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# A Rotaprocating Compressor

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**Abstract** – This paper presents analytical studies of a new compressor design that integrates merits of both the rotary and the reciprocating compressors. The new design replaces the crank mechanism in the conventional reciprocating compressor and results in a construction that allows increases in capacity as compared to the conventional reciprocating compressor with the same swept volume operating at the same motor speed. A mathematical model accounts for the compressor dynamic, valve dynamic, thermodynamic processes, mechanical considerations has been formulated. The predicted torque characteristics have been compared with that of the reciprocating compressor. The effects of stroke length and the stroke number per motor revolution on the compressor performance have also been analyzed and discussed. Results shows that while the new compressor offers possibility of higher capacity at small size and low motor speed, it leaves much room for improvement on its torque characteristics.

## NOMENCLATURE

| | |
|---|---|
| $a$ | acceleration, m/s$^2$ |
| $f_1, f_2$ | columb frictional coefficients |
| $F_k$ | frictional force acting along the guide pin or keyway, N |
| $F_r$ | radial force to prevent piston from revolving, N |
| $F_z$ | net axial force acting along vertical z direction, N |
| $L_s$ | stroke length, m |
| $l$ | offset distance from guide pin center axis to piston internal profile, m |
| $ma$ | inertia load, N |
| $m_B$ | equivalent masses on the piston pin, kg |
| $n$ | number of strokes per unit motor revolution |
| $P_c$ | pressure in compression chamber, N/m$^2$ |
| $r$ | internal groove pitch radius, m |
| $v$ | velocity, m/s |
| $V_c$ | chamber volume, m$^3$ |
| $V_{cl}$ | chamber clearance volume, m$^3$ |
| $Z_{cl}$ | clearance length, m |
| $Z_s$ | stroke displacement, m |
| $\theta(t)$ | motor revolution angle as a function of time, rad |
| $\beta_f$ | angular displacement of frictional force $F_f$, rad |
| $\beta_n$ | angular displacement of normal force $F_n$, rad |
| $\beta_m$ | angular displacement of shaft driven force $F_m$, rad |
| $\varpi$ | shaft angular velocity, rad/s |

## INTRODUCTION

Over the past centuries, many types of compressor have been designed for applications in refrigeration, air conditioning and gas compression. These include various versions of the conventional reciprocating compressors to the more recent rotary and scroll types.

Many new designs have since been introduced. These include quadro-flex compressor [1], Groll compressor [2], helical blade rotary compressor [3,4], orbital vane compressor [5], linear compressors [8,9,10] and swing compressor [11]. It is however, neither of these recent designs nor the conventional compressors have proven to possess formidable attributes to lead the present market place. While industries and research centers are continuously strive to further improve the performance of the existing machines, at the same time, new designs have also been introduced.

In this work, a new mechanism integrating merits of both the **rotary** and **reciprocating** compressor, namely a **rotaprocating** mechanism has been used to replace the conventional crank mechanism in a reciprocating compressor see figure 1. As compared to the reciprocating compressor, this new design allows the compressor operating at the same motor speed with the same pump dimensions to deliver capacity of multiple of that of the reciprocating compressor, depending on the "groove" design. The new design also allows a more compact machine to be fabricated.

### The Advantages of the New Design

As compared to the reciprocating compressor, the advantages of this new compressor are:

- Improved capacity.

The main advantage of this new compressor is that it allows the same motor to be used for compressors with different capacities. This gives greater flexibility in commonization of the compressor parts.

- Compactness

It allows an inline construction of the motor and the pump unit, as that in cases of most rotary compressors, and hence reduces the size and increases the compactness.

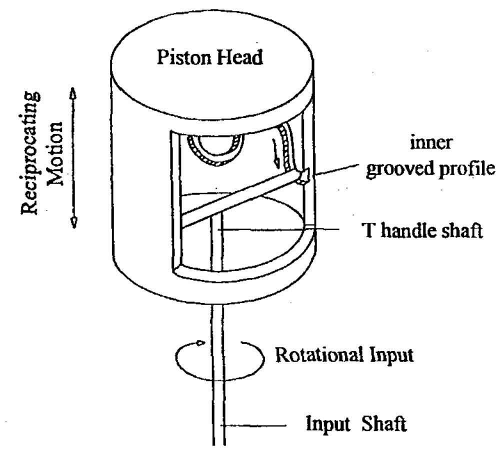



Figure 1 : Working Principles of Rotaprocating Mechanism

- Simple construction with fewer moving parts

The new mechanism has fewer parts and hence lower materials cost required.

