SIZING AIR VESSELS FOR WATER HAMMER PROTECTION IN WATER PIPELINES

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المخلص

تدرس هذه الورقة سلوك التدفق العابر على امتداد خطوط أنابيب المياه نتيجة حياد المضخة، بما يشمل تحديد أحجام الغرف الهوائية اللازمة بما في ذلك حجم الهواء وحجم السائل داخل كل غرفة وأقطار المنافذ والمداخل للغرف. هذا العمل أخذ في الحسبان النموذج الرياضي الذي قدمه "ستيفنسون"، عام 2002، حيث تبني سلوك التدفق العابر عديم الاحتكاك في نموذجه. في هذه الدراسة، تم تعديل النموذج الرياضي لاستيعاب بند الاحتكاك حيث تم تحليل وتقييم النموذجين تحت ظروف مختلفة من التشغيل والهيئة الهندسية باستخدام برنامج أعد لهذا الغرض. وقد طرحت حالة دراسية، حيث أختبر حياد مضخة في منظومة أنابيب مياه طويلة وطور العمل ليشمل آثار القطر وطول خط الأنابيب وضغط الرفع الساكن ومعدل التدفق المستهدف على خصائص الغرفة لكلا النموذجين المدروسين. وكان تغير حجم الغرفة واضحاً وخطياً مع معدل التدفق لحالة التدفق عديم الاحتكاك في حين أنه أكبر ويأخذ شكلاً تربيعياً في حالة نموذج التدفق الاحتكاكي. جميع الأحجام المطلوبة للغرفة كانت في حدود القيود المفروضة بواسطة البرمجية التجارية " بنتلي– هامر"

ABSTRACT

This paper studies the transient flow behavior along water pipelines due to pump trip, including determination of the required air vessel sizes, the volumes of air and liquid inside the vessels, and the diameters of the vessel entrance and exit ports.

This work has considered the mathematical model presented by Stephenson, 2002, where the frictionless transient flow behavior is adapted. The mathematical model is modified to accommodate the frictional term. Both models are solved and evaluated under different geometrical and operating conditions. A software is developed and employed where both results, with and without friction, are documented and compared.

A case study is considered, where a pump trip is experienced in a long water pipeline system. The work is extended to include the effects of the diameter and length of the pipeline, static pressure head, and desired flow rate on the vessel characteristics. This is done for both considered models, where the variation of the vessel size is pronounced and linear with flow rate for frictionless, while it is larger and parabolic for the frictional model. All required vessel sizes were within the limitations imposed by the commercial Bentley-Hammer software.

KEYWORDS: Transient Flow; Water Hammer; Pressure Surges; Pipelines and Pumping Systems; Pump Trip; Air Vessels.

INTRODUCTION

Generally, any event that causes a sudden change in the velocity of the fluid in a pipeline will generate transient pressure waves. Pressure transients can be positive or negative, where the magnitude of these surges can attain a value of many times of the normal operating pressures. The most common sources of transient pressures are pump operation, pump power failure, control valve operation, and pipeline rupture. In case of pump power failure, the initial wave is a negative or reduced pressure wave, which travels from the pump discharge side toward the end of the pipeline. A variety of controlling methods are available to mitigate transient pressures, generally, falling into three categories; alteration of pipeline profile and diameter, valve and pump control procedures, and surge control devices. Different surge control devices are employed, such as surge tanks, surge pipes, air valves, and air vessels [1,2].

Air vessels, Figure (1), generally alleviate negative pressures more effectively than other forms of water hammer protection units, and they can maintain a positive pressure in the line at all stages following the pump trip [3]. This is accomplished by forcing water out of the vessel into the cavity. The vessel's compressed air forces water moving from the air vessel into the pipeline, allowing the water column traveling up the pipeline to maintain its momentum. Friction and other head losses tend to reduce the water velocity and therefore the subsequent oscillations. Thus, some degree of flow throttling is often used in conjunction with the cushioning effect of air vessels [4,5].

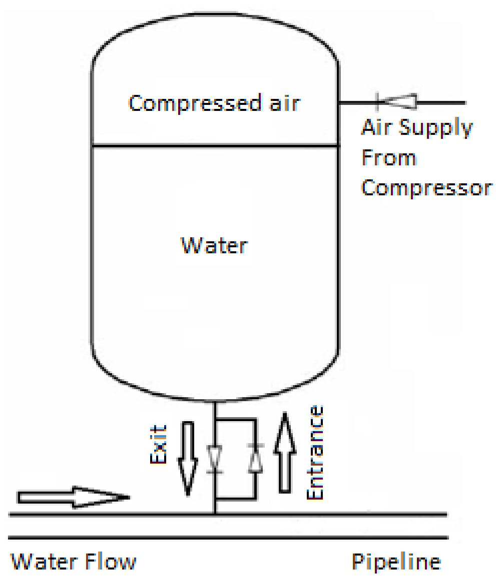

Figure 1: Schematic of air vessel

STEPHENSON'S MATHEMATICAL MODEL

Stephenson's model is based on the rigid theory and on neglecting effects of friction [6]. The rigid model theory suggests the following relationship between decelerating head on a water column and the flow deceleration;

$$h = -\frac{L}{g}\frac{dV}{dt} \tag{1}$$

Where h is the average head difference across the water column, L is the length of the pipeline, g is the gravitational acceleration, – dV/dt is the flow deceleration term, and V is the velocity of flow measured positively in the direction of flow. By the integration of equation (1) with the use of the continuity equation, the required volume of water that the air vessel would force into the pipeline behind the water column, can be given by;

$$\forall_w = \frac{ALV_o^2}{2gh} \tag{2}$$

Where A is the cross-sectional area of the pipe and Vo is the initial water velocity. For rapid expansion, usually in small air vessels, the air expansion process may be considered adiabatic process with negligible heat exchange with surroundings. However, for slower expansion related to relatively large air vessels, heat exchange is expected to take place with surroundings, where the process tends to be isothermal one, Graze and Horlacher [7]. Referring to the perfect gas law, for an isothermal air expansion process, that properly occurs in such large pipeline applications, with employing equation 2, the following relation can be arranged as follows [8];

$$\forall_O = \left(\frac{H_O}{h_{min}} - 1\right)\left(\frac{ALV_O^2}{2gH_O \frac{h_{min}}{2H_O}}\right) \tag{3}$$

Where Ho is the initial absolute operating pressure head of the air in the vessel with an initial air volume of $\forall_o$, and h_{min} is the most severe pressure head drop of the air in the vessel.

