Experimental and Numerical Investigations of the Effect of Net Positive Suction Head on Water Hammer In Pipeline Systems

Maher Abdul Ameer, Audai Hussein Al-Abbas

Abstract: *In the present paper, the effect of the Net Positive Suction Head (NPSH) for the centrifugal pump in a simple pipeline system has been carried out eliminating water hammer. Both the experimental tests and numerical calculations were performed in this study to investigate the transient flow effects when the discharges change abruptly in the system. The phenomenon of transient flow is generally occurred when the sudden opening of the downstream valves are used in the pumping station or due to introducing additional parallel branches of pipelines that contain pumps. Experimentally, two operational tests with different NPSH were conducted to show the relations between the head (H) and discharge (Q). The results of these tests showed two different types of H-Q curves. The first test produced a flat H-Q curve when the water surface level in the suction tank was 4 m above the center line of the pump. While in the second test the operational case created a steep H-Q curve when the pump was used to lift 6 m from the suction tank. That is likely happened due to the decrease in a cut-off discharge point. The numerical calculations on hypothec simple pipeline system have been done for the transient flow after determining the shape of the characteristics H-Q curves produced from the experimental tests. The Darcy-Waysbach equation was used to calculate the friction losses and obtain the system resistance curve. The classical method of characteristics, which is based on the methodological analysis of the finite difference method, was executed to propagate the wave pressure in each cases examined. The numerical results were clearly showed that the wave pressure produced in the pipeline system with the flat H-Q curve is smaller than that produced in the pipeline with the steep H-Q curve. This means that the first case is significantly reduced the possibility of occurrence the water hammer phenomenon.*

Keywords: Transient flow, net positive suction head, pipeline system, water hammer, centrifugal pump.

I. INTRODUCTION

The transient flow in the pipeline systems of water distribution networks (WDNs) is principally considered a main problem in the design of these systems because this particular problem is negatively leading to water hammer. The WDNs have been existed for a long time. The transient flow has not been clearly considered in design calculations. Basically, the WDNs were early fed by gravity effect from high elevation reservoirs or water towers, and as a result the steady flow conditions are nearly dominated in the system. The most significant problem was to predict the equilibrium distribution of flow under assumed demand conditions.

This is also caused due to a geometrical design of these pipeline systems such as the number, length and diameters of pipes and low pressures as well. Therefore, the lack of transient flow in these networks is considerably needed to be well justified.

Recently, modern water distribution systems are usually fed by numerous pumping stations, which are directly discharging the working fluid (water) into the system. The automatic stopping of pumps and the adjustment of control valves, such as power outages, suddenly varying in discharge, increase or decrease, are significantly generating transient flow conditions. From the literature, the researches on the pipeline systems of Bryan W. Karney [1], and X. Wang, M. F. Lambert and A. R. Simpson [2] were focused on investigation the effects of leakage on pipeline transients. Authors found highly dynamic flow effect in the pipeline systems, and these systems should be mathematically analyzed to resolve the problem. They analyzed transient in pipeline systems with leak by using Laplace transforms. While Don J. Wood [3] studied the minimum and maximum operating pressure which is occurring during transient flow operations. Author concluded that it is essential for a good design and operation to perform a transient analysis for normal startup and shutdown, and for unplanned events such as a pump trip associated with power outage. However, mechanical engineers and technicians must be carefully considered all potential mistakes for the pipeline designs and estimated and eliminated the weak spots. And then they should embark upon a detailed transient analysis to make constructive decisions on how to get the best design of pipeline systems and ensure safe and reliable operations SAVA, [4]. Maher Abdul Ameer [5] used two methods by analyzing the transient flow in two pipeline systems. Firstly, the author used time domain method and frequency domain method on simple network, i.e. without any apparatus. While in the second method he used network system which is containing same complexity such as intakes, valves, and other apparatus. The results showed clear differences for two approaches.

To the best of our knowledge, there is a small number of researches conducted on this pipeline system under transient flow operations. This paper is focused on the available net positive suction head (NPSH$_a$) because it is representing the most significant parameters affecting on the shape of H-Q curves. Both the experimental investigations and numerical modelling are used to estimate the H-Q curves for the centrifugal pump utilized in the present work. This study has been carried out at Pumps Eng. Dept. of Al-Musaib Technical College in Babylon / Iraq.

