Hydraulic transient analysis in pumping stations due to power failure. (Al-Mazak Closed Irrigation System), Kut, Iraq.

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Abstract

This paper deals with the problems which may occur during transient state in pumping station in which pressure changes temporarily so much due to power failure that the network in pumping stations are damaged. A computer program was developed based on method of characteristics discussed by Chaudhry [1]. Some improvements were done in the technique used to meet the requirement of any problems may be take place in pumping stations. The developed program was applied to pumping stations in "Al-Mazak closed irrigation system" to treat the water hammering caused by power failure or sudden shutdown of the end valve. The results are presented for different assumptions and the suggested solutions to avoid any problem may be happened.

Key wards: water hammer, transient state, characteristics method, power failure

التحليل الهيدروليكي العابر في محطات الضخ بسبب فشل السلطة

الخلاصة

هذا البحث يتعامل مع المشاكل الناتجة عن حصول ظاهرة الحالة العابرة في محطات الضخ نتيجة للتغير الحاصل في الضغط نتيجة لانقطاع المفاجئ للتيار الكهربائي مما يسبب اضرار في شبكة الانابيب.

تم بناء برنامج حاسوبي اعتمادا على طريقة الخصائص التي ناقشها Chaudhry [1]. تم تطوير البرنامج ليلبي المتطلبات لمعالجة اية مشاكل تحدث في محطات الضخ. تم تطبيق هذا البرنامج لتحليل الحالة العابرة في محطات الضخ لمشروع المزاك للري المغلق في محافظة واسط لمعالجة ظاهرة الطرق المائي نتيجة لانقطاع المفاجئ للتيار الكهربائي او نتيجة اغلاق مفاجئ للصمام عند نهاية الشبكة. نوقشت النتائج والحلول المطلوبة لتلافي اية مشاكل في تشغيل محطات الضخ.

الكلمات المفتاحية: ظاهرة الطرق المائي، الحالة العابرة، طريقة الخصائص، انقطاع التيار الكهربائي.

1- Introduction

Hydraulic transients occur when the steady state conditions change with time due to power failure or sudden shutoff of the end valve. A disturbance in steady state conditions take place at the point of disturbance, a pressure wave will travel and will be reflected back boundaries such as the valve, until a new steady state was reached, so that piping networks should be designed to withstand the positive and negative pressure caused by transient condition.[1]

The aim of this study is to derive the equations governing the different boundary conditions in any pumping station, and then formulate a mathematical model for the project "Al-Mazak closed irrigation system", based on these equations of the transient state and the boundary conditions in which the method of characteristics was used for solving these equations. The model is then used to account the maximum and minimum pressures occur during the transient state caused by power failure or sudden end valve shutoff, when no control devices such as by-pass, surge tank, or control valves are available.

Table (1) Basic Design Data for Modules

Item	Description	Unit	PS#1	PS#2	PS#3
1	Design flow rate	m^3/sec	0.352	0.652	0.652
2	Number of pumps a- Total b- Operating c- Stand-by		 5 4 1	 5 4 1	 5 4 1
3	Pump rated capacity	m^3/sec	0.088	0.163	0.163
4	Head	m	35	35	35
5	Natural ground level at PS	m	15.3	13.29	14.65
6	Natural ground level at suction side	m	12.6	11.99	12.51
7	Natural ground level at discharge side	m	16.08	14.49	16.15
8	Pump speed	rpm	1450	1450	1450
9	Motor speed	rpm	1450	1450	1450
10	Rated motor power	kW	36.9	69	69
11	Motor inertia WR^2 Pump and motor inertia	kg.m^2	29 33	33 37	33 37

2- Non return valve

Each pump in "Al-Mazak closed irrigation system" is provided by a non return valve at its discharge pipe. The operation of this valve is such that it close whenever the head downstream the valve on the pipeline side, is higher than the head upstream the valve on pump side, thus preventing reverse flow through pump. Closing the non return valve will isolate the effect of the pumps on the boundary condition.[2]

