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Development and Application of Geneva Mechanism for Bottle Washing.

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Abstract :*Manual washing of beverage bottles does not give the desired productivity requirement of industrial setting and in the effort to reduce the environmental impact of waste from industrial production, there is an increasing deeply felt need to recover empty glass and plastic containers. This paper therefore aims at developing a Rig (Geneva Mechanism) for bottle washing in a typical brewery or beverage industry. A test rig was designed, fabricated and employed for a performance evaluation. The rig operates on the intermittent rotary motion from a four slot external Geneva Mechanism and requires manual loading and unloading of bottles. The bottles are loaded on subsequent indexing part of the rotating table and are washed one after another. The analysis of design gave the following results: Centrifugal force on the driven pulley (F_R) = 0.158N; Bearing reaction at an end, B, R_B = 403.42N; Bearing reaction at an end, C, R_C = -152.42N; Radial load due to inertia of driver, F_R = 20.90N; Axial load due to weight of Pulley, W_a = 61.70N; Equivalent dynamic load on the bearing, W_e = 349.31N; Bearing load capacity, W_C = 2306.80N. These forces were related to generate shear force and bending moment diagrams. This work presents a practical application of Geneva mechanism for worktable indexing and bottle washing.*

Keywords: *Geneva Mechanism, Rotary Motion, Bottle Washing, Productivity, Indexing Time.*

I. Introduction

Geneva mechanism is a simple and widely used timing mechanism that provides intermittent motion from a continuously rotating input. It consists of a rotating drive wheel (Driver) with a pin that reaches into a slot of the driven wheel (Geneva wheel) advancing it by one step.

They are cheaper than cams, have good motion curve characteristics compared to ratchets and maintain good control of its load at all times. In addition, if properly sized to the load, the mechanism generally exhibits very long life. It is used in machine tools to index spindle carriers weighing several tonnes, in transfer machines for indexing work piece from one work station to another, as a turret indexing mechanism in automatic lathe, in counting instruments, peristaltic pump drives in integrated circuit manufacturing, intermittent advance of films in motion-picture projectors and discrete motion drives with high load capacity in robotic manipulators.

One of the most important processes in beverage production is bottle washing. The high quality of the product depends largely on how thoroughly the bottles are cleaned immediately before filling.

Manual washing of bottles does not give the desired productivity requirement of industrial setting. On this note, a mechanized system of washing and detoxification is very imperative in order to achieve the desired productivity for industries.

The need to address this problem has led to the design of a mechanized bottle washing and detoxification system using a rotary table propelled by a Geneva mechanism [1]

Krishnakumar et al [2], designed and developed a Geneva mechanism for Film Frame. In their design, a 4-slot mechanism was used in which a driver, (A), rotates at uniform angular velocity and for every revolution of the driver, the Geneva wheel makes a fraction part of the revolution which is a function of the number of slots (fig.1).

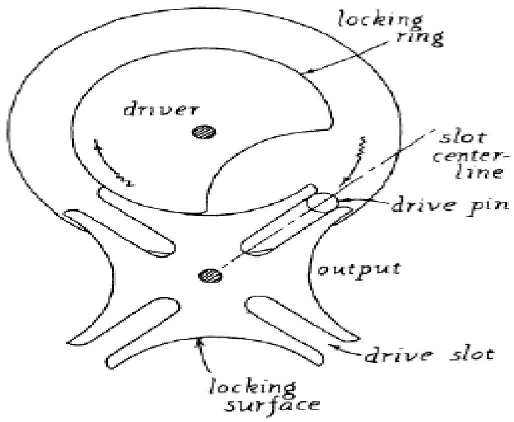


Fig. 1 4-slot Geneva Mechanism

Indexing motion mechanisms find several applications in instruments, watches, projectors, machine tools, printing and pressing machinery, packaging and automatic machinery, transfer lines [3]. A wide variety of applications are derived from the Geneva mechanism such as indexing in automatic machinery, peristaltic pump drives in integrated circuit manufacturing, intermittent advance of films in motion-picture projectors and discrete motion drives with high load capacity in robotic manipulators [4-7]. Geneva mechanism is also used in rotary indexing machine to provide synchronous motion and index a worktable to present parts to workstations for processing [8].

