Chapter 8
Development of a Solar-Powered Treadle Pump

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1 Introduction

Agriculture being demographically the broadest economic sector that contributes significantly to the socio-economic fabric of India, provision of reliable irrigation at low-cost for our marginal farmers who held 67% of arable land, is most sought after field of research by many scientific and nongovernmental agencies. One such aspect is the introduction of the feet-operated treadle pump that utilizes groundwater for irrigation, and is being used in many developing countries across the world with proper design modifications to suit the local requirements.

As shown in Fig. 1, the treadle pump is a manually operated two-cylinder positive displacement reciprocating pump, activated by stepping up and down on paddles which are basically levers driving pistons. It is an improved version of the hand-pump, so as to use comparatively stronger foot muscles to operate the pump for longer periods of time. Basically, it is a rural technology with zero operating costs designed for lifting water from a depth of seven metres or less, mainly for irrigation purposes. To ensure proper ergonomic efficiency, the discharge requirements must be met by cadence being not more than 60 cycles/min and the suction pressure to be generated for lifting water must not require a foot force more than 50% of the body weight for sustaining long periods of operation. Also, the stroke of the foot must be limited within 100−350 mm to prevent overstraining of leg muscles. It has

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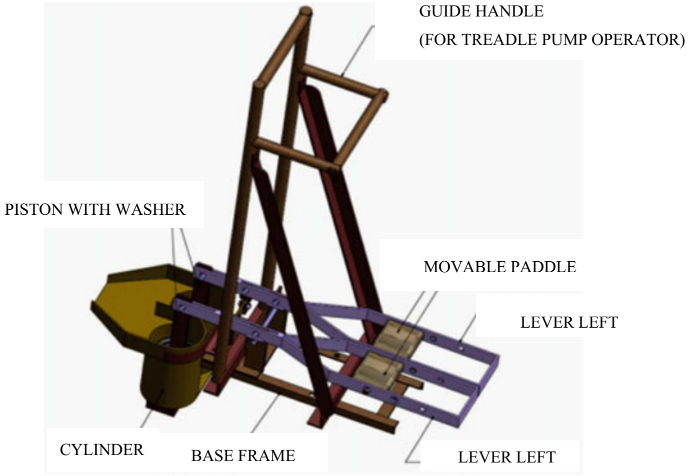


Fig. 1 Manually operated treadle pump [2]

been observed that a healthy adult can produce about 75 W for longer periods, which enables to lift 2–3 m^3/h of water from a depth of 3–5 m at 50% efficiency [1].

This paper focuses on a new design of treadle pump that will utilize solar energy to meet irrigation needs of a tropical country like India.

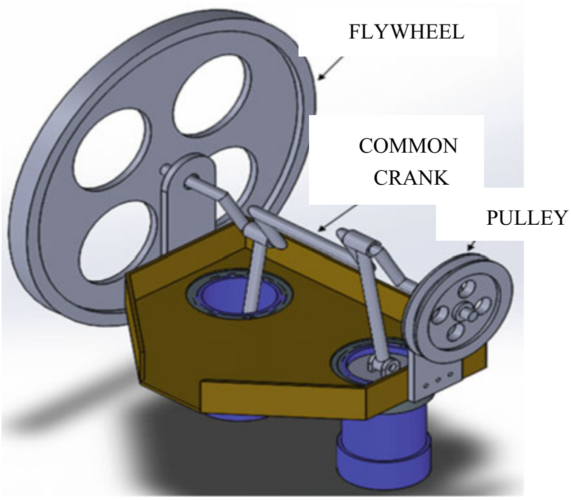
2 Design of a Solar-Powered Treadle Pump

Treadle pump powered by solar energy has been designed with minimum modification to the existing design of a feet-operated treadle pump. For this purpose, the current lever mechanism has been replaced by two slider crank mechanisms coupled by a common crank such that its pistons operate 180° out of phase. The common crank is to be driven by a motor at one end. A flywheel is attached to its other end to maintain a constant angular velocity throughout the operation. This is depicted in Fig. 2.

2.1 Reciprocating Pump

From ground water table data of India [3], it has been found that about half of our country has ground water level in the range of 2–5 m under the ground. For the solar-operated treadle pump to work satisfactorily, it is designed as a reciprocating

Fig. 2 CAD model of the proposed solar-powered treadle pump



pump which will be able to lift water from about 5 m depth and deliver it to the ground level at atmospheric pressure.

