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ABSTRACT

A newly invented compressor, namely revolving vane compressor (RV) has been introduced two years ago (Teh and Ooi, 2006[a], 2006[b], 2006[c]). This paper presents the improved version (Teh and Ooi, 2007) of the RV, called RV-i, where "i" stands for improved. The RV or RV-i is designed with the intention to overcome most of the limitations facing the working characteristics of the rolling piston compressor (RP). In its basic construction, the RV is very similar to RP, but with a rotating cylinder. As compared to RV, RV-i has a fixed vane and the vane forms an integral part of the driving component of the RV-i. The driving component of the RV-i is the component that is driven directly by the motor; in this case, it can be the extended motor shaft that carries the concentric or the motor shaft that is connected directly to the rotating cylinder. In this paper, the RV-i and its operational characteristics as compared to the RP will be presented in detail. As compared to RP, RV-i exhibits many advantages: it has fewer components, geometrically simpler, much lower friction and internal leakage losses, and it can achieve a perfect dynamic balance. The various variants of RV-i will also be shown.

1. INTRODUCTION

The most important factors affecting the success or otherwise, in introducing a new design for an existing product, have always been governed by the following three:

1. production cost
2. performance
3. reliability

Speaking in terms of a refrigeration/air conditioning compressor, a new compressor design is preferably to consist of the following criteria in order to compete successfully with the existing compressors.

1. To achieve a lower production cost, the new compressor is expected to
   i. have less parts and it must be geometrically simpler, and
   ii. be able to accommodate lower machining accuracy/tolerances with less machining processes required.

2. To achieve a higher performance than the existing compressor, the new compressor is preferably to possess
   i. higher volumetric and mechanical efficiencies, and
   ii. it should inherently exhibit lower vibration and hence emit lower noise level.

3. To be more reliable than the existing compressor, the new design should ideally
   i. exhibit lower vibration level,
   ii. consist of lesser or no critical components, and
   iii. possess lower rubbing areas/components.

The RV-i (Teh and Ooi, 2007) compressor that is introduced in this paper possesses the potential to achieve most of the criteria mentioned above.

In the refrigeration and air-conditioning industries, the reciprocating compressor is the first commercially viable compressor for mass production. Its operational characteristics have been very well researched into and understood, and its technology is well established and it is a reliable machine. Although better compressor designs such as those depicted by rotary and scroll compressors have come along, the mature technology of the reciprocating compressor coupled with the vast investment that has already been in place, will mean that the reciprocating compressor is here to stay for many years to come, though its share in market for certain applications is slowly being replaced by other types of compressors.

An introduction of the rotary compressor in early/mid 80s has captured a substantial proportion of the air-conditioning market previously occupied by the reciprocating compressor. This is largely due to many inherent good characteristics of the rotary compressor such as its compactness, lower vibration and noise level, fewer parts and ability to operate at higher speeds.

The rolling piston compressor (RP) is one of the rotary type compressors that has been widely used in household refrigerators and air conditioning systems (Ooi and Wong, 1997, Ooi, 2005). As compared to the well established reciprocating compressor, the rolling piston design is geometrically simpler, has fewer components and exhibits lower vibration and noise level. However, it has high frictional losses and critical parts like the vane and the spring. In this paper, a new compressor design namely the improved revolving vane compressor or RV-i will be introduced and discussed. The initiation of the RV-i can be viewed as a major step forward not only in overcoming many inherent design weaknesses facing the basic construction of the rolling piston compressor (Yanagisawa and Shimizu 1985[a], 1985[b], 1985[c]), but also in potentially creating a new height for the energy efficiency level of a compressor.

2. RV-i: THE IMPROVED REVOLVING VANE COMPRESSOR

2.1 Introduction to Revolving Vane Compressor (RV)

Figure 1 shows the schematic of a rolling piston compressor. In its basic form, it consists of five major components: a roller, an eccentric, a stator/cylinder, a vane and a spring. The roller rolls along the inner cylinder surface while separating the working chamber bounded by vane, roller and cylinder into two; while one chamber undergoes suction, the other, compression and discharge. There are several major areas of contacts that give rise to significant frictional losses and these are at the contact of: (i) roller and eccentric, (ii) roller and cylinder end-faces, (iii) roller surface and vane tip, (iv) eccentric and cylinder end-faces, (v) vane-sides and slot and, (vi) shaft and bearings. Through many years of compressor development and improvement work, the friction losses in many of these contact areas have been reduced mainly through innovative design and to provide better lubrication.