The main disadvantage of Geneva mechanism is the discontinuity in the acceleration at the start and the end of the intermittent motion. At these points, the normal acceleration of the rotating crankpin is transmitted to the wheel with an impact, leading to large jerks and undesirable vibrations in the mechanism [9]. Two undesirable characteristics inherent in this mechanism are (1) high contact stress between the drive pin and the wheel slot and (2) vibratory motion of the driven inertia load. Both of these factors adversely affect performance and life of the mechanism [10]

Several methods have been proposed to decrease the wheel acceleration in order to reduce the inertia force and the consequent wear. Johnson [10] designed Geneva mechanism to minimize drive-pin contact stress and gave a procedure for reducing the undesirable vibratory motion. He proposed that there must be an optimum Geneva wheel diameter that minimizes the pin-slot contact stress. Many authors have considered minimizing the inertial forces on the Geneva wheel using a four bar linkage with the drive pin located at a coupler [11-13]. For an internal Geneva wheel, the jerk and undesirable vibrations can be removed by using a four-bar linkage as a driving mechanism instead of a simple crank [14]. He used compound mechanism to eliminate the non-zero accelerations at the initial and ending stage. Fenton [15] also used Geneva mechanism in series such that the output motion starts and ends with zero acceleration. The acceleration increases gradually at the beginning and ends smoothly at the end of the motion cycle. The undesirable shock loadings typical of the simple Geneva mechanisms were completely eliminated. Another proposed method was to employ certain damping elements in the mechanism to reduce the shock [16-17]. Another method was to change the geometry for the wheel slot. Changing the slot shape from a straight radial line to a curved line results in the elimination of shock loading at the beginning and end of the motion cycles and reduces the wheel peak velocity and peak acceleration values, making the mechanism well suited for high speed applications. Curved slots with designed motion law were applied as in the standard cam mechanism. In this approach, the Geneva mechanism with curved slots was basically thought of as an inverse cam mechanism [18-19]. However, basic equations for the surface geometry of the slots were not discussed in their work resulting in unknown conditions for manufacturing, such as undercutting and double point in the curve. Figliolini and Angeles [20] studied the force transmission of the Geneva mechanism with curved slots. Lee and Huang [21] derived the equation for the geometry of curved slots. Lee and Jan [22] designed the profile of the curves for Geneva mechanism with curved slots, derived the mathematical expression in parametric form for the curves and equations for non-undercutting and developed the criteria for undercutting and double points in the design.

II. Design and Development

The test rig is a 4-slot external Geneva Mechanism for bottle washing consisting of the following features:

1. A water pump connected to pipes and pipe fitting for the channeling of water through the nozzle.
2. A solenoid valve connected to a limit- switch for the regulation of water flow during bottle washing operations.

3. A Geneva mechanism mounted on a Geneva wheel shaft for intermittent positioning of bottles at the washing station
4. A control panel housing the electrical circuit components of the test rig.
5. Bottle holders that grip the bottles at the neck.
6. An electric motor that provides the rotational motion transmitted to the shafts.

