Piston Ring Assembly for a New Natural Gas Vehicle Symmetrical Multistage Wobble-Plate Compressor

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Abstract: Natural gas is an alternative fuel of choice in the market today due to the increase in the price of petroleum, as well as out of environmental concerns. Pressure requirement for a natural gas vehicle (NGV) storage tank is 3000 psig (206 bars). Thus, at NGV refueling facilities, the natural gas need to be stored at a higher pressure in order to refuel the NGV at the pressure required. Compressors are needed in the compression process at the refueling facilities. A new compressor design for natural gas refueling appliance has been developed which is the symmetrical multistage wobble-plate compressor. This compressor design is the newest variation of the axial reciprocating piston compressor. The success of the compressor design in compressing gas depends on the piston ring assembly design. Through this paper, the process of designing the piston ring assembly and considerations taken for this new compressor design were explained. The results presented are those from preliminary tests using air on the working fluid. Real tests on natural gas are to be organised utilising all the experience and lesson learnt from that on air.

Keyword: Compressor, wobble-plate, NGV, piston rings, oil-free, leakage

1. Introduction

An alternative to using gasoline and diesel as vehicle fuel is the use of natural gas. Compressed Natural Gas (CNG) offers fuel cost savings to the vehicle owner, as well as greater efficiency of energy resource utilization, clean burning fuel and lower emissions than gasoline or diesel fuel. The total energy use for Natural Gas Vehicles (NGV) includes not only direct vehicle consumption but also extraction, processing, transportation, distribution and compression of the gaseous fuel. As more public refueling stations are constructed the cost of fueling a natural gas powered vehicle will become more economical and convenient. In line with this, more natural gas vehicle refueling facilities are being built by Petronas (Malaysian national petroleum company) to serve this demand. Currently there are 39 NGV refueling stations nationwide. The government is targeting for 94 refueling stations by the year 2009 to serve a total of 57,000 NGVs. However, using natural gas as the vehicle fuel requires different refueling and storage technology as it involves compressing the gas to a high working pressure up to 210 bars to ensure compact storage.

The reciprocating wobble-plate compressor is one of the axial piston compressor variations. The wobbling motion from the wobble plate is transferred to pistons via connecting rods. A typical existing wobble-plate compressor for air-conditioning systems uses one common piston size for single stage compression (Figure 1). The piston size is adequate to accommodate connecting rods with ball joint connections made using caulking process. A new symmetrical multistage wobble-plate compressor being developed at UTM for natural gas compression application utilizes five-stage compression to achieve 210 bars [1,2,5,6] (Figure 2). This leads to five different piston sizes, with the size getting smaller for higher pressures. The NGV refueling compressor designed is for a medium capacity of 10 Nm^3/hr, and could be used in mini stations for housing blocks, office clusters, vehicle fleet depots or shopping malls. It is not intended for fast refueling such as in full-fledged commercial stations with high throughput, but it would provide a faster refueling than overnight home refueling appliances.

The main components of a new multistage symmetrical wobble-plate compressor for such a mini station would cover the compression system, the driver system, and dispenser and controls. The refueling unit is generally a self-contained, oil-free outdoor appliance that will fill the vehicle gas storage cylinder at a pressure of 210 bars (3600 psig) within 0.5-1 hours, giving an average mass flow rate of 0.67 kg/min. With cascading storage tanks at the mini stations being constantly pressurized, NGVs would not have to wait long to refuel. The specification data of a new multistage symmetrical wobble plate compressor are given Table.1

