Experimental and Computational Investigations of the Effects of Free Valve Motion on Pressure Transient and Ram Pump Performance.

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دراسة تجريبية وحسابيه لتأثير حركه الصمام الحر على السريان المتذبذب
ومعامل أداء المضخة الذاتية

خلاصة البحث:

يتناول هذا البحث دراسة عمليه لإيجاد تأثير حركة الصمام الحر على السريان المتذبذب و على معامل أداء المضخة الذاتية. قد تم توصيل ذراع الصمام الحر إلى ياى من النوع الورقي له طول متغير و بالتالي له معامل مرونة متغير. التجارب المعملية تم إعدادها لفتحات مختلفة للصمام وأيضا لأطوال مختلفة للياي. تم إجراء هذه التجارب لمصدر مياه على علو ٢.٥ م حيث يتم الضخ إلى علو ٤.٣٦ م أما الدراسة الحسابية فقد تم إعدادها اعتمادا على معادلات الطرق المائي لحساب الضغط المتذبذب الناتج من الحركة الذاتية للصمام. و تم حل جميع المعادلات الحاكمة على طول ماسورة مصدر المياه مع الأخذ في الاعتبار العوامل المعروفة عند طرفي ماسورة المصدر ونتيجة لهذه الدراسة فقد تبين وجود تأثير كبير لكل من معامل مرونة الياي وفتحة الصمام على معامل أداء المضخة حيث وجد أنه بصرف النظر عن فتحه الصمام فإن كفاءة المضخة تزداد كلما كــان معامل مرونة الياي المستخدم كبير وكذلك فإن معدل سريان المياه من المضخة يزداد بزيادة فتحه الصمام بصرف النظر عن معامل مرونة الياي. وقد تم تحديد المدى الذي تحدث خلاله هذه الظاهرة و الذي يمكن الاستفادة مه في تصميم وتشغيل هذا النوع من المضخات.

Abstract

An experimental investigation has been conducted to study the effect of free valve motion on the pressure transient and ram pump performance. In this study, the valve stem is attached to a leaf spring, which has a variable effective operating stiffness. The runs have been carried out at different initial valve openings for different spring stiffness. Fixed values have been used for the falling head of 2.5 m, and lifting head of 4.36 m. A computational model based on water hammer wave travel equations is used to verify the pressure transient due to valve self oscillatory motion. The governing equations are solved by the method of characteristics along the drive pipe with the appropriate boundary conditions at both ends of the pipe. The results reveal that the effective spring stiffness and initial valve opening have major effects on the ram pump performance. The stiffer spring, the maximum is the efficiency. Also the elastic spring, the maximum is the ram flow rate. Regardless of the spring stiffness, increasing the valve opening increases the ram flow rate. The design margins have been deduced defining the operational valve instability and/or ram pump performance.

Nomenclature

Latin symbols

A	drive pipe flow area, m^2	P	pressure, N/m^2
d	delivery pipe diameter, m	q	ram flow rate, lit/min
D	drive pipe diameter, m	Q	drive flow rate, lit/min
E	modulus of elasticity, N/m^2	t	time, s
f	Darcy friction factor	u	flow velocity in delivery pipe, m/s
f_r	frequency, Hz	V	flow velocity in drive pipe, m/s
h	ram delivery head, m water	x	x-axis or valve displacement, m
H	source head, m water		Greek symbols
K	Bulk's modulus, N/m^2	η	ram pump efficiency, %
k	spring stiffness, N/m	υ	Poisson's ratio
k_r	spring stiffness ratio	ρ	water density, kg/m^3
l	valve stroke, m		Subscripts
l_r	spring effective length ratio	o	opening
L	pipe length, m	r	ram
		s	spring

1. Introduction

The pressure transient in the fluid flow systems is considered, as a major problem in most industrial processes, however in the other side could be beneficial. The practical problems associated with the pressure transient are concerned with the vibration of structure carrying the fluid flow and in turn noise, fatigue and failure of the flow systems. The other aspect of this phenomenon is the question of control instability. In the other hand this undesirable phenomenon could be exploited as a useful one in the field of pumping water as a ram pump and in the field of heat exchangers as a disobedient of the fouling rate.