Since the system is designed as a reciprocating pump, the calculated safe operating speed such that the pressure at the beginning and middle of the suction stroke does not fall below the vapour pressure of water to avoid cavitation, should be less than 34 rpm. Hence the operating speed, N was taken as 30 rpm, i.e.

$$\mathrm{N} = 30\ \mathrm{rpm} \tag{1}$$

$$\text{Discharge, } \mathrm{Q} = \frac{\pi}{4} 2\mathrm{D}^2 \mathrm{lN}/60 = 99 \times 10^4\ \mathrm{mm}^3/\mathrm{s} = 3.57\,\mathrm{m}^3/\mathrm{h} \tag{2}$$

The relevant data are given in Table 1.

$$\text{Head of the pump, } \mathrm{H} = \mathrm{H_s} + \mathrm{H_d} + \left(\frac{2}{3}\mathrm{h_{fs}}\right) = 5 + 0 + \frac{2}{3}0.41 = 5.27\,\mathrm{m} \tag{3}$$

$$\text{Power output of a pump, } \mathrm{P} = \gamma \mathrm{QH} = 51.34\,\mathrm{W} \tag{4}$$

$$\text{Mean torque to crank at the required speed, } \mathrm{T} = \mathrm{P}/\left(\frac{2\pi \mathrm{N}}{60}\right) = 16.34\,\mathrm{Nm} \tag{5}$$

This torque value is used primarily to design the crank and connecting rod for an initial CAD model. The actual torque required to drive the crank under operating conditions will be determined from inverse dynamics of the system. Hence, the dynamic modelling of the system at hand has been described in details in the next section.

Table 1 Different parameters of the reciprocating pump

S. no.	Parameter	Symbol	Dimension/Value
1.	Length of suction pipe	L_s	5.2 m
2.	Suction head	H_s	5 m
3.	Delivery head	H_d	0 m
4.	Head loss due to friction	h_{fs}	0.41 m
2.	Diameter of the cylinder	D	88.9 mm
3.	Diameter of suction pipe	d	40 mm
4.	Stroke length	l	160 mm
5.	Co-efficient of friction in pipe	f	0.085
6.	Temperature of water	t	28−32 °C

2.2 Dynamic Analysis

Solar-operated treadle pump has been designed as two throw out-of-phase reciprocating pump with common crank. The system is a coupled slider-crank mechanism, whose piston operates 180° out of phase, i.e., active stroke of one piston corresponds to idle stroke of the other, as shown in Fig. 3. Active stroke comprises of suction of ground water through inlet valve and delivery of water above the closed piston valve when the piston moves upward. The idle stroke is when the piston moves down with the inlet valve closed and the water in the cylinder goes to upper part of piston through the piston valve. Hence, when the piston moves up it encounters suction force (F_s) and delivery force (F_d). While moving down it experiences a much lesser amount of force for pushing the piston down into the cylinder (F_i) as the water comes above it through piston valve. Table 2 shows the link parameters obtained from the CAD model.

Maximum force on piston during suction and delivery stroke is given by,

$$F_a = F_s + F_d = \gamma H_s \pi \frac{D^2}{4} + m_{water}(g + a) \approx 320N \tag{6}$$

Maximum force on piston during idle stroke is as follows,

$$F_i = m_{water}(g + a) \approx 10N \tag{7}$$

where, ‘a’ denotes the maximum acceleration of the piston and the mass of water inside the cylinder during the pump operation. Note,

Fig. 3 Two-throw out of phase reciprocating pump with common crank

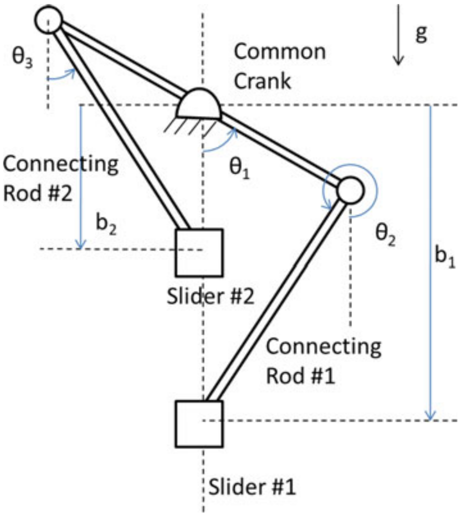


Table 2 Link properties of the two-throw reciprocating pump with common crank

Sl. no.	Link Parameter	Symbol	Dimension/Value
1.	Crank length	l_1	80 mm
2.	Centre of mass of connecting rod from revolute joint between it and the crank	r_2	134 mm
3.	Length of the connecting rod	l_2	250 mm
4.	Moment of inertia of crank about its axis	I_1	2.267×10^{-3} kgm^2
5.	Moment of inertia of connecting rod perpendicular to its axis, passing through its centre of mass	I_2	3.42×10^{-3} kgm^2
6.	Young's modulus of connecting rod	E	210 GPa
7.	Mass of connecting rod	m_2	0.48 kg
8.	Mass of piston	m_3	0.4 kg

