristic lines. Mathematically, these lines divide the x-t plane into two regions, when a may be dominated by two different kinds of solution, i.e., the solution may be discontinuous along these lines. Physically they represent the path travels, by a disturbance. For example, a disturbance at point A (**Fig.(3)**) at time to would reach point P after time Δt.

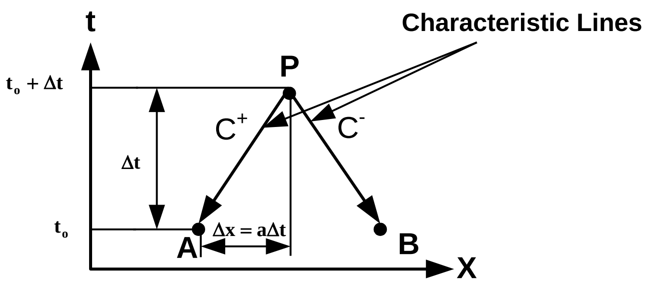


Figure (3) Characteristic lines in X-t plane

Referring to **Fig.(3)**; let the conditions at time $t=t_o$ be known. These are either initially known (i.e., at t=0, these are initial steady-state conditions) or have been calculated for the previous time step. We want to compute the unknown conditions at $t_o+\Delta t$. Referring to Fig.(3), we can write along the positive characteristic line AP.

$$dQ = Q_p - Q_A \quad \text{......(14)}$$

$$dH = H_p - H_A \quad \text{......(15)}$$

Similarly, we can write along the negative characteristic line BP.

$$dQ = Q_p - Q_B \quad \text{......} \quad (16)$$

$$dH = H_p - H_B \quad \text{......} \quad (17)$$

The subscripts in Eq.(14) through Eq.(17) refer to the locations on the X-t plane. Substituting Eq.(14) and Eq.(15) into Eq.(10) and Eq.(16) and Eq.(17) into Eq.(12), computing the friction term at the points A and B, and multiplying throughout by Δt, we obtain:

$$(Q_p - Q_A) + \frac{gA}{a}(H_p - H_A) + \frac{f\Delta t}{2DA} Q_A |Q_A| = 0 \quad \text{......} \quad (18)$$

And:

$$(Q_P - Q_B) - \frac{gA}{a}(H_P - H_B) + \frac{f\Delta t}{2DA} Q_B |Q_B| = 0 \quad \text{......} \quad (19)$$

Equation (17) can be written as:

$$Q_p = C_p - C_a H_P \quad \text{......} \quad (20)$$

And equation (18) as:

$$Q_p = C_n + C_a H_P \quad \text{......} \quad (21)$$

In which

$$C_p = Q_A + \frac{gA}{a} H_A - \frac{f\Delta t}{2DA} Q_A |Q_A| \quad \text{......} \quad (22)$$

$$C_n = Q_B - \frac{gA}{a} H_B - \frac{f\Delta t}{2DA} Q_B |Q_B| \quad \text{......} \quad (23)$$

And

$$Ca = \frac{gA}{a} \quad \text{......} \quad (24)$$

Note that Eq.(20) is valid along the positive characteristic line AP and Eq.(21) along the negative characteristic line BP.

4. Method of Analysis

The dynamic and continuity equations (Eq.(1) and Eq.(2)) describe the transient-state flows in oil pipelines. These equations can be integrated only by numerical methods since a closed-form solution is not possible because of the presence on nonlinear terms. The method of characteristic presented in Sec.3 may be used for the numerical integration of these equations.

However, as the friction losses in long oil pipelines are large compared to the potential surge, a first- order approximation of the friction term $fQ|Q|/(2DA)$, of Eq.(10) and Eq.(12).

In the Predictor-Corrector scheme, a first-order approximation is used to determine the discharge at the end of the time step. This predicted value of the discharge is then used in the corrector part to compute the friction term.

Referring to **Fig.(3)**. Let us assume that the conditions (e.g., pressure and flow) are known at time t_o and that we have to determine the unknown conditions at P.

For the predictor part, integration of Eq.(10) and Eq.(12) by using a first-order approximation yields

$$\mathbf{Q_p{}^* - Q_A + C_a(Hp^* - H_A) + RQ_A|Q_A| = 0} \quad \textbf{(25)}$$

$$\mathbf{Q_p{}^* - Q_B - C_a(H_P{}^* - H_B) + RQ_B|Q_B| = 0} \quad \textbf{(26)}$$

In which $R = f\Delta t/(2DA)$. In these equations, the notation of section (3) is used except that an asterisk is used to designate the predicted values of various variables.

Eq. (24) and Eq. (25) can be written as:

$$\mathbf{Q_P{}^* = C_P{}^* - C_a H_P{}^*} \quad \textbf{(27)}$$

$$\mathbf{Q_P{}^* = Cn^* + C_a H_P{}^*} \quad \textbf{(28)}$$

In which $C_P{}^*$ and $C_n{}^*$ are equal to the right-hand sides of Eq.(22) and Eq.(23) respectively. Elimination of $H_P{}^*$ from Eq.(27) and Eq.(28) yields:

$$\mathbf{Q_P{}^* = 0.5(C_P{}^* + C_n{}^*)} \quad \textbf{(29)}$$

Note this value of $Q_P{}^*$ may be used in the corrector part to calculate the friction term. Integration of Eq.(10) and Eq.(12), by using a second-order approximation and by using $Q_P{}^*$ for computing the friction term, yields:

$$Q_P - Q_A + C_a(H_P - H_A) + 0.5R\left(Q_A|Q_A| + Q_P^*|Q_P^*|\right) = 0 \qquad (30)$$

