High Performance Valve System

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HIGH PERFORMANCE VALVE SYSTEM

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ABSTRACT

This paper covers the design and performance of a radically new valve system now being used in semi-hermetic compressors. The discharge valve, trademarked the "Discus Valve," is made from a high temperature polymeric material. This valve, together with a unique laminated steel valve plate and a ring-type suction valve, provides large flow areas, low clearance volumes, and high efficiency.

The development of this valve system required the determination of the long-term physical integrity of the valve material and the design of suitable spring and retention systems. The valve plate development required optimization of performance, durability and machinability. In addition to details of these developments, the paper describes the long-term reliability tests run on components and assemblies to guarantee that reliability as well as performance goals were met.

INTRODUCTION

In the commercial refrigeration market, reliability is of primary importance, since the failure of a single compressor can result in thousands of dollars worth of spoiled merchandise, and it is not uncommon for the compressors to operate virtually non-stop for 10 years or more. Discus compressors were developed to meet this market's demand for reliability together with improved energy efficiency. The driving force for efficiency improvement is increasing electrical costs. A typical supermarket will spend $40,000 per year for its refrigeration requirements, and high efficiency compressors such as the Discus pay for themselves in very short order.

Prior to Discus compressors going into production in September of 1982, many semi-hermetic compressors were built with conventional finger-type suction and discharge valves, seated on a series of circular ports in the valve plate. This is a very reliable design that inherently has high clearance volume due to unswept volume in the discharge ports. Development of Discus valving was intended to minimize clearance volume, while maintaining ample suction and discharge port areas.

DESIGN GOALS

Design goals for Discus compressors are:

1. <u>Reliability</u> - The reliability of Discus compressors had to be consistent with requirements in the marketplace. The design life was targeted at 15 years.
2. <u>Efficiency</u> - Minimum acceptable BTU/WATT-HR values were established for low, medium and high temperature operation, for all refrigerants.
3. <u>Adaptability</u> - Discus compressors were developed by applying new valving to existing running gear. The intent was to apply the concept to four different lines of compressors. Since these four compressor lines had eight different bore sizes, it was of major importance that the design be flexible.
4. <u>Capacity</u> - Proportional capacity increases were expected for all models, but specific capacity levels were not targeted.
5. <u>Cost</u> - Cost targets were established for all models based on a dollar per cylinder figure.
6. <u>Refrigerants</u> - All four compressor lines had to be capable of operation on refrigerants 22, 502 and 12.
7. <u>Capacity Control</u> - Three of the four compressor lines had to have capacity control capability. This capacity reduction was to be achieved by blocking the return gas on two lines while the third line used a discharge gas bypass method.

8. Sound and Vibration - When compared on a capacity weighted basis, Discus compressors were to have no higher sound or vibration levels than existing product.

9. Agency Approvals - Underwriters' Laboratories and Canadian Standards Association approvals were required on all models.

10. Size - No dimensional increases were allowed over present models and connection locations had to remain the same.

DESIGN APPROACH

The Discus design concept was simply to develop a valve plate assembly that when applied, results in minimal clearance volume and at the same time minimizes inlet and discharge flow restrictions. In the conventional valve plate design shown in Figure 1, it is possible to minimize flow restrictions, but in the process, clearance volumes are increased and the porting can get so congested that discharge valves start to block suction ports and vice versa. The Discus design is shown in Figure 2. Concentric porting is used to make maximum use of the available area. The discharge port adds little to the overall clearance volume. This is accomplished by using a multi-layered valve plate having conical-shaped discharge valve seats. The seats are impressed to a level where the discharge valve, in the closed position, is flush with the bottom surface of the valve plate. The ring-type suction valve seat is concentric to the discharge valve seat and located so as to achieve maximum flow area for minimum span. This is the key to the high operating efficiencies of Discus compressors. It is also a key to a major reduction in the number of components needed to cover the four lines of compressors. Only one discharge valve and two suction valves are used on the 36 compressor models in the four lines.

DEVELOPMENT DETAILS

As one might expect, taking a design concept such as the Discus through development and into production was no simple task. It took approximately five years to achieve our goals. I would like to review the Discus design in detail and discuss some of the development problems that were encountered along the way. The major Discus components are shown in Figure 3.

A. Valve Plate

Discus valve plates are unique furnace brazed assemblies made from three main stampings and varying numbers of internal supports. After brazing, the assemblies are heat-treated. Final machining consists of grinding the top and bottom surfaces and turning the conical discharge valve seat. Cleanliness is extremely important for reliability, so valve plates are power flushed internally after final machining.