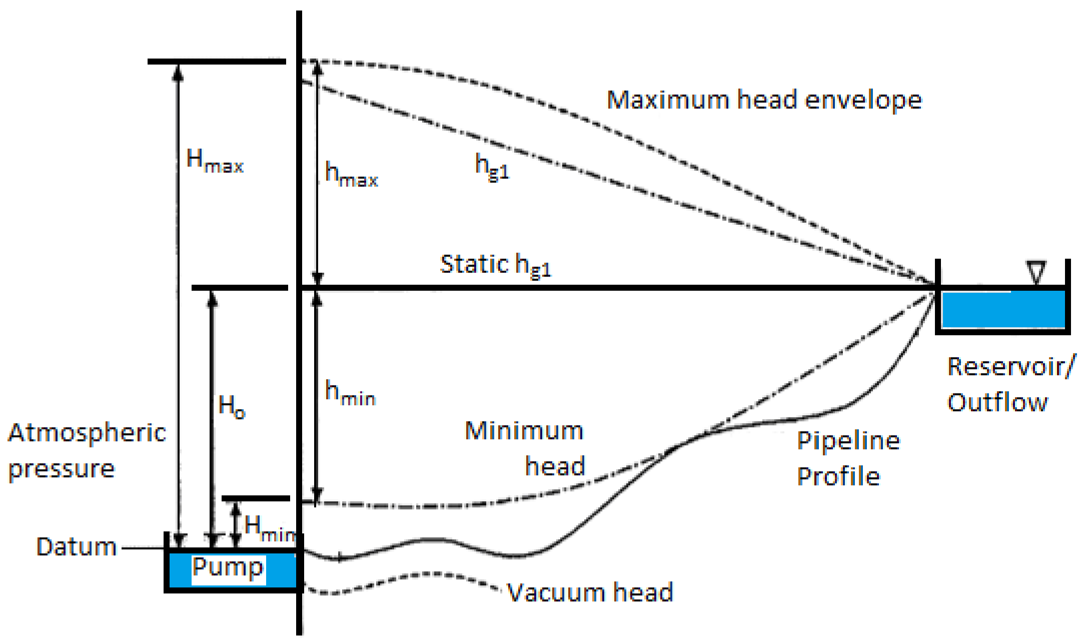

Figure 2: Pipeline profile with definitions of pressure heads [6].

OUTLET PIPE SIZE

The outlet pipe from an air vessel can be throttled to reduce the outflow volume and to more rapidly decelerate the water column in the pipeline. The outflow from the vessel is out of phase with the minimum pressure, so the maxima of the two pressure drops, air expansion and outflow head loss, cannot be summated. The maximum rate of outflow occurs immediately after pump trip, when the air in the vessel is still at operating pressure. Referring to the minor head loss representation and employing the continuity equation, the pressure head loss takes the following form [8];

$$h_l = KV_o^{\,2}(D_p/D_e)^4/2g \tag{4}$$

Where K is the loss coefficient accounting for the entrance contraction and expansion losses plus pipe work losses. Referring to the field practice cited in a number of references, this is typically taken to be 2. D is the pipe diameter, with subscript e referring to the exit vessel pipe and with p is related to the main pipe.

The permissible head drop in the main pipe should be less than the static head Ho, and often only a small fraction of Ho to avoid negative heads further along the line. The actual permissible head drop should be determined from the pipeline profile, which is as typically shown in Figure (2), the head sag curve is convex down and the pipeline profile convex up, so such a case, a head loss of Ho/2 is more typical. Then solving for D_e /D_p, we obtain the following;

$$D_e/D_P = (2V_O^{\,2}/gH_O)^{1/4} \tag{5}$$

INLET PIPE SIZE

The return surge wave, the reversal of the water column in the main pipe, compresses the air in the air vessel, gradually decelerating the water column. The maximum compression of the air, which coincides with the maximum pressure in the pipeline, occurs when the water column has stopped. On the other hand, the maximum head loss into the air vessel occurs when the water column is at maximum velocity, i.e., It is out of phase with gas compression, so it can be designed to be a maximum, subject to not exceeding the gas compression head. This head loss can be further utilized to decelerate the water column, requiring less air to cushion the surge at its furthest extent. From equation 1, and since the average decelerating head would be $h_{\min}/2$, the maximum reverse flow velocity in the main pipe can be arranged to be;

$$V_r = V_o - \frac{2gh_{\min}\cancel{V}_o}{LQ_o} \tag{6}$$

Where Q_o is the initial volume flow rate. In the operational mode under a protection air vessel in the system, one may desire to limit the critical pressure in the line. For example, if the maximum head rise in the line is to be limited to 0.1Ho, thus we may write;

$$h_{\max} = 0.1H_o = \frac{2V_o^{\,2}}{2g}\left(\frac{D_P}{D_i}\right)^4 \tag{7}$$

Employing equation (6), the air-vessel inlet-diameter ratio could be written as;

$$\frac{D_i}{D_p} = \left[\frac{10V_O^{\,2}}{gH_O}\right]^{1/4} = \left(\frac{10}{gH_o}\right)^{1/4}(V_o - \frac{2gh_{\min}\cancel{V}_o}{LQ_o})^{1/2} \tag{8}$$

In terms of h_{max}, the instantaneous air volume in the vessel can take the following relationship;

$$\cancel{V} = \frac{ALV_o^{\ 2}}{gh_{max}} \tag{9}$$

Accordingly, equation (8) can be simplified and the air-vessel inlet-diameter could be written as follows;

$$D_i = \frac{1}{\sqrt{2}}\left[\frac{V_O^{\ 2}}{gh_{max}}\right]^{1/4} D_p \tag{10}$$