And:

$$Q_P - Q_B - C_a(H_P - H_B) + 0.5R\left(Q_B|Q_B| + Q_P^*|Q_P^*|\right) = 0 \qquad (31)$$

Eq.(30) and Eq.(31) may be written as:

$$Q_P = C_P - C_a H_P \qquad (32)$$

And:

$$Q_P = C_n + C_a H_p \qquad (33)$$

In which:

$$Cp = Q_A + C_a H_A - 0.5R\left(Q_A|Q_A| + Q_P^*|Q_P^*|\right) \qquad (34)$$

$$C_n = Q_B - C_a H_B - 0.5R\left(Q_B|Q_B| + Q_P^*|Q_P^*|\right) \qquad (35)$$

By eliminating H_P from Eq.(32) and Eq.(33), we obtain:

$$Q_P = 0.5(C_p + C_n) \qquad (36)$$

Now H_P may be determined from either Eq.(32) or Eq.(33).

5. Mathematical Representation of a Pump

The discharge of a centrifugal pump depends upon the rotational speed, N, and the pumping head, H; and the transient-state speed changes depend upon torque, T, and the combined moment of inertia of the pump, motor, and liquid entrained in the pump impeller.

By defining the non dimensional parameters:

$$\left.\begin{aligned} \alpha &= \frac{N_{pu}}{N_R} \\ \beta &= \frac{T_{pu}}{T_R} \\ \upsilon &= \frac{Q_{pu}}{Q_R} \\ h &= \frac{H_{pu}}{H_R} \end{aligned}\right\} \qquad (37)$$

In which the subscript "R" refers to the rated conditions.

6. Complete Characteristic Curve

The pump operation at a pumping system, before and after power failure may be divided into three zones as follows; see **Fig.(4)**:

1. Normal pumping zone, where the pump operates at normal conditions, at this zone the discharge, head, speed and torque are considered positive.
2. Energy dissipation zone, where the discharge is negative (reverse flow), and the pump speed and torque are still positive.
3. Turbine zone, where the speed, discharge and torque are negative.

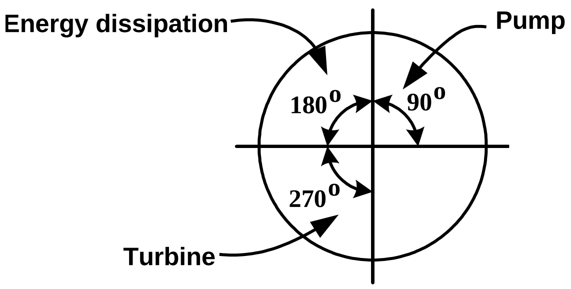


Figure (4) Zone of pump operation

7. Differential Equation of Rotating Masses

The accelerating torque for a rotational system is equal to the product of the angular acceleration and the polar moment of inertia of the system. Since there is no external torque acting on the pump following power failure, the decelerating torque is the pump torque. Hence,

$$T = -WR^2 \frac{dW}{dt} \qquad (38)$$

$$T = -WR^2 \frac{2\pi}{60} \frac{dN}{dt} \qquad (39)$$

In which WR^2 = combined moment of inertia of the pump, motor, shaft, and liquid entrained in the pump impeller.

On the basis of Eq.(37), Eq.(39) may be written as:

$$\beta = WR^2 \frac{2\pi N_R}{60T_R} \frac{d\alpha}{dt} \quad \text{.......... (40)}$$

In this Equation,

$$T_R = \frac{60\gamma H_R Q_R}{2\pi N_R \eta_P} \quad \text{.......... (41)}$$

By using an average value of β during the time step, this equation may be written in finite-difference form as:

$$\frac{\alpha_P - \alpha}{\Delta t} = \frac{60T_R}{2\pi WR^2 N_R} \frac{\beta + \beta_P}{2} \quad \text{.......... (42)}$$

Which may be simplified to:

$$\alpha_P - C_6\beta_P = \alpha + C_6\beta \quad \text{.......... (43)}$$

In which

$$C_6 = \frac{-15T_R\Delta t}{\pi WR^2 N_R} \quad \text{.......... (44)}$$

8. Characteristic Equation for Discharge Pipe

Referring to **Fig.(5)**, the following equation can be written for the total head at the pump:

$$H_P = H_{suc} + H_{pu} - \Delta H_{pv} \quad \text{.......... (45)}$$

$$\Delta H_{pv} = C_v Q_p{}^2 = C_v Q_p \left| Q_p \right| \quad \text{.......... (46)}$$

In which H_{suc} =height of the liquid surface in the suction reservoir above datum, H_{pu} =Pumping head at the end of the time step, C_v =Coefficient of head losses, and H_{pv} =head loss in the discharge valve.

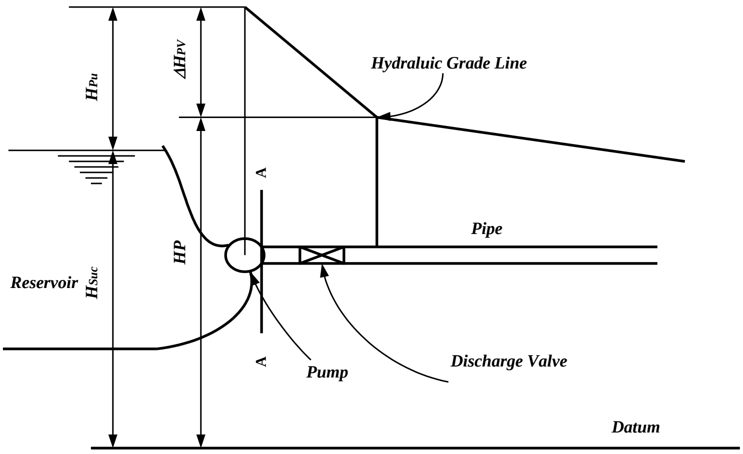


Figure (5) Centrifugal pump

As the suction line is short, it may be neglected in the analysis. Therefore, we need only the characteristic equation for the discharge line, i.e., for section A-A.

$$\mathbf{Q_P = C_n + C_a H_P} \quad \text{.......... (47)}$$

Since there is no storage between the suction reservoir and the section A-A, then the continuity equation can be written as:

$$\mathbf{Q_P = n_p Q_{pu}} \quad \text{.......... (48)}$$

In which n_p =Number of parallel pumps, and Q_{pu}: Flow rate of pump.