- Reliable

The established piston-cylinder and valve technologies of the reciprocating compressor are retained and hence reduces the reliability problem commonly associated with new products.

Figure 1 illustrates the basic construction of the rotaprocating mechanism. In its simplest form, the new mechanism comprises of two main components: (a) a piston with a cycloidal groove cut in its inner surface and (b) a T-shaft with driven end connected to motor shaft. The driving end is confined in the groove on the piston inner surface. During the operation, when the motor shaft rotates, the T-shaft will rotate and since the piston is prevented from rotation, a linear displacement of the piston is resulted. It can be seen that the T-shaft is only a single component and it replaces the entire crank mechanisms in the reciprocating piston compressor. At the same times, the delivery capacity may increase by multiple fold if the profile of the groove is designed to have multiple strokes per motor revolution (n = 1, 2, 4,...). In general, increases the stroke to two per motor revolution should have effect to double the capacity, while the operating condition and the dimensions of other parts remain unchanged.

## Mathematical Modeling

Figure 2 shows a schematic of the pump unit of the rotaprocating compressor. For performance evaluation, a mathematical model has been formulated for the new compressor. The reciprocating compressor with similar dimensions has also been modeled for comparison purposes.

In present study, for the rotaprocating compressor, a cycloidal curve profile has been cut in the inner piston wall. This profile is chosen based on its appropriate dynamic characteristics for medium to high-speed operation. The cycloidal motion has no abrupt changes of acceleration at the dwell ends. Thus giving finite jerk at the transition point. It must be born in mind that the jerk free groove is almost impossible in practice. The kinematics equations of this motion is,

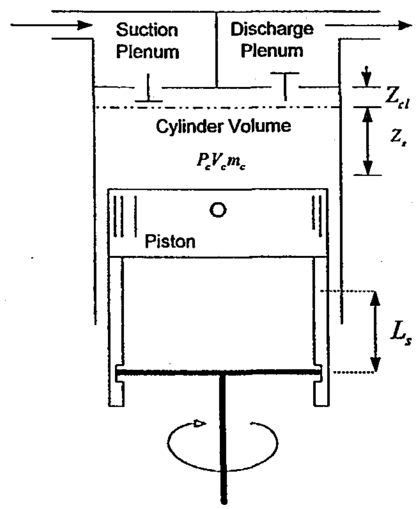



Fig 2: Schematics of a rotaprocating compressor.

$$Z_s(t) = L_s\left[\frac{n\theta(t)}{\pi} - \frac{1}{2\pi}\sin 2n\theta(t)\right] \qquad \text{for rise stroke } 0 < \theta(t) < \frac{\pi}{n} \tag{1}$$

$$Z_s(t) = L_s\left[2 - \left(\frac{n\theta(t)}{\pi} - \frac{1}{2\pi}\sin 2n\theta(t)\right)\right] \qquad \text{for return stroke } \frac{\pi}{n} < \theta(t) < \frac{2\pi}{n} \tag{2}$$

The velocity $v$ , acceleration $a$ and jerk $j$ could be obtained by taking the time derivative of $Z_s$, $v$ and $a$ respectively. The volume expression is given by,

$$V_c(t) = V_{cl} + \frac{\pi D^2}{4} Z_s(t) \tag{3}$$

## FORCE ANALYSIS OF ROTAROCATING MECHANISM

The various forces associated with the new mechanism are shown in Fig 2. These are given below:

(a) The gas pressure load on the piston face area is given as

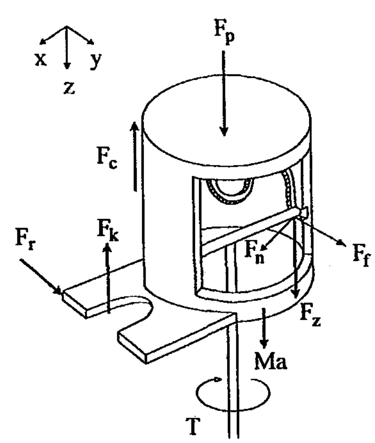



Fig 3: Free body diagram

$$F_p = (p_c - p_a)A_p \tag{4}$$

(b) The frictional force acting along the piston wall is given as:

$$F_c = \mu A_t \frac{V}{h} \tag{5}$$

(c) The frictional force along the internal groove surface which is in contact with the tip of the T- shaft, assuming a Coulomb type friction:

$$F_f = f_1 |F_n| \tag{6}$$

(d) The frictional force rubbing along the key-way is:

$$F_k = f_2 \left| \frac{-2r(\cos\beta_n + f_1 \cos\beta_f)}{(r+l)\cos\beta_m} \right| F_n \tag{7}$$