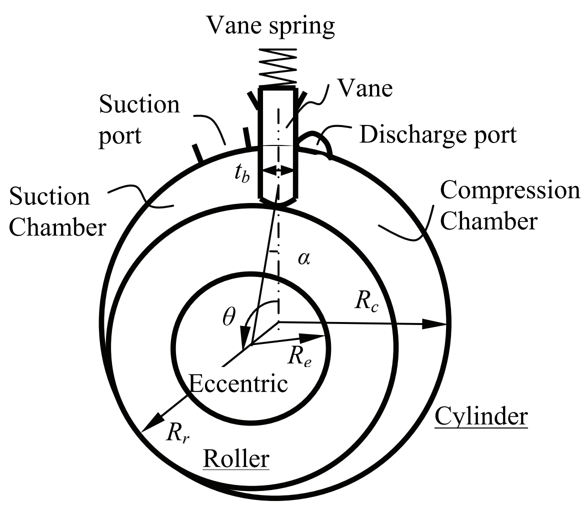

Figure 1 A schematic of a rolling piston compressor

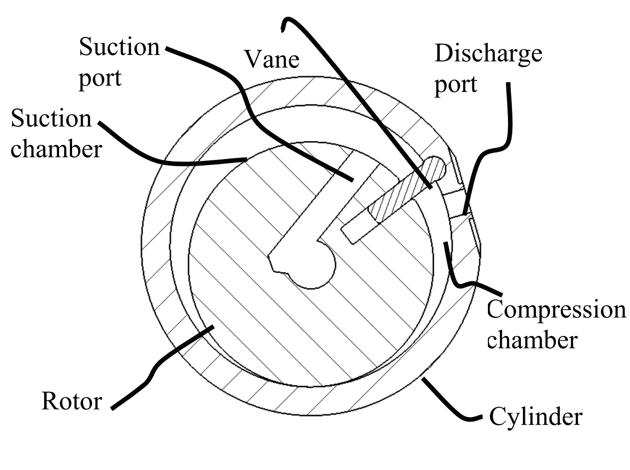

Figure 2 A schematic of a revolving vane compressor

The existence of friction in the areas mentioned above restricts the compressor to operate at speeds beyond the current limits, as higher speeds will result in significant friction, which will not only reduce the compressor performance but it also leads to compressor failures. The next question is if these friction losses can be further reduced or best still, completely eliminated. For the current design state of the rolling piston compressor, its design and improvement work has almost reached the stagnation level which, without significant changes in its current design, no significant change in performance can be expected.

Let us take a step back to reconsider the fundamentals. Generally speaking, the friction that occurs between any two components is caused by the difference in the speed of these components. For an example, in the case of the rolling piston compressor, the friction between the cylinder and the roller is caused by the existence of a high relative velocity between them. Hence it is obvious that to reduce the friction, the relative velocity of the rubbing components must be reduced, best still, eliminated. Therefore, to reduce the friction between the rotating and the stationary components (such as the roller and the cylinder), one idea is to also rotate the "stationary" component. Using the latter idea, here is the birth of the revolving vane compressor, where the cylinder is set in rotation.

Figure 2 shows the schematic of a compressor in which the cylinder is set in rotation during the compressor operation. It is called revolving vane compressor or in short RV. It can be seen that RV has fewer components than the rolling piston counterpart. In its basic form, RV has three major components: a concentric (an integral part of the shaft, as oppose to eccentric in the case of the rolling piston compressor), a vane and a rotating cylinder, and it does away with the roller and the spring. During the operation, the motor shaft rotates and hence rotates the concentric, the vane then turns with the shaft, and since one end of the vane is attached (by a pin joint) to the cylinder, the vane then rotates the cylinder.

