The Influence of Air Vessel Volume on the Delivery Flow Rate and Efficiency of a Hydram Water Pumping System

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ABSTRACT - *Air vessels have extensively been used in hydrams with the intention to improve their efficiencies but without having any specific air vessel sizing criteria to achieve the optimal sizes and better efficiencies. As a result, different air vessel sizing practices had been used to manufacture the unnecessarily costly and heavy hydrams. In this study, both the theoretical and experimental studies were conducted in an attempt to investigate the existing relationship between an air vessel size (volume wise) and the delivery flow rate and even the system efficiency while using a standard size of a known hydram size. In the study, different air vessel specimens whose capacity sizes V_s ranged from 0.12 to 19.6 litres were tested with a 40 mm size conventional hydram. The findings showed that an air vessel size V_s was affecting the delivery flow rate q and hence the system efficiency E_{vf} of a hydram pump. When V_s was too small nearing a zero volume, the system efficiency became poor and as V_s was increased beyond a value above zero there was a corresponding gradual increase in the efficiency E_{vf} until a certain critical value was attained whereby a further increase in V_s had only an insignificant increase in the efficiency E_{vf}. Conclusively, an efficiency model and an air vessel size selection criteria were established in this study. The experimentally tested air vessel specimens showed better efficiencies of $E_{vf} \geq 60$ % when the ratio (V_s / V_d) ranged between 50 and 500.*

Key Words: Hydram, water hammer, surge pressure, beat frequency, air chamber pressure air vessel volume

1. INTRODUCTION

For several decades now, an air vessel has been believed to be a vital component in improving the pumping efficiency of a conventional hydram water pumping system (Rajput, 2009; Ojha et al., 2011). However the matching of an appropriate air vessel volume to a hydram pump of a particular size had so far been irrationally handled as there were several differing opinions from several hydram pump researchers. For example a researcher Mitchell (1977) had argued that there was no maximum limit for the air vessel size (volume - wise) to be used in hydrams. Likewise in the recent hydram pump developments, a designer named Bamford, (Bamford, 2002) was reported to have made a hydram pump which could pump water without necessarily using an air vessel in its operations. The latter had posed a big challenge on the necessity of using air vessels in hydrams. Apart from the use of air vessels in hydrams, it was also learnt that air vessels were widely used in several other engineering applications like in turbine power generating systems and large water pumping stations for the purpose of absorbing shock energy from the sudden pressure changes or surges and also making temporary fluid energy storages (Jaeger, 1977; Parmakian, 1963 and Fox, 1977. Until recently when Bamford, (Bamford, 2002) introduced an improved hydram design which could operate without using an air vessel, there was a traditional believe that an air vessel was an essential component in a hydram water pumping system for its viable operations. As a result, the usability of an air vessel in hydram operations was now questionable and needed to be revised. An in depth study was launched and this had mainly focused at studying the basic role and influence of an air vessel component to a hydram water pumping system. This study had an objective of acquiring a better scientific explanation which could assist in judging as to whether an air vessel had any significant influence on the hydram operations or not and if it had, how did it influence and in which areas. The forthcoming sections of this paper are therefore presenting the relevant studies conducted on an air vessel used in hydram operations and the established relevant mathematical models and relationships which emerged between an air vessel volume V_s and the hydram pump performance in terms of its discharge volume V_d , delivery flow rate q and the system efficiency $E_{vf.}$ Some other presented sections include the applied methods for experimental data collection, results, discussions and conclusion.

2. LITERATURE REVIEW AND ANALYSIS OF THE AIR CHAMBER PRESSURE OF A HYDRAM WATER PUMPING SYSTEM

A model set up of a conventional hydram water pumping system was devised as shown in Fig. 1 whereby water was allowed to flow from a continuous supply source and fed into a supply tank which was raised up to a height H from the hydram base. A controlled flow with a uniform velocity v was discharged from the tank and conveyed to the hydram pump via the drive pipe whose diameter was D1 and having a pipe length L.

