Design and Development of a Variable Rotary Compressor

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ABSTRACT

The following paper describes the design and development issues with a variable capacity control 2-stage rolling piston compressor. The control mechanism was based on a gas-bypass control strategy. During the development, design issues surfaced that were not significant in the case of the full capacity version of the compressor. When the capacity was reduced, loss generation mechanisms remained constant. These resulted in higher discharge temperatures and lower compressor efficiencies. Improvements were made to reduce losses in line with capacity reductions.

1.0 INTRODUCTION

The development of a rotary variable capacity compressor is discussed in this paper. An initial concept was developed and tested. It was found at the time, that there was significant room for improvement. This paper discusses the modifications that were made to improve the functional performance of the compressor.

2.0 BASE COMPRESSOR

The base compressor was a fixed capacity two stage sliding vane compressor, as shown in Fig. 1. Compression to an intermediate pressure occurred in the outer compression chambers. After passing through an intermediate chamber, a second stage of compression was carried in the inner volumes.

Suction ports in the two first stage vanes were kinematically timed to optimize flow into the suction cavity. Two discharge reed valves in the first stage allowed flow into an intermediate cavity. Ports in the two second stage vanes allowed flow into the second stage. Compression occurred and the flow was exhausted through discharge ports.

The capacity of the base compressor was given by the following equation. It was dependant on the suction volume of the first stage.

\[ m_{\text{comp}} = \eta_{\text{suc}} \cdot \rho_{\text{suc}} \cdot V_{\text{stag1}} \cdot \text{RPM} \]

3.0 VARIABLE MECHANISM

The ratio of the first to second stage volumes was approximately 2.2. Given this ratio, it was determined that a 40% variable compressor could be developed using gas by-pass from the intermediate plenum. If full by-pass were accomplished, the compressor capacity would now be:

\[ m_{\text{comp}} = \eta_{\text{suc}} \cdot \rho_{\text{suc}} \cdot V_{\text{stag2}} \cdot \text{RPM} \]
As can be shown in Fig. 2, the effect of creating a flow path from the intermediate plenum to the suction chamber was to reduce the intermediate plenum pressure. This in turn limited the capacity of the compressor to the second stage suction volume.

The valve that was used is a by-pass valve, the function of which is described in Mabe [1]. It regulates the bleed back based on the suction pressure seen by the A/C system.

The compressor was built and tested without any modifications. The compressor displayed worrying efficiency trends at the lower capacities, see Fig. 3. Based on the discharge temperature measurements and COP, it was obvious that more work was required to improve the efficiency.

The efficiency of the baseline compressor was competitive at 100% capacity, Huang [2]. The by-pass strategy at 40% capacity revealed glaring deficiencies in the compressor design that were otherwise not observed.

By the end of the study, the performance of the compressor was improved significantly although it did not quite reach the performance of the competing technology. These improvements involved:

a) improving gas flow within the compressor
b) reducing friction sources

4.0 Flow Improvements

A previous study of flow through the first stage discharge valve had shown that it was going to be a problem.

Two transducers were used. They were placed axially above the discharge ports as shown in Fig. 4. Another transducer was placed in the intermediate plenum.

4.1 Throttling at valve: In the baseline configuration, pressures ran as high as 100 PSIG in the compression chamber, over-shooting the intermediate chamber pressures by 30 PSI.

By raising the reed stop height by 50% and increasing the port area by 20%, the port throttling was substantially reduced. The over-compression was reduced to an average of 15 PSI.

As shown in Fig. 3, these changes resulted in a marked improvement in the efficiency.

4.2 Stiction: The phenomenon of valve stiction can be seen graphically in Fig. 5. The valve was not instrumented for motion, however the motion can be inferred from the pressure measurements. At valve opening, there is a change in the two chamber pressures measured. This is to be expected, the valve nearest the port should have the lower pressure due to a Bernoulli effect as the flow adjusts.

Efforts were made to reduce stiction. Increasing the port size helped in two ways. The total force...
on the valve (=pressure x area) increased, while the contact area (=valve area - port area) was reduced.

In addition to this, miniature slots were machined under the valve to help reduce stiction, similar to Sabha [3]. In this case however, the function of the slot was purely to reduce stiction. The slot groove depth was on the order of a few microns deep.

![Outline of valve](image)

**Fig. 6 Stiction Reducing Port**

As can be seen in Fig 5, all the efforts to reduce stiction did not result in significant improvements in the performance. Neither the time to open nor the overcompression were substantially reduced.

Khalifa and Liu [4] discussed in depth the stiction phenomenon. It would have been beneficial to analyse the effects of oil viscosity. However, oil viscosity reduction was limited by lubrication and sealing requirements at other operating conditions (i.e. high pressure/low speed operation). Lower contact area between the valve and the port reduced marginally the valve time opening.

4.3 Squeeze: As the aspect ratio increases in the compression cylinder, there is a noticeable separation in the pressure profile. With discharge ports mounted in the rear of the compressor for design efficiency, this was always an inherent design problem. As can be seen however in Fig. 5, the total inefficiency due to squeeze was not substantial. It can be seen, that the problem occurred in both the baseline and improved designs.

Neiter et al [5] discuss this phenomenon in detail. They suggest a tapered cutout to reduce over compression due to the squeeze effect.

5.0 Friction Reduction

Friction was known to be a significant contributor to the performance of the compressor. Previous studies had shown friction to account for 15% of power consumption at 100% capacity. Based on the initial test results, there was evidence that this component of losses was not reducing in line with the flow reductions.

5.1 Vane Tip: The baseline vane tip saw several contact modes, ranging from point contact, linear contact to sharp counterformal contact as shown in Fig 7a. It was known that this scenario was not good for the vane friction or vane following, Kawahara et al [6].

![Shape of Vane Tip](image)

**Fig. 7 Shape of Vane Tip**

The vane tip was re-shaped so that at every angle of rotation, a consistent counterformal contact was achieved, Fig. 7b. The advantages included:

- The largest counterformal contact that is geometrically possible, improving the ability to form a lubrication layer
- More even wear across the entire tip surface and more even surface velocities

5.2 Vane Side: The slot and vane width were precision machined to avoid cocking of the vane in the slot. This tight tolerance however created unnecessary high friction loads due to oil shear.
A large groove was machined into the side of the vane on the low load side of the vane. This detail, although only a few microns deep would double the gap width between the vane and slot wall, thus reducing the overall oil shear friction.

![Diagram of friction reducing vanes](image)

In this paper, we saw several instances where slight design modifications had a significant impact on the efficiency of a compressor when operating at part load (variable mode). The design steps that made the compressor significantly more competitive were clearly defined.

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Compressor turned on

Valve starts to open, Pi drops

Fully open valve, Pi and Ps drop together

Suction hits control press

Clutch cycles off

Time

Fig. 2 Valve Operation and Effect on Intermediate Pressure

Fig. 3 Compressor Discharge Temp at 40% Capacity Vs. Baseline Technology
Fig. 5 Baseline Versus Improved PV Chart for Discharge Port