An Aerodynamic Investigation of a Centrifugal Compressor for HCFC123

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ABSTRACT

The experimental investigation of a centrifugal compressor is carried out to improve HCFC123 compression performance. A CFC11 compressor applying HCFC123 shows a lowered adiabatic head rise, efficiency, and volume flow. Another CFC11 compressor with HCFC123 and a modified impeller shows a result comparable to the CFC11 compression case. A tandem impeller is designed to cover the volume flow increase for HCFC123. The performance of this impeller is sufficient to absorb the volume flow increase and additionally some efficiency improvement is obtained.

INTRODUCTION

Although CFC11 is widely used for centrifugal chillers, its production will be stopped because of its high ozone-depleting potential. As the HCFC123 is the best possible subsutitution refrigerant for CFC11, the development program for a HCFC123 centrifugal chiller was planned and implemented. During the program, testing was carried out to obtain the character of a 300 RT CFC11 centrifugal chiller filled with HCFC123. As a result, we experienced a decrease in capacity of ten odd per cent and an energy consumption increase between 5 and 10%. The direct cause of this performance degradation was presumed to be the difference in thermodynamic properties of CFC11 and HCFC123. The theoretical compression work and volume flow rate, per unit refrigerating capacity of HCFC123, are higher than those of CFC11. The heavier molecular weight of HCFC123 was considered to cause this degradation in compressor performance leading to overall lower chiller performance.

Akitani and Nakaya [1] experimented with CFC11 and HCFC123 in the same compressor. They concluded that an HCFC123 compressor with a CFC11 compressor performance level could be realized by an appropriate impeller design. With this background, an aerodynamic investigation was made to improve the compressor performance for HCFC123. The aim of this investigation is not only to improve energy consumption, but to develop a compact machine. HCFC123 compression performances with a CFC11 impeller and with a modified impeller are presented. The performance with a tandem impeller for higher flow rate is presented, too.

NOMENCLATURE

Had = adiabatic head rise
M = Mach number
Qs = suction volume flow rate (evaluated by stagnation density)
w = relative velocity
ηc = total to total stage efficiency
ηt = stage efficiency including mechanical losses

Subscripts
d = design point
1s = impeller inlet tip
2 = impeller exit

EXPERIMENTAL PROCEDURE

Experimental Apparatus

The experiments were made with a centrifugal compressor test system, consisting of a single stage compressor, a torque meter, a d-c motor, speed-up gears, a system of heat exchangers, and flowpiping. The outlook and the cycle of the system are shown in Figs. 1 and 2, respectively. Refrigerant is not condensed but chilled to the compressor inlet temperature by a brine cooled heat exchanger. This gas cycle test system has some merits. Heat control equipment is simplified and energy consumption is lowered. Added to this, only a small amount of refrigerant is necessary. A cross-sectional view of the centrifugal compressor is shown in Fig. 3. Two types of impellers were used in this compressor. They are described in the following section. The compressor casing was covered with insulating materials to eliminate the heat leakage effect on the total temperature measurement.

Measuring Method

The total pressure and the total temperature, at the suction and discharge sections, were measured with Kiel-type probes and total temperature probes consisting of copper-constantan thermocouples. The performance of the compressor stage was evaluated on the basis of the total-to-total rating. The flow rate was controlled with a valve between the heat exchangers and was measured with an orifice-type flow meter in the suction pipe.

RESULTS AND DISCUSSIONS

HCFC123 Compression by a CFC11 Compressor

As the molecular weight of HCFC123 is 11% heavier than that of CFC11, it is easier to compress HCFC123. This difference between refrigerants is considered to cause several problems when HCFC123 is applied in a CFC11 compressor. These problems are:

1. Decrease of the choke flow rate caused by the lower sonic velocity of HCFC123.
2. Drop in impeller efficiency caused by the increase of the diffusion ratio ($w_1 s/w_2$).
3. Diffuser pressure recovery drop caused by severe velosity distortion at the impeller exit.

All these problems can be solved by an impeller correctly designed for HCFC123. It is, however, interesting to see how great is the performance deterioration. The combination of a fully shrouded model impeller and a parallel wall vaneless diffuser was used. The design pressure ratio of this combination is 3.5 . The impeller had 14 blades with backswept angles of 50 deg. from the radial direction at the exit and a diameter of 265 mm (10.43in.). The diameter of the vaneless diffuser was 376 mm (14.8 in.), and the width of the diffuser was the same as the impeller exit width. The impeller is shown in Fig. 4 . The results of experiments are shown in Fig. 5. The compressor inlet pressure for CFC11 ranged between 20 and 30 kPa (2.9 to 4.4 psia) and that for HCFC123 between 18 and 28 kPa (2.6 to 4.1 psia). The compressor inlet temperature was between 276.2 and 279.2 K (37.4 to 42.8 °F) for both refrigerants. The difference in performance is summarised in Table 1, based the levels of CFC11. The choke flow rate for HCFC123 decreased by 8% and this reduction nearly corresponded to the drop in sonic velocity. The head rise at the surge point dropped 6% and the maximum efficiency dropped by 7% for HCFC123. The flow rate at the surge point for HCFC123 was lower than that for CFC11, contrary to anticipation, and the ratio of choke flow rate to surge flow rate reduced only by 3%. The unexpectedly wide flow range seemed to suggest that increase of the impeller diffusion ratio and the deteriorated impeller exit flow distribution did not directly lead to an earlier stall onset of the impeller and diffuser. A comparison of the flow situation in the impeller is made in Table 2. The diffusion ratios were calculated at Qs/Qsd = 0.95 by one-dimensional analysis [2]. The diffusion ratio for HCFC123 is 15% larger than that for CFC11 and the inlet tip Mach number is 8% higher. The inlet tip Mach number for HCFC123 is 1.16, considered to be a high level for a radial impeller. This high Mach number could be a cause for the performance drop of HCFC123 through the shock wave generation added to the effect of increase of the impeller diffusion ratio. To clarify the dominant effect, another experiment was made and its results are reported in the following subsection.