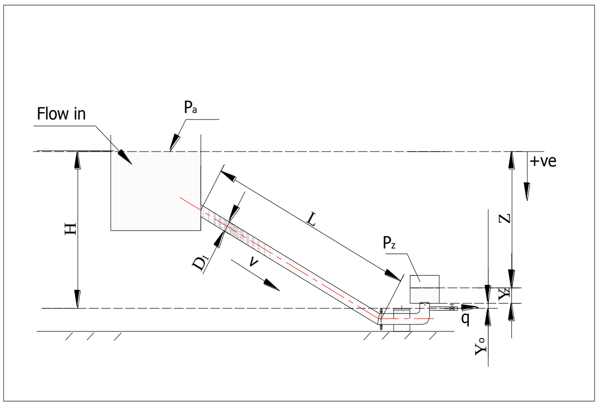

Fig. 1: A setup of a conventional hydram water pumping system

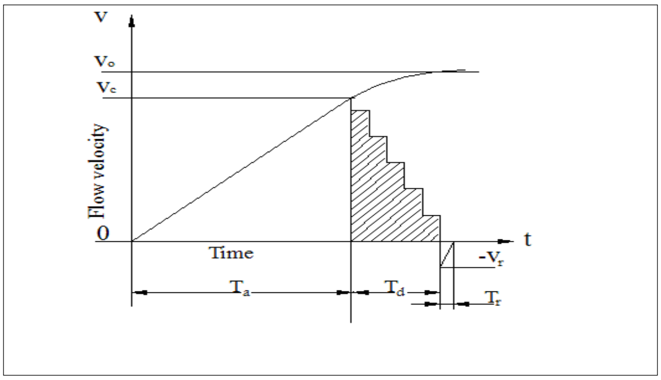

Chart. 1: Characteristics of the flow velocity v at the bottom end of a drive pipe during one operation cycle of a hydram water pumping system.

2.1 The Conceptual Model on the Relationship between an Air Vessel Volume V_s and Delivery Dicharge Volume V_d

When an air vessel was coupled to a hydram water pumping system, it was expected to impart two major functional operations to the system and these were: (1) The air vessel caused the captured air inside it to undergo air compression and hence building up a high pressure at its air chamber during the cyclic water hammer actions which occurred in the hydram operations. (2) The air vessel shown in Fig. 2 served as a temporary reservoir of water which was gradually allowed to go out to a delivery pipe during the cyclic operations of a hydram pump at the expense of the conserved high pressure existing in the air chamber of an air vessel.

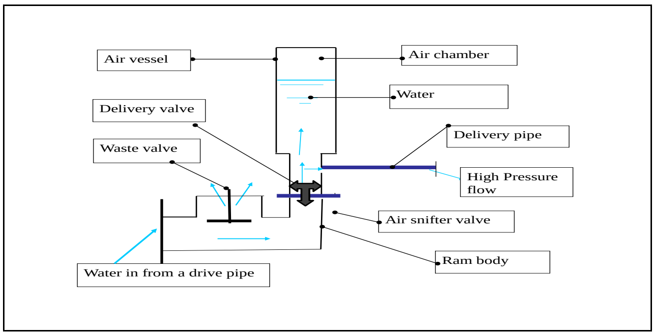

Fig. 2: A conventional air vessel of a hydram pump

In a normal operation of a hydram pump, water from a drive pipe is sent to a delivery pipeline under high pressure at only a small time interval of a cycle period and that time interval is denoted as T_d as shown in Chart 1.

The rest of the cycle time in particular the time T_a is a duration for which no flow is admitted into the delivery pipeline from a drive pipe. This is what creates the pulses of water pumping on the delivery side of a hydram pump which has no air vessel coupled into its system. However when a hydram pump has an air vessel, it will have a secondary source of water supply which will be an air vessel. The flow from the air vessel will be released to a delivery pipeline at the time T_a and due to that, a continuous flow pattern throughout the cycle period will emerge. This flow of water from the air vessel to a delivery pipeline will depend on the net head difference existing between the heads in the air chamber and that on the delivery side. It is from this perspective point of view that the presence of an air vessel in a hydram water pumping system is believed to affect the hydram operations as it interferes with its delivery flow pattern by changing it from the pulses flow pattern having high flow velocities into the continuous flow pattern of low flow velocities

Before building any model, the compression of air in the air vessel was critically studied and related to a scientifically well known case of a piston compressing air inside a cylinder of a positive displacement compressor as outlined by Kirillin et al. (1981). In a hydram pump, it was conceptualized that during the water hammer action, a volume of water V_d would be admitted into the air vessel under a high pressure to form a column of water which could compress the air enclosed in the air chamber volume. This column of water was assumed to be immiscible with air and was expected to behave like a piston which was a force provider inside the cylinder of a compressor. The latter was expected to compress air in the air chamber of an air vessel.