Mechanical friction losses in piston assembly amount to approximately 40% in the "piston-ring-liner" tribological system. The high friction in piston rings makes it very important for proper design to minimise these losses. Traditionally, piston rings for reciprocating motion were made from cast iron. In this case of using cast iron with steel counterface, lubricating with oil becomes necessary to reduce friction and as part of the cooling system. Lubrication oil film also functions to prevent leakage between piston ring and cylinder liner. However, in the case of compressing natural gas into vehicle tank, it is preferable not to have oil lubrication so as to avoid oil contamination of the gas that would in turn affect fuel performance.
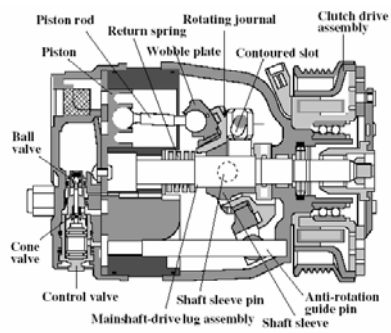
**Figure 1 Existing Wobble-plate Compressor
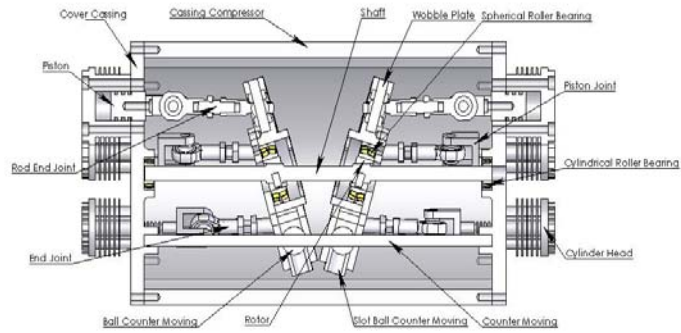(Single Side Compression)**

**Figure 2 The New Multi-stage Symmetrical
Wobble-plate Compressor**

**Table.1 Specification data of a new multistage symmetrical wobble plate compressor**

Input Data	Reference Value				
	First Stage	Second Stage	Third Stage	Fourth Stage	Fifth Stage
Cylinder diameter	39 mm	28.25 mm	20.47 mm	14.83 mm	10.74 mm
Suction Pressure	50 psi	113.4 psi	257.176 psi	583.26 psi	1322.79 psi
Discharge Pressure	113.4 psi	257.176 psi	583.26 psi	1322.79 psi	3000 psi
Stroke	47.96 mm				
Fluid	Natural gas				
Rotating Speed	1500 rpm				
Power	19 kW				
Tilting Angle	16°				
Pressure ratio	2.698				
Capacity	10 Nm³/hr				

## 2. Material Selection for Piston Rings Assembly

The piston ring assembly are the most critical parts to be designed to ensure good sealing with lower friction, low leak rate, long life and high efficiency. In these high pressure and temperature conditions, material selection is very critical for the piston ring and liner. The polytetrafluoroethylene (PTFE) based material with additive fillers was adopted for this compressor. The PTFE based material is very suitable for oil-free/unlubricated slidings because this material has low friction and wear, low thermal expansion, low shear strength to ensure rapid transfer to the counterface and low friction, leading to longer maintenance intervals, and wide range performance suitable for many operating conditions if filled with fillers i.e. carbon, graphite, bronze, molybdenum disulphide (MoS₂), fiber glass, etc [3]. For successful operation, the cylinder liner as the counterface to the piston rings must be designed suitable to the piston ring material. The liner was designed as smooth as possible with surface roughness ra = 0.088 and Vickers Hardness of 124 MPa.

