8th WEDC Conference: Water and waste-engineering in Asia: MADRAS: 1982

APPLICATION OF VECTOR MECHANICS IN A MANUALLY OPERATED TURBO-PUMP

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INTRODUCTION

In view of present energy crisis that has gripped the entire industry and agriculture, this country has been forced to hunt for the alternate sources of energy such as solar, wind-power, tidal, bio-gas, geo-thermal, nuclear, water power, waste heat, plant fuel and so forth. These resources, some of these being available naturally in abundance, can be converted into useful sources of energy, if appropriate technology based on sound engineering principles and modified to suit local conditions, can be developed. Yet another resource overflowing and crying to be utilized in this country is, of course, the manpower. In this write-up, an attempt has been made to use the manpower by applying high technology in devising precise mechanisms or mechanical systems, that would otherwise have required the already depleted fuels like petroleum products or electricity.

The objective of this research problem, therefore, was to develop a machine operated by manpower to lift water for drinking or for irrigation at a head under three metres and a discharge between 15-20 cubic metres per hour. At the same time, the machine had to be portable and cheap, so that it could be used by a small farmer, who could not afford an internal combustion engine or electric motor driven pump. For this purpose, a turbo-pump (an axial flow pump or a centrifugal pump) with mechanisms to transmit power efficiently from two men to the turbo-pump, has been developed and fabricated as shown in Fig. 1, 2 and 3.

FEATURES

The main features of manually operated turbo-pump are :

1. A speed increasing gear-drive.
2. A 4-bar linkage.
3. A turbo-pump (centrifugal).

Fig. 1. Manually operated turbo-pump (portable).

To obtain a high speed required to run the turbo-pump, a speed increasing gear train has been designed. Although a worm gear drive would have been preferable to keep the number of gears at a minimum, the fabricated machine uses six spur gears giving an over-all gear-ratio of 1 to 120. The gears have 120-20, 80-20 and 110-22 teeth respectively and all having a diametral pitch of 3.78 per cm. The gear shafts are placed in five brackets and run in ball bearings. The maximum final speed at the driven shaft was approximately 2400 r.p.m.

To transmit the manual power, which is in the form of to and fro motion, a 4-bar linkage, called "crank and rocker mechanism" was provided. The linkage has two handles for two men to operate. The lengths of the linkages, crank or driver, follower or rocker, connecting rod and the fixed link or frame were computed to be 10

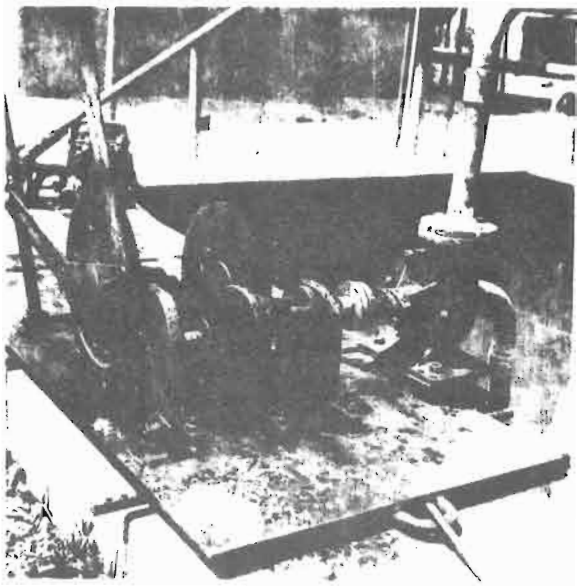

Fig. 2. Manually operated turbo-pump (close-up).

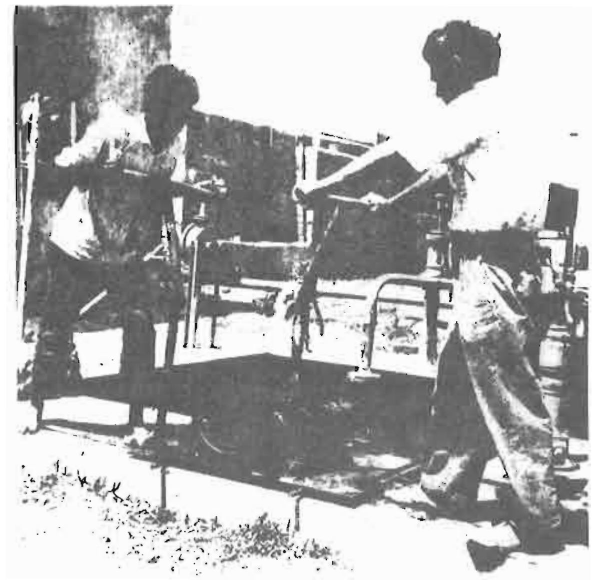

Fig. 3. Manually operated turbo-pump (in operation).

cm., 20 cm., 38 cm. and 46 cm. respectively. The follower link is extended to form a handle, one metre long. The driving gear is connected by a pin to the crank of the linkage. With this arrangement, two men could easily operate the first driving gear at a speed of 10-20 r.p.m., which gives an impeller speed of 1200 to 2400 r.p.m.