The hydraulic ram pump is a motorless low flow rate pump, which uses a large amount of water falling from a low height to lift a small amount of that water to a much greater height. The hydraulic ram pump can be used where small quantities of water are required, such as domestic water supplies or livestock watering. It is also practical where conventional power for pumping water is not available, such as remote areas where the installation of electric service would be uneconomical or where an internal combustion would be impractical.

Many investigators have studied the design, installation, and operation of the hydraulic ram pumps. Abdel-Rahim and Awad (2001) have studied the effect of valve stroking on ram pump performance. They used a predefined valve motion driven by cam motor arrangement to obtain the characteristics necessary for the hydrodynamic design of ram pump valve. Their arrangement has been made to enable the study of the effects of valve stroking and frequency by replacing cam profiles and changing cam rotational speed. They concluded that the cam frequency has a great influence on ram pump performance and it is found that the frequency range, at which the ram flow is maximized, depends on the cam profile. The shorter, the full opened stroke; the maximum is the efficiency. Also the longer, the full opened stroke, the maximum is the ram flow rate.

Jennings (1996) and Smith (2000) have cited the technical characteristics of the ram pumps. They stated that the ram pump could be used to lift water as high as *120 m*, and provide flow rate as much as *180 lit/min*. The ram pumps could be installed under a source head as minimum as *1 m* with drive flow rate as less as *8 lit/min*. The drive pipe length and diameter should be chosen such that a proper flow velocity in the drive pipe is attained to achieve a pressure transient enough for the pump operation. The drive pipe length has another effect on the operation of ram pump. Such effect arises from the necessity of water column elasticity that would be adequate to sustain the pressure transient.

Thomas (1994) and Curtis and Tyson (1999) concluded that the ram pump efficiency could reach *60%* and the ram and drive flow rates ratio could be as high as *1/7*. In general the criterion for ram pump operation is to maximize the ram flow except if the drive flow source is limited the maximum efficiency operation is sought. The operational valve frequencies were suggested to be from *0.3* to *1.5 Hz* for proper operation.

Numerous authors have investigated the fluid transients during water hammer in hydraulic line. Jones and Wood (1974) studied the pressure transient in straight pipe with axial boundary motion. They concluded that the pressure surge magnitude is significantly higher than that predicted neglecting the boundary motion.

Stuckenbruck, *et al.* (1985) studied the axial wave propagation speeds in both fluid and pipe wall. They simplified the six-equations model with three wave speeds namely radial, and axial stresses and axial fluid waves to four-equations model with two wave speeds, axial stress and fluid waves. They neglected the radial inertia and used the Poission coupling to transform the tangential strain caused by fluid pressure to axial strain.

Wiggert, *et al.* (1985) investigated analytically and experimentally the effect of pipe support on the pressure transients. They showed significant pressure transient alteration when

the pipe support at the elbow becomes more flexible due to the motion of the elbow which is driven by the axial stress in the pipe and by the fluid pressure.

Liou, (1991) used two-equation model and showed that the maximum transient pressure is strongly related to the inherent flow characteristics of the valve and closure period. Elansary, and Contractor, (1994) used four-equation model to examine the optimum valve closing characteristics on the pressure transient. They didn't incorporate the structural damping in their model and assumed rigidly supports at the pipe ends. They recommended the use of advanced valve operating system that can simulate the optimum valve closure to alleviate the pressure pipe transients.

Choy, et *al.* (1996) solved the two-equation model by method of characteristics and finite difference formulation and their results were transformed to the frequency domain using Fast Fourier Transform. The two methods gave almost the same conclusion that the pressure component at its fundamental frequency is a strong function of the valve closure time, and piping size. The second harmonic has a little effect on the pressure transients.

Stephenson, (1997) studied the effect of air valves and pipework on water hammer pressures. He concluded those local effects such as branch pipe size, rigidity of pipe wall, coatings, and surrounding fill could have great influence on water hammer transients.