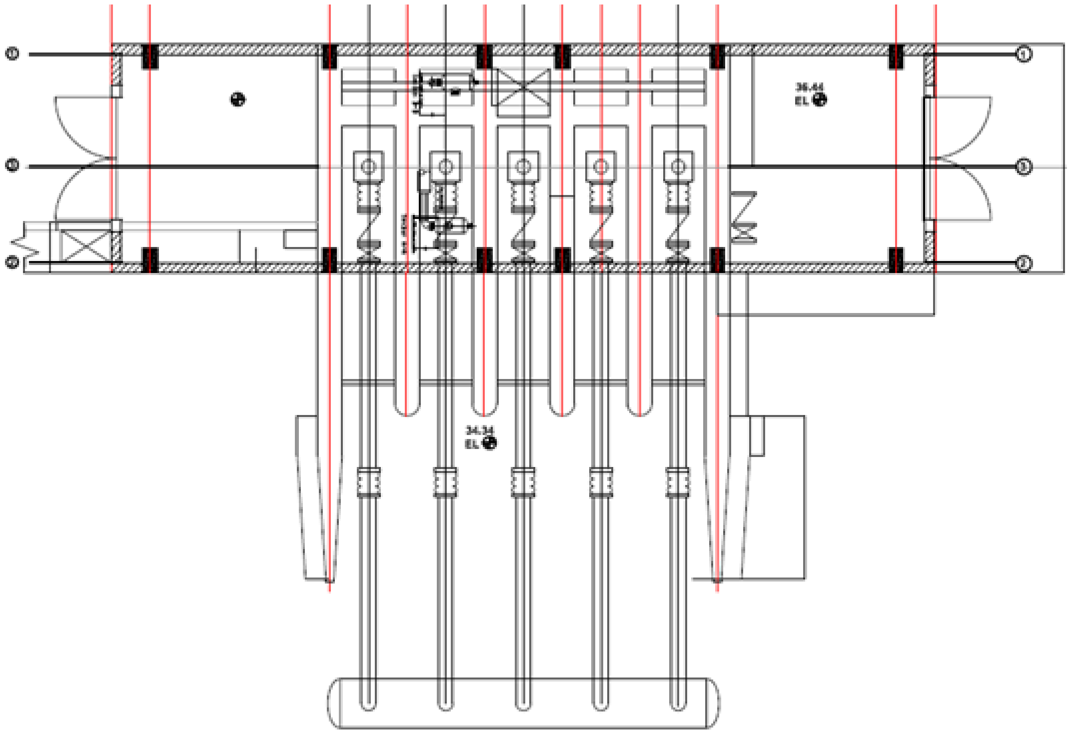

Figure (1): Schematic diagram of pumping station

3- The Mathematical Model

The pumping stations layout shown in figure 1 and pumps specifications are listed in table 1. The mathematical model based on the application of the transient state equations in elastic pipes. These equations are based on Newton's second law of motion and continuity equation. The assumptions are found in literature on fluid transient Chaudhry [1], Wyile and Streeter [2], and Paramakian [3].

The equation of motion can be written as

$$\frac{1}{\rho}\frac{\partial P}{\partial x} + v\frac{\partial v}{\partial x} + \frac{\partial v}{\partial t} + g\,sin\theta + \frac{f\,V|V|}{2D} = 0 \qquad (1)$$

And the continuity equation is

$$\rho a^2\frac{\partial v}{\partial x} + \frac{\partial P}{\partial t} + v\frac{\partial v}{\partial x} = 0 \qquad (2)$$

The two equations form a pair of hyperbolic differential equations in two independent variables, distance x and time t, and two dependent variables velocity V and pressure P.

In equations (1) and (2) the variables: wave speed (a), fluid density (ρ), pipe diameter (D) and friction factor (f) are called system parameters and do not change with time, and they may be a function of distance (x), fluid properties, flow conditions and pipe properties. It is well known that the friction factor varies with the Reynolds number and the relative roughness. Many formulas for calculating the friction factor during the steady state are available, while the variation of friction factor with the Reynolds number (Re) and the relative roughness during transient state is unknown and no satisfactory method for calculation the friction factor during transient state available. It has become common practice that the same formulas for evaluating friction factor in steady state are assumed to be valid during transient state. In dealing with the transient problems Chaudhry [1] and Streeter & Wylie [2] considered the variation of the friction factor with Reynolds number during transient is to be small, and use constant friction factor equal to the steady state friction factor, in the transient state. The wave speed (a) in an elastic pipe is a function of the bulk modulus of elasticity of fluid, the elastic properties of the pipe walls, pipe wall thickness, pipe diameter, density of the fluid and the pipe support conditions.[4]

Parmakian [3] shows that the convective acceleration term $v\frac{\partial P}{\partial x}$ and $v\frac{\partial v}{\partial x}$ in transient flow equations (1) & (2) are very small compared with the local acceleration terms and may be neglected. In addition when dealing with a piping system having a small angle of inclination with horizontal and the term (g sin θ) is very small and can be neglected.