   Maher Abdul Ameer, Pumps Eng. Dept./Al-Musaib Technical College/ Foundation of Technical Education/ Babylon, Iraq.
   Audai Hussein Al-Abbas, Pumps Eng. Dept./Al-Musaib Technical College/ Foundation of Technical Education/ Babylon, Iraq.

II. SYSTEM CHARACTERISTICS AND THEORETICAL CONCEPTS

A. The H-Q concept

The form of the pump curve graph is geometrically the same that of the mathematical 'X-Y' graph. On the horizontal line (X-axis), the flow rate (Q) is given in m^3/sec., while the vertical line (Y-axis) shows the head (H) in meters. The pump characteristics curve for the H-Q relation is expressed by a second-order polynomial, as simply presented in the following equation: $h_p = AQ^2 + BQ + C$, in which the coefficients A, B, and C are determined by using of experimental data and numerical approaches. For the precise system head curve, it is important to select an appropriate pump in order to deliver the required capacity. Manufacturers published pump H-Q curves for this particular purpose. Since these pump curves are usually applied to a particular impeller and pump design. Different manufactures showed a slight difference in the performance and operation for the same type and size of pumps. Therefore, several manufactures pump curves should be checked to establish a realistic and cost-effective criterion for the pump selection. The performance of H-Q curve has several types or several shapes. This variety in number of shapes is due to many reasons and parameters such as normal rising curve, drooping curve, and a steeply rising curve. Figure **(1)** is clearly showed the typical shape of the H-Q curve for the centrifugal pump, which is having the radial flow, the shut-off head and cut-off discharge that defined by Randall W. Whitesides, CPE, PE [6].
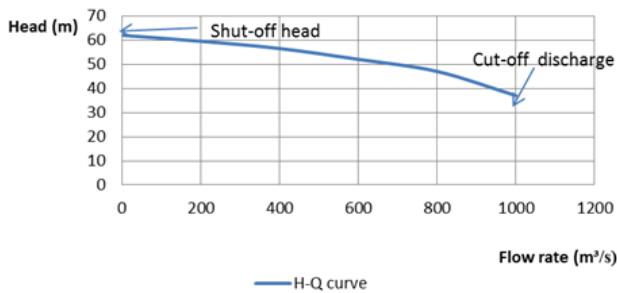

Figure (1) Typical shape of the H-Q curve for the centrifugal pump [6]

B. The NPSH concept

The concept of NPSH can be classified to have two terms:
1- The net positive suction head required ($NPSH_r$) by the pump to prevent the inspection of cavitations and for safe and reliable operation.
2- The net positive suction head available ($NPSH_a$) by the suction system of the pump.

The Hydraulic Institute defines the $NPSH_r$ of the pump as the NPSH that causes the total head to be reduced by 3% due to flow blockage from cavitations' vapor in the impeller vanes. The rated pump head is not achieved when the $NPSH_a$ equals to the $NPSH_r$ of the pump. For varying suction heads, the $NPSH_r$ can be determined by testing the geometrically similar pumps which are operated at constant speed and discharge. Large variations in the $NPSH_r$ are occasionally observed in tests of theoretically identical pumps. Small pumps are more sensitive to casting imperfections than large ones. Mathematically, the $NPSH_r$ cannot be explicitly defined in clear equations. But in practical design it depends on the value of $NPSH_a$. For head values from 2 to 3 ft., it is recommended that the $NPSH_a$ is greater than the $NPSH_r$.[6-10].

According to the latest edition of the Hydraulic Institute Standards, the $NPSH_a$ is defined as:

$$NPSH_a = \pm h_S - h_L + h_A - h_V \qquad (1)$$
$$NPSH_a > NPSH_r \qquad (2)$$

Where, h_S represents the static suction head (+ve) or static suction lift (-ve) in meters, h_L is the suction line losses such as friction, entrance, and fittings, and it is measured in meter, h_A represents the absolute pressure at the liquid's free surface in meter, and h_V is the vapor pressure of liquid at pumping temperature converted to meter of liquid.