DEVELOPED MATHEMATICAL MODEL WITH FRICTION

Here, the flow frictional effect is considered to know the consequence on the upsurge and down surge pressures and the required time to damp such surges. Referring to equation (1) and Darcy's frictional head loss model, the pressure head loss in transient flow is [8];

$$h = \frac{fL}{2gD}V_O^{\ 2} - \frac{L}{g}\frac{dV}{dt} \tag{11}$$

and the required time to damp such pressure surges is;

$$\int dt = \frac{L}{g}\int \frac{dV}{\frac{fL}{2gD}V_o^{\ 2} - h} \tag{12}$$

Where f is the friction factor, other variables are as defined above. Now, using the partial fraction integration to integrate the right-hand term, the above equation becomes;

$$T = \sqrt{\frac{LD}{2gfh}}\ \ln \frac{\sqrt{\frac{2gDh}{fL}} + V_O}{\sqrt{\frac{2gDh}{fL}} - V_O} \tag{13}$$

The time T over which the deceleration occurs is related to the volume of water in the air vessel that would be forced into the pipeline in the vacuum region behind the water column. Considering the water volume in the vessel is equal to half of the vessel volume, then the water volume in the vessel, we find;

$$\cancel{V}_w = \frac{V_O A}{2}\sqrt{\frac{LD}{gfh_{min}}}\ \ln \frac{\sqrt{\frac{gDh_{min}}{fL}} + V_O}{\sqrt{\frac{gDh_{min}}{fL}} - V_O} \tag{14}$$

Substituting equation (2) into equation 14, the required initial air volume $\cancel{V}_o$ can be obtained as follows;

$$\cancel{V}_o = \left(\frac{H_o}{h_{min}} - 1\right)\left(\frac{V_o A}{2}\sqrt{\frac{LD}{gfh_{min}}}\ \ln \frac{\sqrt{\frac{gDh_{min}}{fL}} + V_o}{\sqrt{\frac{gDh_{min}}{fL}} - V_o}\right) \tag{15}$$

AIRV- SOFTWARE

A computer program is written in the visual basic language, covering the above two mathematical models, with and without frictional effects. This software is titled as "AirV-

Software" which is capable to calculate various pressure heads along the pipeline under different conditions. In each case, the volume of the required air vessel(s) for the protection of the system is evaluated including obtaining both volumes of air and water inside the desired vessel. Referring to Stephenson's frictionless model and the developed software [6], Figure (3) presents the dimensionless volume of the air inside the air vessel, where the volume increases roughly parabolically with the dimensionless allowable minimum head. Figures (4) shows the initial volume of the air vessel, for both Stephenson's and AirV-software. According to the results, both have same trend with good agreement especially for low minimum head range.

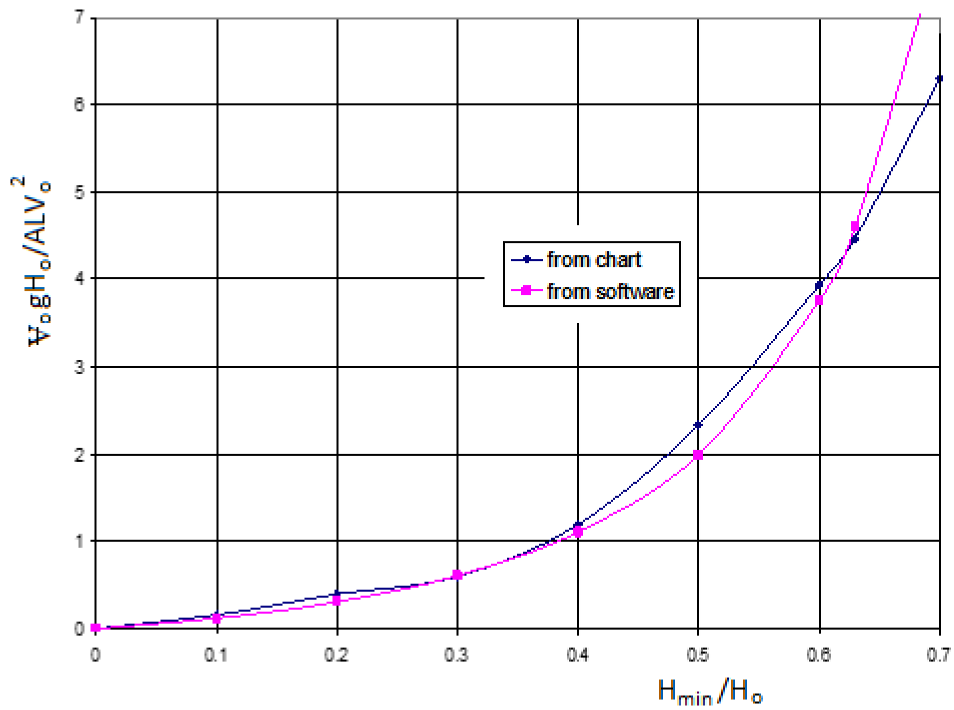

Figure 3: Dimensionless volume of the air inside the air vessel.

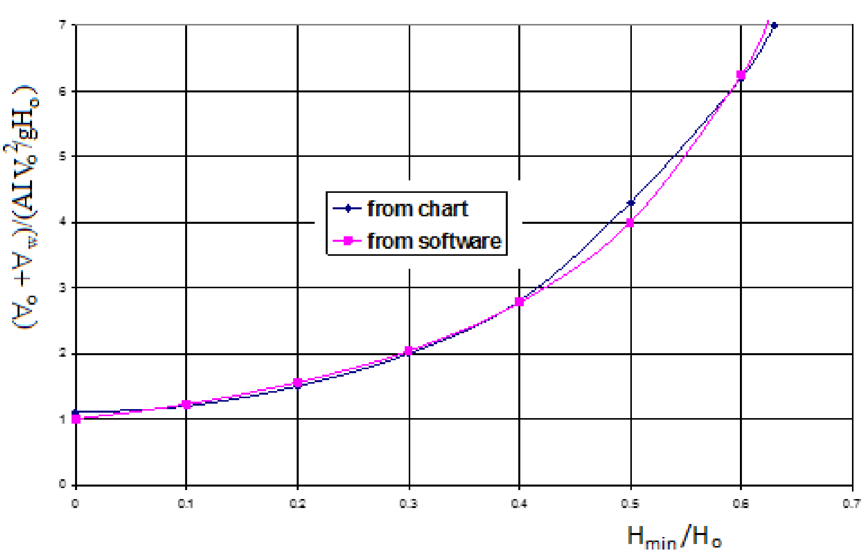

Figure 4: Dimensionless initial volume of the air vessel.

