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Investigating The Effect Of Valve Submersion Depth On The Flow Rate Of Sonic Pump

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INVESTIGATING THE EFFECT OF VALVE SUBMERSION DEPTH ON THE FLOW RATE OF SONIC PUMP

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Ivan A. Loukanov[α], Jacek Uziak[Ω]

Abstract - This paper investigates the effect of the valve submersion depth under the water level in the well on the flow rate of a sonic pump furnished with a spring loaded poppet valve. An equation governing the retardation of water column (*WC*) in terms of valve head losses, valve submersion depth and the depth of pumping is derived and analyzed. Graphs showing the WC retardation in terms of these parameters are presented. It is found that the above parameters have considerable effect on the flow rate of the pump when operating on shallow wells. In order to improve the flow rate when pumping from such wells, valves with low head losses have to be used and submersed at a depth of 3 - 4m. It is also found that the above results do not apply to pumps operating on deep wells because the *WC* retardation attains the gravitational acceleration and the flow rate improvement become negligible.

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I. Introduction

The major pumping elements used in sonic pumps are one-way valves of various designs. For shallow wells up to 30 m and pump resonance frequency up to 20 Hz one valve is usually used, while for deep wells and operating frequency of 20 to 50 Hz, the number of valves may be increased to seven (Usakovskii, 1973). Such arrangement reduces the static and dynamic loads on each valve and increases the flow rate and pressure developed by the pump. Since valves are the only mechanical elements involved in the pumping process the valve design is of great importance for the performance of sonic pumps (Usakovskii, 1973; Virnovskii & Tzinkova, 1966; Loukanov, 2007). In the above studies it is observed that the valve design and valve parameters determine the pump performance. However, in the research of Usakovskii (1973) and Virnovskii & Tzinkova (1966), old valve designs were employed. Since spring loaded poppet valves are readily available today it is therefore required to investigate the suitability of these valves for sonic pump applications. Therefore the objective of this paper is to investigate the effect of the valve submersion depth below the water level in the well on the flow rate of a sonic pump taking into account the valve head losses, the depth of pumping and the valve design parameters.

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II. Material and Methods

The design schematic of a spring loaded poppet valve is shown in Fig. 1.

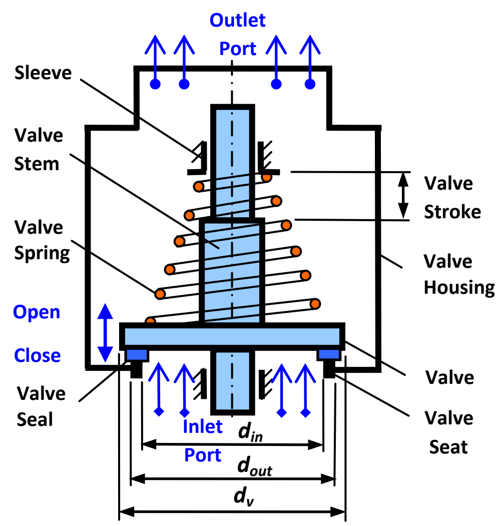

Fig.1 : Schematic of spring loaded poppet valve

In this design water enters through the inlet port, passes through the valve housing and around the valve and leaves from the outlet port towards the oscillating pipe (Fig. 2). As the name of the valve suggests the valve itself is kept generally closed by a preloaded helical compression spring. When the valve opens water flows around the valve through the valve housing experiencing some head losses due to sudden change of the interior cross section as well as due to vortices around the housing corners and the spring. When a poppet valve is employed in a sonic pump it oscillates together with the pipe as shown in Fig. 2. During this process the valve is subjected to a number of forces; some of them are pushing the valve to open while others are assisting it to close. According to Loukanov (2010) the major design parameters of a spring loaded poppet valve are: valve diameters (inlet - d_{in} and outlet - d_{out}), valve stroke (s), valve spring constant (k_v), spring preload (Δ), valve mass (m_v), valve body inlet area (A_{in}), maximum stroke limiting area ($A_{max,\ s}$) and valve body limited area ($A_{v,b}$). Depending upon the size of the valve these parameters vary considerably and determine the performance ability of the poppet valve.