The equations (1) & (2) can be written in terms of piezometric head and discharge as dependent variables. The pressure and discharge are:

$$P = \rho g(H - z) \tag{3}$$

and

$$v = \frac{Q}{A} \tag{4}$$

Since the slope of the pipe is neglected, then $\frac{\partial z}{\partial x} = 0$. Liquid density variation with pressure is too small and can be neglected. Hence, the partial derivatives of pressure with respect to time and distance become:

$$\frac{\partial P}{\partial t} = \rho g \frac{\partial H}{\partial t} \tag{5}$$

And

$$\frac{\partial P}{\partial x} = \rho g \frac{\partial H}{\partial x} \tag{6}$$

If the pipe walls are considered slightly deformable, then the variation of pipe cross section area (A) due to the variation of pressure may be neglected, and the partial derivatives of velocity w.r.t distance and time can be written as:

$$\frac{\partial v}{\partial x} = \frac{1}{A}\frac{\partial Q}{\partial x} \tag{7}$$

And

$$\frac{\partial v}{\partial t} = \frac{1}{A}\frac{\partial Q}{\partial t} \tag{8}$$

Incorporating the above simplifications in transient flow equations (1) &(2) can be rewritten as

$$\frac{\partial Q}{\partial t} + g\,A\,\frac{\partial H}{\partial x} + \frac{fQ|Q|}{2DA} = 0 \qquad (9)$$

And

$$\frac{\partial H}{\partial t} + \frac{a^2}{gA}\frac{\partial Q}{\partial x} = 0 \qquad (10)$$

Equations (9) & (10) governing the transient state in elastic pipes and they are quasi-linear hyperbolic partial differential equations. These equations cannot be solved analytically. A numerical technique however made accurate methods for analyzing the transient state conditions in simple and complex piping systems possible. The method of characteristics is the one most widely used in solving transient problems.

4- The Method of Characteristics

The method of characteristics utilizes a special property of hyperbolic partial differential equations to find their numerical solutions. For a system of hyperbolic partial differential equations there are two characteristic directions in the x-t plane in which the integration of the partial differential equations is reduced to the integration of a system of ordinary differential equations.

The advantages of this method are that it is a method of solution which allows the direct inclusion of friction losses. It affords ease in handling the boundary conditions and in programming complex systems. It is a general method, i.e., the program once written, may be used for analyzing any system having the same boundary conditions, and the transient state conditions obtained by using this method are close to the actual situation. The only restriction in this method is that the flow must be one dimensional; the wave speed is constant during the transient state and the time increment chosen to satisfy the stability conditions.

The two quasi-linear, hyperbolic, partial differential equation (9) & (10) are transformed by the method of characteristics in to four ordinary differential equations which are solved numerically using Newton Raphson method.[2]

5- Power Failure to Pump

When the power to a pump motor is suddenly cut off a rapid deceleration in the speed of the motor and pump will take place immediately and cause a rapid change in flow condition. As a result of this change a low pressure wave will be transmitted in the discharge pipe and a wave of high pressure will transmitted in the suction pipe. After a short time the flow in the discharge pipe will be reduced to zero and reverses through the pump. If it is not provided with a non return valve, causing reverse rotation to the rotating elements will takeplace. As the speed of the pump increases in the reverse direction, it causes greater resistance to the reversal of the flow, which causes high pressure to develop upstream of the pump and low pressure downstream of the pump.

These pressures accompanying the transient state which follows the power failure in centrifugal pump system are the most extreme that the system must withstand.

The pump operation at pumping station, before and after the power fails may be divided into three zones as follows:[5]

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

When dealing with the transient states for a system involving pumps, the relation between the pump discharge (Q), speed (N), torque (T), and head (H), under different operating conditions must be taken into consideration. For a given geometrically similar pumps the homologous relations may be presented as:[2]

$$\frac{H}{N^2 D^2} = Constant$$

$$\frac{N}{Q D^3} = Constant \qquad (11)$$

$$\frac{T}{N^2\, D^2} = Constant$$

When restricted to particular unit, the diameter of the impeller is considered a constant then

$$\frac{H}{N^2} = Constant$$

$$\frac{N}{Q} = Constant \qquad (12)$$

$$\frac{T}{N^2} = Constant$$

By defining the non dimensional parameters

$$\alpha = N/N_r \quad , \beta = T/T_r \quad , \nu = Q/Q_r \quad \text{and} \quad h = H/H_r \qquad (13)$$