RESULTS AND DISCUSSIONS

The airV- software and Bentley-Hammer software are used to evaluate the considered transient flow behavior [9]. The developed software, airV, is verified and the results are compared and discussed. Here, a published hydraulic-system case study is introduced in order to apply both flow mathematical models. The work is extended to include the effect of the diameter and length of the pipeline, static pressure head, and amount of the flow rate.

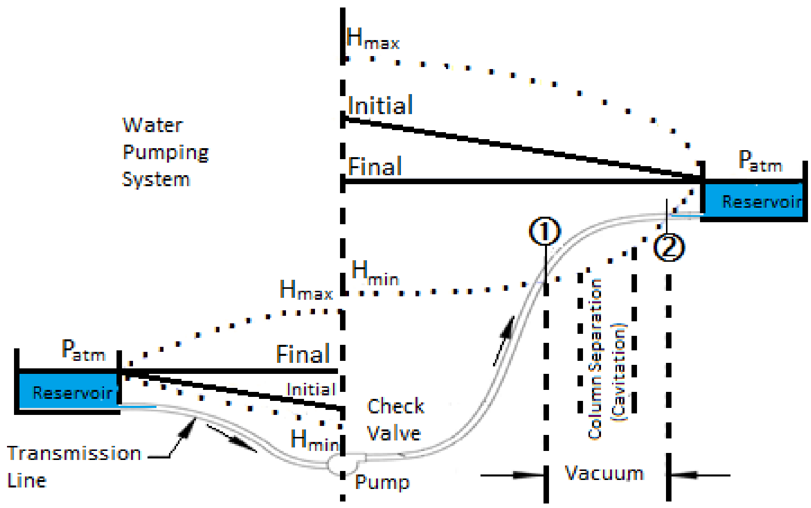

Figure 5: Case study for a pumping system suffers from a pump failure [6].

Case Study

Let us consider the case study presented elsewhere [6], Figure (5). The hydraulic system including a pumping station with negligible inertia effect, a pipeline of 900 mm in diameter and 18,000 m long, with a static head of 410 m, and conveys water at an initial velocity of 1.4 m/s and a temperature of 20°C. The air vessel characteristics are calculated below to limit both, the minimum pressure head to 40% of the static head, or 164 m water head, and to limit the maximum pressure head to 40% above the static head, or 574 m water head.

Effect of Friction

Referring to Figure (6), considering the studied head ratio range of 0.0-0.7, the trends of the peak-related pressure results indicate that the required volume of the vessel for the frictionless model is always larger than that required for the frictional model. For the above considered conditions, the required maximum volume of the vessel is 81.64 m^3 for the frictional model, where the maximum air and water volumes in the vessel are 57.14 and 24.5 m^3, as indicated in Figures (7 and 8), respectively.

However, for the desired minimum pressure head, the trends are vice versa, where the required volume of the vessel for the frictionless model is always smaller. The dimensionless required volumes of the air vessel are approximately 0.21 and 0.18, the required, air volumes are 0.1 and 0.12, and the required water volumes are 0.37 and 0.42, for frictional and frictionless models, respectively.

These results confirm the role of the frictional presence, where it is leading to damp the peak pressure heads and to lower more the bottom pressure heads. Hence, the frictional process has an advantage for the hydraulic systems that suffer from high transient pressure waves, while the pipe friction is not favorable for systems with negative pressure waves, as frictional contribution is towards lowering the pressure more.

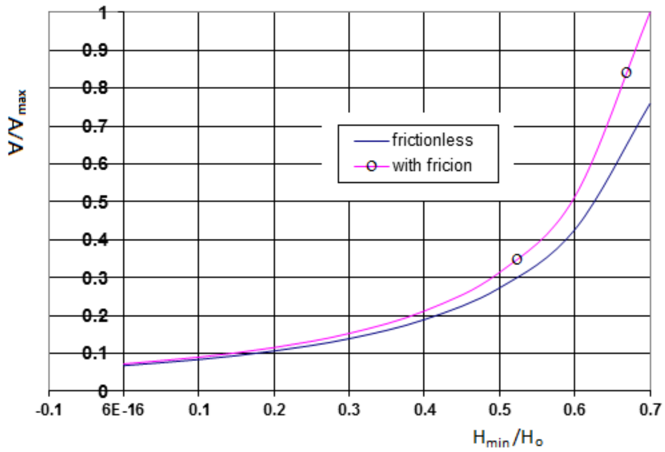

Figure 6: Dimensionless volume of the air vessel for both models; with and without friction.

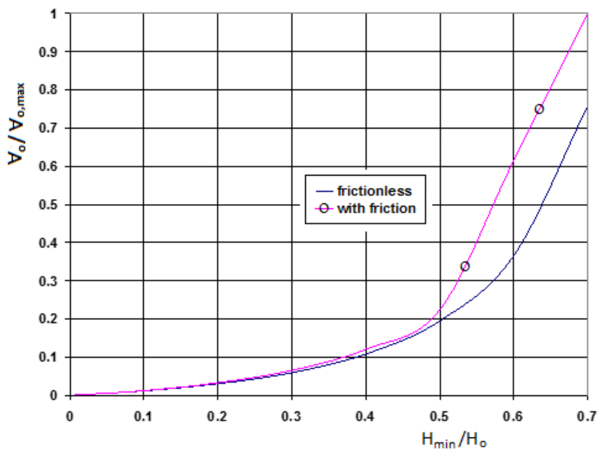

Figure 7: Dimensionless volume of the air inside the air vessel for both models with and without friction.