9. Solution of Pump Equations

Now, assuming that the points corresponding to α_P, υ_P, h_P, and β_P lie on these straight lines, then:

$$\frac{h_P}{\alpha_P^2 + \upsilon_P^2} = a_1 + a_2 \tan^{-1}\frac{\alpha_P}{\upsilon_P} \quad \text{.......... (49)}$$

$$\frac{\beta_P}{\alpha_P^2 + \upsilon_P^2} = a_3 + a_4 \tan^{-1}\frac{\alpha_P}{\upsilon_P} \quad \text{.......... (50)}$$

In which a_1, a2, a3 and a_4 are constants for the straight lines representing the head and torque characteristics, respectively.

10. Boundary Conditions

Since the oil pumping station contains of upstream and downstream tanks for the boundary conditions at upstream and downstream introduced Eq.(32) and Eq.(33) are used to determine the conditions at the interior sections. Eq.(32) is used for the downstream boundaries and Eq.(33) for the upstream boundaries.

10-1 Centrifugal Pump

To develop the boundary conditions, we have to solve Eq.(43), Eq.(45), Eq.(46), Eq.(47), Eq.(48), Eq.(49), and Eq.(50) simultaneously. By eliminating H_P, ΔH_{pv}, and Q_P from Eq.(45), Eq.(46), Eq.(47), and Eq.(48) and by using Q_R and H_R as reference values, the resulting equation may be written as:

$$\mathbf{npQ_R\upsilon_P = C_n + C_aH_{suc} + C_aH_Rh_p - C_aC_vQ_R{}^2\upsilon_P|\upsilon_P|} \quad \textbf{(51)}$$

By substituting for h_P from Eq.(49) into Eq.(51) and for β_P from Eq.(50) into Eq.(43) and simplifying, we obtain:

$$\mathbf{F_1 = C_aH_Ra_1(\alpha_P{}^2 + \upsilon_P{}^2) + C_aH_Ra_2(\alpha_P{}^2 + \upsilon_P{}^2)\tan^{-1}\frac{\alpha_P}{\upsilon_P} - npQ_R\upsilon_P - C_aC_vQ_R{}^2\upsilon_P|\upsilon_P| + C_n + C_aH_{suc} = 0} \quad \textbf{(52)}$$

$$\mathbf{F_2 = \alpha_P - C_6a_3(\alpha_P{}^2 + \upsilon_P{}^2) - C_6a_4(\alpha_P{}^2 + \upsilon_P{}^2)\tan^{-1}\frac{\alpha_P}{\upsilon_P} - \alpha - C_6\beta = 0} \quad \textbf{(53)}$$

Then a better estimate of the solution of Eq.(52) and Eq.(53) is:

$$\mathbf{\alpha_P{}^{(2)} = \alpha_P{}^{(1)} + \delta_{\alpha p}} \quad \textbf{(54)}$$

$$\mathbf{\upsilon_P{}^{(2)} = \upsilon_P{}^{(1)} + \delta_{\upsilon p}} \quad \textbf{(55)}$$

To calculate $\alpha_P{}^{(2)}$ and $\upsilon_P{}^{(2)}$ are found in [1].

$$\mathbf{\delta_{\alpha p} = \frac{F_2\dfrac{\partial F_1}{\partial \upsilon_P} - F_1\dfrac{\partial F_2}{\partial \upsilon_P}}{\dfrac{\partial F_1}{\partial \alpha_P}\dfrac{\partial F_2}{\partial \upsilon_P} - \dfrac{\partial F_1}{\partial \upsilon_P}\dfrac{\partial F_2}{\partial \alpha_P}}} \quad \textbf{(56)}$$

$$\delta \upsilon_P = \frac{F_2 \frac{\partial F_1}{\partial \alpha_P} - F_1 \frac{\partial F_2}{\partial \alpha_P}}{\frac{\partial F_1}{\partial \upsilon_P}\frac{\partial F_2}{\partial \alpha_P} - \frac{\partial F_1}{\partial \alpha_P}\frac{\partial F_2}{\partial \upsilon_P}} \quad \text{.. (57)}$$

Differentiation of Eq.(52) and Eq.(53) yields the following expressions for these derivatives:

$$\frac{\partial F_1}{\partial \alpha_P} = C_a H_R \left(2a_1\alpha_P + a_2\upsilon_P + 2a_2\alpha_P \tan^{-1}\frac{\alpha_P}{\upsilon P} \right) \quad \text{.................................... (58)}$$

$$\frac{\partial F_1}{\partial \upsilon_P} = C_a H_R \left(2a_1\upsilon_P - a_2\alpha_P + 2a_2\upsilon_P \tan^{-1}\frac{\alpha_P}{\upsilon P} \right) - n_p Q_R - 2C_a C_v Q_R^2 |\upsilon_P| \quad \text{...... (59)}$$

$$\frac{\partial F_2}{\partial \alpha_P} = 1 - C_6 \left(2a_3\alpha_P + a_4\upsilon_P + 2a_4\alpha_P \tan^{-1}\frac{\alpha_P}{\upsilon P} \right) \quad \text{................................ (60)}$$

$$\frac{\partial F_2}{\partial \upsilon_P} = C_6 \left(-2a_3\upsilon_P + a_4\alpha_P - 2a_4\upsilon_P \tan^{-1}\frac{\alpha_P}{\upsilon P} \right) \quad \text{.................................... (61)}$$