In which the subscript r refers to the rated conditions and the equation (12) can be written as

$$h/\alpha^2 = \text{const.}, \quad \alpha/\nu = \text{const.}, \quad \text{and} \quad \beta/\alpha^2 = \text{const.} \qquad (14)$$

In order to avoid having h/α^2, α/ν, and β/α^2 reach infinity when α, and ν equal to zero, $h/(\alpha^2+\nu^2)$ are normally used instead of h/α^2, $\beta/(\alpha^2+\nu^2)$ instead of β/α^2, and $\theta = \tan^{-1}(\alpha/\nu)$ instead of α/ν. If $h/(\alpha^2+\nu^2)$ is plotted as ordinate against $\theta = \tan^{-1}(\alpha/\nu)$ as abscissa, the resulting curve is termed the complete characteristics head curve. Similarly, if $\beta/(\alpha^2+\nu^2)$ is plotted versus $\theta = \tan^{-1}(\alpha/\nu)$ the resulting curve is termed the complete characteristics torque curve. These two curves are used to determine the relation between pumps discharge, speed, torque, and head during different operation conditions[1].

Pumps manufacturers usually supply the characteristics curve of normal operation zone and very few of them supply the complete characteristics curve. If these curves are not available for a pump, then the curves for a pump of the same type with approximately the same specific speed may be used. In case study presented in this work (Al-Mazak closed irrigation system) the complete characteristic curves of the plant pumps whose specific speed is 5.1 were not supplied by the manufacturer. The curves used were those of a pump specific speed 4.94 stating that they are sufficiently close to those of specific speed of 5.1 that their use is acceptable.

To use pump characteristic curves in a mathematical model, points on the curve are stored at intervals of $\Delta\theta$ from 0^o to 360^o .Each segment of the curve between two points may be approximated by a straight line. If a sufficient number of points are stored, then the error introduced by the segmented straight lines is negligible. Figures (2), (3), (4), and (5) show the characteristic curves for the pumps used in Al-Mazak closed irrigation system. The data used to plot figures 2 to 5 were taken from the data sheet supplied by pumps manufacturing company [8].