MATHEMATICAL MODEL OF FLUID MOTION

To suit the local operating conditions, the basic mathematical model applicable in design of the turbo-pump was modified as shown in the following discussion. From the field of vector mechanics, the angular momentum equation (1) was used with reference to Fig. 4 to compute the net torque required :

$$\int_{c.s.} \vec{r} \times d\vec{F}_s + \int_{c.v.} \vec{r} \times \vec{B}\, d\forall = \frac{\partial}{\partial t}\int_{c.v.} \vec{r} \times \vec{V} \rho\, d\forall + \int_{c.s.} \vec{r} \times \vec{V} \rho \vec{V} \cdot d\vec{A} \quad \text{---(1)}$$

where, $\vec{r}$ = position vector from the reference origin to the point where the force is acting,

$\vec{F}_s$ = surface force vector,
$\vec{B}$ = body force vector,
$\forall$ = volume,
ρ = mass density
t = time,
c.s. = control surface,
$\vec{V}$ = velocity vector,
$\vec{A}$ = surface area vector,
c.$\forall$. = control volume fixed in space and bounded by c.s.

In equation (1), the integrand of the first term ($\vec{r} \times d\vec{F}_s$) gives the moment around the origin attributed to the force $d\vec{F}_s$ at the control surface. The integrand of the second term is the moment about the reference point / origin due to the body force acting on the infinitesimal volume element $d\forall = dm/\rho$ (m = mass of the matter). The integrand of the third term is angular momentum of the infinitesimal mass element $\rho\, d\forall$. The integration gives the total angular momentum of the mass within the control volume. The last term is the rate of efflux of angular

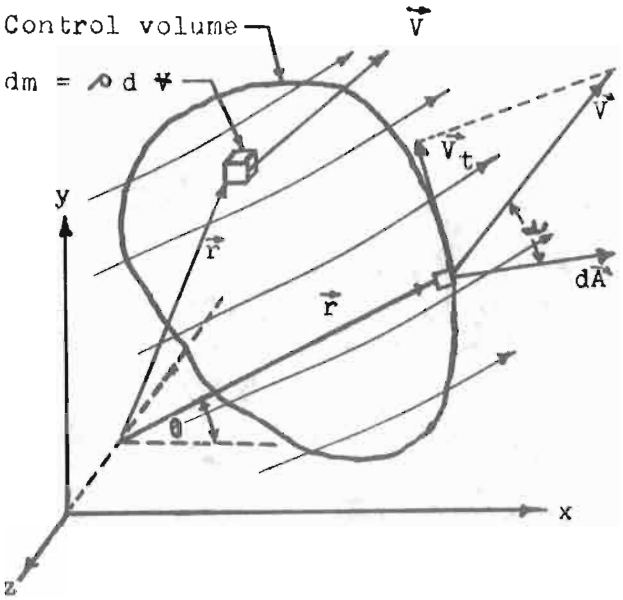

Fig. 4. Angular momentum of the mass within the control volume about z-axis.

momentum through the control surface.

In the present problem, however, the flow is assumed to be steady, the body forces negligible and since the control volume is fixed, the second and third terms of Eq. (1) vanish. Taking z-axis about which rotation of flow takes place, we have

$$\vec{T}_z = \int_{C.S.} \vec{r} \times d\,\vec{F}_s$$

$$= \int_{C.S.} (\vec{r} \times \vec{V})_z\ (\rho\,\vec{V}.\,d\,\vec{A})$$

where $\vec{T}_z$ represents the net torque acting counter-clockwise on the control volume about the z-axis and

$$(\vec{r} \times \vec{V})_z = r\,V_t\ ,$$

$$\rho\,\vec{V}.\,d\,A = \rho\,V\cos\alpha\ d\,A$$

where V_t is the component of the velocity vector and perpendicular to the z-axis and α is the angle between the velocity vector and the area $d\,\vec{A}$. Then

$$T_z = \int_{C.S.} r\,V_t\,V\cos\alpha\ d\,A \quad \text{--- (2)}$$

Let us assumet that the entire flow enters the control volume at an area A_1 and leaves at an area A_2 over each of which ρ, V and $\cos\alpha$ are uniform over the entrance section 1 and exit section 2. Also define $(r_2\,V_{t_2})$ as the mean value of $r\,V_t$ over A_2 and $(r_1\,V_{t_1})$ as the mean value over A_1, then

$$r_2\,V_{t_2} = (1/A_2)\int_{A_2} r\,V_t\ d\,A\ ,$$

$$r_1\,V_{t_1} = (1/A_1)\int_{A_1} r\,V_t\ d\,A$$

The continuity equation gives,

$$\rho_1\,A_1\,V_1\cos\alpha_1 = \rho_2\,A_2\,V_2\cos\alpha_2$$

$$= \rho_1\,Q_1$$

Here, Q_1 is the volumetric flow rate at A_1. Eq. (2) now becomes

$$T_z = \rho_1\,Q_1\,(r_2\,V_{t2} - r_1\,V_{t_1}) \quad \text{--- (3)}$$

Fig. (5) shows the velocity diagram for a radial flow impeller of the turbo - pump, in which $\vec{v}$ refers to the impeller (relative velocity) and $\vec{V}$ denotes as usual the absolute velocity relative to earth.