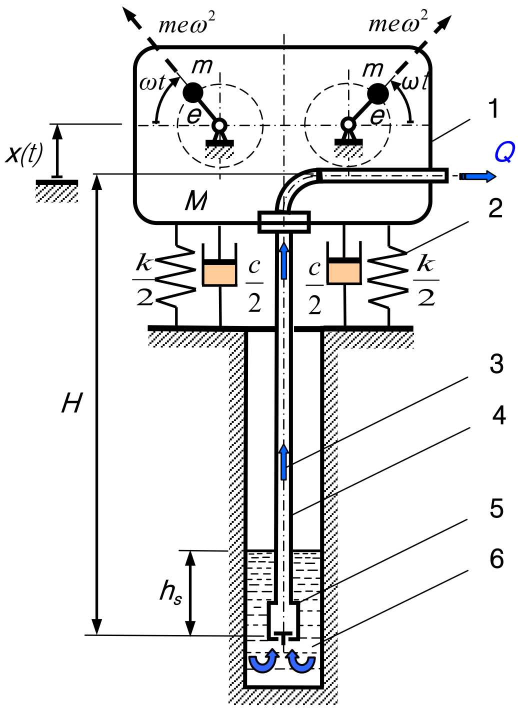

Fig.2 : The dynamic model of a sonic pump: 1 – dual shaft shaker, 2 – spring suspension system, 3 – water column (*WC*) in the pipes 4, 5 – spring loaded poppet valve, and 6 – aquifer of the well

In this study the dynamic model of a sonic pump is presented in order to understand the behaviour of the spring loaded poppet valve as well as the effects of valve head losses and valve submersion depth on the pump flow rate.

Fig. 2 shows the dynamic model of a low frequency sonic pump furnished with one spring loaded poppet valve. The valve is attached to the bottom end of the oscillating pipe and submersed at a depth h_s under the water level in the well. The pump is modelled as one degree-of-freedom oscillating system and for simplicity of the analysis it is assumed that both the pipe and the water column (*WC*) in the pipe are solid bodies. Therefore, the compressibility of *WC* and the elastic properties of the pipe are neglected. In addition to that, the valve itself could move together with the valve housing that is with the pipe as well as relative to the pipe and relative to the *WC*.

In Fig. 2, the following nomenclature is introduced:

- *x(t)* - denotes the absolute displacement of the oscillating system of mass *M*, which includes the masses of shaker, pipes, valve housing and the mass of *WC* in the pipes;
- *H* – is the depth of pumping being equal to the height of *WC*, assuming that water is discharged at the upper end of the well;
- h_s – stands for the depth of valve submersion below the water level in the well;

Since the pipes, valve housing and the shaker are connected together they undergo the same displacement, velocity and acceleration.
The equations governing the resonance vibrations of the oscillating system for one period, such as pipe (valve) displacement, velocity and acceleration are found to be (Loukanov, 2007):

$$x_p(t) = X_{\max}\sin\omega t,$$
$$\dot{x}_p(t) = \omega X_{\max}\cos\omega t = V_{\max}\cos\omega t \quad , \qquad (1)$$
$$\ddot{x}_p(t) = -\omega^2 X_{\max}\sin\omega t = -a_{\max}\sin\omega t$$

where

$x_p(t) = x(t)$, $a_{\max} = \omega^2 X_{\max}$ and $V_{\max} = \omega X_{\max}$ are the absolute displacement, maximum acceleration, and the maximum velocity respectively. The expression $X_{\max} = \dfrac{me}{2M\varsigma\sqrt{1-\varsigma^2}}$ gives the resonance amplitude and the parameter ς denotes the damping factor of the oscillating system.

The equations governing the absolute motion of the *WC* when it is controlled by the gravitational acceleration are as follows

$$x_{wc}(t) = -\frac{gt^2}{2} + V_s t + X_s,$$
$$\dot{x}_{wc}(t) = -gt + V_s, \qquad (2)$$
$$\ddot{x}_{wc}(t) = -g.$$

where

$$V_s = \dot{x}_p(t_s) = V_{\max}\cos\omega t_s = \frac{a_{\max}}{\omega}\cos\omega t_s \qquad (2a)$$

is the velocity of the *WC* at the point of separation *S* (Fig. 3), and

$$X_s = \delta_{st} = \frac{g}{\omega^2} \qquad (2b)$$

is the static spring deflection of the suspension system.