C. Pump characteristics, system characteristics and operating point

All pumps must be designed to comply with or meet the needs of the pipeline system. The needs of the system are recognized using the term 'Total Dynamic Head', or TDH. For the purpose of hydraulic analysis, the pump system is any defined set of single or multiple variables fluidic or physical conditions or static arrangements. In fact, a centrifugal pump will operate at the point of intersection of the H-Q curve and system curve. Also, the mathematical expressions that define these curves can be established. To clearly define this system curve, it mathematically follows the total head loss. All selected fully turbulent flow rates will be taken the form [$H_L = H_S + CQ^2$], and the pump characteristic is having quadratic polynomial equation [$H_P = A + BQ + DQ^2$]. Therefore, it stands to reason that the simultaneous solution of these two expressions would result in a numerical demarcation of the pump expected operation point. The system curve depicts a system piping circuit resistance to flow at various flow rates. The H-Q curve is dependently equipment, whereas the system curve is independently equipment. Figure **(2)** shows the operating point on the pump performance curve [6].
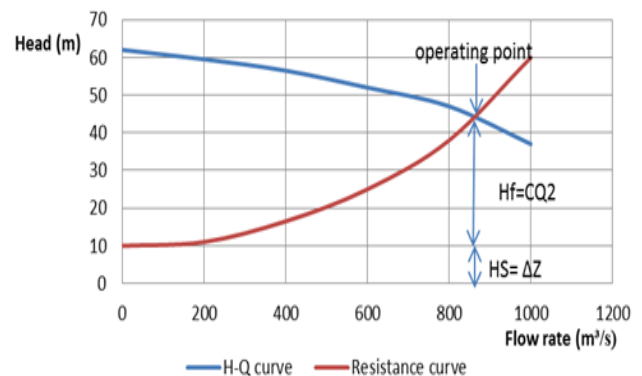

Figure (2) Pump characteristics, system characteristics and operating point [6]

III. EXPERIMENTAL WORK

The experimental setup of this work was conducted to establish the H-Q curve from the available pump in the laboratory of Pumps Engineering Department at Al-Musaib Technical College. The effect of NPSHa on the shape of H-Q curve is experimentally tested. This was done by installing the centrifugal pump with various values of NSPHa. The characteristic of centrifugal pump is a single stage in-line, GRUNDFOS, LM/LP, 1480 rpm. This type of pump operates on various NPSHa. Figures **(3 and 4)** show the pump used and layout of the lab. equipment used in the present experimental investigations.

Figure (3) Pump used in Lab experiments

Figure (4) Layout of the Lab equipment used in experiment

Based on the water level available in the suction tank, two different tests are carried out in this study. In the first test, the values of the $NPSH_a$ were from 11 to 12.0m. This means that the water level in the suction tank was 4 meter above the center line of the pump. The results of this test showed that the H-Q curve is relatively flat, and the head is slightly changed with pump capacity, i.e. generally the results were practically unexciting in the first test. While in the second test the centre line of the pump was 6 meter above the water level in the suction tank. Typically, the increase in the suction lift head is significantly reduced the value of $NPSH_a$ (see equation 1) and as mentioned in [6,7, and 10]. According to this fact, this experiment showed establishing a steep H-Q curve, and there is a large decrease in the head between the shut-off head and developed far-field head. Also, it was obviously noted that there is a minimum change in the H-Q curve with increasing the pressure, as shown in figure **(5)**. In figure **(5)**, the $NPSH_r$ values are also recorded for each value of $NPSH_a$. The curve of this parameter can be established for several installations (see equation 2). In general, the change of this parameter ($NPSH_r$) showed a considerable change on the shape of H-Q curve from flat to steep curve.
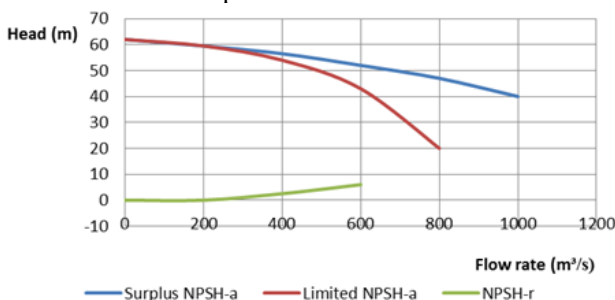