In the present work, experimental and computational investigation have been conducted to exploit the pressure transient induced by self-vibratory valve motion to lift small amount of water to an elevation higher than that of the source. The induced valve oscillation has been obtained by slowly opening the valve up to a certain value at which the valve begins and sustains a self-vibratory motion. Also the valve spring stiffness has been adjusted by changing the length of an external leaf spring to study the effect of valve natural frequency on the valve dynamics and in turn on the ram pump performance. In the computational model, two partial differential equations governing the traveling water hammer wave in the drive pipe flow have been formulated by the method of characteristics along with the equation of the pulsating ram flow. An ordinary differential equation governing the valve dynamics has been implemented in the down stream boundary condition of the drive pipe. The system of equations with the appropriate boundary conditions has been solved by time marching scheme.

2. Experimental Work

The experimental work is carried out on a laboratory hydraulic circuit equipped with the required instrumentation for data acquisition and measurements.

2.1. Experimental Set-up

The experimental set-up shown in Fig.(1), consists of the flow circuit arrangement along with the required instrumentation. The water flows from an overhead reservoir ($5 m^3$ volume and $2.5 m$ above the level of the valve) to a $2''$ drive pipe of $6 m$ length .The pipe is equipped with a $2''$ valve shown in Fig.(2) with the external leaf spring. Under transient pressure due to valve motion an amount of water (q_{ram}) is rammed through a vertical $1''$ delivery pipe to a ram delivery head of $4.36 m$. The delivery pipe is equipped with a non return valve at its lower end to admit the upward flow only.

The $2''$ valve has a measured spring stiffness of $67 kN/m$. where the added external leaf spring, of mild steel material, has $38 x 5 mm$ cross section. The spring full length is taken as $155 mm$ to have the same value of stiffness as the valve spring. Four spring effective lengths have been used with different valve openings. For each spring, the valve begins to oscillate at a certain initial valve opening. The pressure signal is collected on one channel of the Data Acquisition System (DAS), described below.
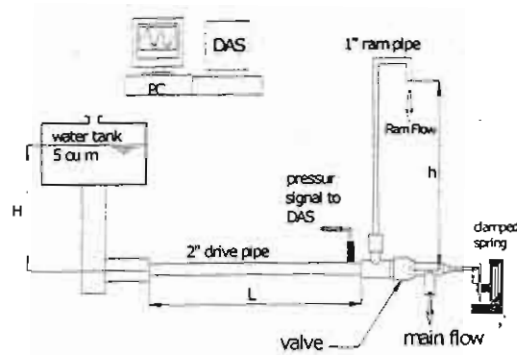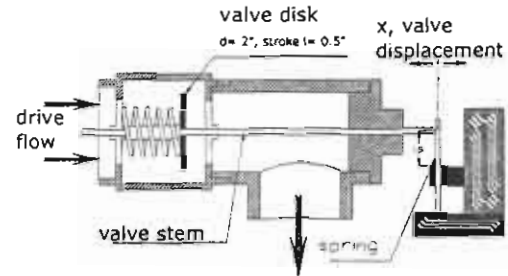
Fig.(1) Experimental test rig

Fig. (2) Drive valve structure

2.2. The Investigated Parameters

The objectives of this study is to examine the effect of valve parameters on the pressure transient in the drive pipe, ram pump flow, and drive flow. The investigated valve parameters are the initial valve opening and valve stiffness, which is achieved by changing the external leaf spring length. The tested length ratios are $l_r=1$, ¾, ½, and ¼ of the spring full length. These length ratios correspond to spring stiffness, ratios of that of the valve, as $k_r=1$, 2.5, 8, and 45. For each length, the valve-opening ratio has been changed from the inception of the valve oscillation up to the disappearance limit. At each run, the drive and delivery average flow rates are measured and the transient pressure is sampled by the DAS.