(e) The normal contact force $F_n$ is obtained as:

$$F_n = \frac{ma + F_c - F_p}{\left[2(\sin\beta_n + f_1 \sin\beta_f) \mp f_2 \left| \frac{-2r(\cos\beta_n + f_1\cos\beta_f)}{(r+l)\cos\beta_m} \right| \right]} \tag{8}$$

(f) The shaft driven force $F_m$ is given as

$$F_m = \frac{-F_n(\cos\beta_n + f_1 \cos\beta_f)}{\cos\beta_m} \tag{9}$$

The $\beta_f$, $\beta_m$ and $\beta_n$ are the angular displacement of frictional force, shaft driven force and normal force respectively on the T- shaft as shown in Figure 3. The final equations for torque are given in equations (10), (11) and (12).

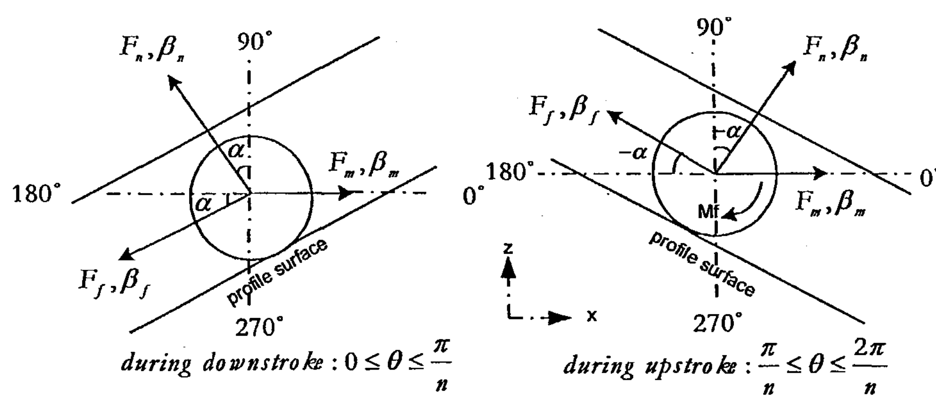



*Fig 4 : Force interaction on the T shaft*

(a) The torque due to interacting effects of gas pressure, inertia load and frictional losses is given as

$$T = \frac{-2r(ma + F_c - F_p)(\cos\beta_n + f_1\cos\beta_f)}{\cos\beta_m\left[2(\sin\beta_n + f_1\sin\beta_f) \mp f_2\left|\frac{-2r(\cos\beta_n + f_1\cos\beta_f)}{(r+l)\cos\beta_m}\right|\right]} \tag{10}$$

(b) The desirable torque due to effect of inertia alone can be determined by neglecting the friction and gas pressure $(F_p = F_c = F_k = F_f = 0)$. Thus become

$$T' = \frac{-r \cdot ma\cos\beta_n}{\sin\beta_n\cos\beta_m} \tag{11}$$

(c) The desirable torque due to effect of gas pressure alone can be determined by neglecting the inertia and friction, assuming that moving parts are massless with $(ma = F_c = F_k = F_f = 0)$.

$$T'' = \frac{rF_p\cos\beta_n}{2\sin\beta_n\cos\beta_m} \tag{12}$$

Equations (10- 12) allow the desirable torque to be estimated and hence a suitable motor can be selected. The input power on shaft $\dot{W}$ is obtained by assuming a constant motor angular speed of $\omega$:

$$\dot{W} = T\omega \tag{13}$$

The inherent nature of rotaprocating motion has no side forces between the piston and the cylinder walls.