Figure (5) Two types of operation for the same pump using two different $NPSH_a$

IV. NUMERICAL ANALYSIS AND METHODOLOGY

In the numerical analysis of the hydraulic transient flows of this study, the approximate equations are obtained by neglecting the spatial variation of *V* and *P* whenever both space and time varying terms appear in the same equation. In general, the spatial variations are much less significant in determining the solution behavior than the time varying terms, Bruce E. Larock [11]. The essence of the method of characteristics is the successful replacement of a pair of partial differential equations by an equivalent set of ordinary differential equations. The first equation is the momentum equation:

$$\frac{\partial V}{\partial t} + \frac{1}{\rho}\frac{\partial p}{\partial s} + g\frac{dz}{ds} + \frac{f}{2D}V|V| = 0 \qquad (3)$$

The second equation is the continuity equation

$$a^2\frac{\partial V}{\partial S} + \frac{1}{\rho}\frac{\partial P}{\partial t} = 0 \qquad (4)$$

The development of the method begins presuming that the pair of the momentum and continuity equations may be replaced by some linear combination of these equations. Using *l* as a constant linear scale factor, sometimes called a Lagrange multiplier, one possible combination is as follows:

$$\lambda\{\frac{\partial V}{\partial t} + \frac{1}{\rho}\frac{\partial p}{\partial s} + g\frac{dz}{ds} + \frac{f}{2D}V|V|\} + \{a^2\frac{\partial V}{\partial S} + \frac{1}{\rho}\frac{\partial P}{\partial t}\} = 0 \qquad (5)$$

The scale factor *l* is a linear and constant as required, *a* is constant, and the succeed combination of equations (3) and (4) and rewrite equation (5) with *l* = - *a* as a replacement for the second equation. By dividing the equations by the wave speed *a,* the result is a pair of ordinary differential equations rather than partial differential equations, as mentioned below:

$$\frac{dV}{dt} + \frac{1}{a\rho}\frac{dp}{dt} + g\frac{dz}{ds} + \frac{f}{2D}V|V| = 0 \qquad (6)$$

$$\frac{dV}{dt} - \frac{1}{a\rho}\frac{dp}{dt} + g\frac{dz}{ds} + \frac{f}{2D}V|V| = 0 \qquad (7)$$

Equation (6) valid only when $\frac{ds}{dt} = +a$, and equation (7) valid only when $\frac{ds}{dt} = -a$. This is the reason why the name of this procedure is the method of characteristics. The numerical solution of these characteristics equations needs to work with a graph which is having s as the abscissa and t as the ordinate, referred to the s-t plane. Figure **(6)** shows the s-t plane that is related to the simple pipeline system. The s-coordinate is the distance along the pipe from the upstream end.
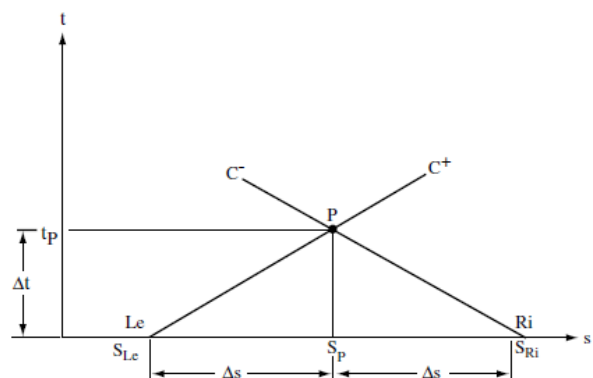