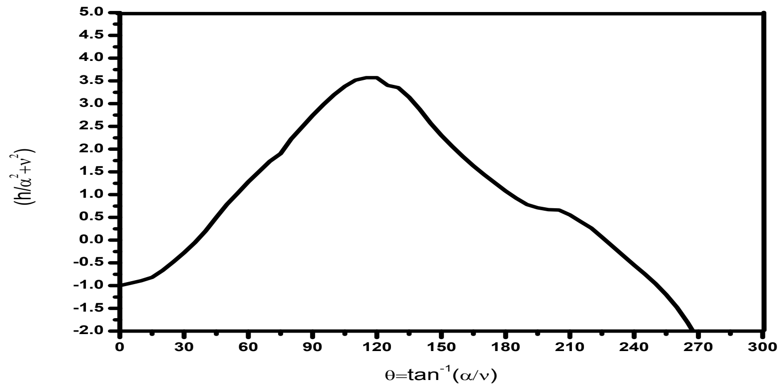

Figure (2): The characteristic curve for head in pumping station # 1

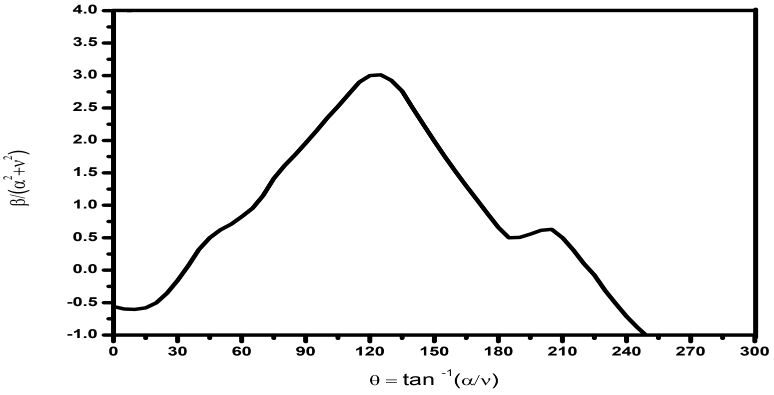

Figure (3): The characteristic curve for torque in pumping station # 1

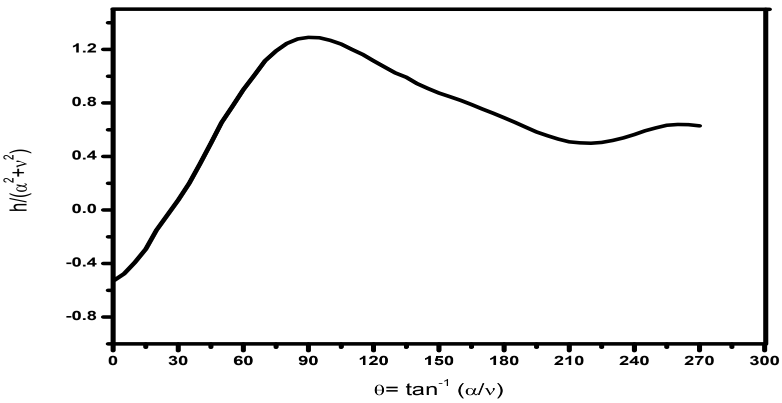

Figure (4): The characteristic curve for head in pumping station # 2 & 3

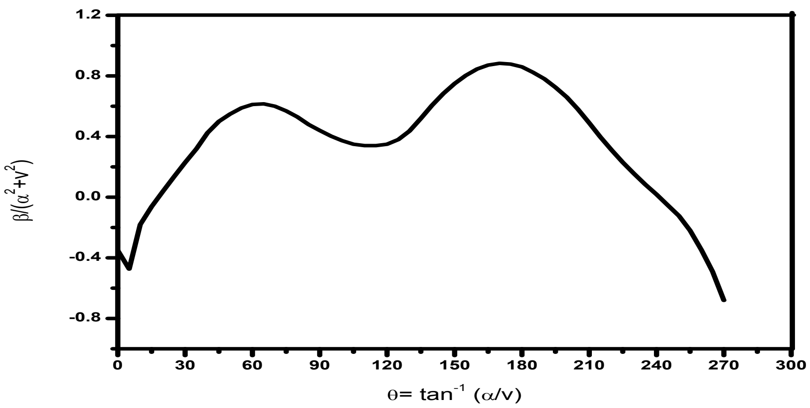

Figure (5): The characteristic curve for torque in pumping station # 2 & 3

6- Pump speed change

Speed change of the rotating elements of a pump following power interruption or pump start, result from unbalanced torque applied which is equal to the product of the mass moment of inertia of the rotating elements and the angular acceleration[6,7]

$$T = -I \frac{2\pi}{60} \frac{dN}{dt} \tag{15}$$

Where I= Combined moment of Inertia = W R^2 / g

W= Weight of rotating parts.

R= Radius of Gyration of rotating parts

$\frac{2\pi}{60} \frac{dN}{dt}$ = angular acceleration

Equation (15) may be written in a dimensionless form as

$$\beta = -I \frac{2\pi}{60} \frac{N_R}{T_R} \frac{d\alpha}{dt} \tag{16}$$

Using an average value of β during each time step, the finite difference form of the above equation becomes

$$\frac{\alpha_p - \alpha}{t_p - t} = -\frac{60\, T_R}{2\pi I N_R} \frac{\beta + \beta_p}{2} \tag{17}$$

For programming usage, the equation of the torque curve may be approximated and stored in the same way as shown in section 4 .

7- The equation for surge tanks

Derivation of the equation representing the surge tank boundary condition, with the flow considered positive into the tank, the following equation may be written as

$$Z_p = H_p - h_{orf}$$

$$Z_p = H_p - C_{orf} Q_{ps} |Q_{ps}|$$

Where

Z_p = water level in the surge tank

$h_{orf} = C_{orf} Q_{ps} |Q_{ps}|$ = head loss through the orifice

C_{orf} = orifice head loss coefficient

Q_{ps} = flow into the surge tank

8- Application of the model

8.1 System parameters

System parameters are those variable that may change with distance and remain constant with time. They include pipe diameter, fluid density, friction factor, and the speed of a pressure wave in the system.

The choice of pipeline friction factor was discussed early in this work. In transient analysis of a system two extreme cases are critical. For pump failure, a new pipeline with lower friction factors is the critical case while the aged pipeline with higher friction factor represents the critical case for pump start-up. Pipe roughness height, k, is generally taken as 1.5 mm for the aged steel pipe. The roughness height for the new pipe depends on the material, manufacturing, and method of connecting pipe segments. Precise information are not available on these and it seemed prudent to investigate the friction factor for a range of k values from 0.06 mm to 0.15 mm for the new pipes. Miner losses were also taken in the consideration and the equivalent friction factors are 0.014 and 0.021 for new and aged pipes, respectively. In (Al-Mazak closed irrigation system) pump station No. 1 is 0.018, and for pump stations No. 2 & 3 are 0.022.