The rotating cylinder causes a significant reduction in the relative velocities between all the rotating components which are in contact with the cylinder, and therefore reduces the friction between these components. Of the six major areas of friction occur in RP as mentioned in Section 2, there leaves only the last two items, item (v) and (vi), which are: (v) vane-sides and slot and, and (vi) shaft and bearings. The friction loss in items (i) to (iv) has almost been entirely eliminated. The introduction of the new RV, RV-i, reduces the friction losses further. It completely eliminates the friction in item (v), i.e. between the vane-sides and the slot, and it also significantly reduces the friction in item (vi), i.e. between the shaft and bearing. This part will be discussed in detail in the later section of this paper. The reduction in frictional losses between these moving components gives rise to many advantages in terms of increasing compressor performance and reliability. For example, the mechanical efficiency increases as a result of reduction in friction, and this is shown in Table 1. In Table 1 comparison of mechanical efficiency between rolling piston compressor and the revolving vane compressor are made. It can be seen that the RV-i has a highest mechanical efficiency

Table 1: Comparison of mechanical efficiencies among various compressors

S/n	**Compressor Type**	**Mech. Efficiency (%)**	**Investigators**
1.	Rolling Piston	85 – 90	Ozu and Itami, 1981
2.	Rolling Piston	93.3	Matsuzaka and Nagatomo, 1982
3.	Rolling Piston	92.5	Wakabayashi *et al*., 1982
4.	Rolling Piston	92.0 – 92.2	Sakaino *et al.*, 1984
5.	Rolling Piston	91.8	Ishii *et al.*, 1990[a]
6.	Scroll	92.3 – 92.7	Hayano *et al*., 1988
7.	Scroll	92.3 – 93.4	Ishii *et al*., 1990[b], 1992, 1994
8.	RV	95.1	Teh and Ooi, 2008
9.	RV-i	96.5	Teh and Ooi, 2008

One of the advantages of the rotating cylinder is that the fluctuation of the motor torque over a complete compressor cycle is significantly dampened. This is because the rotating inertia of the rotating cylinder helps to reduce the torque required during the compression stage, by providing part of the torque needed through its rotating inertia.

This feature provides a fresh venue for further optimizing the RV compressor performance. The torque variation of the compressor with a rotating cylinder (of the RV compressor) to that of the stationary cylinder is shown in Figure 3. The figure also shows the expected variation of torque for RP. It can be seen that the peak torque required for RV compressor is lower than that of the RP equivalent.

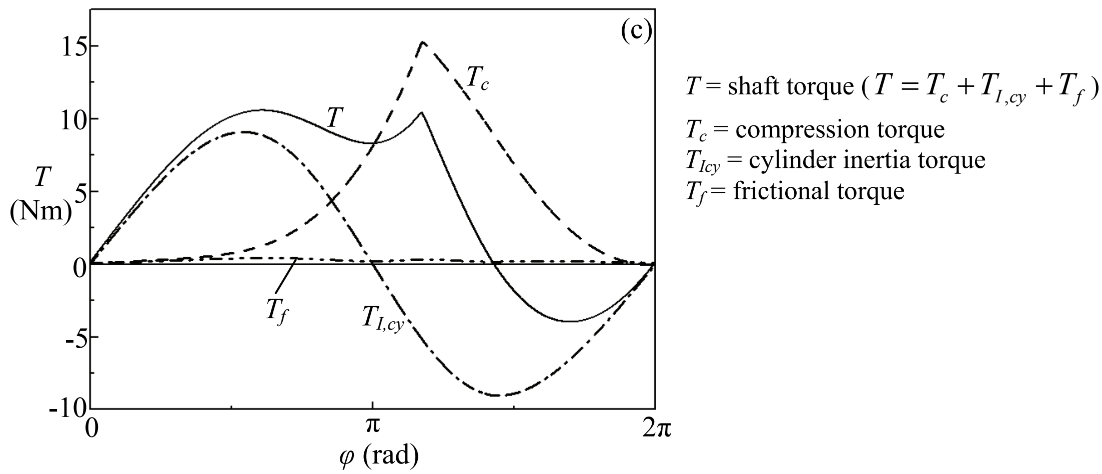

Figure 3: Variation of shaft torque and its components for revolving vane compressor: (T_c also represents the typical qualitative shaft torque in rolling piston compressors.)

The rotating cylinder also carries the whole valve assembly to rotate with it. Simulation results show that the centrifugal force acting on the valve reed during the rotation has the effect of "softening" the valve reed and allows it to have a larger opening at the initial discharge process than that in the conventional non-rotating valve and hence reduces the discharge flow losses (Teh and Ooi, 2006[b]) and improves the energy efficiency. This softening effect of the valve reed also implies that the use of a thicker valve reed is possible, which is needed in some applications to increase the reliability of the valve reed, without penalizing the compressor performance.