2.3. Instrumentation

The pressure in the $2''$ drive pipe is measured by a piezoelectric transducer, model no PCB101A05, of full-scale range $0-250$ psi. The transducer signals are collected by one channel on Data Acquisition System (DAS), LeCroy 8013A main frame with 6810 module of sampling rate up to 5 megasample/s with a full memory of 512 ksample.
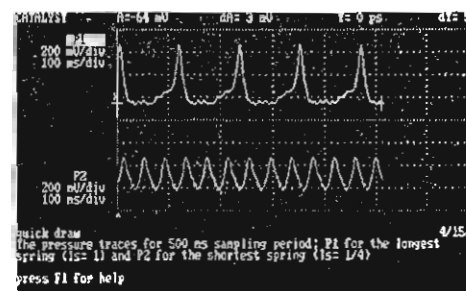
The DAS is connected with an IBM PC with a GPIB card. The commands for arming, triggering, and acquiring the data are sent from the PC via the Catalyst® software. The average of both drive and driven pulsating flows are measured by the aid of collecting tanks.

The DAS and Catalyst® software are configured in the present work as; sampling rate of 2 k sample/s with full-scale voltage up to 10 v.

A typical screen of Catalyst® software is displayed in Fig.(3) showing the pressure transient signals. The upper trace (P1) presents the pressure signal for the longest spring, $k_r=1$ and valve opening of 16%. The lower trace (P2) for the shortest spring, $k_r=45$ and valve opening of 31%. The menu bars show the configuration parameters used in this particular run.

Fig.(3) Sample of Catalyst® software for pressure wave form

2.4. Experimental Data Processing

The Catalyst® software triggers and acquires the signals in mellivolts for pressure as

shown in Fig.(3). The data could be displayed on the PC screen for visual processing of signal amplitude and frequency, and/or stored in Catalyst® format files. This data is converted to ASCI format files for further processing.

3. Computational Work

In this investigation the water hammer transient in piping systems is computationally investigated by the method of characteristics, Wylie, and Streeter, (1978). The equations governing the fluid flow transient in the drive pipe are derived based on the dynamic equilibrium during the transient. The system of equations constitutes two hyperbolic partial differential equations of first order. The method of characteristics was used to transform the two equations into ordinary differential equations. The characteristic values were calculated and the corresponding characteristic functions were solved numerically over the characteristic grid. The pulsating ram delivery pipe flow differential equation is used as a boundary condition at its junction with the drive pipe. The drive pipe downstream end boundary condition constitutes an ordinary differential equation governing the vibration of the valve while the upstream boundary condition is of constant reservoir head. The pipe was clamped at both ends.

3.1. Governing Equations

Liou (1991) introduced the following two-equation model for fluid variables: P, pressure and V, fluid flow velocity.

$$\frac{\partial P}{\partial t} + K' \frac{\partial V}{\partial x} = 0 \tag{1}$$

$$\frac{\partial V}{\partial t} + \frac{1}{\rho} \frac{\partial P}{\partial x} + \frac{\tau_w}{A} \frac{\pi D}{\rho} = 0 \tag{2}$$

Where: the wall shear stress is

$$\tau_w = \frac{f}{4} \rho \frac{V^2}{2} \tag{3}$$

And the modified bulk's modulus is

$$K' = K \Big/ \left\{ 1 + \left[\left(\frac{K}{E} \right) \left(\frac{D}{t} \right) \right] \left(1 - v^2 \right) \right\} \tag{4}$$

The two partial differential equations (1,2) are transformed along the characteristics to an ordinary differential equation as

$$\left. \begin{array}{l} \pm \dfrac{1}{\rho c} \dfrac{dp}{dt} + \dfrac{dV}{dt} + \dfrac{fV|V|}{2D} = 0 \\[2ex] \dfrac{dx}{dt} = \pm c \end{array} \right\} \tag{5}$$