$$a = (2\pi N/60)^2 \times l_1 \tag{8}$$

For the dynamic modelling of the mechanism, the links were assumed to be sufficiently rigid during the operation. Applying Euler−Lagrange's formulation [4–6], the equations of motion of the solar-powered treadle pump are derived as:

$$\mathbf{M}\ddot{\mathbf{q}} + \mathbf{h} = \boldsymbol{\tau} + \mathbf{J}^{\mathbf{T}}\boldsymbol{\lambda} \tag{9}$$

where,

the generalized mass matrix, denoted by **M** is given by,

$$\mathrm{M} \equiv \begin{bmatrix} \mathrm{I}_1 + 2\mathrm{m}_2\mathrm{l}_1^2 & \mathrm{m}_2\mathrm{l}_1\mathrm{r}_2\mathrm{c}_{1-2} & \mathrm{m}_2\mathrm{l}_1\mathrm{r}_2\mathrm{c}_{1-3} & 0 & 0 \\ \mathrm{m}_2\mathrm{l}_1\mathrm{r}_2\mathrm{c}_{1-2} & \mathrm{I}_2 + \mathrm{m}_2\mathrm{r}_2^2 & 0 & 0 & 0 \\ \mathrm{m}_2\mathrm{l}_1\mathrm{r}_2\mathrm{c}_{1-3} & 0 & \mathrm{I}_2 + \mathrm{m}_2\mathrm{r}_2^2 & 0 & 0 \\ 0 & 0 & 0 & \mathrm{m}_3 & 0 \\ 0 & 0 & 0 & 0 & \mathrm{m}_3 \end{bmatrix} \tag{10}$$

where $c_{1\text{-}2} \equiv \cos(\theta_1 - \theta_2)$, $c_{1\text{-}3} \equiv \cos(\theta_1 - \theta_3)$ and the vector of generalized coordinates q, is given by

$$\mathbf{q} = \begin{bmatrix} \theta_1 & \theta_2 & \theta_3 & \mathrm{b}_1 & \mathrm{b}_2 \end{bmatrix}^{\mathrm{T}} \tag{11}$$

Since the degree of freedom of the system is 1 and the generalized coordinates are 5, we have 4 constraint equations namely,

$$\mathrm{l}_1 \cos\theta_1 + \mathrm{l}_2 \cos\theta_2 = \mathrm{b}_1 \tag{12}$$

$$\mathrm{l}_1 \cos(\pi + \theta_1) + \mathrm{l}_2 \cos\theta_3 = \mathrm{b}_2 \tag{13}$$

$$\mathrm{l}_1 \sin\theta_1 + \mathrm{l}_2 \sin\theta_2 = 0 \tag{14}$$

$$\mathrm{l}_1 \sin(\pi + \theta_1) + \mathrm{l}_2 \sin\theta_3 = 0 \tag{15}$$

Differentiating Eqs. 12–15, we get the Jacobian matrix, **J** of Eq. 9, as

$$\mathbf{J} \equiv \begin{bmatrix} \mathrm{l}_1 \sin\theta_1 & \mathrm{l}_2 \sin\theta_2 & 0 & 1 & 0 \\ -\mathrm{l}_1 \sin\theta_1 & 0 & \mathrm{l}_2 \sin\theta_3 & 0 & 1 \\ \mathrm{l}_1 \cos\theta_1 & \mathrm{l}_2 \cos\theta_2 & 0 & 0 & 0 \\ -\mathrm{l}_1 \cos\theta_1 & 0 & \mathrm{l}_2 \cos\theta_3 & 0 & 0 \end{bmatrix} \tag{16}$$