3. Gas Pressure and Forces Acting on the Piston Rings

Figure 3 illustrates the simplest form of piston ring/seal (*a*), and a cross-sectional view of the ring, installed in the groove of a piston, located in sealing position (*b*), and in a neutral position (*c*), but in contact with the cylinder liner. When the ring is in the operating position the external face of the ring presses against the wall of the cylinder due to its built-in tension, which forces the open ends of the ring to stretch outwards, thus establishing the so-called primary seal contact. Once pressure is applied the ring is pushed against the side wall of the opposite side of the piston groove where it reaches its secondary contact position on the downstream side. Forces acting during compression for each piston or stage are shown below (Figure 4):
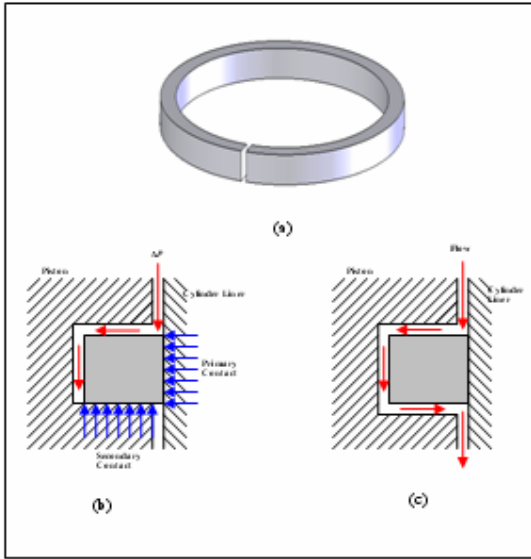
Figure 3 Piston ring pressure and gas flow.
(*a*) Piston square ring (*b*) Ring in sealing position
**(*c*) Ring in neutral position**
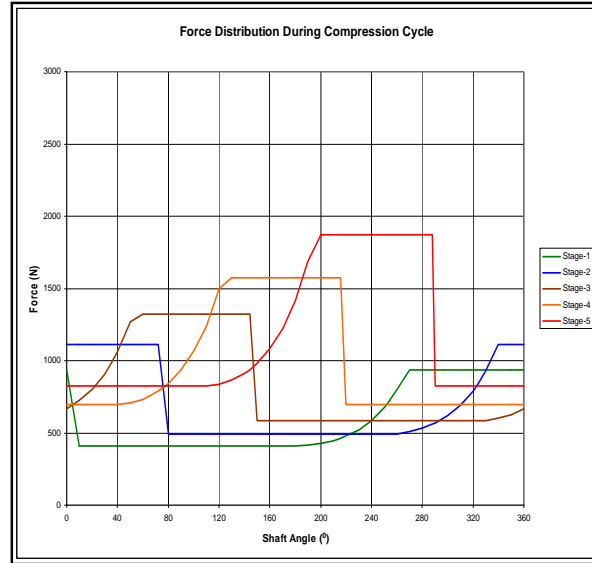
Figure 4 Force Distribution During Compression

4. Gas Leakage of Piston Rings

The design of reliable piston rings for oil-free applications with good characteristics as regards friction and leakage is presented here. Basically the main function of piston rings is to prevent gas leakage under compression with low friction and wear. Usually, each piston has a minimum of two piston rings installed. To minimize side force effects and to take the weight of the piston, rider rings are also incorporated in the whole assembly, preventing rubbing between the piston and the cylinder liner. The performance of piston rings depends on many parameters involved during sliding. Some of the major variables can be identified as: load or mean effective pressure, piston velocity, piston and liner material, gas composition, operating temperature, and surface finish.

Predicting gas leakage during operation is very complicated and difficult to measure. Theoretical estimations may be used for some of the solutions [4]. In an oil-free gas compressor there are three possible paths of gas leakage through the rings (Figure 5): (a) Between the rings and the surface of the cylinder liner, (b) Between the rings and the groove of the piston, and (c) Through the gaps of rings.
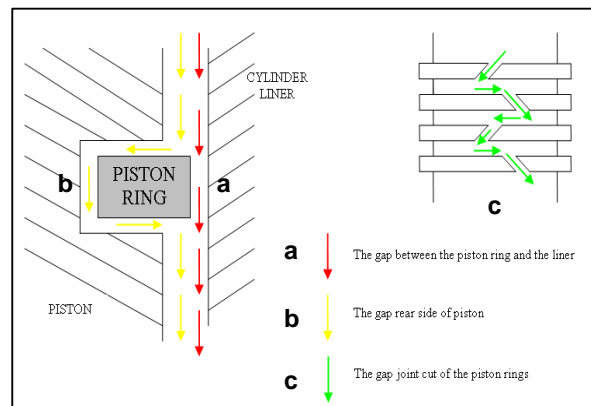
Figure 5. Three possible paths of gas leakage through the rings

For the piston with several rings the total leakage mass flow ($\dot{m}_i$) through the ith ring become:

$$\dot{m}_i \quad = \quad \dot{m}_{ai} + \dot{m}_{bi} + \dot{m}_{ci} \qquad \qquad \text{...(1)}$$