The torque characteristic of the rotaprocating compressor will be compared with the conventional reciprocating compressor. The torque equations for crank mechanism of the reciprocating compressor are:

(a) The torque at the crankshaft to overcome gas pressure force is:

$$T' = -F_p r\sin\omega t\left(1 + \frac{r}{l}\cos\omega t\right) \tag{14}$$

(b) The torque to overcome the inertia load:

$$T'' = \frac{m_B}{2}r^2\omega^2 \quad -\frac{r}{2l}\sin\omega t + \sin 2\omega t + \frac{3r}{2l}\sin 3\omega t \tag{15}$$

(c) The torque due to overall effects excluding the frictional losses is obtained by summing up equation (16) and (17).

$$T = T' + T'' \tag{16}$$

(d) The side force acting on the piston can be determined as

$$F_{14} = \left[F_P + m_B\ddot{x}\right]\left(\frac{r}{l}\sin\omega t\right)\left(1 + \frac{r^2}{2l^2}\sin^2\omega t\right) \tag{17}$$

## RESULTS AND DISCUSSIONS

The torque variation for a reciprocating compressor is shown in Figures (5) and (6). Fluctuations in torque may cause the vibrations that result the structure born noise in the compressor. The simulation is carried out for a reciprocating compressor with stroke length of 18 mm, bore diameter 25.4 mm, motor speed 3000 rpm and the piston weight of 59.6g with refrigerant R134a as the working fluid.

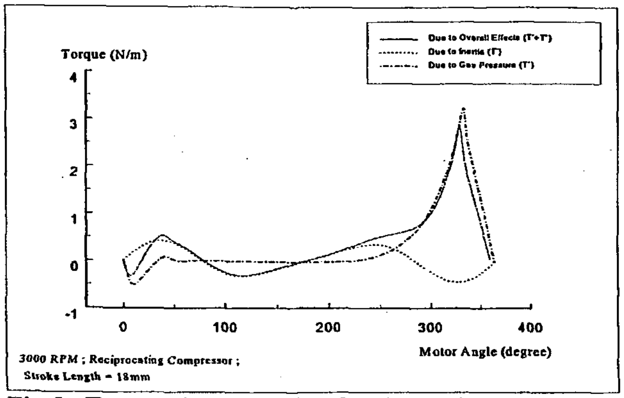



Fig 5 : Torque characteristic of reciprocating compressor

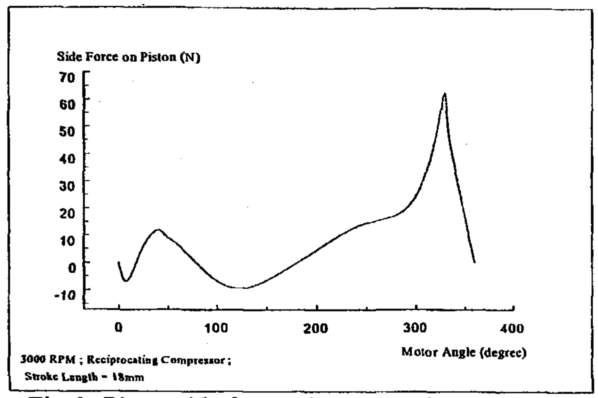



Fig 6 : Piston side force of reciprocating compressor

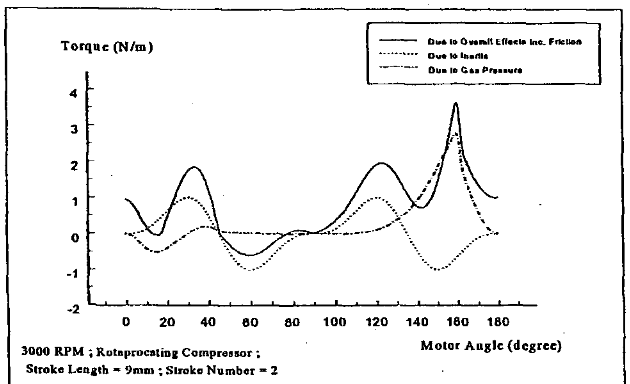



Fig 7 : Torque characteristic of rotaprocating compressor

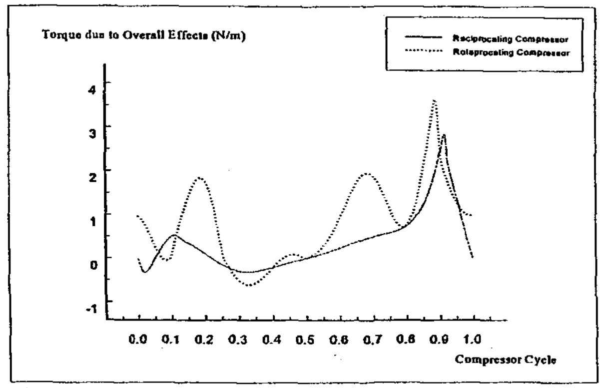



Fig 8 : Torque comparison between two compressors

Figure 7 shows the variation of various torques for the rotaprocating compressor, with shorter stroke length of 9 mm and 2 strokes per motor revolution to ensure the same capacity with the reciprocating compressor. Both compressors share the same dimensions for pump unit to enable a fair comparison. The variations of driving torque for these two compressors are shown in Figure 8.