The other vectors of the equation are the vector of convective inertia terms, denoted by **h** and the vector of external forces, denoted by **τ**. Their expressions are as follows:

$$\mathbf{h} \equiv \begin{bmatrix} \mathrm{m}_2\mathrm{l}_1\mathrm{r}_2\mathrm{s}_{1-2}\dot{\theta}_2^2 + \mathrm{m}_2\mathrm{l}_1\mathrm{r}_2\mathrm{s}_{1-3}\dot{\theta}_3^2 \\ -\mathrm{m}_2\mathrm{l}_1\mathrm{r}_2 s_{1-2}\dot{\theta}_1^2 \\ -\mathrm{m}_2\mathrm{l}_1\mathrm{r}_2\mathrm{s}_{1-3}\dot{\theta}_1^2 \\ 0 \\ 0 \end{bmatrix} \tag{17}$$

where $s_{1\text{-}2} \equiv \sin(\theta_1 - \theta_2)$ and $s_{1\text{-}3} \equiv \sin(\theta_1 - \theta_3)$

$$\boldsymbol{\tau} \equiv \begin{bmatrix} \mathrm{T} & \mathrm{m}_2\mathrm{g}\mathrm{r}_2 \sin\theta_2 & \mathrm{m}_2\mathrm{g}\mathrm{r}_2 \sin\theta_3 & \mathrm{m}_3\mathrm{g} + \mathrm{f}_1 & \mathrm{m}_3\mathrm{g} + \mathrm{f}_2 \end{bmatrix}^{T} \tag{18}$$

where ‘T’ is the driving torque and ‘f_1’ and ‘f_2’ are the hydraulic forces acting on piston 1 and 2 respectively.

The Lagrange multipliers (denoted by $\boldsymbol{\lambda}$) corresponding to the constraint equations account for the constraint forces between the crank and connecting rod, and the piston and the connecting rod. The Coulomb’s friction between the leather washer on the piston and the cylinder, depends on the force normal to the direction of motion between them. Therefore, the frictional forces can be taken into account by including coefficient of friction within the Jacobian matrix.

The maximum torque value (Fig. 6) and the reaction forces at the various joints were used to design the crank and connecting rod for sufficient strength.

3 Fabrication and Assembly

All the parts of the treadle pump are generally made of plain carbon steel. For safe design, the factor of safety was taken as 3.5. The treadle pump was designed to run for six hours daily. Also being a rurally assembled machine, it may have loose fits, clearances in its parts, which result in mild shocks while in operation. To maintain comparable discharge at par with the manually operated treadle pump the length of the crank was calculated as 80 mm. The theoretical discharge as calculated in Eq. 2 is 3.57 m^3/h. The maximum torque on the crank obtained from the dynamic model of Sect. 2.2 is 32 Nm (Fig. 6). Accordingly, the crank and connecting rod diameters were calculated using the knowledge of Machine Element Design [7] as 16 mm. The length of the connecting rod should be at least 2 times the length of the crank to prevent higher reaction forces between the cylinder and the piston. Thus, its length was taken as 200 mm, which is 2.5 times of that of the crank.

The pistons and washers used were that of hand-pumps because of their ready availability in a local market. Since the leather washers experience higher wear and tear, they should be easily replaceable. Hence, the size of cylinders in treadle pump was also standardized according to the locally available hand-pump parts, which is 88 mm (inner diameter). The mass moment of inertia of the flywheel required to maintain the required speed for a coefficient of fluctuation of speed by 0.2 was calculated as 6.52 kgm^2. Proper supports were necessary at both sides which housed the bearings and supported the crank, pulley and flywheel. The components were manufactured and assembled as shown in Fig. 4 (Table 3).

The DC geared motor used to power the crank has output speed of 90 rpm. Its speed was reduced to the operating speed of 30 rpm by a V-belt pulley drive with speed ratio of 1:3. The current drawn by the motor during the testing were between 3 and 8 A, where its efficiency ranged from 0.70 to 0.78 as per the motor specifications sheet. Since the motor input voltage was 24 V, we connect two solar panels in series. Most PV panels have temperature coefficient for maximum output power of about $-0.5\%/^{\circ}C$. Since the operating temperature was 40 °C, the maximum power output of each panel was estimated as 115 W.