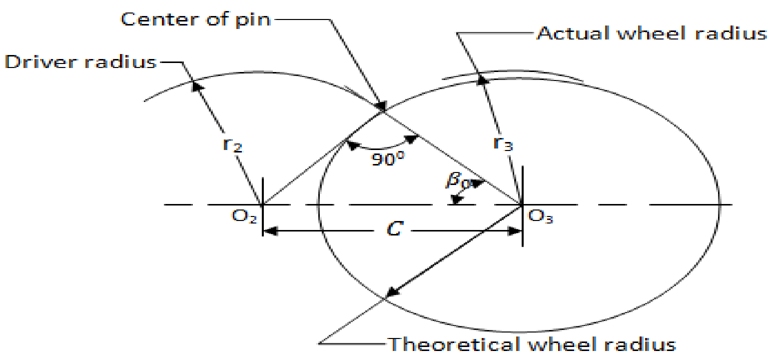
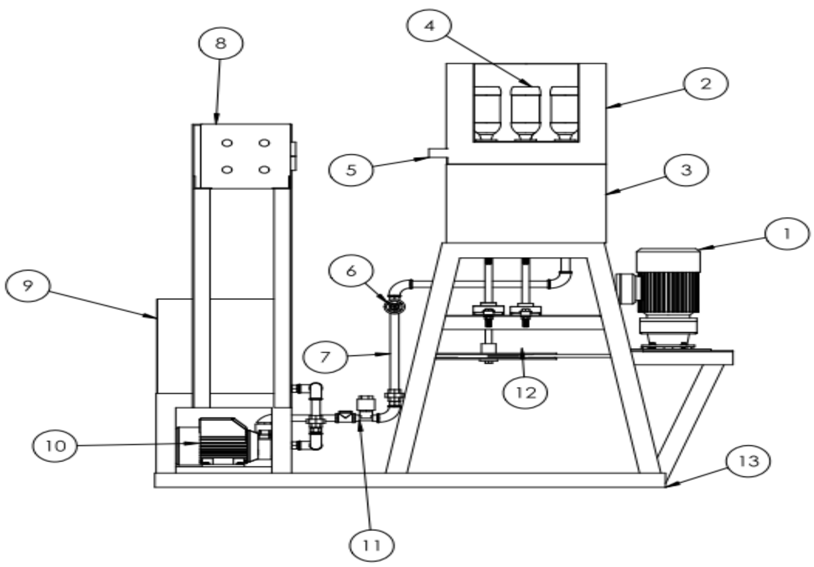


Fig. 2 Design of Geneva Mechanism



S/N	COMPONENT NAME
1	Electric Motor
2	Upper Chamber
3	Lower chamber
4	Bottle
5	Limit Switch
6	Gate Valve
7	Water Pipe
8	Control Panel
9	Water Tank
10	Electric Pump
11	Solenoid Valve
12	Pulley
13	Machine Frame

Fig. 3 Test Rig Assembly

The design is initiated by specifying the Crank (driver) radius, the roller diameter and the number of slots (fig.2), as follows:

The Crank (Driver) radius, r_2 = 60mm

The roller (pin) diameter = 12mm

The number of slots, n = 4 slots

The angle β is half the angle subtended by adjacent slots, that is;

$$\beta_0 = \frac{360^0}{2n} \qquad (1)$$

$$\beta_0 = \frac{360^0}{2 \times 4} = 45^0$$

Where n is the number of slots in the Geneva wheel.

$$Sin\beta_0 = \frac{r_2}{c} \qquad (2)$$

$$C = \frac{r_2}{\text{Sin } \beta_0}$$

$$C = \frac{60}{\text{Sin } 45^0}$$

Where C is the center distance between the driver shaft and Geneva wheel shaft.