As argued by Kirillin et al. (1981), if a compression process occurs rapidly in the compressing cylinder which is properly insulated such process could be regarded as an adiabatic process. In this case no heat transfer was allowed to occur. On the other hand if a compressing cylinder had a thermostat so that the temperature could be maintained constant like having a cooling water jacket to surround the cylinder, then such compression process could be considered to be an isothermal compression process.

In practice, Kirillin et al. (1981) had realized that even if there could be a thermostat like that of a water jacket enveloping the walls of a compressing cylinder in order to ensure the existence of an isothermal compression process, still a real isothermal compression would not be attained because the rate of heat transfer was finite and the compression was quickly accomplished in the compressing cylinder. Due to that reason, it was difficult to have an ideal adiabatic compression due to the practical difficulties involved in ensuring that there was no heat exchange taking place during such compression process. Hence, a more rational or real compression process taking place under the above mentioned situation was a polytropic compression process whereby:

$$PV^n = C \tag{1}$$

In this case a polytropic index n was assigned to such process in such a way that, $1.0 < n < 1.4$. When the compression process was adiabatic and frictionless $n = c_p / c_v = 1.4$ while when it was purely isothermal then $n = 1.0$. With regard to an air vessel of a hydram, such vessel was coupled to a hydram which was installed in the field and expected to operate at an open air environment. Such air vessel was practically not surrounded by any thermal insulation material which prevented the occurrence of any heat transfer from the vessel to its surroundings. Hence such compression process could not fit to be regarded as an adiabatic compression process. That being the case then, its compression process was assumed to be of a polytropic type as mentioned in Equation (1) whereby the index n was taken to be an average value between 1.0 and 1.4 so that $n = 1.2$.

When the hydram operation had attained a stable cyclic condition, it was assumed that the amount of water V_d admitted into the vessel (Fig. 2) must be equal to the amount being discharged out of the vessel and sent to the delivery pipe in every operation cycle. Thus the air compression and expansion processes taking place inside the air vessel specifically at the air chamber was assumed to obey an ideal gas law relationship which is stated as:

$$P_o V_o^n = P_2 V_2^n = P_{2m} V_{2m}^n = P_3 V_3^n \tag{2}$$

P_o is the pre - charge pressure, P_2 is an intermediate pressure when the delivery flow rate $q > 0$, P_{2m} is the maximum (peak) pressure and this occurs when $q = 0$. P_3 is a delivery pressure at a delivery head h_3.

Another assumption considered is that, after a discharge of water from the vessel, the status of pressure and volume of air at the air chamber will be P_3V_3. When a fresh water volume V_d is admitted into the air vessel from a drive pipe in order to start another new cycle, the new charge will cause an air displacement due to the air compression in the vessel and as a result, both the pressure and volume of air at the air chamber will vary from P_3V_3 to $P_2 V_2$. In this case, $V_3 - V_2 = V_d$ or $V_3 = V_d + V_2$. Based on the latter relationship and with the use of a gas law relationship of equation (2), an intermediate pressure P_2 will thus be expressed as:

$$P_2 = \frac{P_3}{\left(1 - \frac{V_d}{V_3}\right)^n} \tag{3}$$

Where P_3 = delivery pressure and V_3 can be determined from $P_o V_o$ as follows:

$$V_3 = \left(\frac{P_o}{P_3}\right)^{1/n} V_o \tag{4}$$

The above volume V_o is a volume to be obtained after the pre-charging of air vessel. Pre-charging is a normal process done during the initiation of the automatic pumping of a hydram pump. In this case an air vessel should be expected to have a certain internal volume denoted as V_s before been engaged in the compression process of air. This volume V_s will initially be expected to be filled with air at an atmospheric pressure P_a before the starting of an air compression process inside such vessel. Again it is assumed that the pre-charging process is of a slow action so that there will be no significant temperature rise occurring during such compression and therefore the following relationship could then be applicable:

$$P_a V_s = P_o V_o \tag{5}$$

Whereby

$$P_o = P_a \left(\frac{V_s}{V_o}\right) \tag{6}$$

If dV = volume of water admitted into the air vessel at its pre-charging stage, then volume V_o could also be expressed as $V_o = V_s - dV$ and hence equation (6) could be simplified to be:

$$P_o = P_a \left(\frac{1}{1 - \frac{dV}{V_s}}\right) \tag{7}$$

Volume dV could also be expressed in terms of the discharge volume V_d in such a way that, dV = xV_d whereby x is the number like 1, 2, 3 etc designating the number of pumping cycles done to pre–charge an air vessel before attaining the pressure P_o. Thus the pressure P_o could further be expressed as:

$$P_o = P_a\left(\frac{1}{1-\frac{xV_d}{V_s}}\right) \quad (8)$$

P_a = atmospheric pressure at the ambient temperature.