Where + ve sign for the positive characteristics and − ve sign for the negative characteristics and c is water hammer wave speed

$$c = \sqrt{K'/\rho} \tag{6}$$

The upstream boundary condition is given by

$$P_i = P + \rho V^2/2 \tag{7}$$

The valve linear motion gives the downstream boundary condition, which describe the valve opening ratio x/l where l is the valve stroke length. For $0<x/l<l$. The valve oscillation equation is given by

$$m'\frac{d^2x}{dt^2} + c'\frac{dx}{dt} + kx = -P_v A \qquad (8)$$

Where: m' is the virtual mass, c' is the damping coefficient, k is the valve spring stiffness, P_v is the instantaneous pressure at the valve, and A is the valve disk area. The initial condition is $x = x_0$, where x_0 is the initial valve opening. If the valve disk hits the valve seat, i.e. at $x = 0$ or $x = l$ the disk is assumed to rebound by an initial velocity equal to fraction of impact velocity at previous time. This fraction is the coefficient of restitution and it depends on the material of the valve disk and seats, and is assumed as small value 0.2 since the disk seats are made of rubber. The ram boundary condition is obtained by the ordinary differential equation describing the oscillating column of water as:

$$\frac{du}{dt} = g\frac{h}{L_r}\left(\frac{P}{\rho gh} - l\right) + \frac{f}{2d}u^2 \qquad \text{for all } u \geq 0 \qquad (9)$$

4. Results and Discussion

The experimental measurements have been collected and post processed to highlight the effects of the investigated parameters. The amplitude and the fundamental frequency of pressure transient in the delivery pipe have been obtained by applying the Fourrier transform to the experimental pressure waveforms.

The experimental and computational pressure waveforms are displayed in Fig.(4). The computational pressure waveform is plotted for the steady state solution. It is found that the transient solution die out after about l sec, that is why the upper time axis begins at l sec, while the experimental pressure wave form is always measured after enough time such that the steady oscillation has been reached. From the figure there is a good match between the measured and computed pressure waveforms.

Fig.(4) Sample of computational and experimental results for pressure wave form

The valve opening represents a measure of steady flow rate through the pipe and hence the flow velocity. Therefore the increase of valve opening eauses an increase of the equivalent oscillating mass of fluid in the pipe what is so called virtual mass or apparent mass. The inertia force of the oscillating water column in the pipe is governed by the virtual mass of this column. The oscillation restoring force depends on the water column length i.e., the drive pipe length and the valve structure elasticity. For equivalent lumped systems, the frequency of the system is given by: $f = 1/2\pi\sqrt{k'/m'}$ where: f is the equivalent lumped system frequency, k' is the equivalent lumped system stiffness, and m' is the virtual mass. The frequencies of the pressure oscillation will inversely depend on the virtual mass in the drive pipe flow where the column stiffness is kept constant since the pipe length is invariant in this study. The elasticity of the system is ehanged by an external leaf spring with variable length attached in parallel to the valve stem. From the simple beam theory the leaf spring stiffness is inversely proportioned
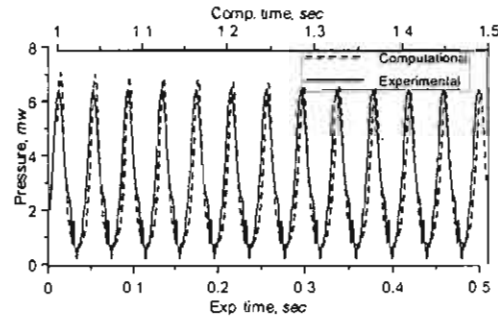
with its length. Therefore the increase in the external leaf spring length will decrease the oscillation frequency.

From the above discussion, it can be drawn that, the increase of valve opening causes an increase in the virtual mass while the system stiffness remains unchanged for a given spring stiffness. This argument explains the decrease of the pressure frequency by increase of the valve opening as shown in Figs.(5).