Another advantage of the rotating cylinder is that it aids the improvement in the volumetric efficiency by providing effects similar to that caused by the rotating impeller of a centrifugal pump. During the operation, the cylinder rotates at high speeds, causing the entire gas field in the working chamber to rotate. This rotating cylinder induces "centrifugal" forces to the gas field causing the gas being thrown radially outward towards the outer circumference of the working chamber. This effect can result in a higher gas density at the outer periphery of the working chamber. The higher density gas means more mass is being induced and hence improves the volumetric efficiency of the compressor. This "centrifugal" effect is further assisted by locating the suction gas path in the shaft centre and allows the incoming gas to flow radially outward through the radial opening at the concentric into the suction chamber.

2.2 From RV to RV-i

In the RV compressor, the friction still occurs at the vane sides and its slot as well as at the shaft bearings. The vane-slot friction is mainly caused by the contact forces between the vane-sides and the inner wall of the slot. The contact forces, on the other hand, are the product of the gas pressure forces acting across the vane protrusion and the area of the vane side protruded out of the slot. Since the vane is freely placed in the slot, the gas pressure force will press the vane hard onto the inner slot wall and solid to solid contact occurs between the inner slot wall and the vane-sides. Therefore the magnitude of the friction loss caused by the rubbing in the vane slot is proportional to the pressure differential across the vane and the vane side area that protrudes out of the slot. Under such a situation, significant friction loss will occur if the compressor operates under a high pressure differential between the suction and the discharge.

Fortunately, the situation mentioned above can be prevented if the vane is being held rigidly by the driving component of the compressor, in this case the concentric. Figure 4 shows the schematic of the improved version of RV, namely RV-i. It shows that the vane is now formed an integral part of the concentric. It also shows that the vane slot has been relocated to the cylinder and its design has also been modified to accommodate the two degree-of–freedom motion of the newly introduced rigid vane. These design modifications have brought fore a very significant performance improvement. Firstly, the vane-slot friction is no longer dependent on the gas pressure forces across the vane protrusion and in fact, the vane slot friction is almost entirely eliminated. That is to say that it is now possible to design a compressor such that the contact of the vane and the new vane slot (on the cylinder) is merely a seal tight with almost negligible contact forces. Secondly, since the vane side and slot contact forces are no longer dependent on the vane protrusion area, a compressor design that has a longer vane protrusion and hence a short cylinder, is now preferred. This is because for a given compressor displacement volume, a fat and short compressor gives a lot lower bearing force than a tall and thin design. Therefore, in the RV-i design, it is expected that the bearing force which is taken up by the shaft bearing will reduce significantly and hence further improves the mechanical efficiency. Figure 5 shows that the short and fat compressor has a lower total bearing force and hence it is expected that it has a lower bearing friction.

This “fixed vane” feature allows the compressor to operate effectively at high efficiency even though the pressure differential across the vane is very high, such as those in large CO_2 compressors operating at transcritical operational conditions. The short and fat compressor also allows the compressor design to use only one set of bearings and allow the other end of the compressor to overhang similar to that in the case of the scroll compressor. This design would reduce the frictional losses even further.

In the RV-i, both the concentric and the cylinder rotate about their own axes, and the vane is rigidly fixed onto the concentric, hence during the operation of the compressor, there is no component rotating with varying radial positions, and therefore, a perfectly dynamic balance condition can be achieved. This feature is expected to reduce the vibration and noise level of the compressor.