If $|\delta\alpha_P|$ and $|\delta\upsilon_P|$ are less than a specified tolerance (e.g., 0.001). Then $\alpha_P^{(2)}$ and $\upsilon_P^{(2)}$ are solutions of Eq.(52) and Eq.(53). Otherwise, $\alpha_P^{(1)}$ and $\upsilon_P^{(1)}$ are assumed equal to $\alpha_P^{(2)}$ and $\upsilon_P^{(2)}$, and the above procedure is repeated until a solution is obtained.

Note: The boundary conditions derived in section 10.1 are first to used to compute Q_P^* for the predictor part at Eq.(48). Then this value of Q_P^* is used to compute Cnn from Eq.(35) and the same boundary conditions are used again to determine Q_P and H_P in the predictor part before proceeding to the next time step.

10-2 Constant-Head Reservoir at Downstream End

If the entrance losses as well as the velocity head are negligible **Fig.(6)**, then:

$$H_p = H_{res} \quad \text{.. (62)}$$

In which H_{res} =height of the reservoir water surface above the datum, Eq.(21) for the upper and thus becomes:

$$Q_P = Cn + C_a H_{res} \quad \text{.. (63)}$$

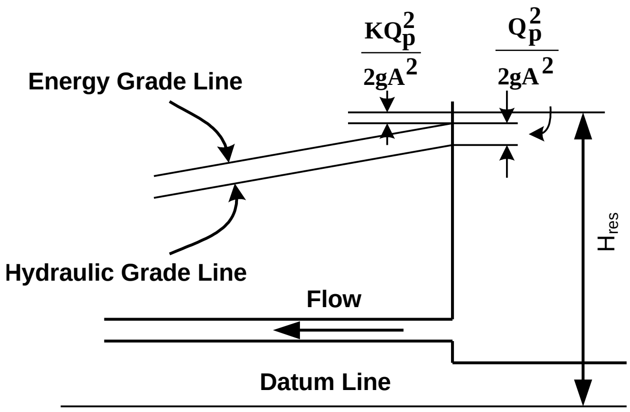


Figure (6) Constant-Level downstream reservoir

Note: The boundary conditions derived in section 10.2 are first to used to Q_P^* for the predictor part at Eq.(63). Then, this value of Q_P^* is used to compute C_{PP} from Eq.(34) and the same boundary conditions are used again to determine Q_P and H_P in the predictor part before proceeding to the next time step.

11. Stability Conditions

The convergence criterion for the finite-difference equations (30 and 31) are given by the expression:

$$\frac{\Delta T}{\Delta x} \leq \frac{1}{a} \quad \text{......} \quad (64)$$

This is called Courant's stability condition. The adjusted pressure wave speed may be written as:

$$a_{adj} = a(1+\psi) \quad \text{......} \quad (65)$$

where:

a_{adj} :is the adjusted wave (m/Sec)

ψ: is a permissible variation in the pressure wave speed which is less than a maximum limit of "$\pm15\%$" of the wave speed [1]:

Utilizing the concept, Eq.(64) may be written as:

$$\Delta T \leq \frac{L_i}{a_i(1+\psi)_i N_i} \quad (i=1, 2,\ldots m) \quad \text{......} \quad (66)$$

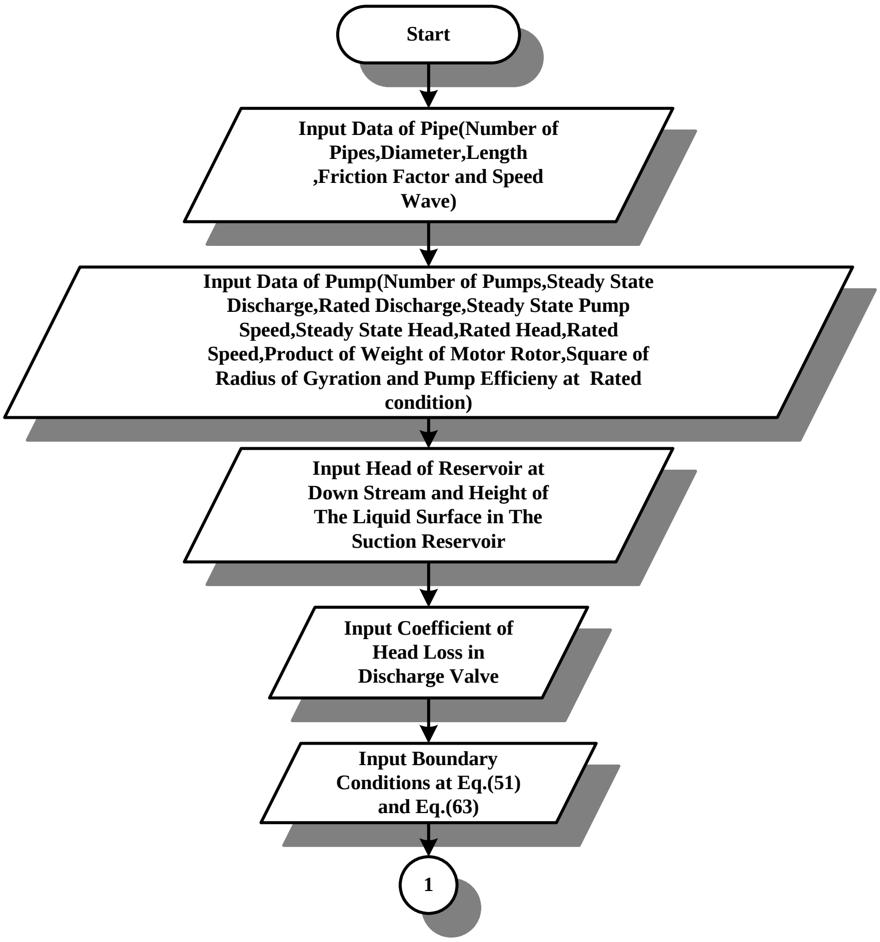
where:

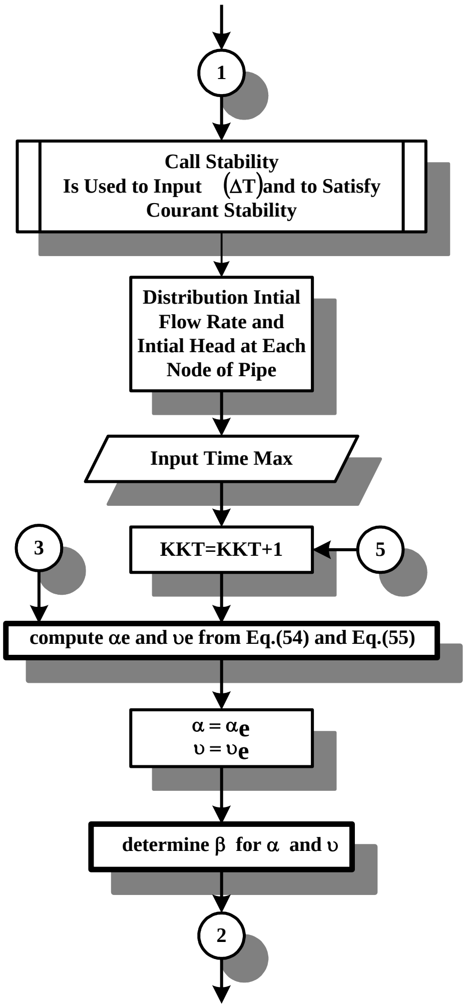
N: number of reaches

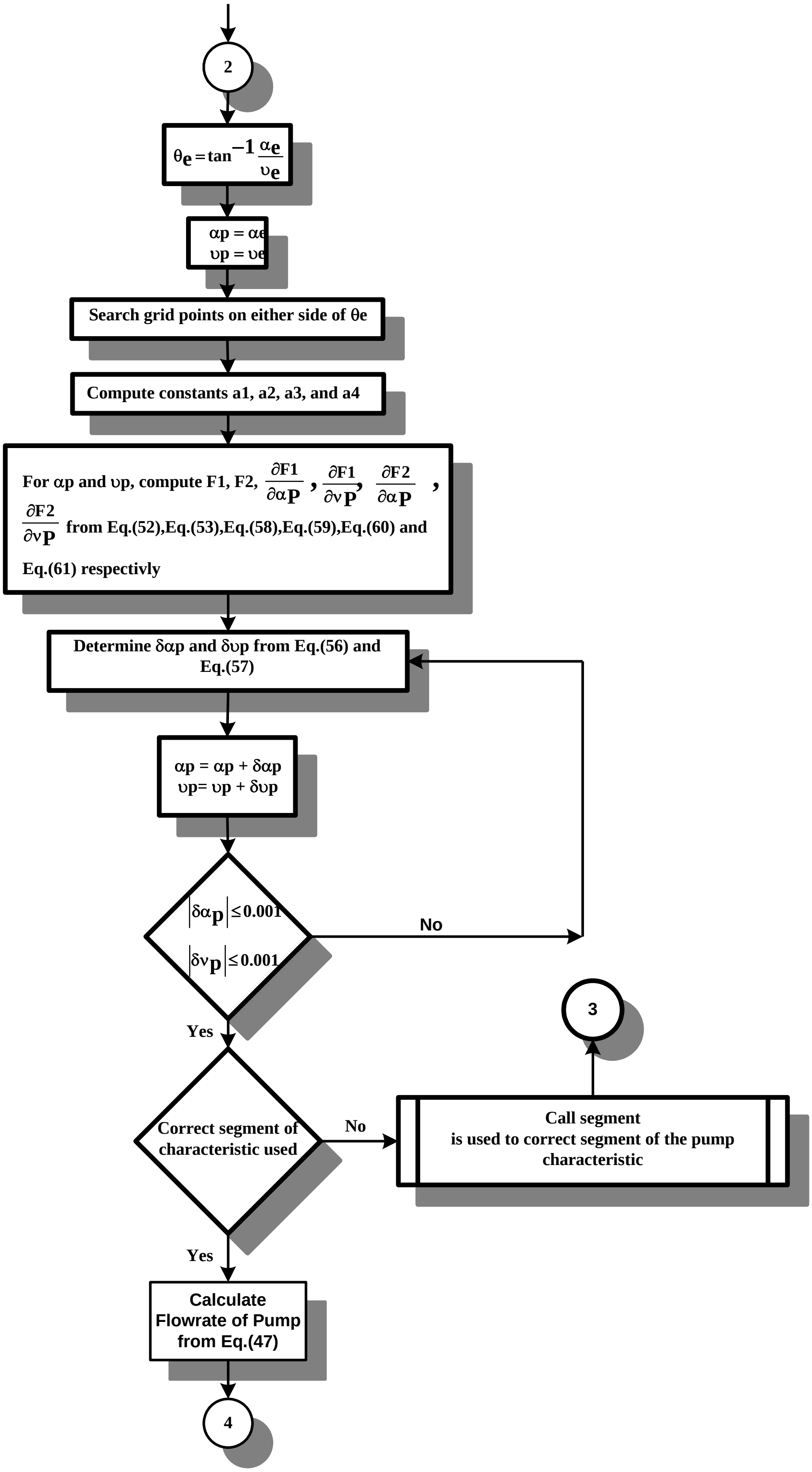
m: pipes number in the system

12. Computer Program

By using (Q-Basic) language, a computer program, which named (program Oil Hammer), is developed to solve numerical one-dimensional linear partial differential equations, unsteady state with a predictor-corrector scheme of characteristic methods. The flow chart of the computer program is shown in **Fig.(7)**.




1
Call Stability
Is Used to Input (ΔT)and to Satisfy
Courant Stability
Distribution Intial
Flow Rate and
Intial Head at Each
Node of Pipe
Input Time Max
3
KKT=KKT+1
5
compute αe and υe from Eq.(54) and Eq.(55)
α = αe
υ = υe
determine β for α and υ
2


2
θe = tan⁻¹ αe/υe