Theoretically, equation (8) could reveal a few critical comments as follows; when an air vessel size or volume V_s is very large and must be pre-charged with a small volume of water dV= V_d, then, mathematically dV / $V_s \approx 0$. The latter will lead to the pressure P_o been equal to the atmospheric pressure P_a. However when $P_o = P_a$, it implies that there will be no air compression process taking place at all in the air vessel of a hydram pump and as a result, the delivery valve will not acquire any adequate pressure from the air chamber to close. This closing of a delivery valve is an important condition for the triggering of the automatic or cyclic operation of a hydram pump. For a viable cyclic or automatic operation to occur in a hydram pump, the pressure P_o must be slightly greater than P_a so that a delivery valve can be closed by a force emanating inside the air vessel . Again, when V_s = 0, the quantity dV / V_s will mathematically become undefined. Since this study is dealing with a hydram system having an air vessel, the argument with V_s = 0 will not be dealt with in this paper. It is generally expected that with the selection of a good range of the air vessel volume V_s sizes, the normal air compression studies could be done. Mathematically, the volume V_s could thus be expressed as:

$$V_s = V_d\left(\frac{1}{\left(\frac{P_o}{P_3}\right)^{1/n}\left(1-\left(\frac{P_3}{P_2}\right)^{1/n}\right)}+x\right) \quad (9)$$

From a separate analysis done to study the influence of a parameter x in the above equation, it was found that x had no significant influence on V_s especially at low values where x was equal or less than 10. By neglecting the influence of x in the equation (9), an important expression had resulted as followed:

$$\frac{V_s}{V_d} = \left(\frac{1}{\left(\frac{P_o}{P_3}\right)^{1/n}\left(1-\left(\frac{P_3}{P_2}\right)^{1/n}\right)}\right) \quad (10)$$

The above equation implied that the ratio of the size of an air vessel to the cyclic discharge of a hydram pump volume wise was related to the ratio of the pressures occurring at the air chamber of an air vessel and this gave a certain dimensionless number.

2.2 Analysis of the Pumped Delivery Discharge V_d

An examination of how the velocity v varies with time at the bottom end of a drive pipe is indicated in Chart 1. In the presented chart, the variation of flow velocity v is traced as from the beginning of a new operation cycle when time t = 0 up to the end of the cycle when t = $T_a + T_d$. In a hydram pumping process a water hammer action becomes an essential function as it causes the generation of a high pressure wave to occur in the system and this takes place at a certain critical velocity denoted as v = v_c (Krol 1976 and Tacke 1987). The effective flow of water from the drive pipe to the delivery pipe occurs when the pressure in the drive pipe is high and this will last for a period of time T_d. The time T_a is a duration when the flow from the drive pipe goes out to a waste by passing through a waste valve. When this is happening there will be an increase in velocity from 0 to v_c. The time denoted as T_r is a very short period and it is a time taken for the recoil of the flow to occur at the bottom end of a drive pipe. Thus the total time taken for one operation cycle T can be approximated as: T = $T_a + T_d$.

Likewise, the delivery discharge volume V_d which occurs during the time t = T_d could be predicted by the expression:

$$V_d = A_1\int_{T_a}^{T_a+T_d} v(t)dt$$

Or,

V_d = A1 ($\sum \Delta t \sum_{Ta}^{T-Tr} v$ (t)) .