Figure (6) The *s-t* plane for simple pipe flow

A. The Finite Difference Representation

The numerical solution of the characteristics equations (6) and (7), for the transient flow in the pipeline system, must be written in the finite difference form, and theses equations are supposed to have the following forms:

$$\frac{Vp-VLe}{tp-0} + \frac{g}{a}\frac{Hp-HLe}{tp-0} + \frac{f}{2D}\,VLe\left|VLe\right| = 0 \quad (8)$$

$$\frac{Vp-VRi}{tp-0} - \frac{g}{a}\frac{Hp-HRi}{tp-0} + \frac{f}{2D}\,VRi\left|VRi\right| = 0 \quad (9)$$

In point of fact two significant assumptions made in developing these equations. First the velocity at the beginning of the time interval is replaced rather than an average velocity over the interval, which adequately represents the frictional effect. The unknown value v_p in the friction term of the difference equations would become nonlinear and requires an iterative solution. The second assumption is the steady- state friction coefficient can sufficiently represent friction losses in the transient flow. The assumption of non-transient constant friction coefficient in transient flow analysis has always been approximated. The use of the steady-state Darcy-Weisbach (*f*) implies that the flow in the pipe is behaving as an entirely rough flow.

By replacing t_p-0 with Δt in the above-mentioned equations, the mathematical analysis will be applied for more than the first time interval, and multiplying these equations by Δt gives:

$$C^+ : (V_p - V_{Le}) +$$
$$\frac{g}{a}(Hp - HLe) + \frac{f\Delta t}{2D}\,VLe\left|VLe\right| = 0 \quad (10)$$

$$C^- : (V_p - V_{Ri}) -$$
$$\frac{g}{a}(Hp - HRi) + \frac{f\Delta t}{2D}\,VRi\left|VRi\right| = 0 \quad (11)$$

In the selection of a spatial interval in the s-direction finite difference solution, the number of sections of the pipeline will be divided, as shown in figure (7). Figure (7) shows the grid of characteristics.
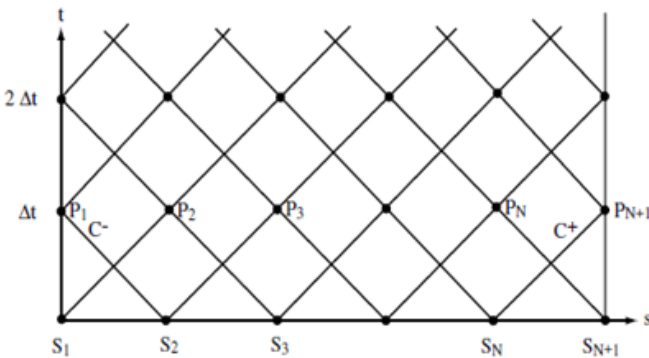

Figure (7) The characteristics grid for a single pipe

Grid points along the s-axis represent points spaced (Δs) that are along the pipe axis, and the values of H and V at these points are initial conditions. These initial conditions are usually a set of values of H and V which describe the steady flow in the pipeline and at the moment of beginning the transient flow. With the known values from points L_e and R_i, the above equations can be solved simultaneously in order to obtain the values of H_P and V_p through the time interval. The boundary conditions at s=0 and s=L must be used in conjunction with the appropriate C^+ or C^- equations to compute

the values of H_{P1} and H_{PN+1}. This is completed the solution for all the values of H and V at time interval ($t=\Delta t$). This process is continuously repeated as the head is marching in the s-t plane.

Finally, any change in the velocity or head at the point in the pipeline cannot be sensed by another point in the pipeline until the pressure wave has had time that propagates at the wave speed to that section. Figure (**8**) shows where and when a disturbance at S can be sensed at subsequent times. The values of H and V at the point P can only be contained within the zone formed by the subtended (C^+) and (C^-) characteristics.
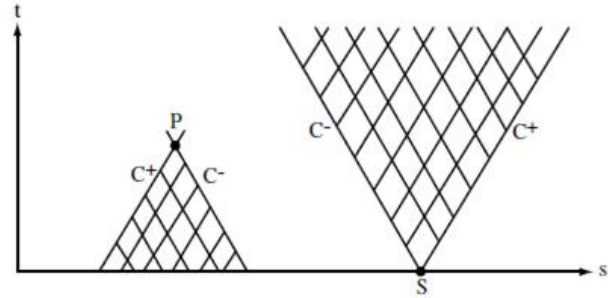