The speed of pressure wave in the pipe is a function of fluid density, and bulk modulus of elasticity, pipe diameter and wall thickness, pipes Young's modulus, and type of supporting. Using typical value of the bulk modulus and Young's modulus the wave speed is about 960 m/s[2]. The value of wave speed and time increment, Δt, determine the number of reaches into which a pipeline is divided for computational purposes. It is therefore customary to adjust the wave speed by up to +/- 15% in order that an integer number of reach is obtained. In this case it was only necessary to adjust the speed down to 900 m/s. this allowed using 8 reaches, each 250 m long, and a time interval of 0.25 seconds in pipeline for pumping station No. 1 and 5 reaches each 195 m long, and time interval of 0.15 second in pipeline for pumping stations No. 2 & 3.

8.2 Pump characteristics curves

Transient analysis of pumping station requires characteristics curves covering pump operation in four quadrants representing θ= $\tan^{-1}$ (α/ν) values from 0 to 360^o. Manufacturers usually supply the curves of normal operation zone, θ=0 to 90^o, and it is customary to choose from few complete characteristic curves that are published in the engineering literature. Chaudhry [1] has published the complete characteristic curves of four pumps whose specific speeds are 0.46, 1.61, 2.78, and 4.94. the specific speed can be calculated using the following equation

$$N_s = \frac{N_R\sqrt{Q_R}}{(g\ H_R)^{0.75}}$$

(18)

The pumps of (Al-Mazak closed irrigation system) have H_R=35 m, Q_R=0.163 m^3/s, N_R=1450 rpm=151.848 rad/s. The specific speed as defined by Chaudhry [1] is 5.1 which is very close to N_s= 4.94 whose complete characteristics curves were used for this study. The other data used in program are listed in Table (2)[8]

Table (2) Data for pumping stations[8]

Item	Description	Unit	PS#1	PS#2	PS#3
1	Head losses at suction line and fittings	m	0.65	0.69	0.69
2	Head losses at discharge line and fittings	m	1.9	2.1	2.1
3	Friction loss in pipeline	m	6	7.5	7.5
4	Design static head	m	35	35	35
5	Friction factor for new pipes		0.013-0.014	0.016-0.017	0.016-0.017
6	Friction factor for aged pipes		0.021	0.024	0.024
7	Pipe diameter	mm	900	900	900
8	Pipe wall thickness	mm	11.91	11.91	11.91