Fig. 4 Assembled solar-powered treadle pump

Table 3 Specifications of DC motor and PV panel

Sl. no.	Component	Specifications	Value
1.	Geared DC motor	Rated power	100 W
		Rated speed	3600 rpm
		Gear ratio	40:1
		Input voltage	24 V
2.	Photo-Voltaic panel (All specifications at STC: Insol. 1000 W/m^2, T 25 °C)	Maximum power	125 W
		Rated voltage	17.2 V
		Rated current	7.30 A

4 Performance Testing

Since the testing was performed in a laboratory without the provision of a deep well from which the ground water can be lifted using the pump, the setup was designed to measure the performance parameters of the pump shown in Fig. 5.

It is advisable to use a 24 V cell in between the motor and solar panel to maintain a steady voltage supply. Otherwise, the voltage fluctuates. The voltage and current drawn by the motor gave us the input power. The actual discharge of the pump was measured. Also, the net suction head, which included the suction pipe friction loss and minor loss due to bends and valves in suction manifold was calculated. This gave us the output power of the pump as $P_{out} = \gamma QH$ [8]. The theoretical discharge of the pump was also calculated from the speed to find out the volumetric efficiency.

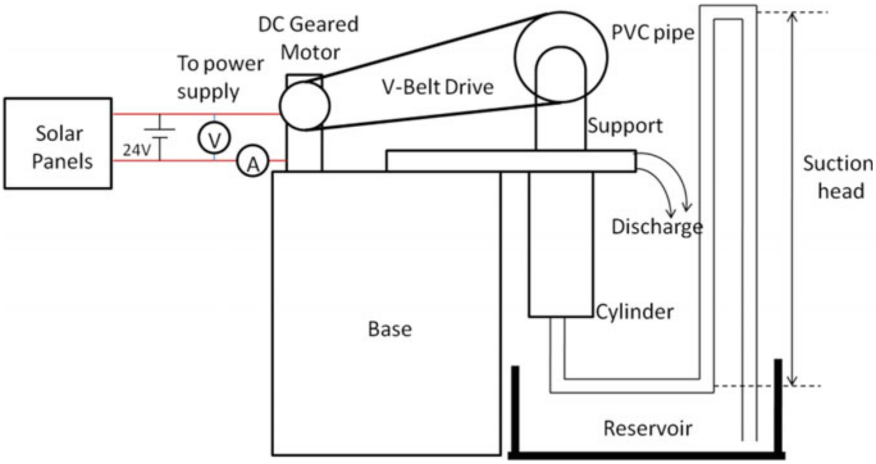


Fig. 5 Experimental setup

The performance testing shows that the power drawn by the motor to lift water at a rate of 2.9 m^3/h from a suction depth of 3 m is found to be 107 W. When the suction depth increases to 5 m, the discharge decreases to 2.5 m^3/h and the power consumption increases to 180 W. The mechanical efficiency which accounts for the losses in the belt drive, frictional loss between the piston and cylinder due to the compliant leather washer is about 48%, whereas the volumetric efficiency is 65%.

5 Results and Discussion

The torque required to drive the pump at 30 rpm for two complete revolutions of the crank was determined from inverse dynamics. It is depicted in Fig. 6, where the red line denotes the driving torque without friction and the blue one denotes driving torque with Coulomb's friction. The frictional coefficient between the leather washer and plain carbon steel cylinder under wet conditions was taken as 0.4. The mean torque without considering friction was obtained as 16.8 Nm which is close to that calculated in Eq. 5. The mean torque considering friction was 21.3 Nm. This information is crucial for selecting the size of the motor needed to drive the pump. Also, this torque profile was used to calculate the dimensions of the flywheel. The flywheel is quite heavy as the torque fluctuates from 0 to 32 Nm. If the flywheel is not properly designed the mechanism will lock itself when the pistons reach their dead centres simultaneously.

Note that the hydraulic power output increases with the increase in net suction head because the power output is directly proportional to the head ($P_{out} = \gamma QH$). But after sometime the power output decreases because the discharge decreases due to slip. This behaviour of the reciprocating positive displacement pump is seen in