Figure (8) Disturbance propagation in the *s-t* plane

The simultaneous solution of equations (10) and (11) develops the pair of equations to calculate the values of H and V at interior points, as presented in the following equations:

$$V_P = \frac{1}{2}\left[(V_{Le} + V_{Ri}) + \frac{g}{a}(H_{Le} - H_{Ri}) - \frac{f\Delta t}{2D}(VLe\left|VLe\right| + VRi\left|VRi\right|)\right] \quad (12)$$

$$H_p = \frac{1}{2}\left[\frac{a}{g}(V_{Le} - V_{Ri}) + (H_{Le} + H_{Ri}) - \frac{a}{g}\frac{f\Delta t}{2D}(VLe\left|VLe\right| - VRi\left|VRi\right|)\right] \quad (13)$$

B. The boundary conditions

As the water exits from a reservoir, the head (H) assumes the value corresponding to the head of the reservoir water surface that is adding to the head of the operating pump on the line. If the water surface elevation is constant in time, then H is constant, and when the reservoir water surface elevation changes with time, the head will be changed. As the local pipe entrance loss is assumed to be neglected, the above equations (12) and (13) can be represented in the following expressions:

$$H_{P1} = H_0 \quad (14)$$

$$V_{P1} = V_2 + \frac{g}{a} + (H_0 - H_2) - \frac{f\Delta t}{2D}\,V2\left|V2\right| \quad (15)$$

As the reservoir is located at the downstream end of the pipe, the same approach for using the C^+ equation can be used to give a similar expression for $V_{N+1.}$. When the velocity is known at the downstream end of the pipe, this information can be combined with the C^+ characteristic equation for H_{PN+1}. The velocity behavior is as follows:

$$V_{PN+1} = V_0\left(1 - \frac{t}{Tc}\right) \qquad 0 \le t \le T_C \quad (16)$$

$$V_{PN+1} = 0, \qquad t \ge T_C$$

The equation for H_{PN+1} can be found by substituting equation (16) into equation (10) to give:

$$H_{PN+1} = H_N - \frac{a}{g}(V_{PN+1} - V_N) - \frac{a}{g}\frac{f\Delta t}{2D}\,V_N\left|V_N\right| \quad (17)$$

This boundary condition added a complexity to the equation because of having both H_{P1} and V_{P1} in the boundary

condition. Consequently the boundary equation must be solved simultaneously with C- to produce equations for the HP1 and VP1.

The simplest approach that is reasonably appropriate for this calculation is to represent the pump discharge characteristics by a quadratic equation which has the following form:

$$Hp = AP' Q2 + BP' Q + CP' \qquad (18)$$

where Q is the pump discharge and HP is the head increase across the pump. These variables are not identical to those in the C- equation; therefore, some adjustments must be made. Replacing Q with VP1A and HP with (HP1- HSUMP) leading to incorporating HSUMP into CP' and A into AP' and BP' and gives:

$$HP1 = AP\ V2P1 + BP\ VP1 + CP \qquad (19)$$

From the shape of the characteristics curve of the pump, the constants in this equation are having the following limitations: AP < 0, BP < 0, and CP > 0. Simultaneously, solving with C- equation and rearranging leads to the following equation for VP1:

$$\left(\frac{g}{a}A_P\right)V2P1 + \left(\frac{g}{a}B_P -1\right)VP1 + \left(V2 + \frac{g}{a}CP - \frac{g}{a}H_P - \frac{f\Delta t}{2D}V2\,|V2|\right) = 0 \qquad (20)$$

This quadratic equation can now solve for VP1, and as a result the value of Hp1 can be calculated by substituting the last value of Vp1 into equation (19). Determining the parabolic equation coefficients can be done by selecting several points on the pump characteristics diagram and using least square method to find the unknown coefficients. The numerical results of the coefficients are as follows: AP'= -1.05 *10-5, BP'= -6.15 * 10-2, and CP' = 134. Hp = 200 m.