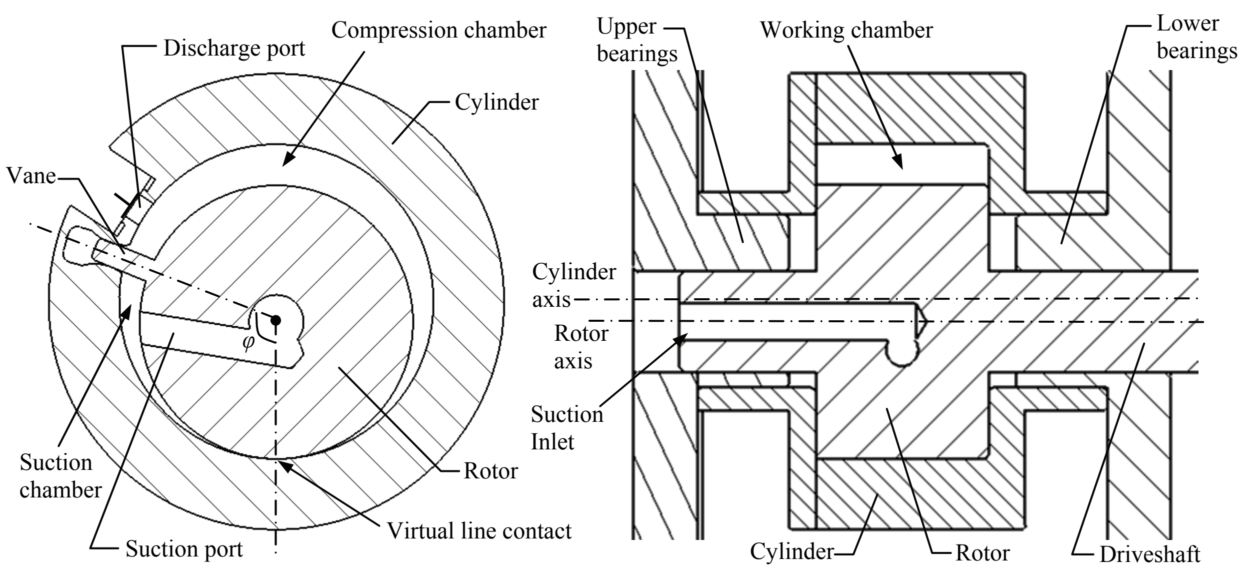

Figure 4: (Left) Front sectional view of RV-i compressor; (Right) Side sectional view of RV–i compressor with journal supports shown

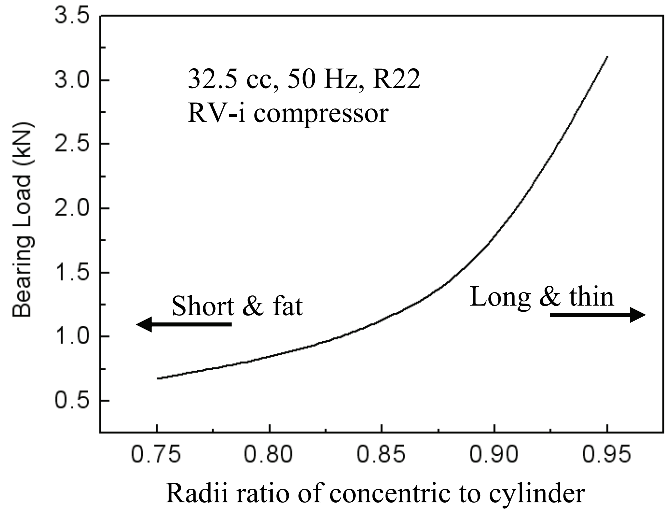

Figure 5: Variations of bearing forces for various geometrical configurations

3. POSSIBLE VARIANTS OF RV-i

In the case of the RV-i as shown in Figure 4, the vane protrusion will be long for the case of a short and fat compressor design, in this case, to accommodate a longer vane protrusion, the cylinder wall may have to be thicker. Though in practice, this may not pose problems since smaller cylinder inertia and reduced material for the cylinder can still be achieved by having a thick cylinder wall with many longitudinal holes in the wall, which can be practically achieved through casting. Alternatively, this can be achieved by fixing the vane on the inner cylinder wall rather than the concentric, as shown in Figure 6(a). For the design shown in Figure 6, it is better to have a slanting vane than the radial vane so that a longer vane slot can be accommodated in the concentric. For such a design, it is better to drive the compressor through the cylinder, as shown in Figure 6(b). Other variants of the RV-i include those with a forward or backward sweep vane, in which the vane can be curve or straight.