The flow through the gaps of the piston rings is presumed as a one dimensional compressible isentropic flow. Therefore the gas leakage through the gap of ith ring would be written as:

$$\dot{m}_{ci} \quad = \quad \frac{A_i . P_{i+1}}{T_{i+1}} \sqrt{T_i \left[1 - \left(\frac{P_{i+1}}{P_i} \right)^{\frac{k-1}{k}} \right]} \qquad \qquad \text{... (2)}$$

where $\quad A_i \quad = \quad \alpha f_i \sqrt{\dfrac{2k}{R(k-1)}}$

But if the flow speed in the gap equals the speed of sound,

$$\dot{m}_{ci} \quad = \quad \frac{B_i P_i}{\sqrt{T_i}} \qquad \qquad \text{... (3)}$$

where $\quad B_i \quad = \quad \alpha f_i \sqrt{\dfrac{2k}{R(k+1)} \left(\dfrac{2}{k+1} \right) \left(\dfrac{2}{k-1} \right)}$

The flow between the piston rings and the cylinder liner and between the piston rings and the piston groove are considered as flow in a thin clearance between two smooth surfaces. This problem can be solved using two dimensional incompressible viscous laminar flow theory. Navier-Stokes equation was used to predict the leakage between the rings and the cylinder liner ($\dot{m}_{ai}$), and the leakage between the rings and the piston groove ($\dot{m}_{bi}$):

$$\dot{m}_{ai} \quad = \quad \frac{\pi D \, \partial_a^3 \, (p_i^2 - p_{i+1}^2)}{24 \, \mu \, R \, T_i \, h} \qquad \qquad \text{... (4)}$$

$$\dot{m}_{bi} \quad = \quad \frac{\pi \, \partial_b^3 \, (p_i^2 - p_{i+1}^2)}{12 \, \mu \, R \, T_i \, \ln \left(\dfrac{D}{D-2b} \right)} \qquad \qquad \text{... (5)}$$

Now $\quad C \quad = \quad \dfrac{\pi D}{24h}$

and $\quad E \quad = \quad \dfrac{\pi}{12 \ln \left(\dfrac{D}{D-2b} \right)}$

The total leakage through ith ring is thus:

$$\dot{m}_i \quad = \quad \frac{A_i . P_{i+1}}{T_{i+1}} \sqrt{T_i \left[1 - \left(\frac{P_{i+1}}{P_i} \right)^{\frac{k-1}{k}} \right]} + \frac{C \partial_a^3 + E \partial_b^3 \, (p_i^2 - p_{i+1}^2)}{\mu \, R \, T_i} \qquad \qquad \text{... (6)}$$

This formula to predict flow leakage through piston rings is usually used for butt joint type of piston rings (Figure 6). Several types of cut joints were designed to minimise leakage. Scarf joint and step-cut joint were popular shapes for use in high-pressure oil-free compressors. These joints are more easily cut, installed and removed compared with the gastight joint construction which costs up to twice as much. In accordance with the sealing function required, piston ring joints can be shaped to any degree of complexity within the bounds determined by the

material properties and ring dimension. To minimise the leakage between piston ring cylinder liners, ring backup springs can attached behind the ring to apply a slight amount of pressure (2-3 psi) and assist the piston ring in establishing the initial seal.
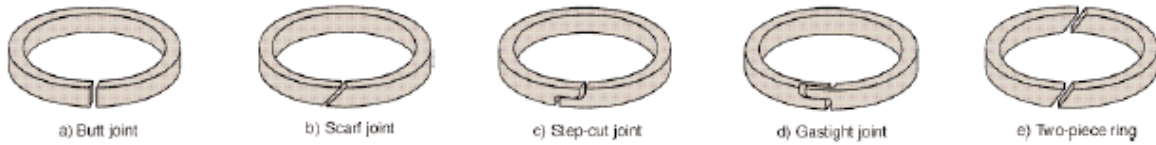

| a) Butt joint | b) Scarf joint | c) Step-cut joint | d) Gastight joint | e) Two-piece ring |