<u>Improvement of Efficiency by Impeller Diffusion Ratio Adjustment</u>

A trial was made to improve the dropped efficiency of HCFC123 compression. The test impeller was a half-shrouded type with 18 blades and was originally designed for CFC11. Design pressure ratio was 3 when a vaneless diffuser was used. Its diameter was 300 mm (11.81 in.) and the exit backswept angle was 30 deg. from the radial direction. The impeller diffusion ratio was controlled by decreasing the original exit blade height. The meridional shape is shown in Fig. 6. A comparison of calculated diffusion ratios and Mach numbers at the design condition is shown in Table 3. The suction volume flow rate at the design point and the rotational number were the same for CFC11 and for HCFC123. The Mach number level was the same as that of the fully shrouded impeller presented in the previous subsection. In this experiment, a channel diffuser was combined to aim for a higher efficiency. The installed impeller and diffuser are shown in Fig. 7. The diffuser outer diameter was 465 mm (18.31 in.) and its inner diameter was 1.1 times the impeller diameter. Compressor inlet temperature and pressure were the same as the case of the fully shrouded impeller presented in the previous subsection.

Experimental results are shown in Fig. 8. Head rise and stage efficiency of HCFC123 and CFC11 were almost the same near the design flow rate. Flow range, however, was not equal. Flow rate at the surge point of HCFC-

123 was about 15% higher than that of CFC11. As is shown in Fig. 5, the unmodified impeller had a lower surge limit for HCFC123 than for CFC11, so the modified impeller was expected to work also at a much lower flow rate. This change suggests that the surge for HCFC123 is triggered by stall of the diffuser and not of the impeller. At the same time, the choke limit flow rate for HCFC123 was larger than for CFC11. These suggest that the diffuser was matched to a little higher flow rate for HCFC123 than for CFC11. To verify this diffuser matching problem, the diffuser inlet volume flow rate was increased a little by reducing the pressure ratio slightly. The rotational number was lowered by 2.5% for this experiment. This slight change reduced the surge flow rate 12%, as expected, and the choke flow rate 5%. Pressure ratio at the surge point dropped 8%, as shown in Fig. 9. The impeller exit absolute Mach number was around 1 for the CFC11 case, and several percentage points higher for HCFC123. This Mach number difference might have some effect on the delicate nature of diffuser matching for HCFC123, but no clear conclusion has yet been made.

Though some complexities arise by the combination of impeller and channel diffuser, basically, the same level of flow rate, head rise and efficiency can be realized by controlling the impeller diffusion ratio. As stated above, the Mach number may have some effect, but it was not so apparent as that of diffusion ratio within this experiment.

EVALUATION OF A TANDEM IMPELLER

The thermodynamic properties of HCFC123 require about a 20% larger compressor suction volume flow rate per unit refrigerating capacity. This is not a difficult problem when there are no design constraints on impeller size. In general, when the volume flow rate is increased without impeller diameter enlargement, relative velocities increase, too. According to the results in the previous section, the efficiency drop by Mach number increase is not remarkable. Therefore, a compressor for HCFC123 may be developed without enlargement and efficiency drop.

Higher efficiency, however, is always desireable, so an experiment with a tandem impeller, which can absorb the volume flow increase without diameter enlargement,was made. A tandem impeller is a combination of an axial impeller and a radial impeller. There are two aerodynamic advantages of this type of impeller, as reported by Klassen et al. [3]. One is that transonic axial compressor technology can be applied to diffuse the inlet transonic flow into a subsonic one with high efficiency. The second is that the boundary layer on the axial impeller scatters into the main flow and a fresh boundary layer develops on the radial impeller. This can reduce the losses caused by boundary layer development.

A comparison was made of a tandem impeller and a radial impellerin a centrifugal chiller using CFC11. The blade exit backswept angles and peripheral velocities of these impellers were the same. The diameter of the tandem impeller was 460 mm (18.11 in.) and the diameter of the radial impeller was 510 mm (20.08 in.). The tandem impeller was more compact than the radial impeller by 10%. Meridional shapes of these impellers are shown in Fig. 10. Impeller inlet tip-relative Mach number of the tandem impeller is shown in Table 4. The Mach number level is the same as those in Tables 2 and 3. These impellers were combined with vaneless diffusers. The tandem impeller is shown in Fig. 11 and the experimental results are shown in Fig. 12. The tandem impeller realized the same flow range, a slightly higher head rise and an efficiency increase of about 3%. When CFC11 is replaced by HCFC123, it is impossible to avoid the increase of flow rate per unit refrigerating capacity and the theoretical compression work increase. Further investigation of tandem impeller technology may provide a possible and effective solution to this problem.