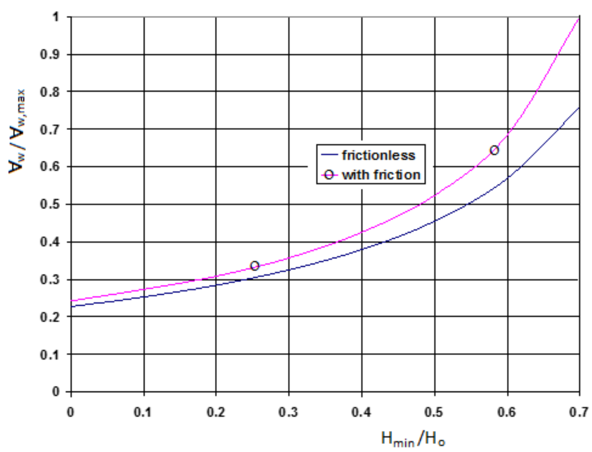

Figure 8: **Dimensionless volume of the water inside the air vessel for both models; with and without friction.**

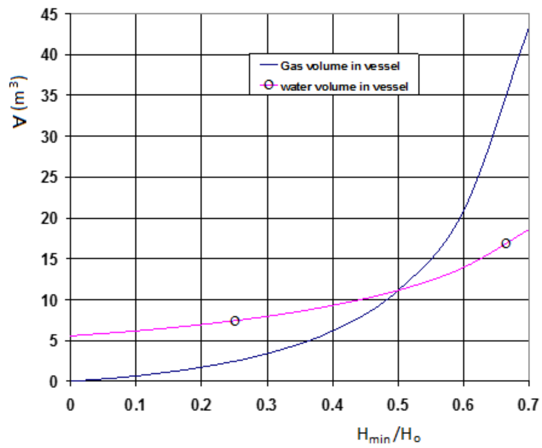

Figure 9: **Air and water volumes versus the relative minimum Static head for frictionless model.**

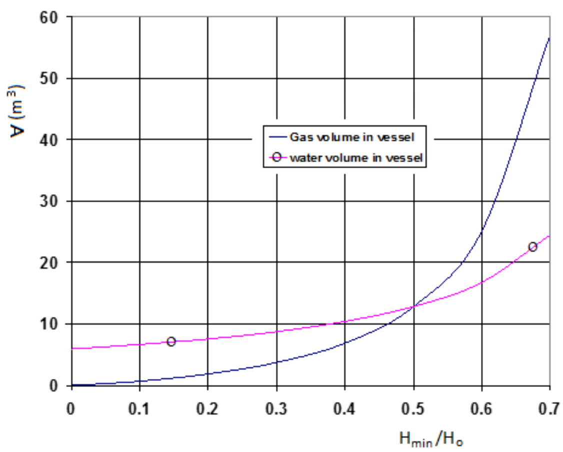

Figure 10: **Total air and water volume versus the relative minimum static head for frictional model.**

Figures (9 and 10) indicate the effect of the desired minimum static head, where for negligible minimum head is desired, H_{min} =0, no compressed air is required inside the vessel for both models, while water is essential by volumes of 5.57 m^3 and 5.94 m^3 for frictionless and frictional models, respectively.

As the minimum static head, H_{min}, increases, the size of the vessel and air volume increase, where there is a neutral point at $H_{min}/Ho = 0.5$ at which air and water volumes are identical. Also, we may see that for high values of static minimum head, the air volume exceeds the volume of water inside the vessel, this could be interpreted as the head increases as the need to absorb the positive pressure wave becomes larger.

Effect of Pipe Diameter

In order to evaluate the effect of changing the pipe diameter, the Hazen-Williams and Chezy-Manning formulas are used, where the Hazen-Williams roughness coefficient, C, is taken to be equal to 100. Figures (11, 12, and 13) introduce the volumes of the vessel variations with pipe diameters. For example, for a system with a pipe diameter of 0.6 m, the air, water, and vessel volumes are 1.57, 2.36, and 3.94 m^3 for frictionless model, respectively. For frictional model, they are 1.8, 2.7, and 4.5 m^3, respectively.

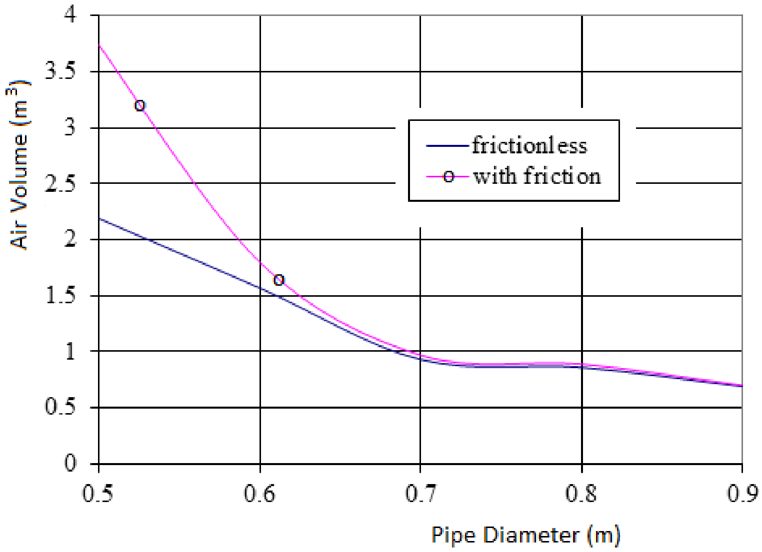

Figure 11: **The volume of the air inside the vessel for both models; with and without friction.**

For fixed volume flow rate, the general trend of the above results indicates that the required size of vessel for with and without friction models decreases with the increase of the pipe diameter. This result seems to be obvious because for fixed volume flow rate, larger pipe diameters lead to have low velocities in the pipe. This leads to less sever surge pressures, which require relatively smaller air vessels. However, for smaller pipe-diameter systems, higher flow velocity makes the negative pressure waves decrease more and this requires larger air vessels to overcome the vacuum occurred.

Referring to Figures (11-13), under the considered conditions, the effect of friction decays for pipe diameters larger than 0.7 m, also the volumes of air and water are equal for the large pipes, where small vessel sizes of less than 2.3 m^3 are required. in this regard, large-diameter pipes seems to behave as vessels themselves.