Using Pythagoras theorem, $C^2 = r_2^2 + r_3^2$

Where r_3 is the radius of Geneva wheel

$$r_3 = \sqrt{C^2 - r_2^2}$$

$$r_3 = \sqrt{84.85^2 - 60^2}$$

$$r_3 = 59.996mm$$

$$r_3 \sim 60mm$$

2.1 Forces Acting on the Driver Shaft

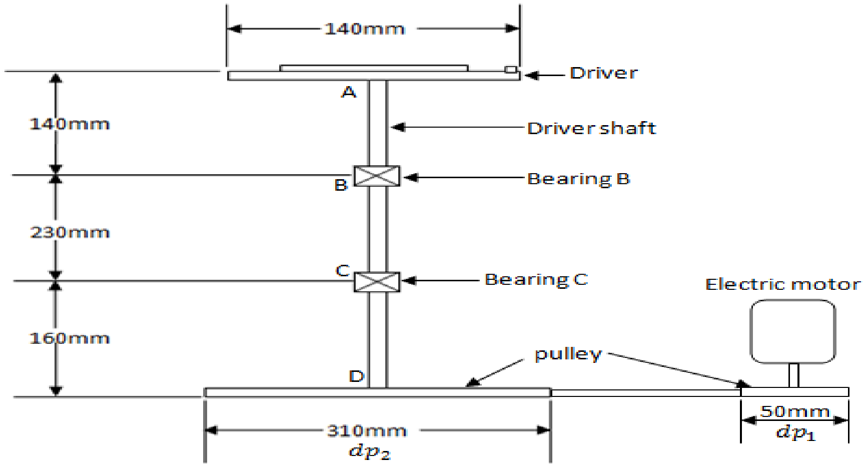


Fig.4 **Schematic Diagram showing the drive mechanism of the machine**

The shaft is r

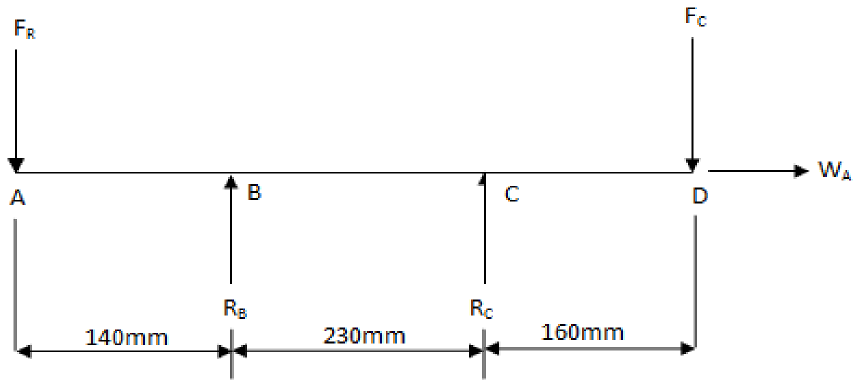


Fig. 5: Force diagram of the driver shaft

Figures 4 and 5 show the drive mechanism and the forces acting on the driver shaft. The forces include the following:

- Centrifugal force F_c of drive belt on driven pulley
- Bearing reaction R_B at point B.
- Bearing reaction R_c at point C
- Radial load F_R at shaft end A due to inertia load of Driver
- Axial Load W_A due to weight of pulley, shaft and Driver.

Computation of these forces enables the presentation of a graphic representation of shaft loading from which shear force and bending moment diagrams were obtained. Reaction forces R_A, R_B and axial load W_A are needed for the specification of bearings while the maximum bending moment on the shaft is an important requirement for shaft diameter computation. Linear dimensions of the shaft in fig.4 and 5 were estimated based on the anticipated length of immersion of the shaft in the chamber lower cover and space requirement for the bearings and driven pulley.

2.2 Length of belt connecting the motor pulley and driven pulley

Length of the belt, L_b is obtained using:[23]

$$L_b = \frac{\pi}{2}(dp_1 + dp_2) + \frac{(dp_1 - dp_2)^2}{4C} + 2C \tag{3}$$

Diameter of motor pulley, dp_2 = 50mm

Diameter of driven pulley, dp_1=117.5mm

Center distance between motor and driven pulley, C = 435mm

Substituting dp_1 , dp_2 and C into the above equation, we have

$$L_b = \frac{\pi}{2}(0.1175 + 0.05) + \frac{(0.1175 - 0.05)^2}{4 \times 0.435} + 2(0.435)$$

$$L_b = 1.1357m$$

2.3 Centrifugal force on the Driven pulley

Centrifugal force F_c acting on the driven pulley attached to the driver shaft is given by: [24]

$$F_c = \frac{M_b V^2}{r}$$

But linear velocity, $V = \omega r$

$$F_c = \frac{M_b \omega^2 r^2}{r}$$

$$F_c = M_b \omega^2 r$$

Where M_b = mass of the belt,

ω = angular velocity of driven pulley,

r = radius of driven pulley

Since Density $= \frac{mass}{volume}$, mass of the belt M_b can be computed using the equation:

$$M_b = \rho_b V_b = \rho_b A_b L_b$$

Where ρ_b = Density of belt material,

V_b = Volume of belt material,

A_b = Cross- sectional area of the belt,

L_b = length of the belt

Mass of the belt, $M_b = \rho_b A_b L_b$

$$M_b = 1000 \times 9 \times 10^{-5} \times 1.1357$$

$$M_b = 0.1022kg$$

For a selected Driver speed, $N_2 = 120rpm$

$$\omega_2 = \frac{2\pi N_2}{60}$$

$$\omega_2 = \frac{2\pi \times 120}{60}$$

ω_2 = 12.566 rad/s.