Fig. 6 Torque required to drive the pump at 30 rpm

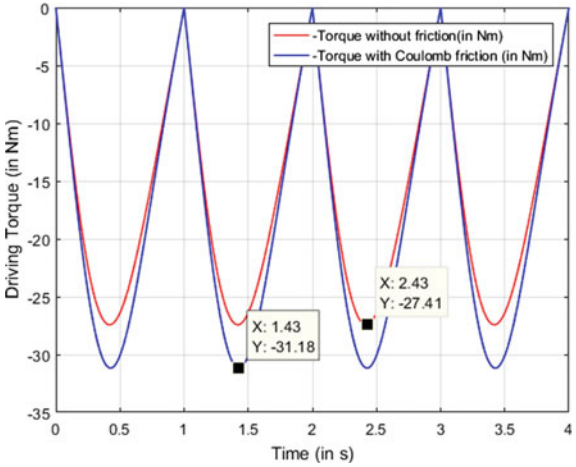


Fig. 7 Net suction head versus hydraulic power output

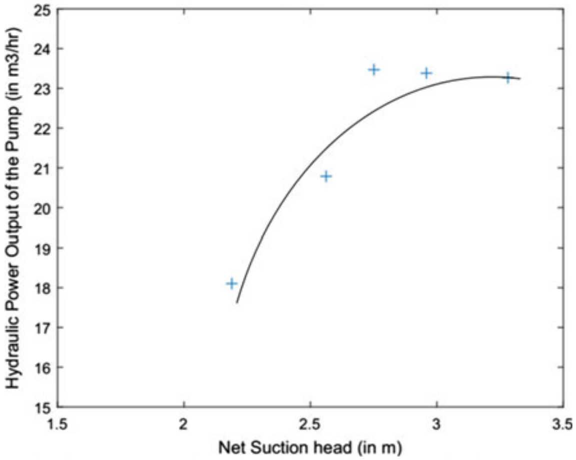
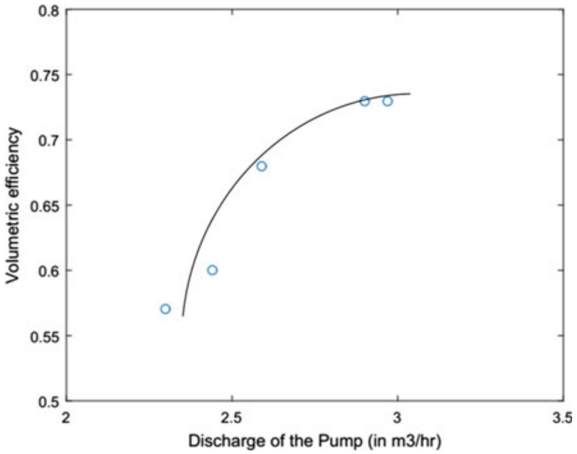


Fig. 7, whereas the volumetric efficiency is seen to be increasing with the discharge of the pump Fig. 8. The discharge increases when the speed of crank increases, which imparts an increased acceleration to the mass of water. Hence, the water is delivered with increased inertia, causing the volumetric efficiency to increase.

Fig. 8 Discharge vsersu volumetric efficiency



6 Conclusions

6.1 Summary

As a conclusion, a treadle pump powered with solar energy will eliminate the drudgery in operation of a conventional treadle pump. The power consumption of the solar-operated treadle pump shows that it can be powered by a maximum of 2 solar panels with capacity of 125 W for suction depth of 3 m and discharge of 3 m^3/h. However, the installation of the solar panels and battery will add cost of about INR 20,000, making the total cost to be about INR 30,000. Considering the fact that the manual operation can be done not for more than 40 min owing to the fatigue of leg muscles, the solar operated one can be a good alternative. This is not only environment friendly but could be used for 4−6 h daily or more depending on the hours of peak sunshine. Therefore, the cost of installation can be shared by 4−5 marginal farmers to irrigate their lands with a nominal maintenance cost. The commercially available systems require diesel as fuel to operate, which not only incurs significant operating costs but also causes water and soil pollution due to leakage of oil.

6.2 Future Work

From the dynamic analysis, we found that the torque required to drive the crank against the hydraulic forces encountered during operation becomes maximum when one of the piston reaches its top dead centre and another its bottom dead centre simultaneously, since they are 180° out-of-phase. If they are designed to be at any other angle, the mass centre of the common crank shifts from the centre and the resulting eccentricity causes objectionable vibrations in the system. Hence, the solar-operated treadle pump can be designed as three-throw reciprocating pump with common crank and the pistons located at 120° out of phase with each other so that they do not reach their respective dead positions at the same time. Dynamic analysis of such system shows lesser driving torque fluctuations so that the need of a heavier flywheel is obviated.

Also, a manual mode of operation can be provided in the solar-powered treadle pump to operate it during lean periods of sunlight. This can be achieved by incorporating a chain and sprocket drive on the common crank to be rotated by feet through pedals after disengaging the V-belt drive from the DC motor. A complete design of such system with easy transition from one mode to another can be taken up as future work.

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