B. Suction Valve

Discus suction valves are made of Swedish stainless steel and are a ring design with 180 degree separated guide tabs. They are free flexing valves with no mechanical stops. Stress analysis was done using finite element analysis along with actual stress measurements taken on valves operating in compressors. It was determined that a safety factor of at least 100% existed at any operating condition within the approved operating envelope. Liquid slugging tests were also run, to determine suction valve adequacy when subjected to pressure loading of up to 2000 PSI.

C. Discharge Valve

The Discus valve is made from a polyimide resin, much the same as you would manufacture a powdered metal part. This material was selected because of its strength, thermal, and wear characteristics and its ability to conform to varying seat contours. We had to generate data on fatigue life, creep characteristics, and compatibility with refrigerants. After evaluating many Discus designs, it was found that a simple shape is optimum for both flow restriction and fatigue properties. Any surface discontinuities result in sizeable reductions in fatigue life.

D. Curved Disk Springs

Curved disk springs are used to load the Discus valve and are made from the same Swedish stainless steel as the suction valves. They are assembled in stacks of three to achieve the desired loading and at the same time limit operating stress. Curved disk springs were far superior to other springs tested because of their compactness and damped operating characteristics. Fatigue life and wear were again major concerns.

E. Wear Disk

A flat disk of Swedish stainless steel is placed between the Discus and the lower curved disk spring. The flat wear disk shields the top surface of the Discus from metal particles.

F. Bridging Retainer

The bridging retainer acts as the valve action control mechanism. It guides the moving parts and controls the amount of discharge valve opening. From the standpoint of wear control, its surface finish is very important to the springs and Discus valve. During normal operation in a compressor, this element sees a maximum load of 160 pounds. During slugging, this load can increase to approximately 900 pounds. Hydraulic fatigue bench tests were used to determine that the bridging retainer and retainer bolts were capable of withstanding this cyclic loading over the expected compressor life.

G. Valve Plate Assembly

A valve plate subassembly is made prior to compressor installation. The subassembly consists of a valve plate, Discus valves, wear disks, springs, bridging retainers, and retainer bolts. A vent hole in the center of the bridging retainer is used to align it to the Discus valve seat. Two tamperproof bolts are used to secure the bridging retainer. Tamperproof bolts are used to discourage disassembly by anyone that might not realize that alignment would be required for proper reassembly.

H. Compressor Assembly

Clearance volume of an assembled compressor is controlled by proper selection of deck gasket thickness. Measuring the piston heights gives the necessary information for selecting the proper deck gasket thickness for a given cylinder bank.

OPERATING CHARACTERISTICS

We will now take a look at the actual operating differences in a Discus compressor as compared to Copeland's conventional designs.

The test data shown in Table I is from a three-cylinder 1750 RPM machine with a displacement of 2120 cubic feet per hour. It is operating on Refrigerant 502 at 105 degrees condensing temperature, -40 degrees evaporating temperature, 65 degrees return gas temperature, and with 5 degrees of sub-cooling. This condition results in an operating pressure ratio of 13.1 to 1.0. Its capacity is 24900 BTU/HR with an energy efficiency ratio of 3.97. That performance equates to 59.7% volumetric efficiency and 59.2% of the theoretical EER. When Discus valving is applied to the same displacement compressor, the capacity increases to 29200 BTU/HR and the energy efficiency ratio increases to 4.46. This is 70.0% volumetric efficiency and 66.5% of theoretical EER. Discus valving increased volumetric efficiency by 10.3% and percent of theoretical EER by 7.3%.

A similar comparison is shown in Table II operating at a pressure ratio of 3.4 to 1.0 on Refrigerant 22. It should be noted that in this case the conventional compressor has a valve plate with more suction and discharge port area than in the R-502 example. The Discus valve plate is the same for both examples. The operating conditions are 130 degrees condensing temperature, 45 degrees evaporating temperature, 65 degrees return gas temperature, and with 15 degrees of sub-cooling. In this case, capacity increases from 167700 to 180000 BTU/HR. Energy efficiency ratio increases from 9.89 to 10.65 BTU/WATT-HR. These numbers represent a 5.5% increase in volumetric efficiency and a 4.7% increase in percent of theoretical EER.

PRESSURE - VOLUME DIAGRAM ANALYSIS

The Discus compressor had superior performance in both cases cited. However, its low temperature improvement was much better than at high temperature. To appreciate this, let's look at pressure-volume diagrams for both valving combinations. This comparison is made at a R-502 low temperature operating condition having an operating pressure ratio of 9.8 to 1.0.

