Investigating the pumping process of a resonance-vibrating pump for medium-depth boreholes

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Abstract: This paper deals with the pumping process of a resonance-vibrating pump, which utilizes the resonance vibrations of one degree-of-freedom oscillating system. The pump is powered by a mechanical shaker consisting of two counter rotating offset masses and operating in resonance. The study investigates the nature of the pumping process and conditions required to achieve pumping action. Equations for the flow rate, pressure developed at ground level or any height above it, the pump efficiency, and the power delivered by the shaker are derived. The analysis of the pumping process revealed that the flow rate of the pump may be maximized either by increasing the acceleration imparted on the oscillating system, and/or by reducing the resonance frequency. It was found that the pressure developed by the pump is independent of the depth of pumping, provided that the same acceleration is imparted, and its efficiency may be increased either by reducing the resonance frequency and/or by increasing the depth of pumping. The preliminary test results about the flow rate and pressure developed at ground level appeared to be close to the values predicted by the proposed theory. Based on the analysis of the theoretical and experimental findings it is concluded that the equations derived in this study may be employed in designing resonance vibrating pumps for a desirable flow rate, pressure, and efficiency in pumping water from a specified depth.

Keywords: resonance vibrating pumps; dual-shaft shaker; spring suspension system; foot valve; oscillating pipe; water column

Pumping water in the desert and remote countryside in the absence of electricity has always been a great engineering challenge. A cheaper and easy means for solving this problem is to use a resonance vibrating pump. These pumps were invented in the early fifties of the 20th century and since then were used for pumping crud oil from as deep as 2000 m, as well as water from 30 to 150 m, (Usakovskii 1973). Unfortunately these pumps are almost unknown today because of the intensive use of submersible centrifugal, screw, and rod-pumps, the latter being specifically designed for pumping natural oil from deep boreholes. The current generation of these pumps is electrically powered, which restricts their use in the non-electrified regions in the desert and semiarid areas of Africa, Middle East, and central Asia. In those regions ground water is available in boreholes at depths of 30–100 m drilled with diameters from 100 to 160 mm but they remain unutilized because of insufficient means to drawn water to the surface. Moreover electrically powered submersible pumps are expensive for developing countries and complex for maintenance by low qualified personnel. This makes the resonance vibrating pumps competitive in terms of price, simple design, and performance as compared with centrifugal or screw pumps used today.

As the name suggests resonance vibrating pumps utilize resonance vibrations of one or two-degrees-of-freedom oscillating systems to achieve pumping action and draw liquids practically from unlimited depths. When operating at speed smaller than 1200 rev/min these are termed vibrating pumps, while those operating at speed greater than 1200 up to 4800 rev/min are known as inertia, acoustic, or sonic pumps, (Dubrovskii 1968; Bodine 1969, and Usakovskii 1973).

It should be noted that the extraction of water by resonance pump is no longer limited to 8–10 m as the case is with the suction pumps and it was found to be independent from the depth of pumping as proved by Usakovskii (1963, 1973). The pumps are efficient and require less power than the conventional pumps as they operate in resonance. Unfortunately these pumps were forgotten and one cannot find them under any pump classification in the literature today.

The resonance pump has been invented by Bodine (1951) and later modified and improved by him and other inventors. The design of one of his US patent #2,553,543, 1951 is shown in Figure 1. It consists mainly of: a primary mover (19), a shaking mechanism (16–18), an oscillating pipe (11), an elastic suspension (13), and a ball valve (15). The operation of the apparatus is as follows; when the shaker rotates at the natural frequency of the mass-spring system, then a resonance occurs, and therefore large amplitudes of displacement, velocity and acceleration are generated. As the pipe oscillates with acceleration greater than the gravity the valve opens and closes periodically allowing water from the well to enter into and travel up the pipe against the gravitational force.

There are many other designs of resonance pumps proposed by Angona (1964), Bentley (1981), James (1953), Kletzkin (1969), Usakovskii (1973) etc., but the design shown in Figure 1 seams to be the simplest and easy to manufacture in the developing countries.