V.   Results analysis

The resistance in the pipes and system can be suddenly changed or in a short time due to a suddenly change in discharge. Figure (9) presents the drop in the head on both flat and steep (H-Q) curves because of the change in the location of operating points, i.e. as presented in resistance curves (1) and (2).
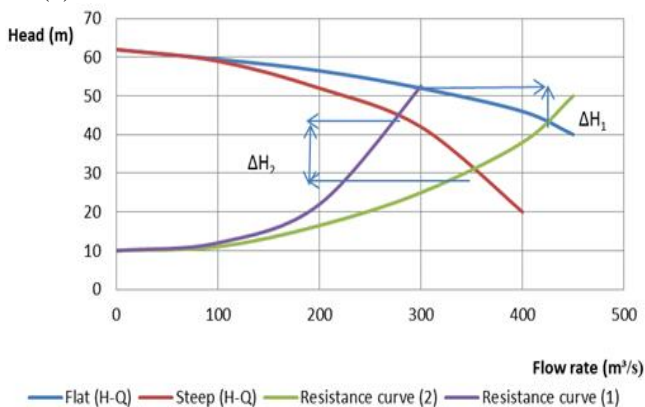

Figure (9) Drop in the head on flat and steep (H-Q) curves due to change in the location of operating points

As shown in figure (9), the sudden drop in pressure occurs on the flat (H-Q) curve (ΔH1) which is smaller than the sudden drop occurs on the steep (H-Q) curve (ΔH2). Hydraulically, this phenomenon means that when the sudden change in the discharge of the pipeline system happens, the transient flow might be occurred in the system driven by

centrifugal pumps that has steep (H-Q) curves. From the interesting point of view, the transient flow analysis used in this paper (method of characteristics) is clearly showed this fact.

In the present study, applying method of characteristics on hypothec simple pipeline system is consisted of using nine pipes and one pump, as sketchily presented in figure (10). Table (1) summarized the pertinent pipe system characteristics. The lengths, diameters, and roughness factors for each pipe number used in this study are listed in Table (1). All the minor losses are neglected to simplify the calculation. Figures (11 and 12) show wave pressures for two different types of pumps (axial flow pump and radial flow pump). The first pump is resulted a flat (H-Q) curve, but in the second one the steep (H-Q) curve is produced. It can be clearly showed that the first type is less wave effects than the second type. This is likely occurred because of the difference in the mechanical design and flow pathway of these pumps.

Table (1) Pipe characteristics for the hypothec pipeline system

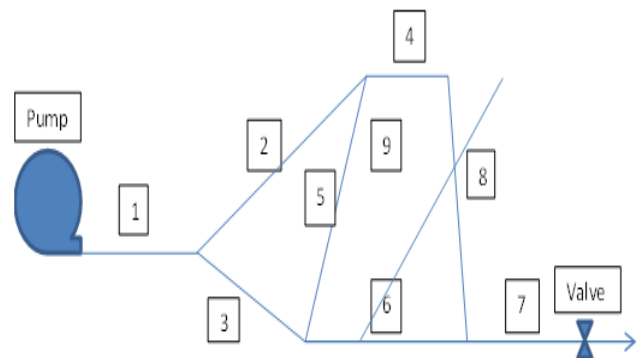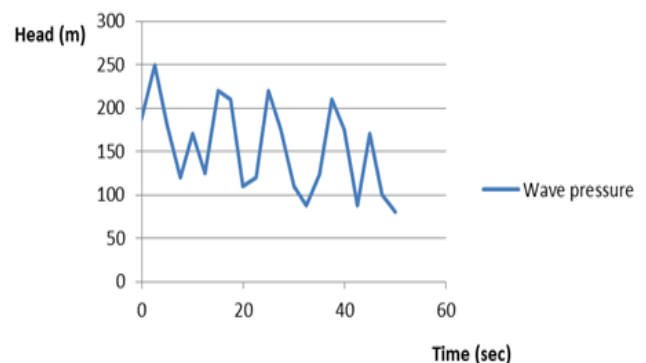| Pipe number	Length(m)	Diameter(mm)	Roughness(f)	Minor loss
1	600	900	0.02	0.0
2	910	780	0.02	0.0
3	600	620	0.02	0.0
4	450	460	0.02	0.0
5	550	460	0.02	0.0
6	670	765	0.02	0.0
7	610	910	0.02	0.0
8	455	610	0.02	0.0
9	490	500	0.02	0.0