Electric motor speed, N_1 = 47rpm

$$D_2 = \frac{N_1 D_1}{N_2}$$

$$D_2 = \frac{47 \times 50}{120}$$

$$D_2 \sim 19.58\, m$$

The Centrifugal force, F_c becomes

$$F_c = M_b \omega^2 r$$

$$F_c = 0.1022 \times (12.566)^2 \times 0.0098$$

$$F_c = 0.158N$$

2.4 Radial load F_R at upper shaft end

The Radial load F_R at the shaft end due to the Driver and Geneva wheel can be expressed as:

$$F_R = M_D \omega^2 r_D + M_G \omega^2 r_G \quad (4)$$

Where M_D = mass of driver, M_G = mass of Geneva wheel,

r_D = radius of driven Driver, r_G = radius of Geneva wheel,

ω = angular velocity of driver shaft, Mass of Driver and locking ring (cam), M_D = 1502g, mass of Geneva wheel, M_G = 454g (measured with Tripple Beam balance, capacity: 2610g

$$F_R = M_D\omega^2 r_D + M_G\omega^2 r_G$$

$$F_R = [1.502 \times (12.566)^2 \times 0.07] + [0.454 \times (12.566)^2 \times 0.06]$$

$$F_R = 20.903N$$

Using a factor of safety of 12. This is the designated value for steel under shock load which is similar to the shock load of the Geneva mechanism.

$$F_R = 12 \times 20.903$$

$$F_R = 250.84N$$

2.5 Reaction loads R_B and R_C

Considering the force diagram, for equilibrium of forces,

$$R_B + R_C = F_R + F_c$$

$$R_B + R_C = 250.84 + 0.158$$

$$R_B + R_C = 250.998$$

Taking moment about B,

$$F_R(0.14) + R_C(0.23) = F_c(0.16 + 0.23)$$

$$(250.84 \times 0.14) + R_C(0.23) = 0.158(0.16 + 0.23)$$

$$R_C(0.23) = -35.056$$

$$R_C = -152.4174N$$

$$R_B + R_C = 250.998$$

$$R_B = 250.998 - R_C$$

$$R_B = 250.998 + 152.417$$

$$R_B = 403.415N$$

Thus, $R_B = 403.415N, R_C = -152.4174N$

2.6 Maximum bending moment (BM_{max})

Referring to fig.5, Bending moments are:

At A, $BM_1 = 0$

At B, BM_2 = 250.84(0.14) = 35.1176Nm

At C, BM_3 = 250.84(0.14 + 0.23) – 403.415(0.23) = 0.02535Nm

At D, BM_4 = 250.84(0.53) – 403.415(0.39) + 152.4174(0.16) = 0

The maximum bending moment, BM_{max} is 35.1176Nm and will be used for computation of shaft diameter.

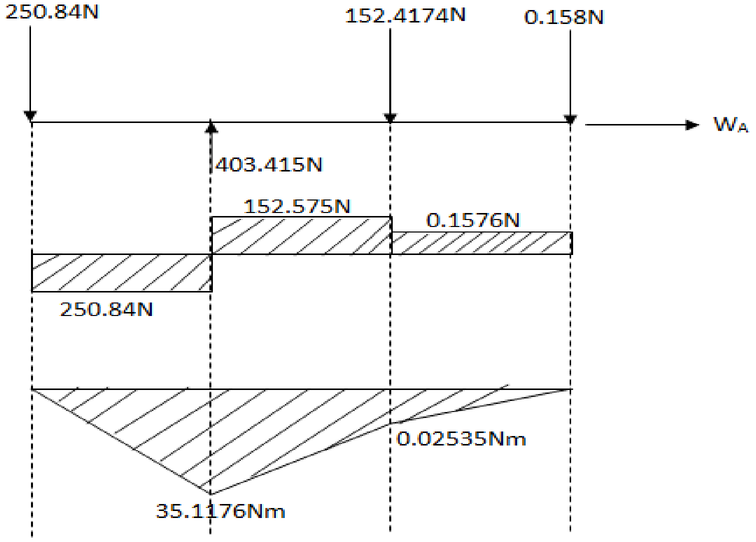


Fig. 6 Shear force and bending Moment diagrams

2.6 Driver Shaft Diameter

The shaft diameter d_s is computed in accordance with ASME code equation for shafting [11]

$$d_s^3 = \frac{16}{\pi \sigma_s}\sqrt{(k_b BM_{max})^2 + (k_t T_m)^2} \qquad (5)$$