The predicated results in Figure 7 and 8 show that the torque is the highest at the end of the compression. The gas pressure is responsible for the maximum torque. However, the inertia of moving mass is also playing a significant role toward the torque variation that is especially obvious in the case for the rotaprocating compressor as shown in Figure 7. The fluctuation in the overall torque values for the rotaprocating compressor is caused by the variations of the inertia torque (Figure 7) which is a related to the internal profile pitch radius, piston mass, acceleration characteristic of the internal profile and the pressure angle. On average, the results shows that the torque values for the rotaprocating compressor is slightly higher than that of the reciprocating compressor in this case because the design of the pump unit used is optimized for the reciprocating compressor at this operational

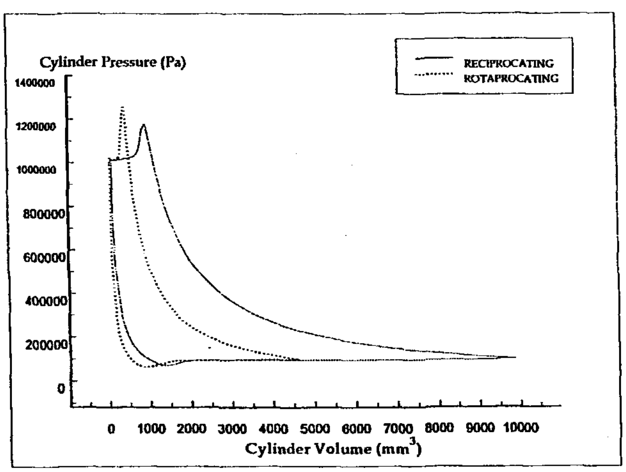



Fig. 9: Pressure Volume diagram in a comparison.

that with the proper design of the valve mechanism, the rotaprocating compressor could produce comparable torque characteristics.

However, the reciprocating compressor is proven with unbalanced side forces due to the inherent characteristics of the crank-slider mechanism as shown in Figure 6. Hence the reciprocating mechanism is normally not recommended for high-speed applications in relating to noise and vibration. On the other hand the rotaprocating compressor, owing to its inherent mechanism, does not have such a problem.

Simulations were also performed to study the effects stroke length and the number of stroke per motor revolution, n from 1, 2 to 4. The same compression capacity is maintained by varying stroke length and maintaining a constant of $L \times n = 18$. The clearance volume is fixed on 1% of swept volume to ensure a fair comparison. Therefore, simulation were performed over three input combinations as follow:

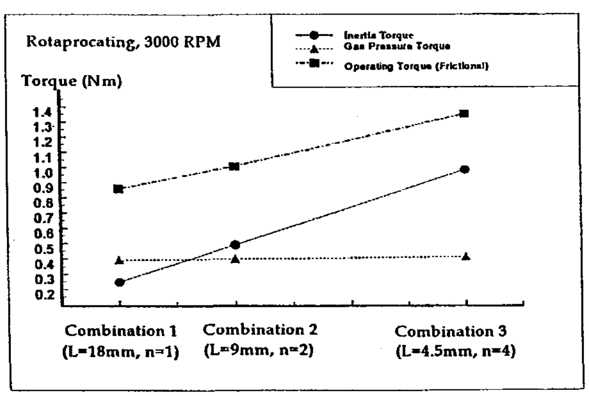
(a) Combination 1 : stroke length $L$=18mm, stroke number $n$=1, ratio $L/n = 18$
(b) Combination 2 : stroke length $L$=9mm, stroke number $n$=2, ratio $L/n = 4.5$
(c) Combination 3 : stroke length $L$=4.5mm, stroke number $n$=4, ratio $L/n = 2.25$