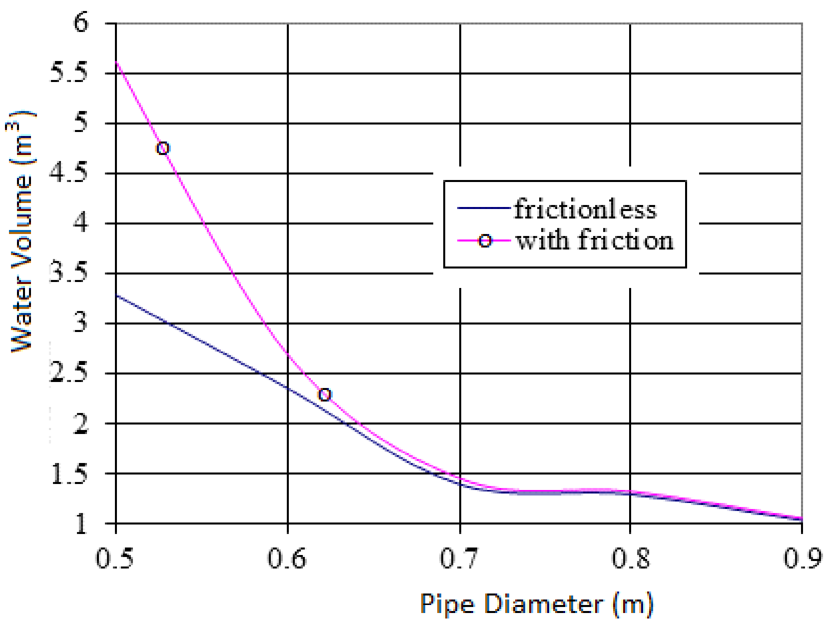

Figure 12: **The volume of the water inside the air vessel for both models; with and without friction**

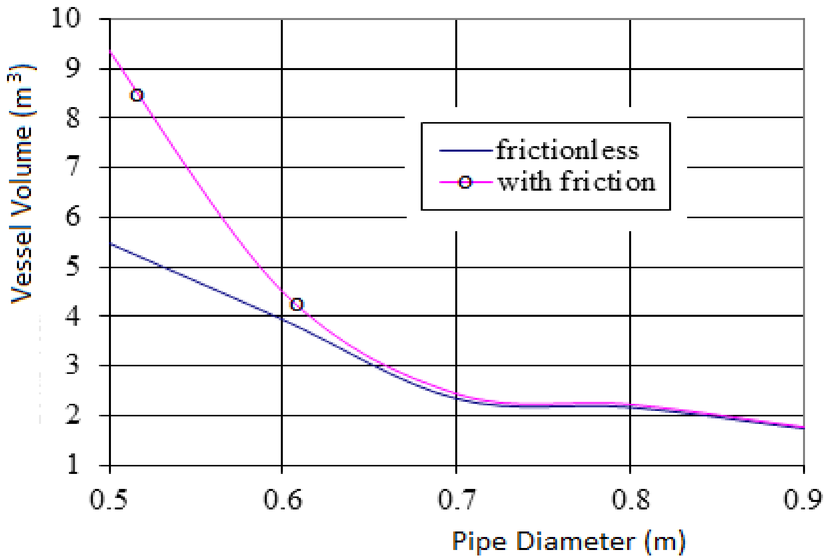

Figure 13: **The volume of the vessel for both models, with and without friction**

Effect of Pipe Length

The trends of the results indicate that the required vessel volume for both models increases almost linearly with increasing the pipe length. Referring to Figures (14, 15), and 16, for L=18,000 m, the air, water, and vessel volumes are 6.19, 9.29, and 15.49 m^3 for frictionless model, and 6.93, 10.4, and 17.33 m^3 for frictional model, respectively. One can see the volume deviation among both models increases with pipe length.

This result seems to be affected by the time that pressure wave takes to travel to the end of the pipe and return to the location where the vessel is located. As the period increases, as the induced vacuum region grows, and the need for more recovery increases through larger air vessels. However, for the up surge wave, the opposite is correct, where; damping of such waves is pronounced as the pipe becomes lengthy.

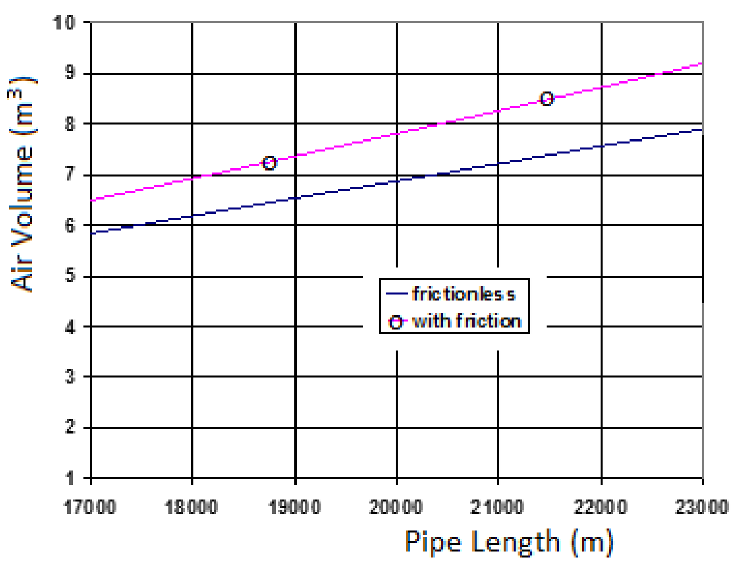

Figure 14: **The volume of the air inside the air vessel for both models; with and without friction.**