Figure (10) Hypothec simple pipeline system

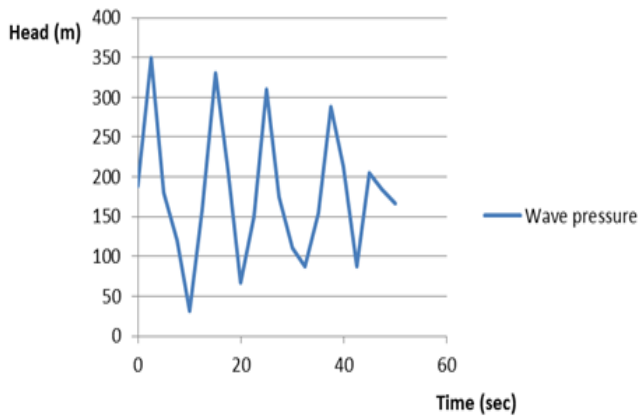Figure (11) Wave pressure for the pump with the flat H-Q curve

Figure (12) Wave pressure for pump with the steep H-Q curve

VI. CONCLUSION

In pipeline systems, the good design and practical operations are essentially required for the transient flow elimination. The appropriate methods of selecting the type and size of centrifugal pump are widely varied. If the entire system is well defined with one set of constant operating conditions, then the selecting of pump type is successfully achieved. The main objective of this research is to give the suitable recommendations in selecting and installation the pumps to eliminate the effects of water hammer. This study gives more flexibility to change the shape of the H-Q curve of the pump by creating a specific requirement to the installation without engaging in the manufacturing process of the pump. And this can be done controlling on the form of the pump characteristics curve as needed in the network and working with the reliability pump. According to the experimental investigations and numerical calculations of this research, the following conclusions and recommendations are given below for the proper setting up of pipeline systems:

1) In practical studies and design, the efforts must be focused on choosing pumps that have the flat H-Q curve, especially in the pipeline systems which include huge lengths and diameters. This is essentially required to eliminate the water hammer phenomenon.

2) For pipeline systems that use the centrifugal pumps, the ground surface level should be more than the water level in order to avoid the suction lift that causes the steep H-Q curve.

3) The suction side must provide a good $NPSH_a$ because of the significant effect of this powerful parameter on the H-Q curve and leads as a result to get the flat shape. This is eliminated the sudden change in pressure due to the impulsive change in discharge.

4) The axial flow pump is very useful in operation. This type of pump is not needed to the high NPSH as the centrifugal pump (radial flow pump) because of its flow mechanisms, and it only needs a minimum submergence.

REFERENCES

[1] Bruce E. Larock and Roland W. Jeppson,' Hydraulics of PipelineSystems', 2000.
[2] X. Wang M. Lambert and A. R. Simpson,' Analysis of Hydraulic Transient in Pipelinewith a Leak Using Laplace Transform,' 2001.
[3] Don J. Wood,' Journal of Environmental Engineering', Vol. 131, No. 8, August 1, 2005.
[4] Report produced for the European Commission, 'Study on improving the efficiency of pumps', SAVA, 2001.
[5] Maher Abdul Ameer,'Numerical Comparison of Transient Flow For Simple Pipeline Systems Using Time Domain and Frequency Domain Method', Al-QadisyaJornal For Engineering Sciences, Vol.5, No.2,209-224, Year 2012.
[6] Randall W. Whitesides CPE, PE, Selecting a Centrifugal Pump bySystem Analysis, 2012.
[7] JacquesChaurette P. Eng.' Tutorial Centrifugal Pump Systems', 2005.
[8] LarryBachus and Angel Custodio,' Know and Understand CentrifugalPumps', 2003.
[9] PareshGirdher and Octo Moniz,' Practical Centrifugal Pumps', 2005.
[10] Robert L. Sank, Ph.D., PE,' Pumping Station Design', 1998.
[11] Bryan W. Karney, 'Journal of Hydraulic Engineering', Vol. 118, No. 7, July, 1992.