The maximum pressure, p_{max}, is increased by increasing the valve opening as shown in Figs.(5). This enhancement in pressure amplitude could be attributed to two folds. The first is the higher asymptotic steady flow velocity, as a result of the higher valve opening. The second, is the increase in valve opening decreases the frequency, as mentioned above, which in turn permits enough time for the dynamic flow velocity to reach a reasonable value of its asymptotic steady value due to the increase of the drive pressure, the ram flow rate, q_{ram}, is increased by increasing the valve opening i.e. by decreasing the valve frequency as shown in Figs.(5).
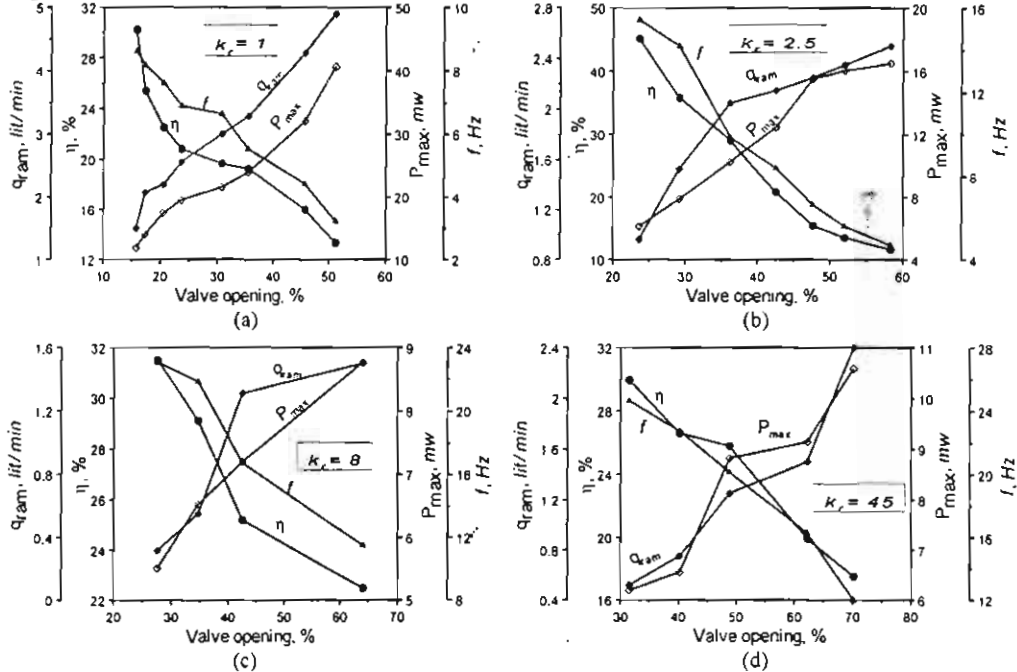

Fig (5) Ram pump performance for tested valve spring stiffness
(a) $k_r = 1$. (b) $k_r = 2.5$. (c) $k_r = 8$. and (d) $k_r = 45$

The ram pump efficiency has several definitions. In this work, the efficiency is defined as the ratio of the produced pumping power ($\rho g q h$) to the equivalent virtual turbine power ($\rho g Q H$), i.e. $\eta = qh/QH$. Increasing the valve opening, as shown in Figs.(5), decreases the ram efficiency However both the drive and delivery flow rates are increased with the increase of the valve opening. the rate of change in drive flow is greater than that in delivery flow. It is clear from the above discussion that the increase in the drive flow is due to both the increase of the flow velocity and the decrease of the valve oscillation. While the ram flow increases with the increase of flow velocity and decreases with the decrease of the valve frequency, i.e. the frequency of pressure pulses.

Figure (6) have been displayed to show the effect of the external leaf spring stiffness on the pressure transient. From Fig.(6a), it is seen that the oscillation frequency is further increased with the increase in spring stiffness as discussed above. The maximum pressure in the drive pipe, displayed in Fig.(6b), increases as the spring stiffness is decreased that is due to the decrease in system frequency.
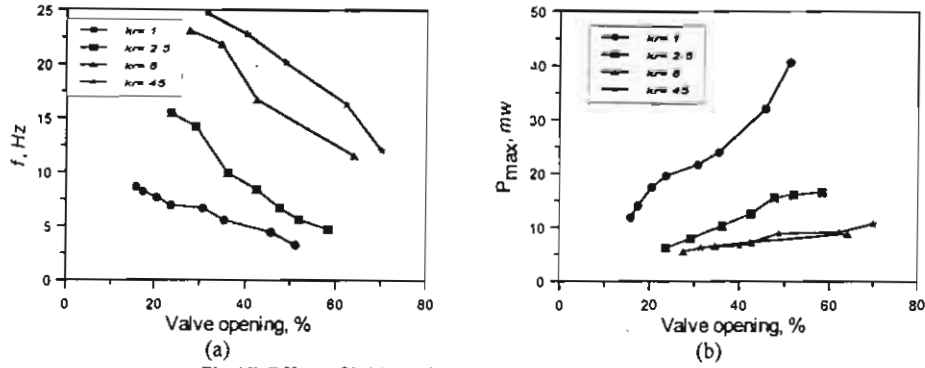