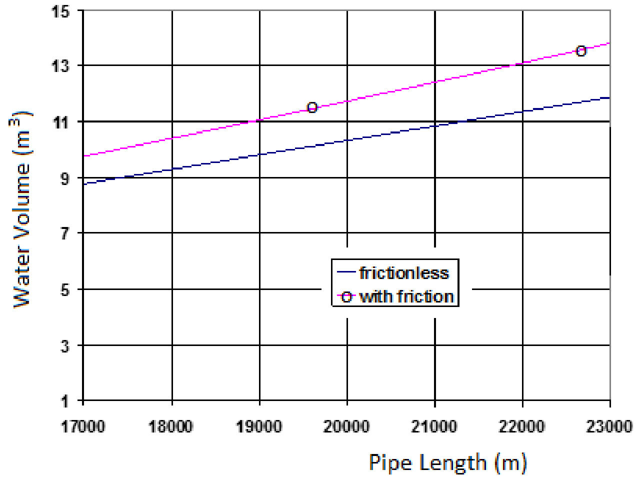

Figure 15: **The volume of the water inside the air vessel for both models; with and without friction.**

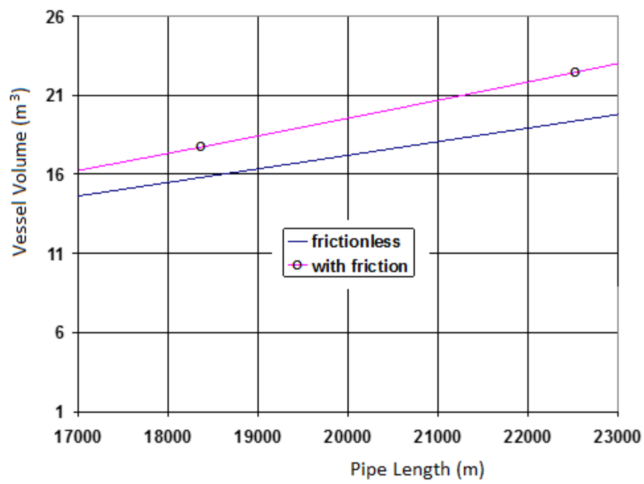

Figure 16: **The volume of the air vessel for both models; with and without friction**

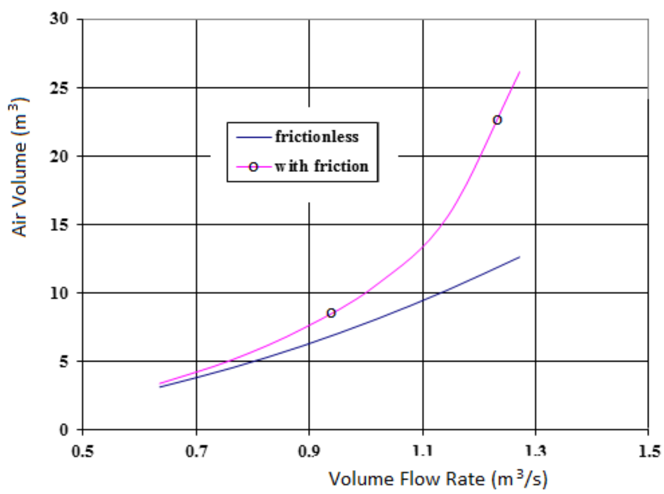

Figure 17: **The volume of air inside the air vessel for both models; with and without friction**

Effect of the Volume Flow Rate

Taking into the consideration the variation of the friction factor, Figures (17, 18), and 19 introduce the variations of the required size of the vessel with change of the volume flow rate. For the water flow rate of 1.14 m^3/s, the required air, water, and vessel volumes are 10.24, 15.36, and 25.61 m^3, respectively, for frictionless model. However, for frictional model, the volumes are 15.6, 23.41, and 39.02 m^3, respectively. Therefore, the required volume of the vessel grows up as the volume flow rate increases, for both models.

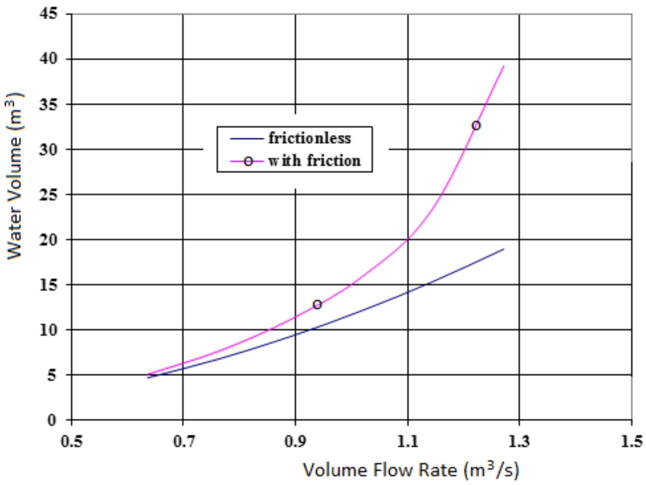

Figure 18: The volume of water inside the air vessel for both models; with and without friction

Here, for the frictionless model, the volumes are nearly linear with the flow rate, where the size of the vessel can take the following approximate straight line relation;

$$\forall = -17.69 + 38.46\ Q$$

Where $\forall$ in m^3, and Q in m^3/s. Referring to this result, no vessel is needed for flow rates equal or less than Q=0.46 m^3/s. However, for the frictional model the required vessel volume takes a parabolic form, which could be related to the relationship among the frictional head loss and the squared flow rate, Q^2.

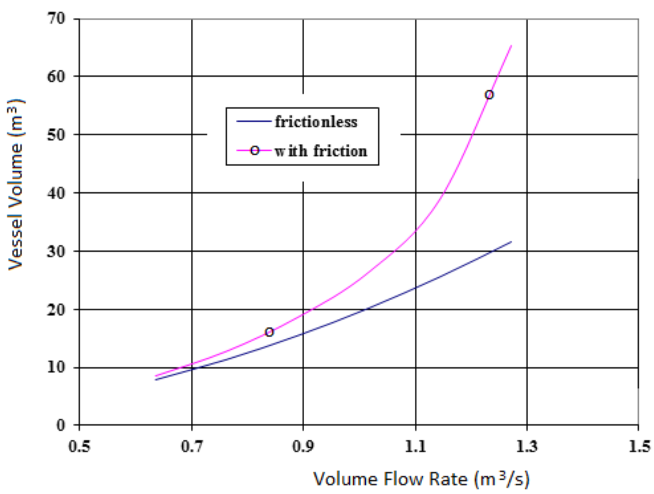

Figure 19: The volume of the air vessel for both models; with and without friction