In Eq. (2a), the variable t_s is the time taken by the oscillating system to move from equilibrium position to the point of separation located in Phase 2 (Fig. 3) and is given by

$$t_s = \frac{1}{\omega}\sin^{-1}\left(\frac{g}{a_{\max}}\right). \qquad (3)$$

According to Loukanov (2007) in deriving the equation for the theoretical flow rate of the pump it is assumed that the absolute motion of the *WC* after the point of separation is governed only by the gravitational acceleration, *g* neglecting the losses in the valve and the pipe as well as neglecting the effect of the valve submersion depth under the water level in the well. Under these conditions the equation for the theoretically predicted flow rate of the pump is found to be

$$Q = 250\pi\, d_{in}^{2}\, x_r\, n \;\; [\ell/\text{min}], \qquad (4)$$

where

d_{in}- is the valve inlet diameter [m], as per Fig. 1

$$x_r = h_{\max} - x_p(t_p) \qquad (4a)$$

is the relative distance between the valve seat and the bottom end of the *WC*, where

$$h_{\max} = \frac{V_s^2}{2g} + X_s \qquad (4b)$$

is the maximum height attained by the *WC*, [m] as seen in Fig. 3; $x_p(t_p)$ is the pipe (valve) coordinate at the time when *WC* is at maximum height, [m], g=9.81 m/s^2 and n is the shaker resonance speed [rev/min].

According to Loukanov (2007) four distinguished phases are identified in the motion of the pipe (valve). Fig. 3 illustrates the phases of the pumping process, the motion of the oscillating pipe (valve) and the motion of the *WC*. Since the resonance vibrations of the system are periodic the investigation is carried out for one period of pipe oscillations. The abbreviations in Fig. 3 *"pipe-TDP"* and *"pipe-BDP"* mean "Pipe Top Dead Position" and "Pipe Bottom Dead Position" respectively. Fig. 3 also shows the method of determination of the relative distance x_r, used in Eq. (4). During operation the valve could be either closed or opened. When closed the valve is moving together with the pipe and the *WC* and when opened it moves relative to the pipe and relative to the *WC*. Previously it was found (Loukanov, 2007) that during the suction period taking place in Phases 2, 3 and 4 the valve relative motion is affected by the water stream passing through the valve housing. Since the valve mass, valve spring constant and the valve spring preload are negligibly small as compared to the comparable parameters of the oscillating system the effect of valve relative motion to the pipe (valve) is neglected in this analysis. Also after separating from the valve *WC* initiates independent upward motion interacting with the valve mainly through the suction effect produced, causing the water flow to blow onto the valve bottom face. The above interactions have significant influence on both the valve relative motion to the pipe and valve relative motion to the *WC* but will be neglected in this study.

To determine when the valve opens and closes during one period of pipe oscillations and what forces are acting upon it, both static and dynamic loading conditions on the valve will be investigated taking into account the valve submersion depth, valve head losses and the depth of pumping. It was previously found that for values of $a_{max} \le 4.04 \times g$ the valve is closed during Phase 1 and moves together with the pipe. At the same time *WC* is retarding with progressively decreasing acceleration directed towards the equilibrium position (Loukanov, 2007).

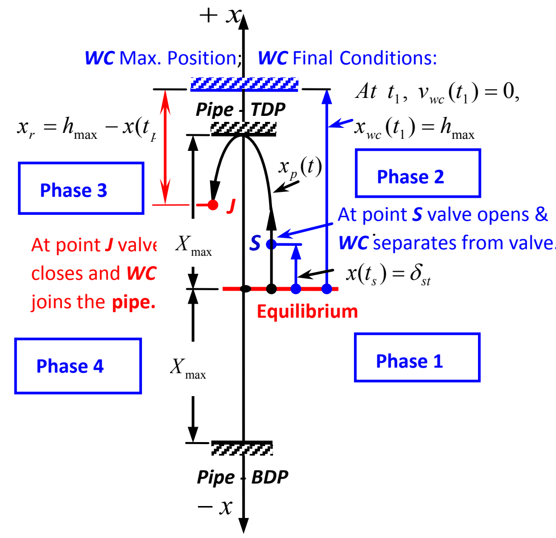

Fig.3 : Phases of the pumping process

During Phase 1 forces acted upon the valve are:

- $G_{wc} = (\rho g H + p_{atm})A_{out}$, [N] - the combined force of the weight of *WC* and the force due to atmospheric pressure acting upon the upper face of the valve, where A_{out} is the upper area of the valve corresponding to a diameter – d_{out}, and ρ is the density of water.
- $G_v = m_v g$, [N] - the weight of the valve, where m_v is the mass of the valve itself along with half of the mass of the valve spring,
- $F_{pl} = k_v \Delta$, [N] - the valve spring preload, where k_v is the valve spring constant and Δ is the primary spring deflection generated during the initial installation of the valve and the spring in the housing,
- $F_{h-st} = (\rho g h_s + p_{atm})A_{in}$, [N] - the combined hydrostatic and atmospheric force acting on the bottom face of the valve, A_{in} is the inlet area of the valve corresponding to the diameter - d_{in},
- $\phi_{wc} = m_{wc}\, \ddot{x}_p(t)$, [N] - the inertia force of *WC* acting downwards on the valve upper face,
- $\phi_v = m_v\, \ddot{x}_p(t)$, [N] - the inertia force of valve acting also downwards, which appears to be negligible as compared to ϕ_{wc}, because of the huge mass difference $m_{wc} >> m_v$, and
- *R*, [N] – is the reaction of the valve seat acting on the valve.

It should be noted that forces ϕ_{wc} and ϕ_v are alternating in accordance with the direction of pipe acceleration $\ddot{x}_p(t)$. As long as the reaction *R* is not zero the valve is closed and moves together with the rest of oscillating parts. Subsequently all the forces acting on the valve are in dynamic equilibrium according to the equation

$$F_{h-st} + R - G_{wc} - G_v - F_{pl} - \phi_{wc} - \phi_v = 0 \qquad (5)$$

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Consider now the upward motion of the oscillating system from equilibrium position towards the point of separation - *S*, as shown in Fig. 3. After passing the equilibrium position the pipe acceleration changes its direction becoming retardation. Then the inertia forces acting on the valve, pipe, shaker and the *WC* change their directions accordingly. On the other hand the resonance amplitude X_{max} of the oscillating system may be considered very small as compared to the depth of submersion ($X_{max} << h_s$), and so it can be assumed with negligible error that the hydrostatic component of the force $F_{h\text{-}st}$ does not vary appreciably.

As long as the reaction *R* of the valve seat exists the valve would be closed and moving together with the pipe. Also it should be mentioned that the inertia forces ϕ_{wc} and ϕ_v are progressively increasing due to an increase in the pipe acceleration $\ddot{x}_p(t)$. It is important to point out that the acceleration of the oscillating pipe is zero at equilibrium and maximum at both *TDP* and *BDP*, being always directed towards the equilibrium. To achieve pumping action the maximum pipe acceleration must be larger than the gravitational acceleration, i.e. $a_{p,max} > g$. At the point of separation *S*, when $t=t_s$ the reaction *R* of the valve seat nullifies and the *WC* begins an independent upward motion in the pipe with constant retardation equal to the gravitational acceleration (Loukanov, 2007). In the interim the pipe is moving up with increasing retardation and when reaching *TDP* it instantly stops and changes its direction of motion towards the equilibrium. After that *WC* reaches its maximum height h_{max} a bit later than the pipe reaches its *TDP* because after the point of separation the pipe's retardation is always larger than the gravitational acceleration $a_p(t)_{retard} > g$. After *WC* attains maximum height it initiates free fall towards the equilibrium with constant acceleration *g*. It should be noted that in the above analysis the effects of valve submersion depth and valve head losses are not taken into consideration.

To summarize, the motion of the pipe (valve) from equilibrium position towards the *TDP* is retarding and takes place in Phase-2, then from *TDP* towards equilibrium (Phase-3) – is accelerating, from equilibrium position towards *BDP* (Phase-4) – is retarding and from *BDP* towards equilibrium position (Phase-1) - is accelerating. In all phases the pipe acceleration is variable, increasing in the directions of *TDP* and *BDP*, and decreasing toward the equilibrium position. It nullifies at the equilibrium position of the system and then changes its direction but it is always directed towards the equilibrium position.