Where $\sum \Delta t$ = T_d ,

$\sum_{T_a}^{T-T_r}$ v(t) (t) = v_c / 2. Hence;

$$V_d = \frac{1}{2}A_1 v_c T_d \quad (11) \qquad (11)$$

Alternatively, V_d could also be attained by multiplying the cross section area A_1 of a drive pipe by the shaded area of a triangle in Chart 1 whose base = T_d and height = v_c The cross section area A_1 = (πD12) /4 where D_1 = diameter of a drive pipe. The time T_d which is the effective water pumping duration is determined as:

$$T_d = \left(\frac{L}{c} + \frac{a_j TH}{h_3} \right) \quad (12)$$

Where aj = waste valve flow coefficient factor, c = acoustic wave speed in a drive pipe, H = Drive head, h_3 = Head due to pressure P_3, L = Length of drive pipe and T = total time taken for one operation cycle. Similarly, the wasted flow volume per cycle V_w could be estimated by multiplying a cross section area A_1 by the area of unshaded triangle in Chart 1 whose base = T_a and height = v_c. The result becomes:

$$V_w = \frac{A_1 v_c T_a}{2} \quad (13)$$

Hence, the total volume of input flow per cycle is predicted to be:

$$V_t = V_w + V_d \quad (14)$$

Alternatively, V_t could also be estimated by multiplying the cross section area A_1 with the areas of the two triangles whereby one triangle has the base of time T_a and another of time T_d while neglecting the base of T_r as it is extremely very small. Hence the volume V_t could then be expressed as:

$$V_t = A_1 \frac{v_c T}{2} \quad (15)$$

The averaged input flow rate per cycle Q_t could be expressed as:

$$Q_t = \frac{V_t}{T} = \frac{V_c A_1}{2} \quad (16)$$

whereby the velocity v_c is determined by the expression:

$$v_c = \frac{a_j gH}{fL} \quad (17)$$

In this case f = beat frequency of a waste valve and g = gravitational acceleration. The factor aj had earlier been proposed by Young (1995) to vary from 0.6 to 0.8 for the best designed waste valve. Combining the equations (11, 12 and 17) the pumped volume of water V_d per cycle is predicted as:

$$V_d = \frac{A_1}{2} \left(\frac{a_j gH}{cf} + \frac{a_j^2 gH^2}{f^2 L h_3} \right) \quad (18)$$

In this case, V_d should be regarded as the volume of water that is temporarily accumulated in the air vessel before been discharged out to the delivery pipe within one operation cycle. Again an average delivery flow rate discharged per one cycle time will be q = V_d / T or q = f V_d where f = 1 / T.

Hence, the average delivery flow rate q is expressed as:

$$q = \frac{A_1}{2} \left(\frac{a_j gH}{c} + \frac{a_j^2 gH^2}{fLh_3} \right) \quad (19)$$

The above stated delivery flow rate q is expected to be in SI units which is m^3 / second and the other variables which are L, H, h_3 will have their dimensions in (m) units.

The efficiency of a hydram fitted with an air vessel could then be defined as a ratio of the available output energy of a fluid passing via an air vessel to the total maximum input energy available in a drive pipe for one operation cycle.

The output energy in the delivery flow after passing through the air vessel per one operation cycle would be Eot = P_3V_d. The total input energy to a drive pipe would be Ein = P_iV_i where P_i is the pressure due to the drive head H and V_i is the total input volume per cycle arising from the flow whose velocity is v_c and which flows through a drive pipe when no losses are yet considered.

Hence:

$$E_v = \frac{P_3 V_d}{P_i V_i} = \frac{(\rho g h_3) qT}{(\rho gH) Q_t T}$$

where P_3 = ρ gh_3, P_i= ρ gH, V_d = qT, V_i = Q_tT and T = cycle operation time.

The latter may be reduced to:

$$E_v = \frac{h_3 q}{Q_t H} \quad (20)$$

The above efficiency expression is sometimes known as the D'Aubuisson's efficiency model whereby it could be noted that there were two dimensionless parameters which are (h_3 / H) and (q / Q_t) and these are used to define the hydram's pumping efficiency. This is an expression which can not directly show the influence of an air vessel size V_s on the hydram's water pumping system. But when looking at equation (9) it could be noted that the volume V_s is linked with the quantities V_d, P_o , P_2 and P_3 with a certain relationship. Pressure P_2 is an intermediate pressure as shown in equation (2) and it is an important pressure in the system as it helps to send the flow out of the air vessel to a desired delivery destination. Analytically, P_2 could be expressed as:

$$P_2 = \frac{P_3}{\left(1 - \frac{V_d}{V_s (r_s)^{1/n}}\right)^n} \tag{21}$$

Since $P_2 = \rho g h_2$, then the head h2 could similarly be expressed as:

$$h_2 = \frac{h_3}{\left(1 - \frac{V_d}{V_s (r_s)^{1/n}}\right)^n} \tag{22}$$

After including all necessary head losses in the system an improved system efficiency expression become:

$$E_{vf} = \frac{h_{3n} q_n}{Q_t H} \tag{23}$$

Where $h_{3n} = h_2 - h_{ft}$ and $q_n = q\ (h_2)$. In the latter expression h_{ft} is the summation of the total head losses occurring in the system and q (h_2) is the flow rate induced by the head h_2. With the use of the above analytical model equation the effect of the variation of the different air vessel sizes or capacities against the system efficiency was investigated. Those various air vessel sizes used had capacities ranging from 0.12 litres to 19.0 litres including a Reference Volume (RV) of 9.80 litres. The latter capacity was taken to represent a value of 100%.RV. The selected V_s specimen samples were varied as follows: 0.12 litres (1.2 % RV), 2.45 litres (25 % RV), 4.9 litres (50% RV), 9.80 litres (100% RV), 14.6 litres (149% RV) and 19.6 litres (200% RV). The 0.12 litres air vessel specimen was the smallest air vessel size aimed to study the effects of V_s when approaching a zero V_s volume while the 14.6 and 19.6 litres vessel size were aimed to study the effects of the oversized air vessel situation. The above mentioned air vessel sizes were carefully prepared and subjected to both the theoretical and experimental studies and the results are presented in this paper.

3. EXPERIMENT METHODOLOGY

In establishing the methods for experimental data collection, the equations (10), (19) and (23) were taken as the control equations for the experimental study (Montgomery, 2001). The next step was to design an appropriate experimental rig to study the characteristics of efficiency against the effects of the variations of the parameters within the above mentioned model equations. In this case a hydram test rig was designed in such a way that it could allow the measurements and acquisition of such data as P_o, P_2, h_3, H, L, f, V_d, V_s, q and Q_t. In the rig, the pressures were measured with the use of the pressure transducers and recorded by a computerized data logging system. The beat frequency f was acquired by taking the average time taken by a sample of operation cycles per time in seconds. The air vessel volume sizes V_s were separately made as unit specimens bearing the measured or calibrated volumes of 0.12, 4.9, 9.8, 14.6 and 19.6 litres.

The calibration of the pressure transducers was done by mounting the transducers onto a specially designed device having a pressure tight piped system. In the calibration process, the pressurized air from an air compressor of a known pressure in bar units was filled into the calibration unit which was having a bourdon gage meter and the resulting electrical current was measured in milliamperes by using a current meter. A series of the measurements of variable pressures against the generated electrical signals from the transducer in milliamperes were measured and recorded. The results of the trends were plotted as shown in Chart 2. The relationship of the trend had shown that there was a linear relationship between the pressure and the generated electrical signal from the pressure transducer and this calibrated data was also found to agree well with the supplied manufacturer's information.

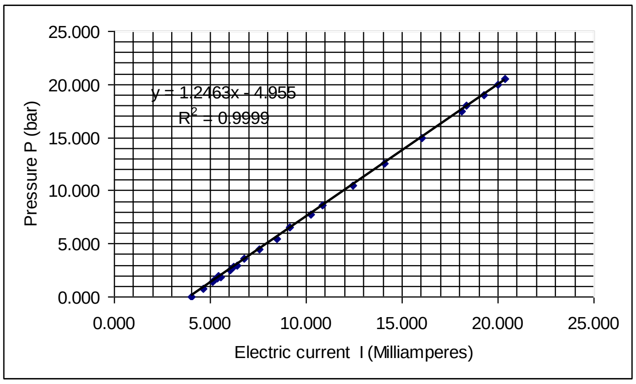

Chart 2: Calibration graph for the measured pressure P (bar) from a pressure transducer against the measured electrical current (Milliamperes)

After calibration process, the transducers were then placed in the experimental test rig system and several trial runs of the system data collection were conducted. The system was run while making all the necessary system adjustments and synchronizations in order to acquire the reliable performance of the system and genuine data measurements.

Based on the mentioned model or control equations, the following experimental tests were conducted:
q versus V_s - when (h_3, f, L,H) were held constant.

E_{vf} versus V_s when (h_3, f, L,H) were held constant.

E_{vf} versus (V_s/V_d) when (h_3, f, L,H) were held constant.