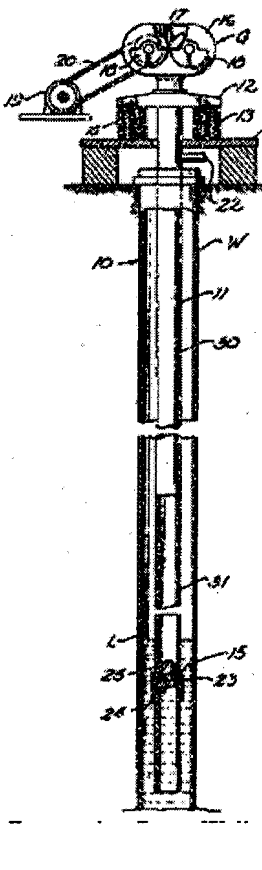

Figure 1. Deep well pump

Unfortunately a proper explanation of the pumping process and the performance of resonance vibrating pump is not available today. Considering the importance of drinking water for the poor people and small farmers in the arid and semi arid areas of Africa a decision was made to investigate the pumping process of these pumps with the strong desire of designing, developing and implementing the resonance vibrating pumps in the desert and remote areas of Botswana.

THEORETICAL CONSIDERATIONS

Parameters characterizing the oscillating system of the resonance pump

The dynamic model of the resonance vibrating pump was assumed to be one-degree-of-freedom system, as shown in Figure 2. The parameters characterizing this model are:

- M – total oscillating mass, which includes the mass of the pipe together with water inside, the mass of the foot valve, shaker and the attachment parts (kg)
- m – total rotating offset mass of the shaker (kg)
- e – eccentricity of the rotating masses (m)
- me – rotating unbalance (kg.m)
- k – stiffness of the spring suspension system (N/m)
- c – damping constant of the viscous damper (N.s/m)
- ω – resonance angular frequency = angular speed of the shaker (rad/s)
- x – displacement of the total oscillating mass (m)
- $\dot{x}$, $\ddot{x}$ – velocity and acceleration of the oscillating mass (m/s), and (m/s^2) respectively
- t – time (s)

The differential equation governing the motion of the oscillating system is

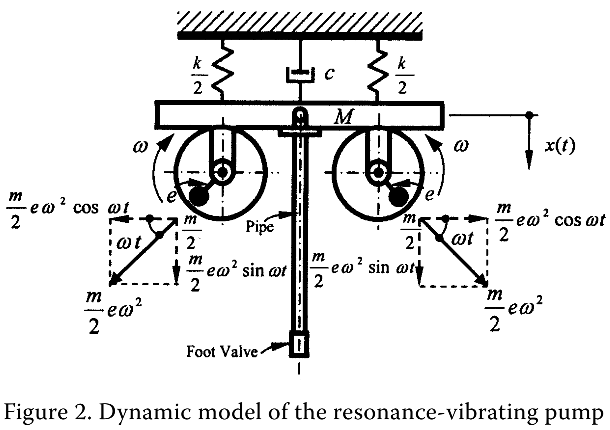

Figure 2. Dynamic model of the resonance-vibrating pump

$$\ddot{x} + 2\zeta\omega_n\dot{x} + \omega_n x = \frac{me\,\omega^2}{M}\sin\omega t \qquad (1)$$

where:

$me\,\omega^2 \sin \omega t$ – excitation force generated by the shaker

ω_n – natural frequency of the oscillating system defined as

$$\omega_n = \sqrt{\frac{k}{M}} \qquad (2)$$

where:

ζ – damping factor of the system given by

$$\zeta = \frac{c}{2M\omega_n} \qquad (3)$$

The steady state solution of the system response in resonance is known as

$$x(t) = X \sin(\omega t - \psi) \qquad (4)$$

where:

X – amplitude of the resonance vibrations,

ψ – phase angle.

The angle ψ indicates that the response of the system in resonance lags the action of the inertia force.

According to HUTTON (1981) and SINGURESU (1995), the resonance amplitude is defined as

$$X = \frac{me}{M} \times \frac{r^2}{\sqrt{(1 - r^2) + (2r\,\zeta)^2}} \qquad (5)$$

where:

$r = \omega/\omega_n$ – frequency ratio.

If Eq. (5) is rearranged to find the maximum value of the ratio MX/me, then the condition for that is

$$r = \frac{1}{\sqrt{1-2\zeta^2}} \qquad (6)$$

Since for inertia type of excitation the resonance takes place slightly to the right of $r = \omega/\omega_n = 1$, then by substituting Eq. (6) into Eq. (5) the maximum value of the ratio MX/me is found to be

$$\left(\frac{MX}{me}\right)_{\max} = \frac{MX_{\max}}{me} = \frac{1}{2\zeta\sqrt{1-\zeta^2}} \qquad (7)$$