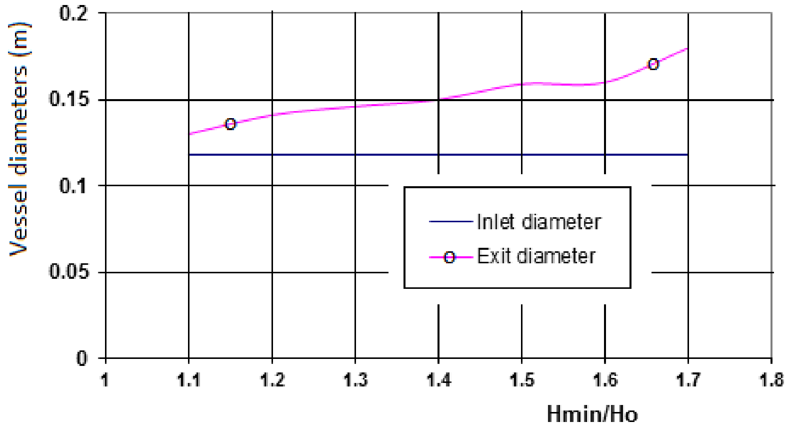

Figure 20: Inlet and outlet diameters of the air vessel with minimum head ratio.

Vessel Inlet and Outlet Ports

For a positive pressure surge, the high pressure wave should be damped, by allowing the liquid in the line entering into the vessel with a carful throttling process. This has what to do with the inlet diameter of the vessel. This is done carefully to avoid another transient source of sudden flow stoppage. In the other side, for the negative pressure surge, the air vessel should deliver the liquid into the line in order to compensate the reduction in the line pressure avoiding the creation of a cavity [8].

Figures (20 and 21) show the trend of the change of the vessel port diameters with the anticipated minimum and maximum static pressure heads, respectively. These results indicate that the vessel inlet diameter is mostly affected by the expected change in the maximum head, while the outlet diameter is affected by the change in the minimum pressure head developed downstream of the pump. These reflect nicely the role of the installed air vessel downstream of the pump.

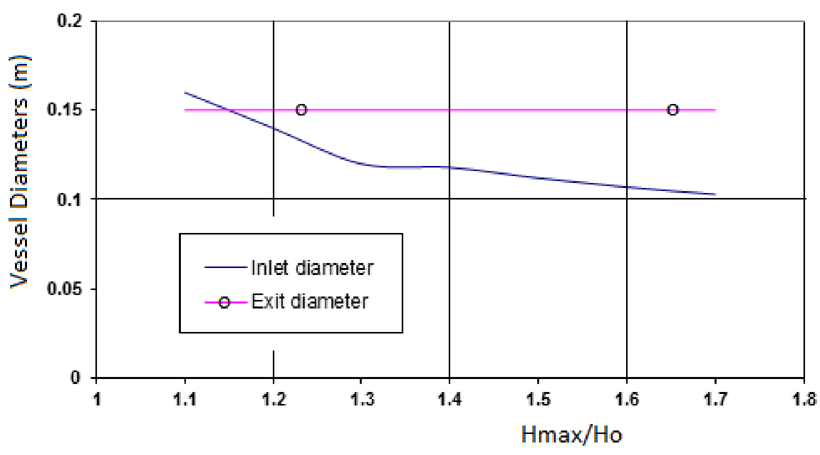

Figure 21: **Inlet and outlet diameters of the air vessel with maximum head ratio.**

BENTLEY-HAMMER SOFTWARE

The commercial Bentley-Hammer software is employed to check whether or not the specified vessel sizes are appropriate for the protection of the studied system under the specified geometrical and operating conditions. This stage is presented elsewhere [8], where all vessel sizes presented above were within the limitations imposed by Bentley software.

CONCLUSIONS AND RECOMMENDATIONS

Based on the results of this study conclusions and recommendations can be summarized in the following points:

- The consequences of pump trip is tremendous, especially in certain pressure-surge enhancement conditions. The frictional up or down pressure-wave role is present, and this frictional effects are decaying as pipe-diameter gets larger.
- Large-diameter pipes lead to less sever surge pressures and to have equal air-water volumes in the required relatively small vessel, where large-diameter pipes contribution is effective in limiting the consequences of surge pressures.
- Long pipes need more linear recovery for down pressure surge, however, for the up surge wave, the opposite is correct; damping of such waves is clear as the pipe length increases. Referring to the frictional and frictionless models, the required linear vessel size deviates more with pipe length increase.
- For the frictionless flow model, the size of the air vessels are nearly linear with the flow rate, where a related formula is obtained. However, for the frictional model the required vessel volume takes a parabolic form. The vessel entrance pipe size is mostly affected by the expected change in the maximum pressure head, while the exit pipe size depends on the change of the minimum pressure head.
- With developed easy AIRV-software, vessel sizes can be obtained under different geometrical and operating conditions, where for example, Bentley-Hammer software could be employed to confirm such results.

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NOMENCLATURE

A	=	cross-sectional area of pipe
De	=	diameter of outlet pipe
Di	=	diameter of inlet pipe
Dp	=	main pipeline diameter

G = gravitational acceleration
H = head above pump level plus atmospheric head in meters of liquid (absolute head)
H_{max}= maximum head above pump level plus atmospheric head
H_{min}= minimum head above pump level plus atmospheric head
H_o= static head above pump level plus atmospheric head
h= head difference along pipeline at given point in time
h_{max}= maximum head above H_0 ($H_{max}=H_o+h_{max}$)
h_{min}= minimum head below H0 ($H_{min}=Ho - h_{min}$)
K= head loss coefficient
γ= gas expansion coefficient
L= length of pipeline
p= pressure
Qo = initial flow rate
$\cancel{V}$ = air vessel volume $= \cancel{V}_o + \cancel{V}_w$
$\cancel{V}_w$ = initial water volume in vessel
$\cancel{V}_o$ = initial gas volume in vessel
T= time to decelerate water column
t= time
Ve= velocity of liquid (water) in outlet pipe from vessel
Vr= return velocity in pipeline
V_o= initial pipeline water velocity
x= distance along pipeline from pump