αp = αe
υp = υe
Search grid points on either side of θe
Compute constants a1, a2, a3, and a4
For αp and υp, compute F1, F2, ∂F1/∂αP, ∂F1/∂vP, ∂F2/∂αP, ∂F2/∂vP from Eq.(52),Eq.(53),Eq.(58),Eq.(59),Eq.(60) and Eq.(61) respectivly
Determine δαp and δυp from Eq.(56) and Eq.(57)
αp = αp + δαp
υp= υp + δυp
|δαp| ≤ 0.001
|δvp| ≤ 0.001
No
Yes
Correct segment of characteristic used
No
Call segment is used to correct segment of the pump characteristic
3
Yes
Calculate Flowrate of Pump from Eq.(47)
4

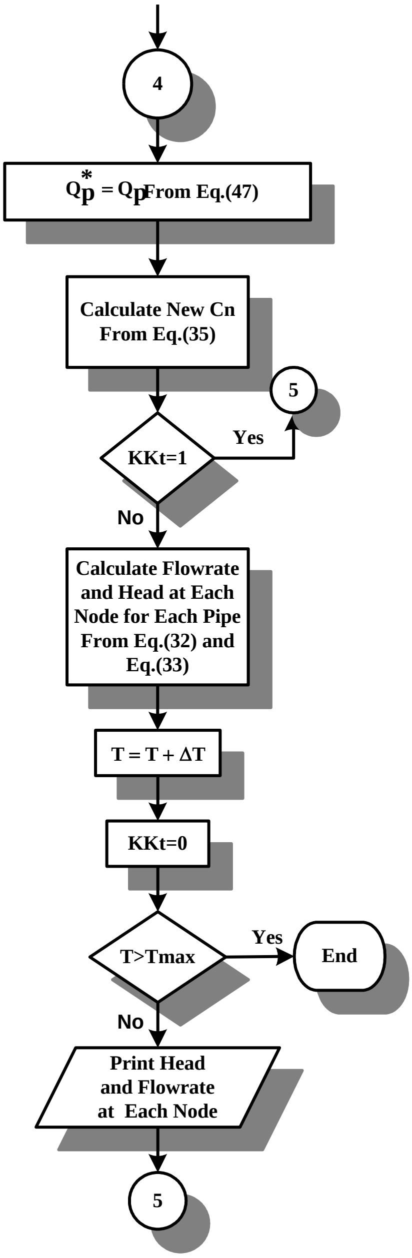


Figure (7) Flow chart of computer program oil hammer

13. Case Study

To determine transient conditions in a pumping system. **Figure (8)** illustrated the dimensions which are as follows:

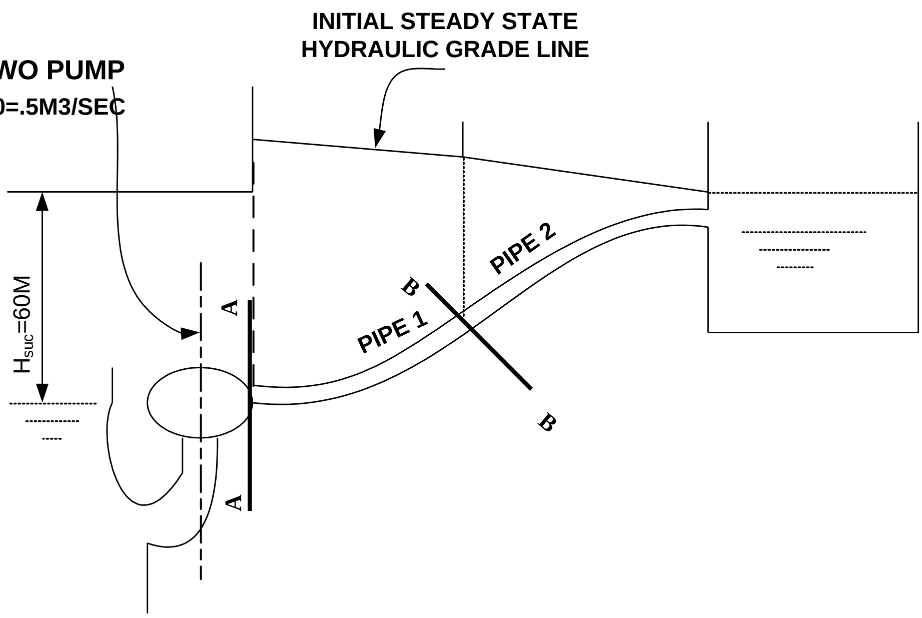


Figure (8) Parallel pump and reservoir at downstream end of the last pipe

Data of the Pump

-Pump Number=2

-Rated Discharge=.25 m^3/Sec

-Rated Head=60m

-Product of Weight Motor Rotor & Square of Radius of Gyration=16.85Kg.m^2

-Efficiency of Pump=.84

Data of the Pipe 1

-Length=450m

-Diameter=.75M

-Speed Wave=900m/Sec for Water & 1291m/Sec for Oil

-Friction Factor=.01 for Water & .037 for Oil

Data of the Pipe 2

-Length=550m

-Diameter=.75m

-Speed Wave=1100m/Sec for Water & 1291m/Sec for Oil

-Friction Factor=.012 for Water & .037 for Oil

14. Result and Discussion

The effect of physical properties of oil compared with water during the power failure in a pumping system shown in **Fig.(9)**, **Fig.(10)**, **Fig.(11)** and **Fig.(12)**. The flow rate and the head at difference section (A-A) and section (B-B).

Figure (9) show the relation between the flow rate and the time for both water and oil. In the water the flow rate will decrease and change is direction (Negative) after (three Sec) and teaks time to become stable while the oil change is direction (Negative) after (3.2 Sec) and becomes stable after (11 Sec) this is due to the effect of viscosity where the kinematic viscosity of oil is 8 times the kinematic viscosity of water.

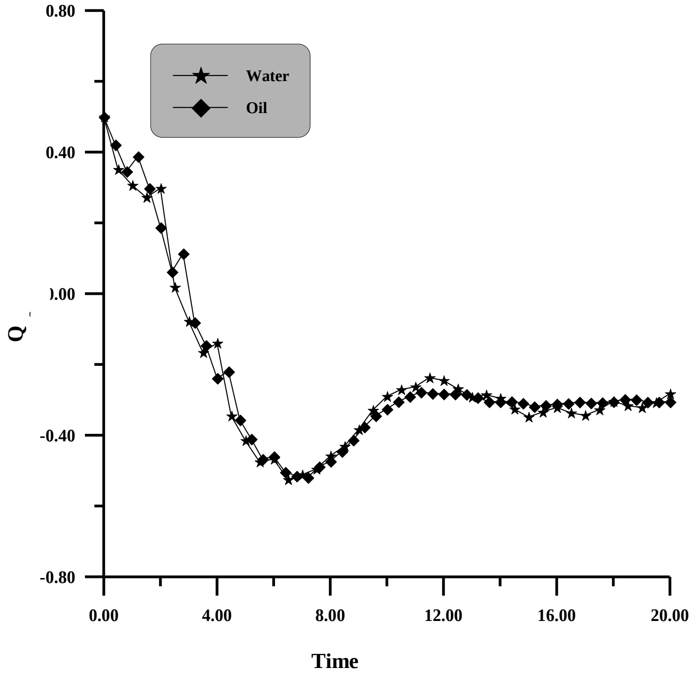


Figure (9) Flow rate distribution with time at sec A-A

While in **Fig.(10)** the water and oil becomes more stable due to viscosity and the friction losses.

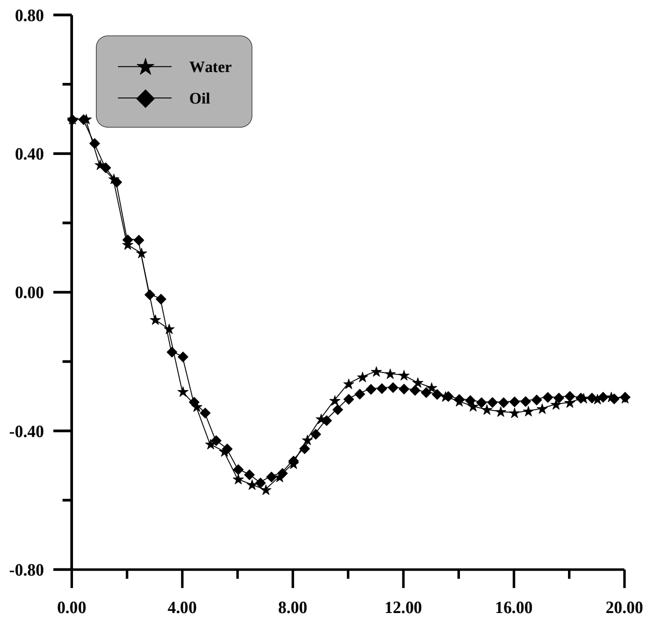


Figure (10) Flow rate distribution with time at sec B-B

Figure (11) shows the relation between the head and the time for both water and oil. The head will fluctuated from (88 m to 6 m) for water while for oil (80.38 m to -2m) this is due to the effect of shear stress and friction losses in the system and this is very clear, in **Fig.(12)** where the head will decrease in both due to the friction losses.