From where the resonance amplitude $X_{\max}$ is found to be dependant on the shaker rotating unbalance, total oscillating mass, and the damping factor of the oscillating system as indicated below

$$X_{\max} = \frac{me}{M} \times \frac{1}{2\zeta\sqrt{1-\zeta^2}} \qquad (8)$$

By differentiating twice Eq. (4) with respect to time the equations governing the velocity and acceleration of the oscillating system in resonance are found to be

$$v(t) = \omega X_{\max}\cos(\omega t - \psi) \qquad (9)$$

$$a(t) = -\omega^2 X_{\max}\sin(\omega t - \psi) \qquad (10)$$

By substituting Eq. (8) into Eq. (9) and (10) maximum amplitudes of the velocity and acceleration are found to be

$$v_{\max} = \omega X_{\max} = \frac{me}{M} \times \frac{\omega}{2\zeta\sqrt{1-\zeta^2}} \qquad (11)$$

$$a_{\max} = \omega^2 X_{\max} = \frac{me}{M} \times \frac{\omega^2}{2\zeta\sqrt{1-\zeta^2}} \qquad (12)$$

By analysing the water column behaviour in the pipe at different magnitudes of acceleration it was concluded that it is supported periodically by the valve and separates from it at a particular instant. This suggests that water column would not be always moving together with the pipe during the entire period of oscillation. Therefore the total mass of the system is considered as composed of the following masses

$$M = (m_{\text{pipe}} + m_{\text{valve}} + m_{\text{shaker}}) \pm m_{\text{water column}} \qquad (13)$$

In regard to Eq. (13) it is assumed that at a particular instant depending upon the magnitude of the imparted acceleration, water column will separate from the pipe (valve), and thereafter perform free vertical motion inside the pipe. This will generate vacuum above the valve forcing it to open and water from the well to enter into the pipe. At later instant when the speed of the pipe and water column become almost equal the valve will close due to the valve spring action and water will join the mass of the oscillating system. Since water is incompressible, whatever amount of water enters the pipe the same amount will leave it at the upper end. When water column is separated from the pipe, the mass of the oscillating system may be considered as composed by the mass of the pipe, valve, and that of the shaker

$$M_p = m_{\text{pipe}} + m_{\text{valve}} + m_{\text{shaker}} \qquad (14)$$

In this regard the effect of sudden drop in the total oscillating mass on the resonance frequency and amplitude is neglected since the oscillating system cannot respond that fast for a small fraction of the period of oscillation. Much longer time is required for these changes to develop, as the process is time dependant and asymptotically attaining the new frequency and amplitude. For simplicity in explaining the following analysis the oscillating system will be considered as represented by the pipe only.

Phases of interaction between the valve and water column

Consider now one period of resonance oscillation of the mass system "Pipe-valve-shaker ± water column". In Figure 3 the "pipe-valve-shaker" and "water column" are depicted as two solid bodies (particles), moving together or individually in accordance of their common or individual equations of motion. For determining the flow rate of the pump it is assumed that the period of one oscillation T is divided into four phases each of them of duration one quarter of the period.

Phase 1

In general, Phase 1 begins at the bottom dead position (*BDP*) of the oscillating system and ends at the equilibrium position (Figure 3). At the *BDP* the oscillating system is characterized by the resonance displacement $x_p(t_p = 3/4T) = -X_{max}$, speed v_p $(t_p = 3/4T) = 0$, and acceleration $a_{max} = \omega^2 X_{max}$. During this phase the valve is closed and the pipe and water column accelerate together as one body towards the equilibrium position. When they reach equilibrium position the displacement $x_p(t) = 0$, the velocity of resonance vibration attains its maximum value $v_{max} = \omega X_{max}$, and the acceleration nullifies. In accordance with the principles of harmonic motion

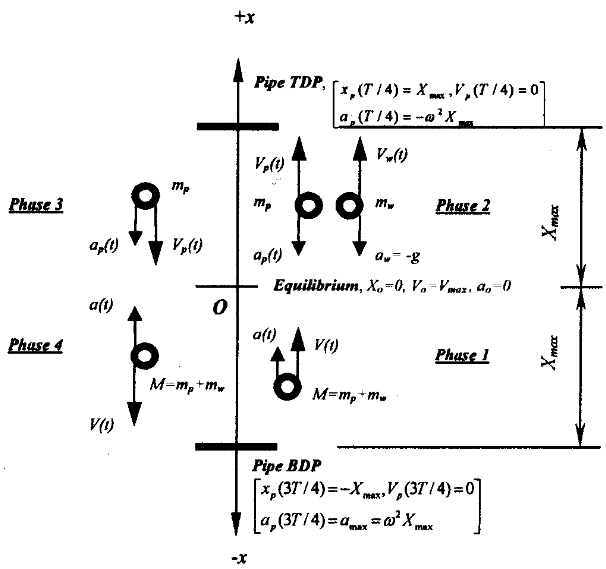

Figure 3. Phases of interaction between the valve and water column

at this point the acceleration changes its direction and becomes retardation during the next phase.