Fig.(6) Effect of initial valve opening on pressure transient;
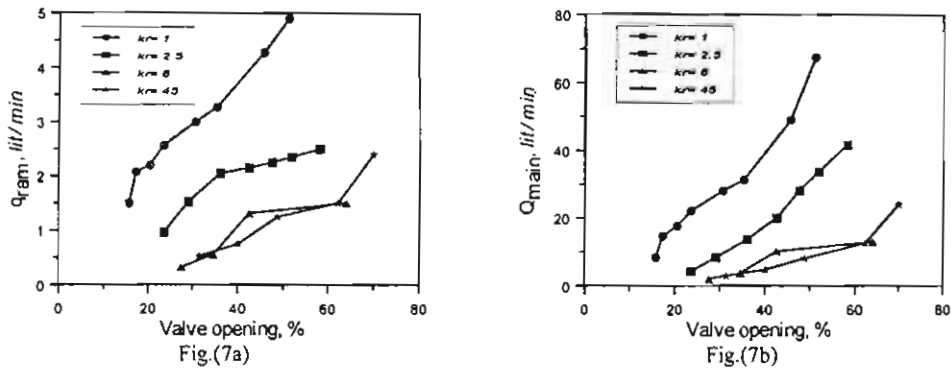(a) system frequency, and (b) pressure amplitude

Figure (7) shows the effect of the external leaf spring stiffness on the ram pump performance. From these figures it is seen that both ram and main flow rates are increased as the spring stiffness is decreased, as illustrated in Figs.(7a, 7b). While both flow rates increase, the ram efficiency decreases as shown in Fig.(7c). That is because the rate of increase in the main flow is greater than that of the ram flow as discussed above. Also, it can be drawn that, to maximize the ram flow rate more elastic spring is required, on the other hand to maximize the ram pump efficiency, stiffer spring is in order. It is up to the designer to choose which parameter to maximize depending on the field requirements.
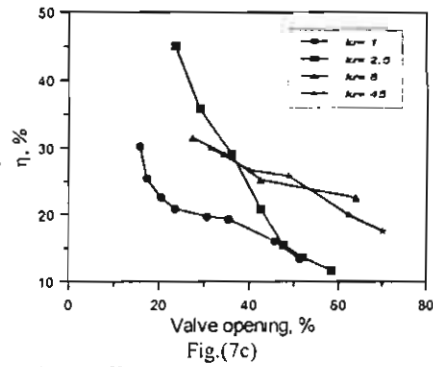

Fig.(7) Effect of initial valve opening on the pump performance; (a) ram flow, (b) drive flow, (c) ram pump efficiency

In fact, the valve oscillation occurs only in a certain range of valve opening what is so called zone of action. Out of this zone, no oscillations were obtained and could be named as zone of silence. Figure (8a) shows the thresholds of this phenomenon on semi-logarithmic axes. The lower and upper thresholds present the inception and die out of the oscillatory motion respectively. From this diagram, it can be seen that the band of operation for each spring is nearly invariant of the spring stiffness. This bandwidth found to be about 40% of the valve opening, while the valve opening of the oscillation inception is decreased with the decrease of spring stiffness. In the limit, as the spring stiffness becomes small enough, the inception of oscillation could be expected at a very small valve opening and vice versa.

For design purpose, the thresholds of frequency and amplitude of pressure transient are illustrated in Figs.(8b,8c). From Fig.(8b), it is seen that the oscillation frequency birth at higher values than its death. In the limit, the two thresholds approach each other at very elastic springs. This could be beneficial in the valve stability design to avoid valve oscillation.