4. RESULTS AND DISCUSSIONS

4.1 The Effect of Air Vessel Size V_s Against the Delivery Flow Rate q

The effect of the variation of air vessel volume V_s against the delivery flow rate q was investigated theoretically and experimentally and the results were presented as shown in Chart 3. In this investigation it was found that the theoretical or model results had a close agreement with the experimental results. In general, an air vessel size V_s was found to have an influence on the delivery flow rate q of a hydram pump. As shown in the graphs, when an air vessel size V_s was too small (as small as 0.12 litres or 1.2 % RV), q was similarly smaller. As V_s was increased from 0.12 litres or 1.2 % RV to 2.45 litres or 25 % RV, there was a rapid increase in q. However, there was a gradual or very small increase in q when V_s continued to increase further from 2.45 litres or 25 % RV to 19.6 litres or 200% RV. The observed behavior was presumably thought to imply that there was a certain minimum air vessel volume which was sufficient in providing the optimum air chamber pressure for the required hydram operations. However, when the air vessel volume was too small it implied that the amount of the entrapped air in that small air vessel volume before the starting of the compressions was also too small to run the hydram operations properly. Due to that little amount of air, the water hammer flows coming at high pressures from the drive pipe and impacting on the air chamber could have little air cushioning effects against those flows. As a result, the vessel could still allow the flows to escape with some higher exit velocities which could eventually be accompanied with higher energy losses.

However in fluid mechanics the loss of energy is dependent on the square of the velocities. The latter energy loss was realized in the reduced delivery flow rate q. As the air vessel volume was increased the amount or mass of the entrapped air was similarly increased until reaching a sufficient level which provided an optimum operational pressure with better air cushioning effects against the incoming water hammer flows and this had resulted in the reduced flow velocities. The delivery flow rate q were increasing.

From the trends it implied that any larger air vessel volume beyond the optimal volume was not having any significant advantage in the hydram's output performance in terms of q. It could also be observed that the variation of the drive head from 2.0 to 4.0 m had resulted in the increased q. Theoretically it implied that the increase in H was corresponding to the increase in the input energy and likewise the corresponding output energy was expected to increase because the delivery head h_3 was maintained constant.

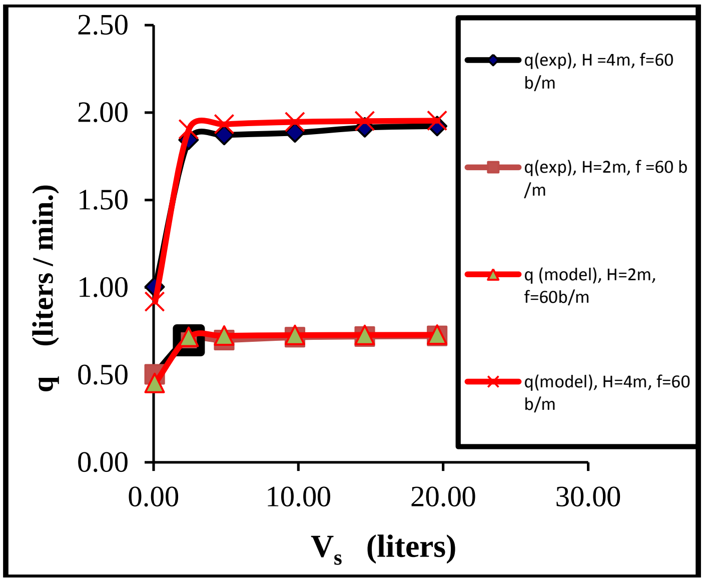

Chart 3: Plots of q (model) &q (exp) versus V_s for h_3 = 45 m, L = 18.5 (m), f = 60 beats /minute and H = 2 & 4 (m).

4.2 The Effect of Air Vessel Size V_s Against the Efficiency E_{vf}

The results shown in Chart 4 shows that there was a specific relationship existing between the system efficiency Evf and the air vessel size Vs.