Phase 2

This phase begins at the equilibrium and ends at the top dead position *(TDP)* of the oscillating system. Since at the beginning of this phase water and pipe are moving together they will retard at the same rate as governed by the equations of motion of the oscillating system. At particular instant t_s – termed time of separation and measured from the equilibrium position, the retardation will become equal to the earth acceleration. At this instant water column will separate from the valve, as it has one-sided support provided by the valve, and therefore will start moving inside the pipe.

The proposed criterion for separation is based on the physical fact that separation will take place whenever the retardation of the pipe becomes greater than that of the water column

$$-\ddot{x}_p(t_s) \geq -\ddot{x}_w(t_s) \tag{15}$$

When separation takes place the pipe system will retard in accordance with Eq. (10) whilst water column will retard at constant rate due to gravity $g = 9.81$ m/s^2, that is

$$\ddot{x}_w(t) = -g \tag{16}$$

As the pipe moves in accordance with the equations governing the resonance vibrations, then water column will be moving inside the pipe as a body thrown vertically with initial conditions determined at the point of separation and its motion will be governed by the equations derived from Eq. (16).

Further the separation of water column from the valve creates vacuum above it since water column acts as a long piston with the pipe being the cylinder, which forces the valve to open and allows water from the well to enter into the volume vacated by the moving column of water.

If the time taken for the water column to reach its maximum height, measured from the point of separation is denoted t_1, and the corresponding time for the pipe $t_p = t_s + t_1$, measured from equilibrium, one can find the locations of both water column and pipe at that instant

$$x_w(t_1) = h_{max} \tag{17}$$

$$x_p(t_p) = X_{max} \sin(\omega t_p) \tag{18}$$

Therefore the maximum relative distance attained between the valve and water column will be

$$X_{rel} = x_w(t_1) - x_p(t_s + t_1) \quad (19)$$

As far as the relative distance increases the valve will be kept open and water from the well will flow into the pipe. This process continues until the relative motion persists and may take place during Phases 3, 4 and 1. This fact can be used to maximize the flow rate of the pump by imparting on the oscillating system as large acceleration as possible. It should be noted that the imparted acceleration would be limited by the pipe endurance limit due to a possible axial fatigue.

Phase 3

This phase begins at the pipe *TDP* and completes at equilibrium position. At *TDP* the pipe's retardation changes to acceleration and hence during this phase the pipe system will accelerate towards the equilibrium position. As far as the relative distance between the valve and water column increases the valve will be kept open and water will be flowing into the pipe. At any point when the relative motion ceases the valve will close due to the action of the valve spring, retaining the entire water column in the pipe. Thus the pipe and water become again one body and will be moving together until the next cycle of separation takes place in Phase 2.

Phase 4

This phase begins at equilibrium position and ends at *BDP* of the oscillating system. In fact the acceleration of any harmonic motion is always directed towards the equilibrium position, so does the acceleration of the pump oscillating system. Therefore during this phase the system retards towards *BDP*. In order to maximize X_{rel} and hence the flow rate of the pump, it is desirable the valve to shut at the end of this phase, when at the same time water column is at its maximum height above equilibrium. This would require an application of an appropriate magnitude of the imparted acceleration. Whenever water column joins the pipe, the oscillating mass of the system will be recovered from M_p to M. In particular at the end of Phase 4 the system will attain the same kinematics' parameters as it has had at the beginning of Phase 1.

Determining the flow rate of the resonance pump

Determination of the time of separation

To achieve a minimal flow rate a particular value of the system acceleration is required, as stated by Dubrovskii (1968), if a_{max} = 9.81 m/s^2 the pump will just about to start raising water from the well.

The analysis of the preceding section revealed that to achieve any value of X_{rel}, water column must separate from the valve during Phase 2 and then perform upward motion inside the pipe. The separation might be achieved if the criterion defined by Eq. (15) is satisfied, that is

$$-\ddot{x}_p(t_s) \geq -\ddot{x}_w(t_s) \quad (20)$$

where:

t_s – separating time measured from the equilibrium position (s)

$\ddot{x}_p(t_s) = -\omega^2 X_{max} \sin(\omega t_s)$ – pipe retardation at the instant of separation (m/s^2)

$\ddot{x}_w(t_s) = -g$ – gravitational acceleration acting upon the water column.