Where k_b = combined shock and fatigue factor applied to bending moment

k_t = combined shock and fatigue factor applied to torsional moment

For a rotating shaft, $k_b = 1.5 \; and \; k_t = 1.0$

ASME code states for commercial steel shafting with keyway,

σ_s(allowable) = $40 MN/m^2$

BM_{max} = 35.1176Nm

T_m is the torsional moment of the shaft and is given by,

$T_m = F_c \times rp_2$

Where rp_2 is the radius of the driven pulley ($rp_2 = \frac{D_2}{2}$)

T_m = 0.158 x 0.00979

$T_m = 1.547 \times 10^{-3} Nm$

Substituting, we have

$$d_s^3 = \frac{16}{\pi \times 40 \times 10^6}\sqrt{(1.5 \times 35.1176)^2 + (1 \times 1.547 \times 10^{-3})^2}$$

$$d_s^3 = 1.27 \times 10^{-7}\sqrt{2\,774.803 + 2.3932 \times 10^{-6}}$$

$$d_s = 0.0188m$$

$$d_s = 18.84mm$$

Applying a factor of safety of 1.7, the shaft diameter becomes

$$d_s = 18.84 \times 1.7 = 32.028mm$$

For this work, a shaft of diameter 32mm is selected being the available stock in the local market.

2.7 Axial load W_a on the Driver shaft

The axial load W_a (fig. 5) on the Driver shaft is very useful data for the calculation of the equivalent load on bearings and for bearing selection.

The axial load on the Driver shaft can be expressed by:

W_a= Weight of Driver Wt_D + Weight of Driver shaft Wt_{sh} + Weight of driven pulley Wt_p.

i.e $W_a = Wt_D + Wt_{sh} + Wt_p$.

Wt_D was obtained using Tripple beam balance, capacity 2610g as 1.5kg.

Wt_{sh} and Wt_p were obtained using a Diamond top loading balance, capacity 10Kg as:

Wt_{sh} = 2.52Kg

Wt_p = 2.15Kg

Substituting Wt_D, Wt_{sh} and Wt_p into the equation for W_a and using acceleration due to gravity g, as $10\ m/S^2$, yields:

$$W_a = [1.5 + 2.52 + 2.15]10$$

$$W_a = 61.7N$$

2.8 Equivalent dynamic load W_e on bearing

Radial bearings are often subjected to simultaneously acting radial and axial loads. If the resultant load is constant as in the case of the Driver shaft, then the equivalent dynamic load W_e can be expressed by the equation:

$$W_e = X \times V \times W_R + Y \times W_a \tag{6}$$

Where V= rotation factor,

X and Y are the radial and axial load factors respectively.

For deep groove ball bearings, $V = 1,\ X = 0.56\ and\ Y = 2$ [24]

W_R is the radial load $(R_B = 403.415N, R_c = 152.4174N)$. The higher of the two values that is $R_B = 403.415N$ is taken for the purpose of bearing safety and uniformity. Thus taking $W_R = 403.415N$ and substituting the values of $X\ and\ Y$ in the above equation results in:

$$W_e = 0.56 \times 1 \times 403.415 + 2 \times 61.7$$

$$W_e = 225.9124 + 123.4$$

$$W_e = 349.3124N$$

2.9 Bearing load capacity W_c

The relationship between the rating life L, the equivalent dynamic load W_e and the bearing load capacity W_c is represented by: [24]

$$W_c = W_e \left(\frac{L}{10^6}\right)^{1/k} \qquad (7)$$

K is a constant with value equal to 3 for ball bearings, and
$W_e = 349.3124N, L = 60 \times N \times L_H\ (revolutions), N = 20rpm, L_H = 40000hours.$