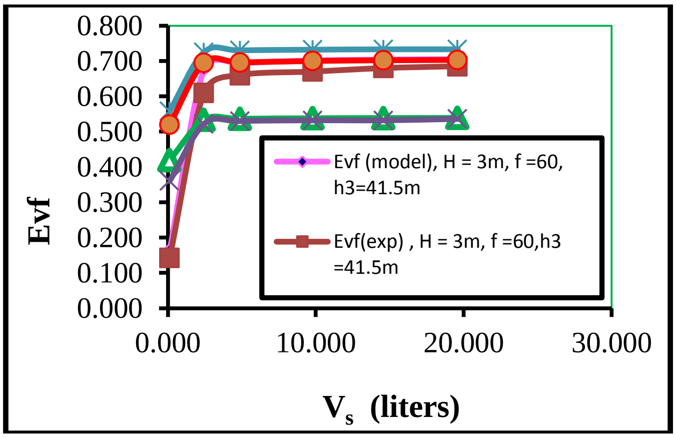

Chart 4: Plot of Evf (model) & Evf(exp) versus Vs for H = 1.5m & 3.0m, f = 60 beats / min, h3 = 41.5m & 26.5 m and L = 18.5 m.

The trends shown in the graphs of Figure 7 appeared to be similar to the trends shown in Figure 6 because Evf was dependent on q since H and h3 were held constant, refer equation (21). In general, the characteristics of Evf (model) & Evf (exp) versus Vs were found to increase with the increase in the air vessel sizes Vs.

4.3 The Effect of Parameter (V_s/V_d) Against the Efficiency E_{vf}

Figure 8 presents the plots of both Evf (model) and Evf (exp) versus Vs /Vd on a common graph chart and they are implicitly involving the air vessel sizes of capacities 2.45 to 19.6 litres. The smallest air vessel size of Vs = 0.12 litres was not included in the plot because of being inconsistent.

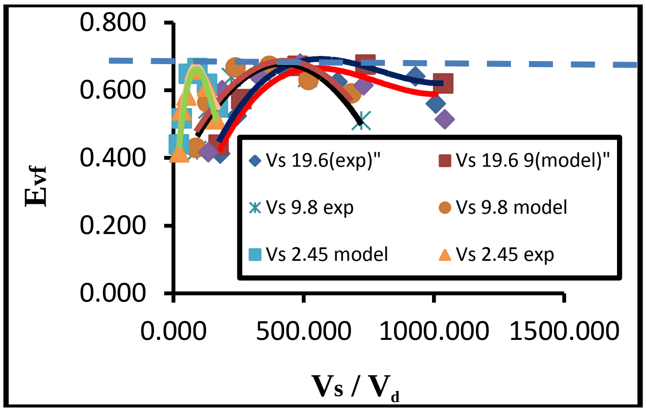

Figure 8: Plot of Evf (model) and Evf exp) versus Vs /Vd for H = 4.0 m, h3 = 50 m, L = 18.5 m and various Vs in litres

In general the plotted trends of Evf (model) and Evf (exp) versus Vs /Vd appeared to have some important characteristics for which the efficiency Evf had their peak efficiencies touching the same horizontal line (dash line) even though the parameter Vs /Vd had some different magnitudes.

The trends showed some good efficiencies which ranged between 0.45 and 0.7. From the above attained behaviour, it was leant that the efficient operations of a hydram pump could still be attained by using the smaller (Vs/Vd) magnitudes or numbers and this provided a higher chance of using the smaller air vessel sizes while maintaining higher hydram efficiencies which were obviously cheaper.

5. CONCLUSION

In order to improve the efficiency of a hydram water pumping system, an air vessel with some significant volume or capacity was found to be a necessary component of a hydram water pumping system because it assisted in the accommodation of a sufficient amount of cushioning air which was essential in the support of operational pressure processes taking place at the air chamber volume of an vessel. This operational pressure was also found to be an important parameter in controlling the water flow velocity, the system energy losses and hence the system efficiency. The study had also revealed that when the air vessel size was too small nearing a zero volume as for a volume Vs = 0.12 liters, the system efficiency was poor as compared to the case with the relatively bigger air vessel specimens having the volumes ranking between 2.45 and 19.6 liters. One of the major reasons accounting for such poor performance was the incapability of such air vessel to accommodate a sufficient amount of air at its air chamber which could support the initiation of the normal pressure operations of a hydram pump. However in order to acquire a better hydram output performance leading to the higher system efficiencies where Evf ≥ 60 (%), the ratio of (Vs / Vd) was supposed to be in between the range of 50 and 500 (refer charts in Figure 8 and this was based on the studies conducted by using the air vessel specimens of sizes 2.45 ≤ Vs ≤ 19.6 liters.

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