Substituting the above expressions into Eq. (20), rearranging and taking the equality sign only, the equation for the separating time is obtained

$$t_s = \frac{1}{\omega} \sin^{-1}\left(\frac{g}{\omega^2 X_{max}}\right) \quad (21)$$

Since at the instant of separation the pipe and water column will still be moving together, the separating displacement and velocity of water column may be determined from the equations governing the displacement and velocity of the oscillating system

$$V_s = v_w(t_s) = v_p(t_s) = \omega X_{max} \cos(\omega t_s) \quad (22)$$

$$X_s = x_w(t_s) = x_p(t_s) = X_{max} \sin(\omega t_s) \quad (23)$$

Determining the location of the point of separation

Substituting Eq. (21) into Eq. (23) gives the location of water column at the point of separation

$$x_w(t_s) = X_s = \frac{g}{\omega^2} \quad (24)$$

Since in equilibrium position the total weight of the oscillating system is supported by the static spring force, hence one can find the resultant stiffness of springs

$$k = \frac{Mg}{\delta_{st}} \quad (25)$$

where:

δ_{st} – static deflection of springs.

Substituting Eq. (25) into Eq. (2) yields

$$\delta_{st} = \frac{g}{\omega_n^2} \quad (26)$$

Comparing Eq. (24) and Eq. (26) and considering that in resonance $\omega = \omega_n$ it may be seen that the location of water column at the instant of separation is provided by the same expression

$$x_w(t_s) = X_s = \delta_{st} = \frac{g}{\omega^2} \qquad (27)$$

Therefore Eq. (27) reveals that separation should always take place at the point of zero spring deflection regardless of the resonance frequency.

Determining water column and pipe equations of motion

Consider now the water column vertical motion taking place inside the pipe. The differential equation governing this motion is defined by Eq. (16)

$$\ddot{x}_w(t) = -g$$

By integrating twice with respect to time and considering the water column initial conditions at the point of separation $t = 0$, $\dot{x}_w(0) = V_s = \omega X_{max} \cos(\omega t_s)$ and $x_w(0) = X_s = g/\omega^2$, the equations governing water column vertical motion are found to be

$$\dot{x}_w(t) = -gt + \omega X_{max} \cos(\omega t_s),$$
$$x_w(t) = -\frac{gt^2}{2} + [\omega X_{max} \cos(\omega t_s)]t + \frac{g}{\omega^2} \qquad (28)$$

Now considering the water column boundary conditions at the point of its maximum height $t = t_1$, $\dot{x}_w(t_1) = V_1 = 0$ and $x_w(t_1) = h_{max}$, and substituting them into Eq. (28) the time taken for the water column to attain its maximum height and the value of that height are found to be

$$t_1 = \frac{\omega X_{max} \cos(\omega t_s)}{g}, \text{ and}$$

$$h_{max} = x_w(t_1) = \frac{[\omega X_{max} \cos(\omega t_s)]^2}{2g} + \frac{g}{\omega^2} \qquad (29)$$

Consider now the oscillating motion of the pipe-valve-shaker mass system. To find out its corresponding position when water column is at its maximum height it is necessary to estimate the time required for the system to reach that position. Taking into account that the time for the pipe motion is counted from equilibrium while the time for the water column motion is measured from the point of separation, the corresponding duration of motion of the pipe is

$$t_p = t_s + t_1 \qquad (30)$$

Thus the pipe (valve) velocity and its location may be obtained from Eq. (9) and Eq. (4) respectively and for the purposes of determining the relative distance between the valve and water column, the phase angle ψ was omitted as it has no effect on that distance.

$$\dot{x}_p(t_p) = \omega X_{max} \cos(\omega t_p)$$
$$x_p(t_p) = X_{max} \sin(\omega t_p) \qquad (9' \,\&\, 4')$$

Determining the relative distance between water column and pipe

If the separation exists the relative distance may be defined as the vertical distance between the lower end of water column and the inner surface of the valve. It specifies the maximum height of a cylindrical volume vacated by the moving column of water into the pipe when water column is at the most distant position from the valve (Figure 4).

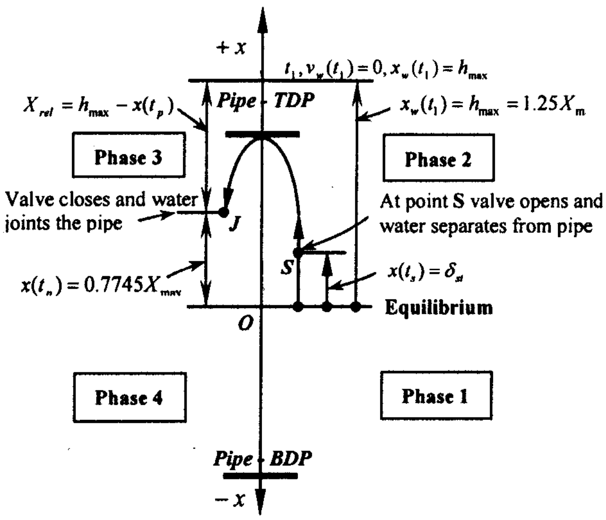

Figure 4. Determination of the relative distance

According to Figure 4 the maximum value of the relative distance is given by

$$X_{rel} = x_w(t_1) - x_p(t_p) = h_{max} - x_p(t_s + t_1) \qquad (31)$$

To find out the location of the valve when it shuts and water column joins the oscillating system, a time ratio was defined.