$$L = 60 \times 120 \times 40000$$

$$L = 288 \times 10^6 rev$$

The bearing load capacity is thus calculated as:

$$W_c = 349.3124 \left(\frac{288 \times 10^6}{10^6}\right)^{1/3}$$

$$W_c = 2306.8N$$

III. Conclusions

The test rig uses Geneva mechanism to index a table intermittently for bottle washing. The test rig was designed, constructed, assembled and used to run the experimental study. As the drive speed of the Geneva mechanism increases, the cycle time, washing time and indexing time decreases while the maximum pin-slot contact force and washing efficiency increases. The washing efficiency of the test rig from 5rpm to 19 rpm increased from 81.57% to 96.89%. It is concluded that at 19rpm, the designed bottle washing machine had washing time of 2.434 seconds and maximum efficiency of 96.89%.

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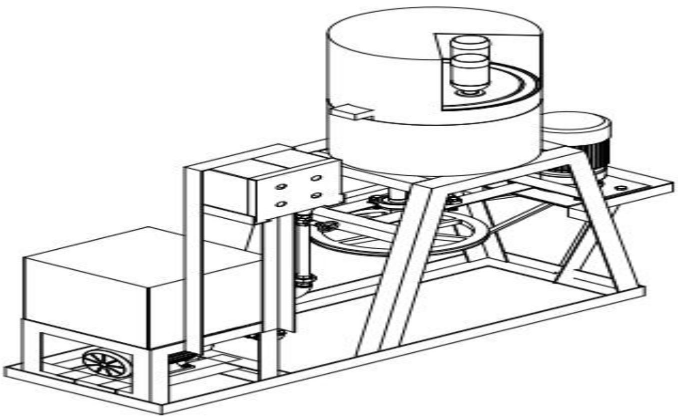
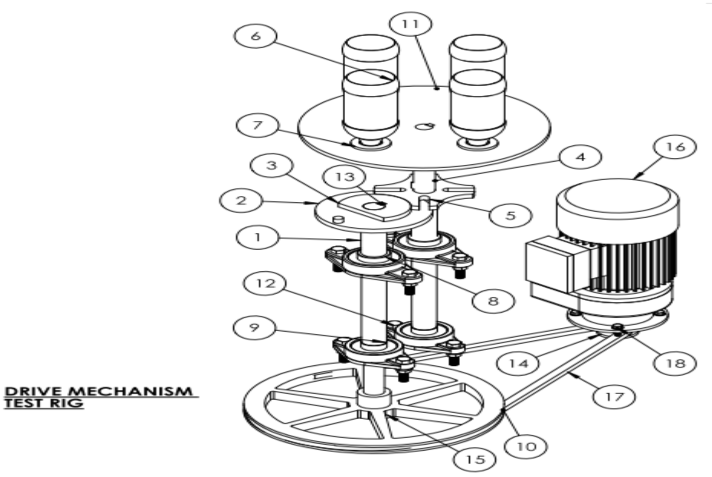


Fig.7 Rig Assembly



ITEM NO.	PART NUMBER	DESCRIPTION	QTY.
1	Driver Shaft	Mild steel Rod	1
2	Driver plate	Mild steel Plate	1
3	Cam	Mild Steel Plate	1
4	Geneva Shaft	Mild steel Rod	1
5	Geneva Wheel	Mild Steel Plate	1
6	Amstel Bottle	Stock	4
7	Bottle holder	Teflon	4
8	bearing 30mm	Stock	2
9	bearing 25mm	Stock	2
10	Driven Pulley	Aluminium Blank	1
11	Table	Stainless Steel Plate	1
12	M10 x 1.5 x 50	Stock	9
13	Key	Mild Steel Rod	3
14	Driver Pulley	Aluminium Blank	1
15	Pulley support	Mild Steel Plate	1
16	Geared electric motor	Stock	1
17	Belt	Stock	1
18	M6 x 1.0 x 20	Stock	4
25	M10 x 1.5 nut	Stock	8

Fig.8 Drive Mechanism