Under these conditions, after the point of separation, there will be two possible interactions between *WC* and the valve. These are:

- *WC* is moving up (Phase-2) acting as a long piston in a cylinder (the pipe) creating a suction force on the upper face of the valve and hence forcing it to open,
- *WC* is moving down (Phase-3 and 4) pushing the valve to close.

The above interactions are to a large extent dependent upon the motion of the pipe and *WC*, and the balance of forces acting upon the valve. Therefore the valve will be either temporarily opened or closed within one oscillating cycle depending upon the balance of forces acting on it. The free motion of *WC* is initiated at the point of separation *S* in Phase 2, as shown in Fig. 3, where the static deflection of the spring suspension system becomes $\delta_{st} = 0$ (Loukanov, 2007).

At this point *WC* separates from the valve and its mass is no longer part of the total oscillating mass *M* of the system. It could be expected that instantaneous change of the total oscillating mass would trigger an immediate change of the resonance frequency of the pipe and consequently change of the resonance amplitudes. But these transformations do not occur since the resonance amplitudes are time dependent and to change them time interval longer than several periods is required.

In the following analysis it is assumed that both the upper end of the pipe and the well are connected to the atmosphere and so the atmospheric pressure acting on both ends of *WC* cancels. Also, the fluid friction between *WC* and the pipe is neglected and the cross sectional area of the pipe is assumed to be equal to the inlet area of the valve. In addition it is considered that water is discharged at the top end of the well and therefore the depth of pumping *H* equals the height of *WC*. Under the above assumptions the equation governing the absolute motion of *WC* is found to be

$$m_{wc}\ddot{x}_{wc} = \rho g h_s A_{in} - m_{wc} g - \rho g h_v A_{in} \qquad (6)$$

The parameters involved in Eq. (6) are as follows:

$\ddot{x}_{wc}$ - acceleration of *WC*, with the $+x$-axis directed up as shown in Fig. 3, [m/s^2];

$m_{wc} = \rho H A_{in}$ - mass of the *WC*, [kg];

ρ - density of water, [kg/m^3];

h_s - valve submersion depth as shown in Fig. 2, [m];

$F_{res} = \rho g h_v A_{in}$ - resisting force due to head losses in the valve housing, [N];

h_v - head losses in the valve housing, measured in meters water head, and

A_{in} - valve inlet area, [m^2].

Upon substitution in Eq. (6) and rearrangement of terms yields

$$\ddot{x}_{wc} = -g\left(1 + \frac{h_v - h_s}{H}\right) = g_1 . \qquad (7)$$

Eq. (7) suggests that the absolute upward motion of the *WC* is retarding with acceleration g_1. The analysis of this equation reveals that the sign of the expression in the parentheses depends upon the difference ($h_v - h_s$) and the depth of pumping *H*.

Accordingly there are four important cases for the practice depending upon the combination of magnitudes of the above parameters.

Case 1: When $(h_v - h_s) < 0$, that is $(h_v < h_s)$, as a result

$$0 < \left(1 + \frac{h_v - h_s}{H}\right) < 1,$$

and therefore *WC* will retard with acceleration smaller than the gravitational acceleration. Then, in accordance with Eq. (2), Eq. (4b) and Fig. 3 the *WC* will attain greater height h_{max}. As a result the relative distance between the bottom end of *WC* and the valve seat will increase and Eq. (4) would yield larger flow rate. In this case the pump performance is improved and to achieve that it requires developing valves with low head losses. The reason is that practically the valve submersion depth is limited to a maximum of 3 – 4 m due to an increased damping effect on the oscillating system (Usakovskii, 1973). It should be noted that the parameter h_s may also vary in case when the well capacity is smaller than the pump capacity. Then the water level in the well would gradually drop to a lower level until a suitable well and pump capacity balance is obtained. As a result the valve submersion depth would also drop although it may become zero when the well and pumping capacity balance is not reached. This would ultimately cease the pumping until water level in the well is recovered.