$$\text{Time ratio} = \frac{t_p}{0.25T} = \frac{2\omega}{\pi} t_p, \text{ dimensionless} \qquad (32)$$

Depending upon the values of the time ratio the following cases may be encountered:

If the time ratio $2\omega t_p/\pi = 1$, then the valve and water column will be situated at the end of Phase 2, and therefore there will be no separation, hence $X_{rel} = 0$. This occurs when $a_{max} = g$.

If $(1 < (2\omega t_p/\pi) \leq 2)$, the separation will be in Phase 2, and valve will shut in Phase 3, hence $X_{rel} \neq 0$.

If $(2 < (2\omega t_p/\pi) \leq 3)$, the separation will be in Phase 2, and valve will shut in Phase 4, hence $X_{rel} \neq 0$.

If $(3 < (2\omega t_p/\pi) \leq 4)$, there will be a separation in Phase 2 but the valve will shut at the beginning of Phase 1 and hence it will be further moving together with the water column. Consequently a further increase in the relative distance will be slowed down and any further increase will be achieved for the expense of increasing the maximum height h_{max}.

It was found by numerical calculations that the maximum value of the relative distance is attained when a resonance acceleration of 4.604 g m/s^2 is imparted on the oscillating system. In this case the valve closes at the *BDP* of the pipe when water column is at its maximum height. Figure 4 shows the locations of the point of separation, the point when valve shuts, and the relative distance between the valve and water column when a_{max} = 2 g. Hence, water column joins the system in Phase 3.

Estimating the pump flow rate

The above analysis reveals that the resonance pump operates in a manner similar to that of a reciprocating pump therefore its flow rate may be defined as

$$Q = A_v X_{rel} \qquad (\text{m}^3/\text{cycle}) \qquad (33)$$

where:

X_{rel} – stands for the stroke of the virtual reciprocating pump.

Considering the number of oscillations (strokes) per minute it becomes

$$Q = A_v X_{rel} n \qquad (\text{m}^3/\text{min}) \qquad (34)$$

where:

$A_v = (\pi\, d_v^2)/4$ – inlet area of the foot valve (m^2)

n – angular speed of shaker (rpm)

The above equations may also be rearranged to give the flow rate in litres per minute

$$Q = 250\pi\, d_v^2 X_{rel}\, n \quad (\text{l/min}) \qquad (35)$$

Pressure developed by the pump at ground level

The pressure developed by the resonance vibrating pump may be defined as

$$p_H = \frac{F_{sp}}{A_v} \qquad (36)$$

where:

$F_{sp} = kx_p(t) = kX_{max} \sin(\omega t - \psi)$ – spring force driving the system in resonance

$k = M\omega^2$ – stiffness of the suspension system,

$A_v = (\pi\, d_v^2)/4$ – foot valve inlet area.

Upon substitution into Eq. (36) the pressure developed by the pump at ground level is obtained

$$p_H = \frac{4M\omega^2}{\pi\, d_v^{\,2}} X_{max} \sin(\omega t - \psi) \quad (\text{N/m}^2) \qquad (37)$$

The analysis of Eq. (37) suggests that there are two pressure pulses delivered during one oscillation cycle, corresponding to the two maximum values of $\sin(\omega t - \psi)$. But it was proved that separation of water column takes place only once per oscillation and therefore this will generate only one pulse of pressure. It is for this reason Eq. (37) has to be divided by a factor of 2.