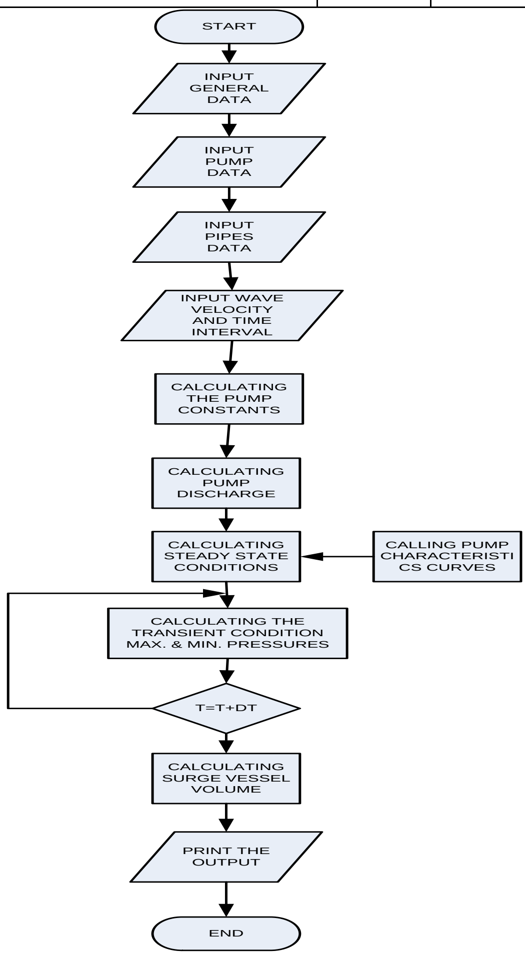

Figure (6): Program Used in Analysis Flow Chart

9- Results and Discussion

9.1 Results for Pumping Station #1

The pumping station #1 consist of 5 pumps (4 working and 1 stand-by) of total rated capacity of 0.352 m3/s, the head losses due to valves and pipe fittings is 1.9 m, and head loss due to friction is 6 m, the friction factors for new pipe and aged pipe are 0.014 & 0.02 respectively. In power failure to pump the maximum pressure was found 20.2 bar, while the minimum pressure was found 0.65 bar. To avoid this increasing in pressure a controlling device must be use such as surge tank or pressure vessel. Figure (7) shows the pressure oscillation with time due to power failure without controlling device provided, while Figure (8) shows the pressure with controlling device (pressure vessel) with time. Figure (9) shows the discharge flow rate based on time with controlling device.

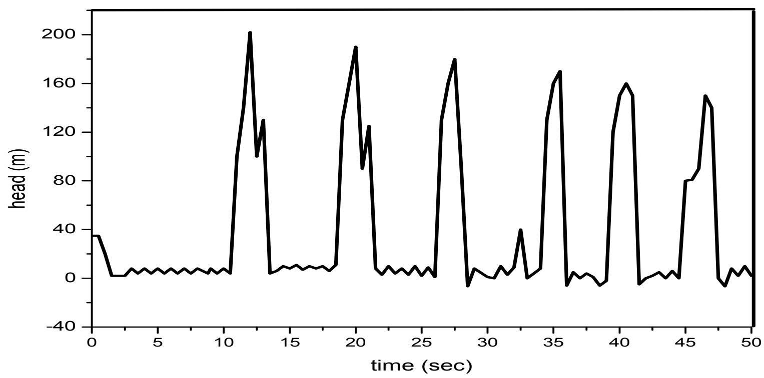

Figure (7): The pressure in pipeline with time due to power failure without pressure vessel for pumping station #1

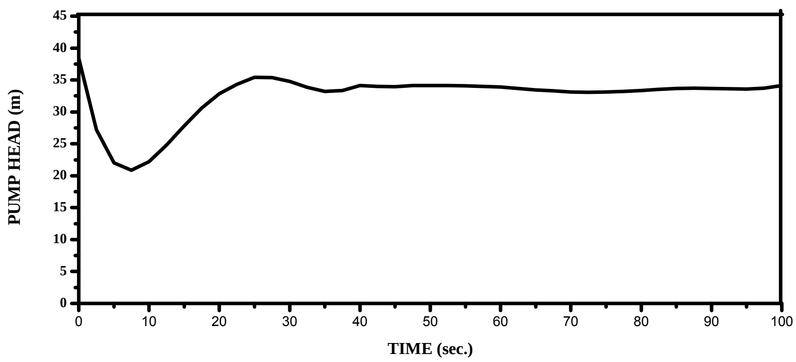

Figure (8): The pressure in pipeline with time due to power failure with pressure vessel for pumping station #1

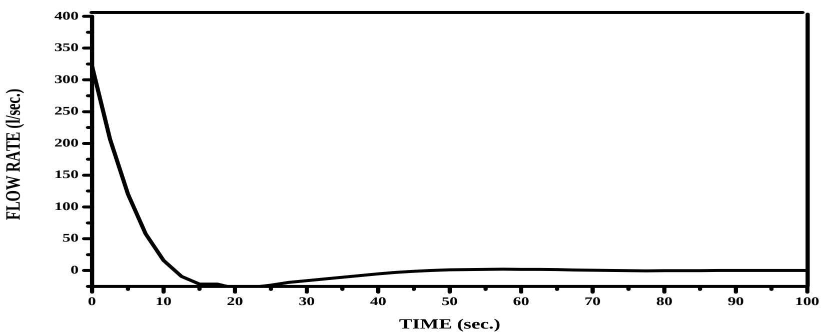

Figure (9): Discharge flow rate with time with pressure vessel connected to pipeline for pumping station #1

9.2 Results for Pumping Stations # 2 & 3

The pumping station #2 & 3 consist of 5 pumps (4 working and 1 stand-by) of total rated capacity of 0.652 m3/s, the head losses due to valves and pipe fittings is 2.1

m, and head loss due to friction is 7.5 m, the friction factors for new pipe and aged pipe are 0.014 & 0.02 respectively. In power failure to pump the maximum pressure was found 20.2 bar, while the minimum pressure was found 0.65 bar. To avoid this increasing in pressure a controlling device must be used such as surge tank or pressure vessel Figure (10) shows the pressure oscillation with time due to power failure without controlling device provided, while Figure (11) shows the pressure with controlling device (pressure vessel) with time. Figure (12) shows the discharge flow rate with time based on controlling device.