The simulation results are tabulated as below:

| | | | |
|---|---|---|---|
| Stroke number $n$ | 1 | 2 | 4 |
| Stroke length $L$(mm) | 18 | 9 | 4.5 |
| Ratio $L/n$ | 18 | 4.5 | 2.25 |
| Piston diameter $D$ (mm) | 25.4 | 25.4 | 25.4 |
| Swept Volume ($m^3$) | $9.212 \times 10^{-6}$ | $4.606 \times 10^{-6}$ | $2.302 \times 10^{-6}$ |
| Max. Pressure Angle, $\alpha$ | 49.96 | 49.96 | 49.96 |
| Avg. Inertia Torque (Nm) | 0.2458 | 0.4917 | 0.9833 |
| Avg. Pressure Torque (Nm) | 0.3899 | 0.3979 | 0.4189 |
| Avg. Total Torque (Nm) | 0.8591 | 1.0069 | 1.3496 |
| Avg. Power (W) | 269.88 | 316.34 | 424.03 |
| Volumetric Efficiency (%) | 75.18 | 75.22 | 75.18 |
| Mass Delivery (g/min) | 79.28 | 79.32 | 79.28 |

Table 1 Effects of stroke length to stroke number ratio toward compressor torque

The simulation results in Table 1 and Fig 10 show that compressor operating torque increases from combination 1 to 3. Results show that the torque increases with the decreasing $L/n$ ratio, implies a poorer throughput. From Fig 11, the gas pressure torque is obviously constant shows the increment of compressor torque has been a vital consequence of increased inertia torque with decreasing $L/n$ ratio.



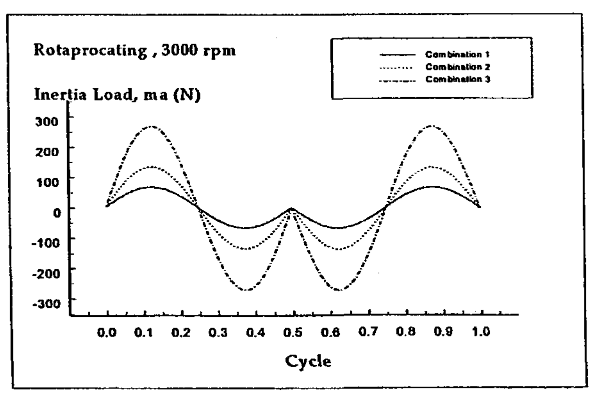




Fig 10 Effects of $L/n$ ratio on compressor torque. Fig 11 Increased inertia load with lower $L/n$ ratio

The consequences of higher inertia load has also caused the characteristic of compressor torque to fluctuate more severely, see Fig 12, which implies a poorer dynamic behaviors. The results shows clearly that with smaller $L/n$ ratio, the rotaprocating compressor has a higher operating torque that reduces the overall compressor throughput.

## CONCLUSIONS

In general, results show that while the rotaprocating compressor has many advantages over the reciprocating compressor, such as the capability of multiple fold capacity for the same pump-unit, more compact because of the inline construction and no unbalance side forces. The new compressor also has the potential of lower materials cost. The greatest advantage comes from the flexibility in higher capacity while using the same motor speed and most of the common components. However, its torque characteristics is less desired as compared to the reciprocating compressor.

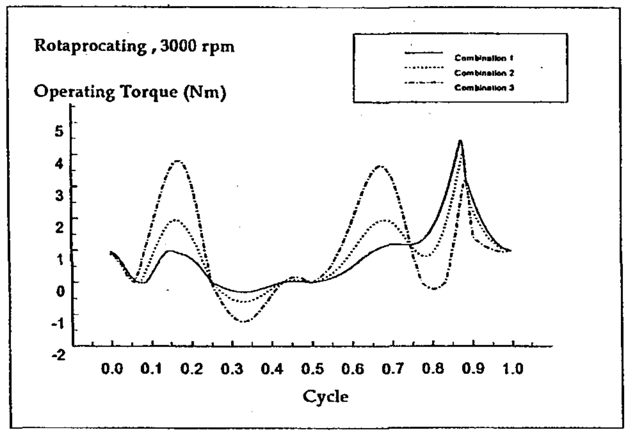



Fig 12 : The torque variations in comparison

Results also concluded that decreasing stroke length to stroke number ratio $L/n$ causes an increase in the inertia torque and resulted in a poorer dynamic characteristic due to higher torque fluctuation.

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