Furthermore to obtain the average pressure developed by the pump the average value of the pressure has to be obtained based on the following mathematical consideration

$$\text{Average of } [X_{max} \sin(\omega t - \psi)] = \frac{2}{\pi} (X_{max}) \qquad (38)$$

Therefore the average pressure generated by the pump at ground level per cycle of oscillation is

$$p_{av} = \frac{4M\omega^2}{\pi^2 d_v^2} X_{max}, \text{ or}$$

$$p_{av} = \frac{2me\omega^2}{\pi^2 d_v^2} \times \frac{1}{\zeta\sqrt{1-\zeta^2}} \qquad (\text{Pa}) \qquad (39)$$

Estimating the efficiency of resonance pump

The efficiency of the resonance pump may be written as

$$\eta = \frac{\rho g(H + H_o)Q}{\text{Input power}} \qquad (40)$$

where:

H – depth of pumping

H_o – delivery head

The input power supplied to the resonance pump is the mean power output generated by the shaker and may be estimated from the following expression

$$P_{\text{The shaker mean power}} = \frac{1}{T}\int_0^T F(t)\, \dot{x}(t)dt \qquad (41)$$

The resultant inertia force generated by the shaker in the vertical direction is given by

$$F(t) = me\,\omega^2 \sin(\omega t) \quad (42)$$

while the velocity of the system is provided by Eq. (10) as $\dot{x}(t) = \omega X_{max} \cos(\omega t - \psi)$.

Upon substitution in Eq. (41) the input power of shaker may be written as

$$P_{\substack{\text{Input}\\ \text{power}}} = \frac{me\omega^4}{2\pi} X_{max} \int_0^T \sin(\omega t) \cos(\omega t - \psi) dt \quad (43)$$

Solving the Integral in Eq. (43) within the specified limits of integration gives

$$\text{Integral} = \frac{\pi}{\omega} \quad (44)$$

By substituting the solution of the integral Eq. (44) into Eq. (43) yields

$$P_{\text{Input power}} = \frac{me\omega^3}{2} X_{max}, \text{ or}$$

$$P_{\text{Input power}} = \frac{(me)^2\omega^3}{4M\zeta \sqrt{1 - \zeta^2}} \quad \text{(Watt)} \quad (45)$$

Now substituting Eq. (45) into Eq. (40) the equation for the pump efficiency is obtained

$$\eta = \frac{2\rho g(H + H_0)\,Q}{me\omega^3 X_{max}} \times 100\%, \text{ or}$$

$$\eta = \frac{4\rho g M Q(H + H_0)\,(\zeta\sqrt{1 - \zeta^2})}{(me)^2\,\omega^3} \times 100\% \quad (46)$$

The analysis of the first part of Eq. (46) reveals that for a desirable flow rate high efficiency can be obtained by reducing the resonance frequency and/or by increasing the depth of pumping $(H + H_0)$. It is evident that the effect of the resonance frequency on the pump efficiency is much significant as compared to the linear effect caused by the depth of pumping.

The Resonance Pump Model

Based of the theory developed a functional model of the resonance pump was designed, constructed and tested. Figure 5 shows the experimental setup used for testing the resonance model pump. The shaker in this design was composed of two small AC motors intended for ceiling fans, which were set to rotate in opposite directions and synchronized by employing timing belt rubber gears. The amount of the unbalanced mass per motor was $m = 0.176$ kg offset at a radius of 0.0895 m. One of the motors was passive and the other one active, delivering 60 W power output at 360 rpm. An electronic control device was used to vary the speed of the active motor to achieve resonance in the oscillating system. The pumped water was measured by a conventional water meter type C-PHB3122 and circulated through a bucket of 20 litre capacity. Table 1 lists the flow rates delivered by the pump for two types of valve: mainly EUROPA and BOSSINI. The valve of the EUROPA design was spring free; while in the BOSSINI design the valves were spring loaded. The use of strainers having 90° angle of taper of two different inlet diameters was intended to improve the flow rate of the pump, but an adverse effect was observed due to an increased damping. This was explained with the limited power output of the shaker, which was unable to

Figure 5. The experimental setup

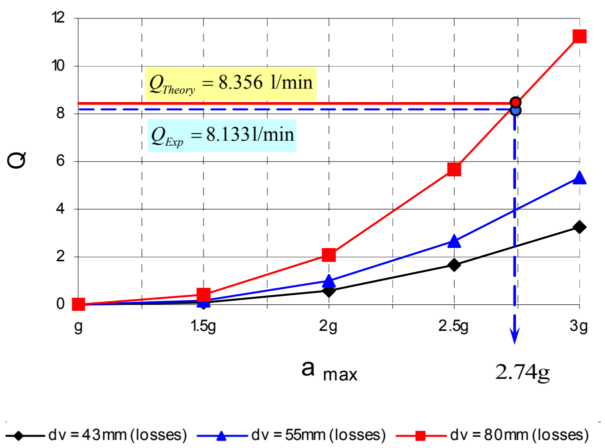

Figure 6. Comparison between theoretical and experimental flow rates for 3-inch valve