Figure 6 Various types of cut joints for piston rings

4. Ringless Labyrinth Plunger Piston – For Last Stage

In case of the last, high-pressure stage where pressure was raised to 3000 psi, many piston rings are needed in order to reduce the load per ring. For this stage, if piston rings are to be included, more than six piston rings would be needed, thus making the use of piston rings not practical because of the complex structure and limited space to install them (since the diameter is very small, 10 mm). Therefore, the established concept of plunger pistons without piston rings was adopted for the fifth (last) stage instead.

Initially the design for this last stage used the plunger piston concept with PTFE cylinder liner. The plunger concept requires tight clearance between the piston and the liner [1]. Piston side force caused scuffing on the liner surface by the plunger piston edges. The liner covered only a portion of the piston during sliding motion especially at bottom dead centre (BDC) where maximum moment occurred at the fulcrum point at point a between the piston and the piston liner as shown in Figure 7. Taking the moments about point a, the force acting on the piston liner at point b is given as:

$$F_b = F_{crti} \sin \phi_i \left(\frac{L_{out}}{L_{in}} \right) \qquad \qquad \dots (7)$$

Where $F_a = F_{crti} \sin \phi_i + F_b$

The optimum clearance between the piston and the liner is 5-6 µm. As PTFE is softer compared to the piston material, this clearance gets bigger as the compressor runs during initial testings as a result of F_a and F_b forces acting on the inner surface of the cylinder liner. Heat generated due friction also caused the liner material to soften. This caused more deterioration due to scuffing by the piston edge and hence enlarging the clearance. Greater clearance increases piston degrees of freedom which in turn lead to more scuffing. Scuffed liner material was pushed in front of the piston head filling the clearance volume available. Thus, it was crucial to reduce F_a and F_b forces acting on the cylinder liner. The maximum values of F_a and F_b at each stage is given in Table 2.

This scuffing problem has been solved by using the piston ring concept in combination with hard-chromed cast iron cylinder liner. Cast iron liner is much harder compared to PTFE and could withstand high temperature. Sealing can also be achieved using piston rings which slide along the smooth hard-chromed surface of the liner bore. One alternative solution is using crosshead concept at piston coupler to minimise the side force effect.
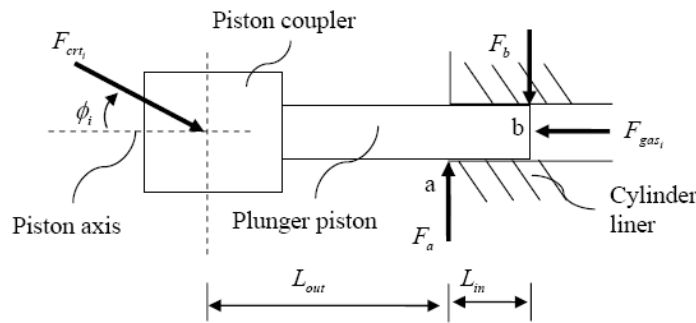

Figure 7 Load acting on the piston

5. Prototype Test Results

A series of tests were done after the compressor has been developed and fabricated. First, mechanism tests were done without pressure loads, to check whether the mechanisms work properly or not. Then pressurised tests with load onto the system were conducted to check the joints and pipings for leakage. Finally, performance tests at full load, with 3 bars suction pressure and varying operating speeds were looked into [7]. Some issues arising during the tests, and measures taken to overcome them, are given in Table 3. The results of pressure versus time the latest test are shown below:
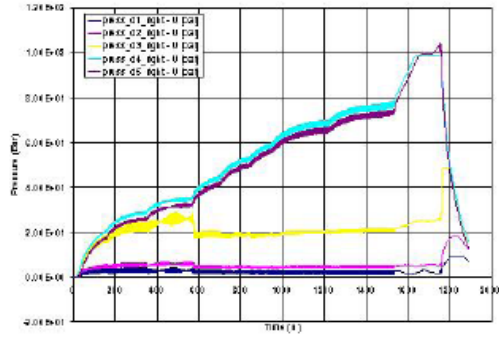
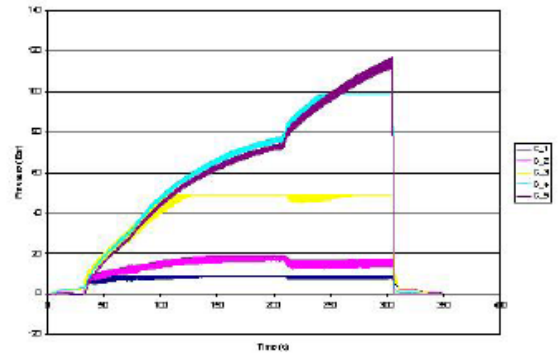**Figure 8 Result test at suction pressure 1 bars and speed 600 rpm**