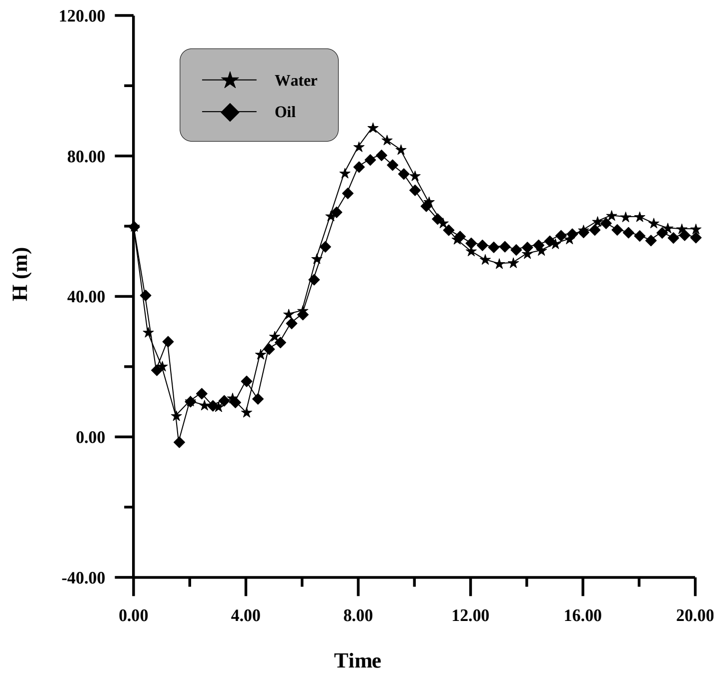


Figure (1q) Head distribution with time at sec A-A

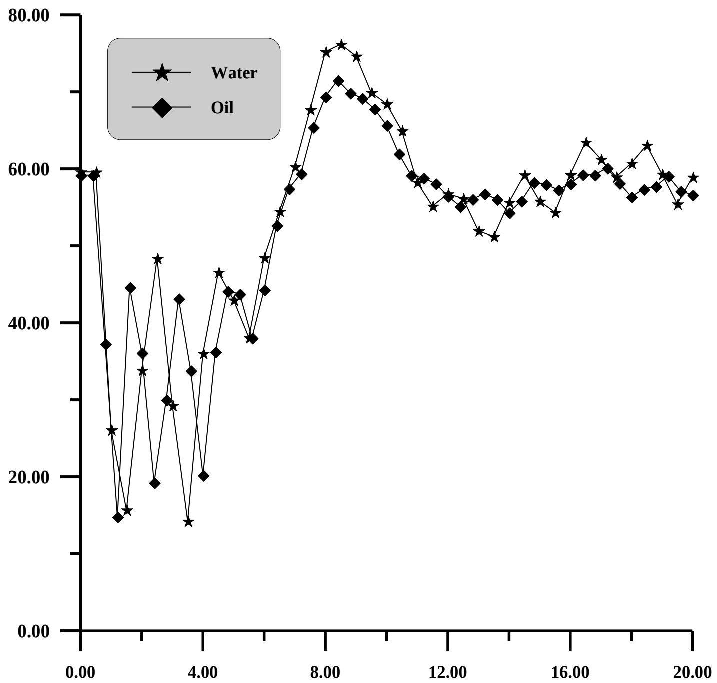


Figure (12) Head distribution with time at sec B-B

Fluid properties effects on piping length (shear force and friction losses) is different between using water and oil, in the water it is found the head becomes zero after pipe length (57.491Km), while in the oil after pipe length (32.809Km).

15. Conclusion

From the previous results, show that the effect of the fluid properties(Viscosity and Density) is very important during the design of oil system also the length of the pipe will decrease between the pumping station due to the highest shear force in the system.

16. References

1. Chaudhry, M. H., ***"Application Hydraulic Transients"***, 1st Edition, Van Nostrand Reinhold Company, New York, 1987.

2. Leino, T., and Linjama, M., ***"Experimental Study on Dynamics of Turbulent Pipeline Flow"***, International Symposium on Fluid Control, Sherbrooke (Qc) Canada, August, 2000.

3. Petter, K., and Kenneth, W., ***"Modelling of Transmission Line Dynamics"***, In Proceedings of the Ninth International Conference on Fluid Power, Linköping, Scandinavian, 1-3 June 2005.

4. Nyman, D. J., ***"Guidelines for the Seismic Design of Oil and Gas Pipeline Systems"***, by Committee on Gas and Liquid Fuel Lifelines of the ASCE Technical Council on Lifeline Earthquake Engineering, New York,1984.

5. Marco, F., and Bruno, B., ***"Pipe System Diagnose and Leak Detection by Unsteady-State Tests"***, Advances in Water Resources, 2003, pp. 95-105.

Notations

The following symbols are used in this paper.

Symbol	Description	Unit
A	Cross section Area	m^2
a	Speed Wave	m^2/Sec
a_{adj}	Adjusted Wave speed	m^2/Sec
C_v	Coefficient of Head Losses	---------
D	Diameter	m
f	Friction Factor	------
g	Acceleration	m/S^2
H	Head	m
H_{pu}	Pump Head	m
H_R	Pump Head at Rated Conditions	m
N	Rotational Speed	rpm
N_{pu}	Rotational Speed of the Pump	rpm
N_R	Pump Speed at Rated Conditions	rpm
Q	Flow rate	m3/Sec
Qo	Steady-State Flow Rate	m3/Sec
Qpu	Pump Flow Rate	m3/Sec
QR	Pump Flow Rate at Rated Conditions	m3/Sec
T	Torque	N/m2
Tpu	Torque of the Pump	N/m2
TR	Torque of The Pump at rated Conditions	N/m2
WR2	Combined Moment of Inertia of the Pump ,Motor ,Shaft and Liquid entrained in The pump Impeller	Kg.m2
Δt	Time Increment	Sec
Δx	Distance Increment	m
ψ	Permissible Variation	------