Case 2: When $(h_v - h_s) > 0$ or $(h_v > h_s)$, then under these conditions

$$\left(1 + \frac{h_v - h_s}{H}\right) > 1.$$

Then *WC* would retard with acceleration greater than the gravitational acceleration. This implies that it would attain smaller maximum height h_{max} and consequently smaller relative distance x_r would be obtained from Eq. (2). Accordingly Eq. (4) would give a smaller flow rate and therefore the flow rate would depreciate. Obviously the above effect becomes insignificant when the depth of pumping *H* is increased considerably. However, when pumping from shallow to medium depth wells (20-60 m) it has to be accounted for. This situation was observed during experiments with a model resonance pump (Loukanov, 2007). The pump was tested with a 55-mm spring loaded poppet valve having head losses h_v=0.43 m and operated at resonance frequency of 5.37 Hz. The oscillating system was subjected to acceleration $a_{max} = 2.0 \times g$, pumping water from H=1.65 m at valve submersion depth h_s=0.25 m. Subsequently the flow rate was measured to be Q = 5.28 ℓ/min while the calculated one was Q = 5.77 ℓ/min.

Now employing Eq. (7) and substituting the values for h_v and h_s the *WC* retardation is found to be

$$\ddot{x}_{wc} = -g\left(1 + \frac{0.43 - 0.25}{1.65}\right) = -1.11g = -10.9 \text{ m/s}^2.$$

Accordingly *WC* attained lesser height h_{max} resulting in a smaller flow rate being calculated from Eq. (4).

Case 3: When $h_v = h_s$, then for any value of *H*,

$$\left(1 + \frac{h_v - h_s}{H}\right) = 1.$$

Under this condition *WC* would retard with acceleration equal to the gravitational acceleration. This is exactly the case described by Eqs. (2), (3), and (4), which do not account for the valve head losses and the submersion depth.

Case 4: When $h_v > h_s$ and $h_s > H$, then the expression in the parentheses become negative

$$\left(1 + \frac{h_v - h_s}{H}\right) < 0.$$

Under these conditions *WC* would be subjected to acceleration larger than the gravitational acceleration and hence would be accelerated contrary to the three cases already discussed. In fact, this case describes the well known principle of the siphon, which is not applicable to sonic pumps since always $H >> h_s$.

Based on the above analysis it is proposed to replace the gravitational acceleration in Eq. (2) and Eq. (4b), with the values of g_1 calculated from Eq. (7) corresponding to the values of h_s, h_v, and *H*, then to determine h_{max}, $x_p(t_p)$ and x_r, and finally to calculate the theoretical flow rate from Eq. (4).

III. RESULTS AND DISCUSSION

The variations of *WC* retardation in terms of parameters involved in Eq. (7) are shown in Figs. 4, 5 and 6, where the gravitational acceleration is included for comparison. All graphs are plotted for valve head losses (h_v) varying from 0 to 1 m, valve submersion depth (h_s) under the water level in the well varying from 0 to 4 m and for three depths of pumping; mainly H=1.65 m, H=10 m and H=20 m.

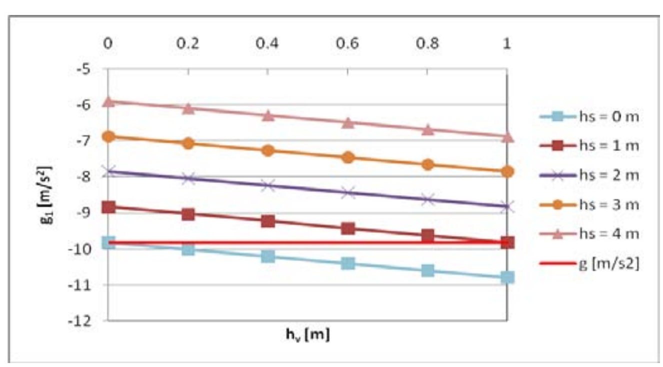

Fig.4 : Retardation of *WC* for *H*=10 m

Fig. 4 illustrates the variation of *WC* retardation (g_1) for depth of pumping H=10m. It is observed that the *WC* retardation is linearly dependent upon h_v and h_s.

Values above the line of gravitational acceleration indicate that *WC* would retard with $g_1 < g$, corresponding to Case 1 and for values below the gravitational acceleration line *WC* would retard with $g_1 > g$, as described by Case 2.