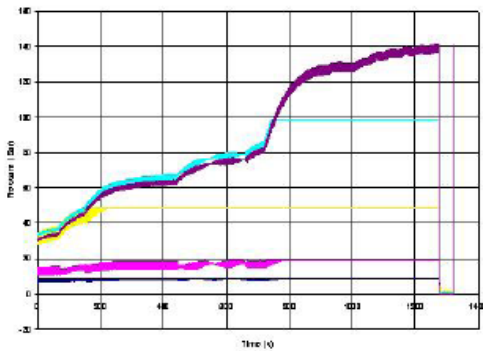**Figure 9 Result test at suction pressure 3 bars and speed 250 rpm**

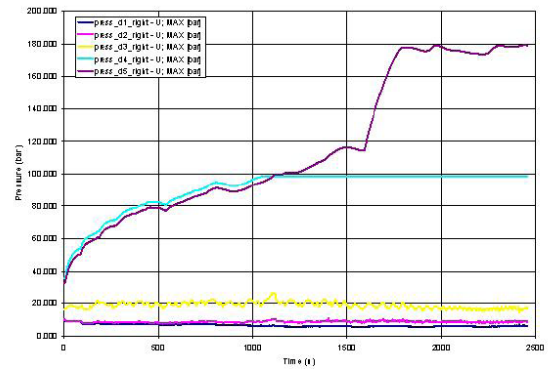**Figure 10 Results of tests at suction pressure of 3 bars and speed of 400 rpm**

Figure 11 Results of tests at suction pressure of 3 bars and speed of 700 rpm

Table 3 Some issues arising after testing the compressor and improving the design

Issues	Solution	
Excessive bending moment due to piston side force	• Change 5th stage piston material to XW41 tool steel • Use crosshead concept to reduce the effect of piston side force on the 5th stage	Bending on Piston
Internal leaking by the piston ring at it's cutting edges (split piston ring is easier to fabricated, installed and cost less)	Change piston ring design to uncut piston ring design	

7. Conclusion

The concept of a multi-stage symmetrical wobble-plate compressor was successfully designed, fabricated, and tested. In an oil-free gas compressor there are three possible paths of gas leakage through the rings; between the rings and the surface of the cylinder liner; between the rings and the groove of the piston and through the gaps of rings. The mathematical models of gas leakage through piston rings assembly can be satisfactorily adopted to describe the working processes in a cylinder. However, there were several design issues regarding the structural integrity of the piston rings i.e. piston side forces, manufacturing and tolerance issues. For the fifth stage the pistons need to be further improved. In overall, the compressor design is promising for natural gas vehicle refueling usage.

8. Acknowledgement

9. References

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10. List of Symbols

A	=	Flow area of valve
b	=	Width or thickness of piston ring
C	=	Diametrical tolerance between the piston and the cylinder liner
D	=	Diameter of cylinder
f	=	Flow area of gap of piston ring
h	=	Height of piston ring
k	=	Isentropic index
$\dot{m}$	=	Mass flow rate
P	=	Gas pressure
R	=	Gas constant
T	=	Gas temperature
α	=	Flow coefficient
δ	=	Average clearance of contacting surface
μ	=	Gas velocity