Table 1. Flow rates delivered by the model pump with and without strainers

Flow rate		Valves					
		EUROPA 1.5 Inch	BOSSINI				
			1.5 Inch (1)	1.5 Inch (2)	2 Inch (1)	2 Inch (2)	3 Inch
Without strainer							
Run 1	(l/5min)	28.3	26.3	26.3	27.7	30	40
Run 2	(l/5min)	29	26.5	26.8	27.5	31	41
Run 3	(l/5min)	29	26.8	25.7	27,5	30.8	41
Average	(l/min)	5.753	5.307	5.253	5.513	6.12	8.133
With strainer							
ø = 90 mm							
Run 1	(l/5min)	26.5	24	24.4	27.2	30	–
Run 2	(l/5min)	26.3	24	24.3	27.3	29.5	–
Run 3	(l/5min)	26.6	24	24.2	27.5	29	–
Average	(l/min)	5.293	4.8	4.86	5.467	5.9	–
ø = 110 mm							
Run 1	(l/5min)	24.5	24.6	24	25.6	26.3	–
Run 2	(l/5min)	24.5	25	23.7	25.5	25.5	–
Run 3	(l/5min)	24.6	25	23.7	25.6	26.1	–
Average	(l/min)	4.907	4.973	4.76	5.113	5.193	–

provide the required acceleration and compensate for the increased energy losses when strainers were employed. Figure 6 shows the theoretical predictions of the flow rate for 1.5, 2, and 3-inch valves versus different accelerations of the oscillating system. In this figure the point of the experimental flow rate is shown, obtained with a 3-inch valve without a strainer, at frequency of 5 Hz, amplitude 21.4 mm, acceleration 2.74g m/s^2, and a depth of pumping H = 1.65 m.

The pressure and efficiency of the pump were found to be 4.55 m water head and 14.5% respectively.

CONCLUSIONS

The theory proposed in this paper provides better understanding of the pumping process and the performance of resonance-vibrating pumps. The suggested methodology and the derived equations are simple and easy to use in designing, analysing, and improving the pump performance. By employing Eq. (35) one can calculate the theoretical flow rates of the pump for different frequencies and imparted accelerations, and modify them later by employing the coefficients of head losses in the pipe system. Eq. (39) yields the average pressure developed by the pump at ground level, which appears to be independent of the depth of pumping, provided that the imparted acceleration is of the same magnitude. From Eq. (46) it is evident that to improve the pump efficiency for a desirable flow rate and given depth of pumping one should reduce the resonance frequency of operation. On the other hand when the pump is set to operate at a particular resonance frequency, the efficiency of the pump will increase with the depth of pumping.

Based on the experimental results and employing the above equations, one can conduct a pump design achieving a desirable flow rate, pressure and efficiency in operating at a particular depth.

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Abstrakt

Loukanov I.A. (2007): **Výzkum pracovního procesu rezonančně vibračního čerpadla při čerpání vody ze středně hlubokých studní.** Res. Agr. Eng., **53**: 172–181.

Příspěvek se zabývá čerpacím procesem rezonančně vibračního čerpadla, který využívá rezonanční vibrace oscilačního systému s jedním stupněm volnosti. Čerpadlo je poháněno mechanickým třasadlem, tvořeným dvěma vzájemně posunutými závažími s opačnou rotací a pracujícími v rezonanci. Ve studijní části se zkoumá povaha čerpacího procesu a podmínky vyžadované pro dosažení čerpací činnosti. Jsou odvozeny rovnice pro průtok, tlak vyvinutý pro přízemní hladinu nebo pro jakoukoliv výšku nad tímto základem, účinnost čerpadla a energii dodanou třasadlem. Analýza čerpacího procesu ukázala, že průtok čerpadla může být maximalizován buď zvýšením zrychlení oscilačního systému, nebo omezením vlastního kmitočtu. Bylo zjištěno, že tlak vytvořený čerpadlem není závislý na hloubce čerpání v případě, že je uděleno stejné zrychlení; jeho účinnost může být zvýšena buď omezením vlastního kmitočtu, nebo zvýšením hloubky čerpání. Předběžné výsledky zkoušek, týkající se průtoku a tlaku vyvinutého pro přízemní hladinu, jsou podobné hodnotám předpokládaným navrženou teorií. Na základě analýzy teoretických a experimentálních poznatků autor dospěl k závěru, že rovnice odvozené v příspěvku mohou být použity při navrhování rezonančně vibračních čerpadel pro požadovaný průtok, tlak a účinnost čerpání vody ze stanovené hloubky.

Klíčová slova: rezonančně vibrační čerpadla; třasadlo se zdvojenou hřídelí; pružinový závěsný systém; patní ventil; oscilační čerpadlo; vodní sloupec

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