In the present problem (Head = 2 m., Q = 12 m.^{3}/hr. = 0.1177 cu. ft./ sec., N = 1800 r.p.m., Effy. = 57.6 %, T = 6.14 kg. cm. = 0.443 ft.-lb$_f$. and B.H.P. = 0.152 H.P.), since the flow at the impeller inlet is in the axial direction, the tangential component of velocity there, $V_{t1} = 0$. Hence, Eq. (3) is further reduced to

$$T_z = \rho_1\,Q_1\,(r_2\,V_{t_2}) \quad \text{--- (4)}$$

Substituting the values of this problem into Eq. (4), where $\omega = (2\pi \times 1800/60)$ radians/sec. = 188.4 rad./sec.

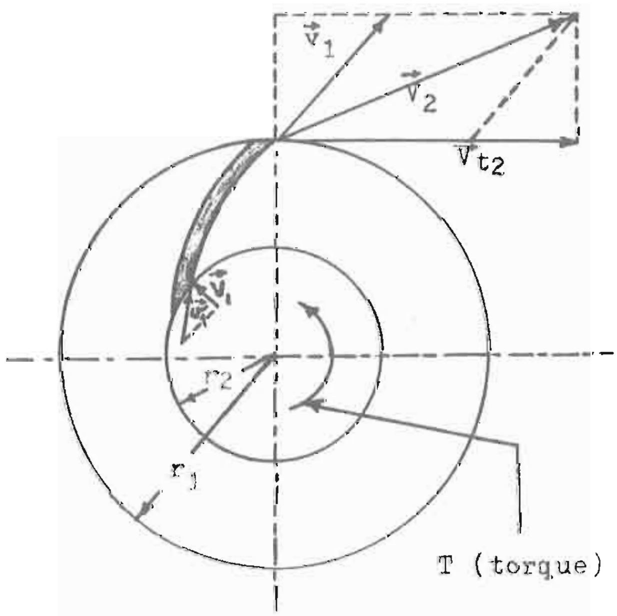

Fig. 5. Velocity diagram for a radial flow impeller of the turbo-pump.

$$0.443 = \frac{62.4}{32.2} \times 0.1177 \times \frac{r_2}{12} \times 188.4 \times (r_2 / 12)$$

$$\text{or} \quad \frac{0.443 \times 32.2 \times 12 \times 12}{62.4 \times 0.1177 \times 188.4} = r_2^2$$

$$\text{or} \quad r_2^2 = 1.488, \quad r_2 = 1.2 \text{ in.} = 3.1\text{cm.}$$

Hence, d_2 = 6.2 cm. and by keeping a ratio of 5:3 between d_2 and d_1, we have $d_1 = 6.2 \times (3/5)^2 = 3.72$ cm. However, due local manufacturing problems, the impeller diameter was doubled and kept to 12.5 cm. The other dimensions computed for the turbo-pump were as follows :

1. Diameter of suction flange = 3.75 cm.
2. Shaft diameter = 1.25 cm.
3. Impeller hub diameter = 2.25 cm.
4. Impeller eye diameter = 7.5 cm.
5. Impeller inlet diameter = 7.5 cm.
6. Vane angle at inlet = 27°
7. Impeller outlet diameter = 12.5 cm.
8. Vane angle at outlet = 35°
9. Passage width at outlet = 0.375 cm.
10. Passage width at inlet = 0.6 cm.
11. Angle of water leaving impeller = 33°.
12. Number of impeller vanes = 12.

The design data of the spiral volute for this impeller were obtained as follows (Table 1) :

Table 1. Volute design data

Reference angle	Volute width	Volute radius
0.0°	2.0 cm.	6.25 cm.
28.4°	2.4 cm.	6.50 cm.
92.6°	3.0 cm.	7.00 cm.
247.6°	4.0 cm.	8.00 cm.
430.6°	5.2 cm.	9.00 cm.

PERFORMANCE

Although the machine developed and fabricated so far is by no means perfect, the accomplished work indicates a potential for commercial development of such a unit for use of small farmers. Approximate discharge against different heads as achieved in field trials are shown in Table 2.

Table 2. Performance of manually operated turbo-pump

Head (metres)	Discharge (cu. m./hr.)
1.00	9.5
1.50	7.5
1.75	3.0
2.00	1.5
2.25	0.7

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