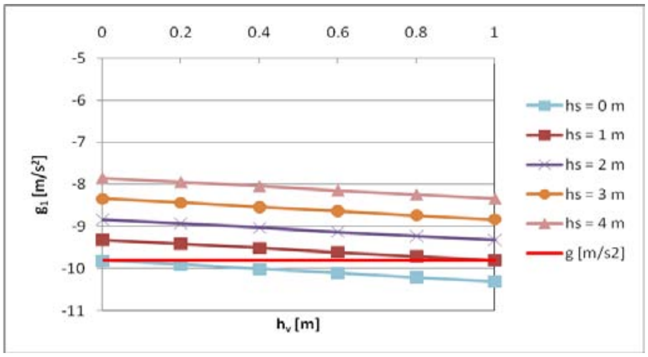

Fig.5 : Retardation of *WC* for *H*=20 m

Fig. 5 shows similar trends of variation of *WC* retardation when plotted for depth of pumping H=20 m. In contrast to Fig. 4 it is seen that the lines become closer to each other and the values of g_1 are grouped within the range of (8.0 – 10.3) m/s^2, while in Fig. 4 these values are within the range (6.0 – 10.4) m/s^2. Also the lines in Fig. 4 are a steeper as compared to the lines in Fig. 5. This indicates that the increased depth of pumping suppresses the effect of valve head losses and valve submersion depth to a large extent forcing the lines to level and approach the line of the gravitational acceleration. The reason being is the term $(h_v\text{-}h_s)/H$, which tends to zero when the depth of pumping approaches infinity that is when $H \rightarrow \infty$ then $a_{wc} \rightarrow g$ and *WC* would retard with the gravitational acceleration irrespective of the values of valve head losses and valve submersion depth. Values of the *WC* retardation $g_1 < g$, imply that *WC* would attain greater maximum height (h_{max}) and therefore larger volume of water will be discharged, which is desirable. On the other hand values of *WC* retardation $g_1 > g$, suggest smaller maximum height and therefore lesser flow rate.

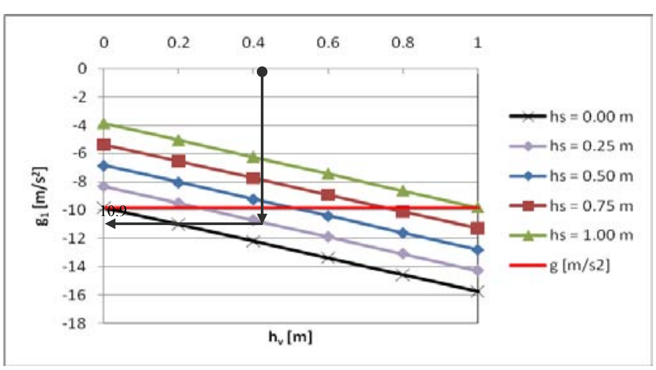

Fig.6 : Retardation of *WC* of model pump for *H*=1.65 m

Fig. 6 depicts the variation of the *WC* retardation of the model pump discussed in Case 2. It is seen from the figure that for the data used (h_s=0.25 m, h_v=0.43 m and H=1.65m) in the illustrative example the graph confirms that the *WC* retarded with acceleration larger than the gravity, (a_{max}=10.9 m/s^2), and therefore the pump delivered smaller flow rate as established during the experiments. To obtain better flow rate it necessitates to submerse the valve deeper than 0.43 m so as to achieve $(a_{max}=g_1) < g$.

IV. Conclusions

In this paper the effects of valve head losses, valve submersion depth and the depth of pumping on the flow rate of a sonic pump are investigated. An equation relating the above parameters is derived, analyzed and graphs based on it are plotted. It is found from the graphs that the parameters involved in Eq. (7) have significant effect on the flow rate of the pump when pumping from shallow to medium depth wells. To improve the flow rate it is suggested to using valves with small head losses and submersing them to a depth of 3-4 m under the water level in the well. The analysis also revealed that an increase in the depth of pumping reduces the negative effect of valve head losses but also the positive effect of the valve submersion depth on flow rate of the pump. Furthermore the greater the depth of pumping the closer the *WC* retardation become to the gravitational acceleration. So the benefits of using valves of small head losses and submersing them deeper in the well would not contribute towards better flow rate when pumping water from very deep wells. To verify the effect of the phenomena based on Eq. (7) a consistent and precise testing on shallow, medium and deep boreholes is required. Possibly this will be the aim of another investigation where valves with low head losses will be developed and tested on a sonic pump under the recommended submersion depths and determine the effects on the flow rate of a sonic pump.

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