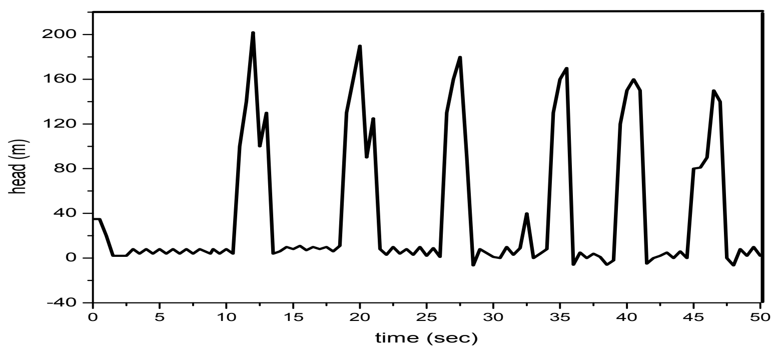

Figure (10): The pressure in pipeline with time due to power Failure without pressure vessel for pumping station #2 &3

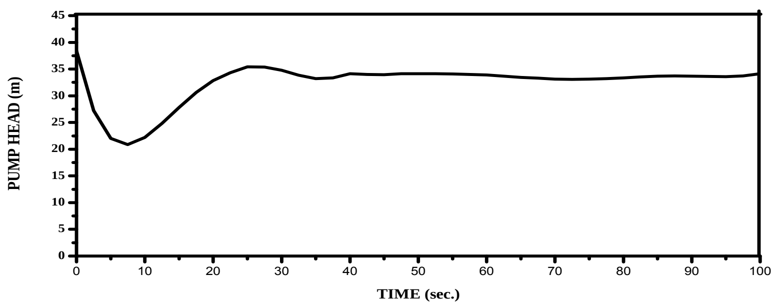

Figure (11): The pressure in pipeline with time due to power failure with pressure vessel for pumping station #2 &3

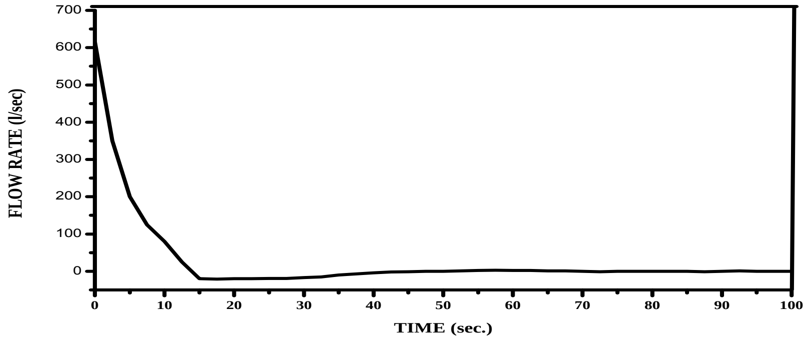

Figure (12): Discharge flow rate with time with pressure vessel connected to pipeline for pumping station #2 & 3

9.3 Conclusions

A surge tank or a pressure vessel must therefore be provided in order to control transient in the pipeline. High head pump installations generally required a pressure vessel rather than surge tank. Many runs were made simulating pressure vessel of various cross section area and height, for pumping station # 1 the pressure vessel of 3 m diameter and 5 m height is acceptable for controlling the transient state, while for pumping stations 2 & 3, the pressure vessel found to be 4m in diameter, 6 m height was sufficient to controlling the transient state.

<u>Nomenclature</u>

a	water hammer wave velocity	m/s
A	cross sectional area of pipe	m^2
D	pipe diameter	m
EL	elevation	
f	friction factor	
k	pipe roughness	m
g	gravitational acceleration	m/s^2
H	head	m
H_R	rated head	m
I	moment of inertia of rotating elements	$kg.m^2$

N	rotational speed	rpm
Nr	rated speed	rpm
Q	flow rate	m^3/s
PS	pump station	
Qr	rated flow rate	m^3/s
T	torque	N.m
Tr	rated torque	N.m
z	pump centerline elevation	m
v	dimensionless parameter of flow rate (Q/Qr)	
h	dimensionless parameter of head (H/Hr)	
α	dimensionless parameter of speed (N/Nr)	
β	dimensionless parameter of torque (T/Tr)	
θ	zone of pump operation	

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