The P-V diagram with conventional valving is shown in Figure 4. To obtain this data, a pressure transducer was mounted through the valve plate at the center of a cylinder. The transducer signal was stored via a processing digital oscilloscope. A timing gear was mounted on the end of the crankshaft and a proximity pickup was used to sense crank angle. The proximity pickup signal was also stored and the combined data established a pressure to crank angle relationship. By processing this data, it is possible to construct the pressure-volume diagram shown. The vertical scale is pressure in PSIA and the horizontal scale represents piston position in the cylinder as well as displaced volume. The two horizontal lines on the diagram are operating pressures, external to the compressor, at the discharge and suction connections. The extreme right-hand point on the diagram represents bottom dead center. Following the upper part of the curve is compression and discharge until you reach top dead center at the extreme left. Reexpansion and intake take place on the bottom part of the curve until you are back at bottom dead center. The areas bounded by various portions of the diagram are proportional to the work associated with each of the four segments of the cycle. The four segments are as follows:

A. Compression (Positive Work)
B. Discharge (Positive Work)
C. Reexpansion (Negative Work)
D. Intake (Positive Work)

In this example, 84.9% of the total work is used for compression, 8.4% is used for discharge, -9.1% is reclaimed from the expansion of high pressure gas trapped in the clearance volume, and 15.8% is used for intake. The time averaged pressures that occurred during the discharge and intake events were 301.0 PSIA and 15.1 PSIA, respectively. The discharge event required 9.5% of the compression stroke, while the intake event took place over 86.5% of the suction stroke.

An analysis of the P-V diagram for Discus operation shown in Figure 5 shows that 88.0% of the total work is used for compression, 6.7% is used for discharge, -7.2% is reclaimed, and 11.6% is used for intake. The time averaged discharge and intake pressures are 278.9 PSIA and 18.7 PSIA, respectively. The discharge event required 10.8% of the compression stroke and the intake event took place over 91.2% of the suction stroke.

The ideal compressor would be one which minimized discharge, intake, and reexpansion work. The result would be to maximize the percentage of total work being used for compression. Table III compares the values from the conventional and Discus P-V diagrams. Reexpansion work is reduced by 1.9% as a result of reduced clearance volume. This reduction causes an increase in volume flow. Discharge work is reduced by 1.7% as a result of lesser restrictive porting. This is substantiated by a 22.1 PSI lower time averaged pressure during discharge and results in higher efficiency. Intake work is reduced by 4.2% as a result of lesser restrictive suction porting as indicated by a 3.6 PSI higher time averaged pressure during intake. This also contributes to higher efficiency. The net effect of these differences is a reduction in input work per pound mass of refrigerant pumped of 7.1%.

RELIABILITY TESTS

All new products such as the Discus are qualified against reliability goals before it is put into production. This evaluation is made on a statistically significant sample size, that includes all final design features and components made from production tooling.

In the case of just one line of Discus compressors, 33 samples were subjected to seven different tests for each refrigerant and temperature range over which it was to be applied. This included such tests as high load and flooded start [1] [2]. Over 300,000 hours of reliability testing was done in this manner to qualify all four lines of Discus compressors.

CONCLUSIONS

Energy efficiencies were achieved with EER increases up to 16%, depending on refrigerant and application. Capacity increases of up to 25% were realized in comparable displacement machines. Field experience has been favorable. A pre-production supermarket installation has been operating trouble free for over two years and has achieved substantial savings in operating costs.