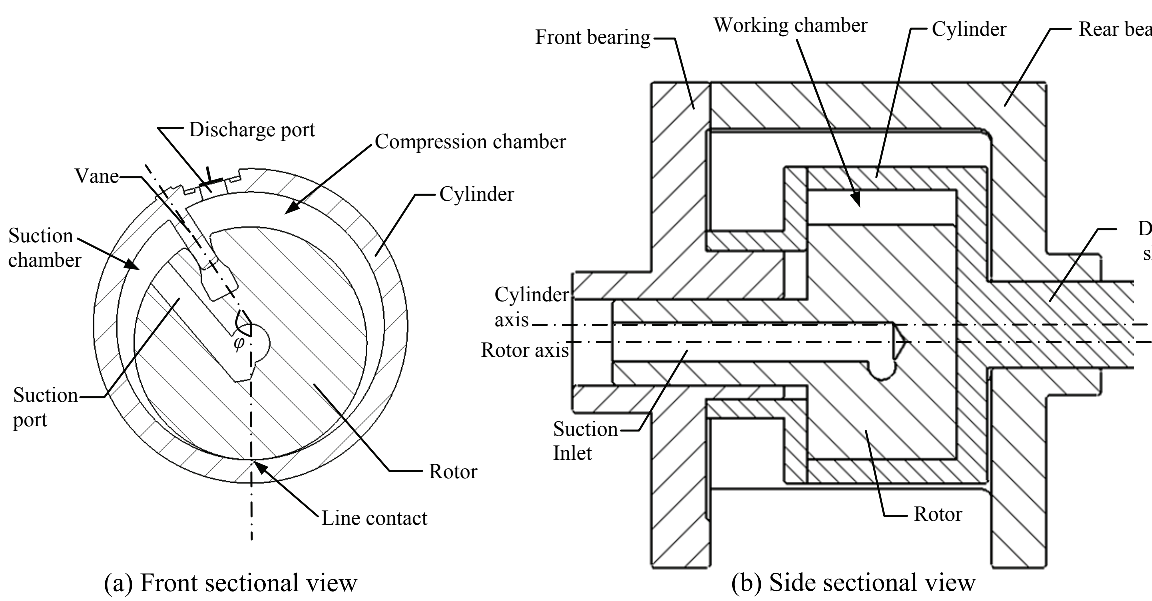

Figure 6: Sectional views of RV-i compressor with the vane fixed on the cylinder. It is better to directly drive the cylinder in this case.

4. CLOSING REMARKS

The comparison between the RP and the RV-i is summarized in Table 2. It is obvious that the inherent operational characteristics of the RV-i (Teh and Ooi, 2006[c]) are superior to that of the RP. Though initial evaluations have confirmed that the RV or RV-i can be a better compressor in terms of cost, reliability and performance, however, before the mass production model of the RV or RV-i rolls out of the production line, a lot of verifications and performance confirmation as well as reliability tests must be carried out and ascertained. Prototype fabrication of the RV is now underway, it is believed that the measurement results of which will take this compressor a step closer to the realization of the commercially viable product.

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Table 2: Comparison between Rolling Piston (RP) and Improved Revolving Vane (RV-i) compressor.

Features	**Rolling-Piston (RP) compressor**	**Improved Revolving Vane (RV-i) compressor**
1. Frictional loss	Higher frictional losses due to rubbing between components at: i. roller and vane tip ii. roller and eccentric iii. roller and cylinder endfaces iv. vane and slot v. eccentric and cylinder endfaces vi. journal bearings	**Lower** frictional losses due mainly at: i. journal bearings (the use of an overhang single bearing is possible)
2. Number of major components	Relatively more main components, such as: i. Cylinder ii. Vane iii. Shaft and eccentric iv. Roller v. Spring	Relatively **fewer** main components, such as: i. Cylinder ii. Vane iii. Shaft and **concentric**
3. Precision requirement	More critical components and require stringent precision and processes: i. Vane – requires hardening to reduce wear at vane tip ii. Roller - requires hardening to reduce wear iii. Eccentric – eccentric offset from shaft requires precision control	**Zero** critical components
4. Costs	i. Higher material costs – due to more components ii. Higher process costs – due to more critical components iii. Higher motor costs – a bigger motor is needed due to: a. more frictional loss b. higher max torque	**i. Lower** material costs – due to **less** components ii. **Lower** process costs – due to : a. **no** critical components b. **no** eccentric iii. **Lower** motor costs – a **smaller** motor is needed due to: c. **less** frictional loss d. **lower** max torque
5. Efficiency	Lower efficiency due to: i. higher frictional losses ii. higher internal leakage due to: a. larger average operating radial clearance between roller and cylinder. b. larger allowable axial clearance at cylinder endfaces.	Higher efficiency due to: i. **lower** frictional losses ii. **lower** internal leakage due to: a. **smaller** average operating radial clearance between concentric and cylinder b. **smaller** allowable axial clearance at cylinder endfaces iii. centrifugal compression due to rotating cylinder and ports' location iv. **lower** discharge loss due to centrifugal softening of valve reed
6. Reliability	Lower reliability due to: i. more critical components ii. more wear	**Higher** reliability due to: i. **no** critical components ii. **less** wear