The maximum pressure thresholds shown in Fig. (8c), indicate that at stiffer springs the amplitudes of pressure transient at both thresholds are quite equal. While at elastic springs, the amplitude of pressure transient dies at much higher values than at its birth. That is due to the aforementioned frequency effect.

To show the minimum and maximum possible ramming flow rates, Fig.(8d) has been displayed. It is seen that the maximum ram flow rate could be obtained by using much elastic springs at the highest attainable valve opening.
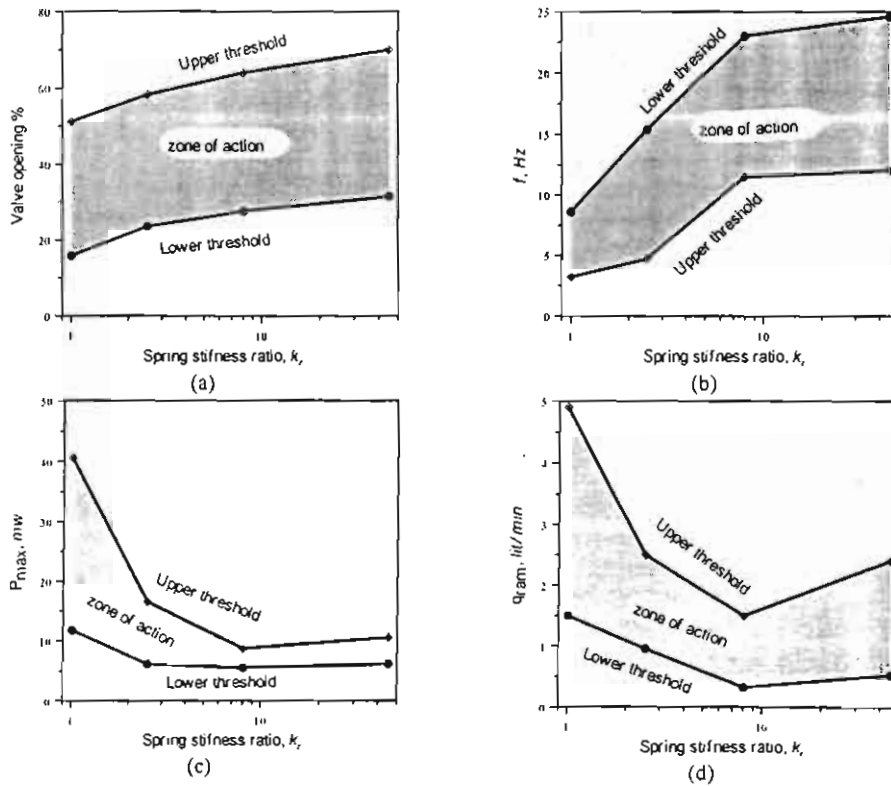

Fig.(8) Diagram of operating limits
(a) valve opening, (b) oscillating frequency, (c) pressure amplitude, and (d) ram flow rate

5. Conclusions and Recommendations

It is concluded from this study that the experimental and computational model results are in good agreement for the prediction of the pressure transients induced by the self-vibratory motion due to valve instability.

The results reveal that the spring stiffness and the initial valve opening have major effects on the ram pump performance. The stiffer spring, the maximum is the efficiency. Also the more elastic spring, the maximum is the ram flow rate. Regardless of the spring stiffness, increasing the valve opening increases the ram flow rate and decreases the pump efficiency.

Diagrammatic plots have been obtained to represent the thresholds and bandwidth of the valve oscillation and/or instability defining the design margins. This could help the designer and the field engineer to avoid the valve oscillation, if valve stability is concerned, or to maximize the outcome of this transient phenomenon if the objective of ram pump is sought.

It is recommended to examine the phenomenon of self-vibratory valve when it is installed in the mid-point of the delivery pipe. A complete computational model for both pressure transient and ram pump performance involving the valve dynamics is required.

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