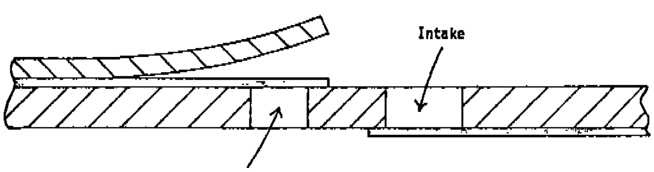

Fig. 1 Conventional Valving Design

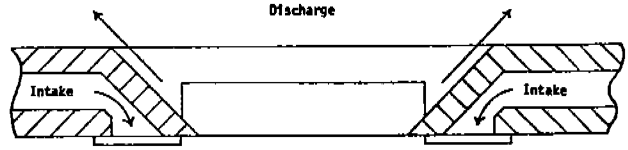

Fig. 2 Discus Valving Design

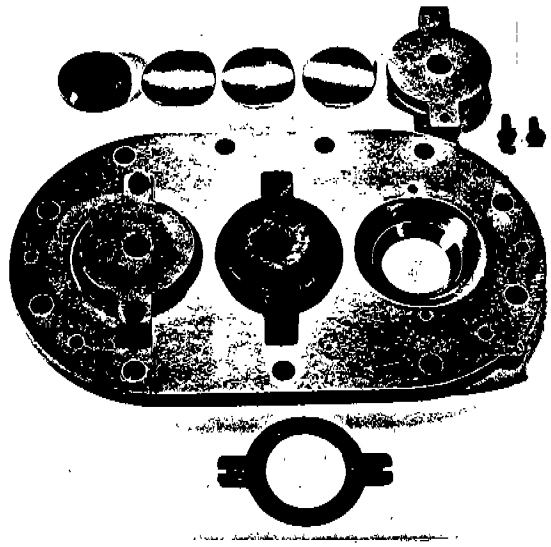

Fig. 3 · Discus Components

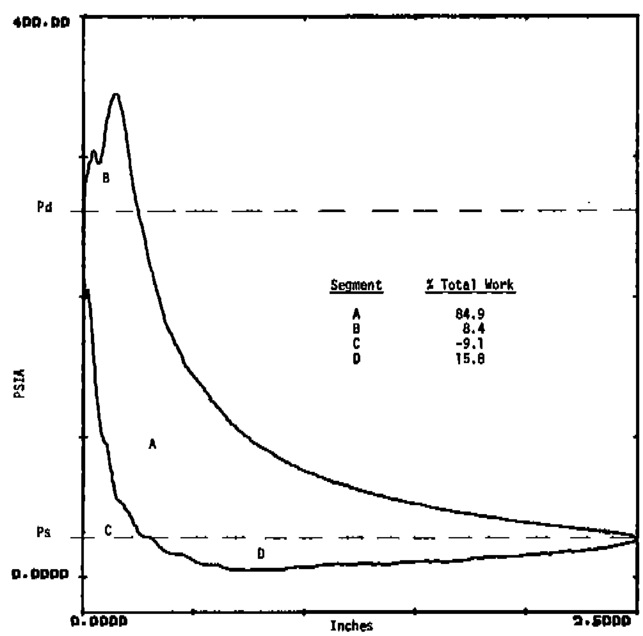

Fig. 4 P-V Diagram - Conventional Valving

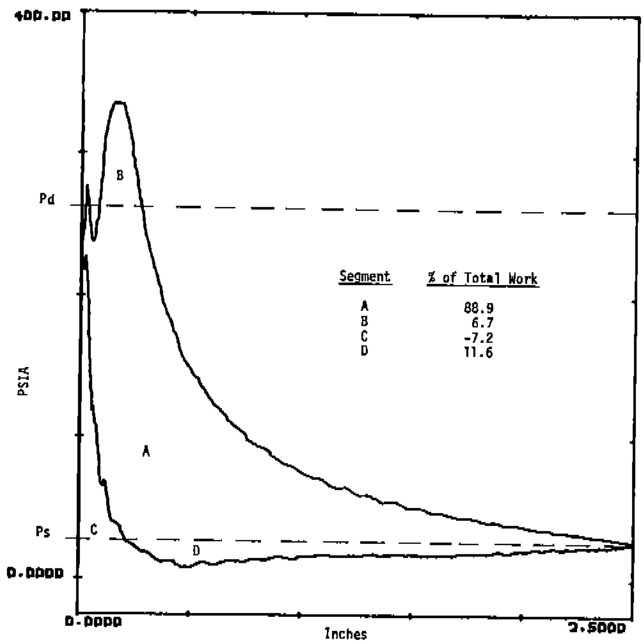

Fig. 5 P-V Diagram - Discus Valving

Table I

Performance Comparison

R-502 @105°/-40°/65° R.G./5° S.C./95° Ambient

	Conventional	Discus	Increase
Capacity (BTU/HR)	24,900	29,200	4,300
EER (BTU/WATT-HR)	3.97	4.46	0.49
Volumetric Efficiency (%)	59.7	70.0	10.3
% of Theoretical EER (%)	59.2	66.5	7.3

Table II

Performance Comparison

R-22 @130°/45°/65° R.G./15° S.C./95° Ambient

	Conventional	Discus	Increase
Capacity (BTU/HR)	167,700	180,000	12,300
EER (BTU/WATT-HR)	9.89	10.65	0.76
Volumetric Efficiency (%)	74.5	80.0	5.5
% of Theoretical EER (%)	60.0	64.7	4.7

Table III

P-V Diagram Comparisons

	Conventional	Discus	Change
Compression (% of total work)	84.9	88.9	+4.0
Discharge (% of total work)	8.4	6.7	-1.7
Avg. Discharge Pressure (PSIA)	301.0	278.9	-22.1
Reexpansion (% of total work)	-9.1	-7.2	+1.9
Intake (% of total work)	15.8	11.6	-4.2
Avg. Intake Pressure (PSIA)	15.1	18.7	+3.6
Input Motor Work (BTU/LB)	38.0	35.3	-2.7

REFERENCES

[1] F. E. Kalivoda and K. W. Yun, "Modeling Mechanical System Accelerated Life Tests," Proceedings of the 1976 Annual Reliability and Maintainability Symposium, IEEE, p. 206.

[2] F. E. Kalivoda, "Accelerated Testing of Compressors," Proceedings of the 27th Meeting of the Mechanical Failures Group, National Bureau of Standards (1977) p. 99.