CONCLUSIONS

1. When a CFC11 compressor was used to compress HCFC123, head rise, efficiency, and volume flow rate decreased.
2. When the diffusion ratio of an impeller was adjusted to the value of the CFC11 case, the performance of HCFC123 compression recovered to the level of the CFC11 case.
3. A tandem impeller with transonic inducer could cover most of the increase in volume flow rate and theoretical compression work which arise from the difference in thermodynamic properties of CFC11 and HCFC123.

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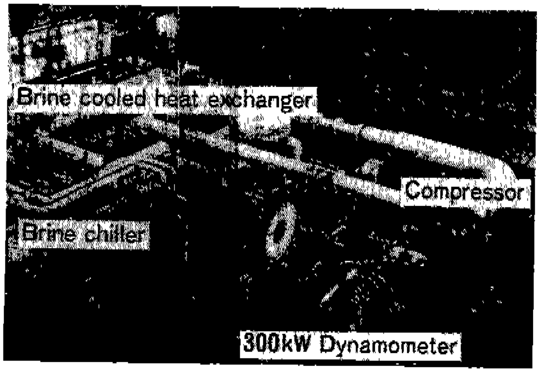

Fig. 1 Test stand

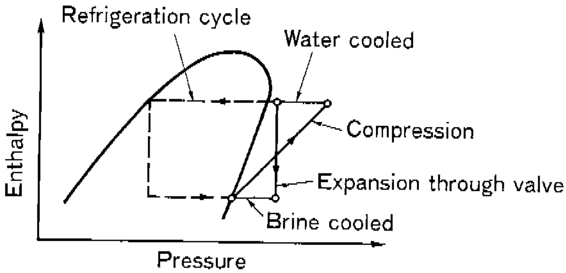

Fig. 2 Test stand cycle

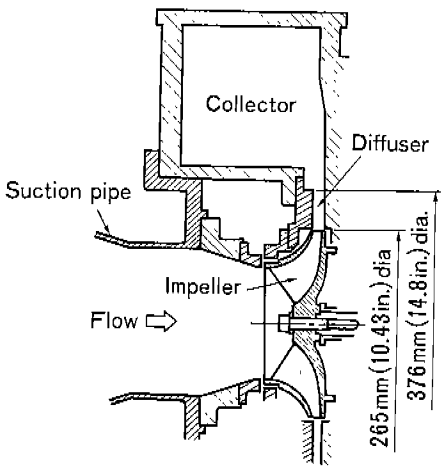

Fig. 3 Cross-setional view of a model compressor rig

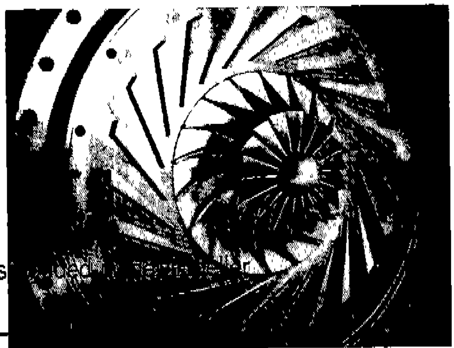

Fig. 4 Fully s

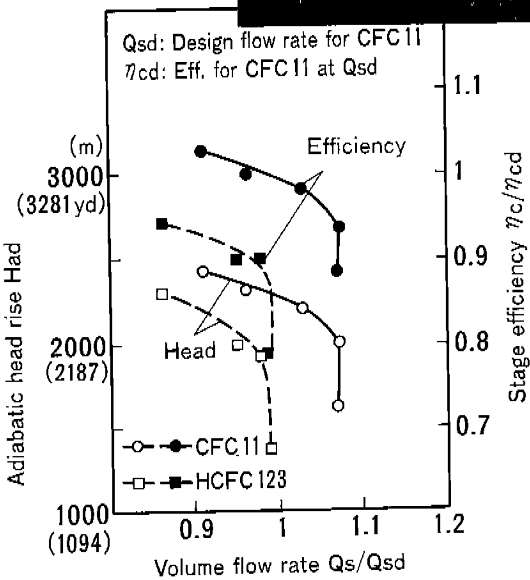

Fig. 5 Comparison of head rise and efficiency (same impeller)

Table 1 Performance of HCFC 123 (CFC 11-based)

Qs chalk	Qs surge	Qs ch./Qs su.	Had surge	ηad max.
0.92	0.95	0.97	0.94	0.93

Table 2 Mach number and diffusion ratio (same impeller)

Gas	M_1 s	w_1 s/w_2 (CFC 11-based)
CFC 11	1.08	1.0
HCFC 123	1.16	1.15

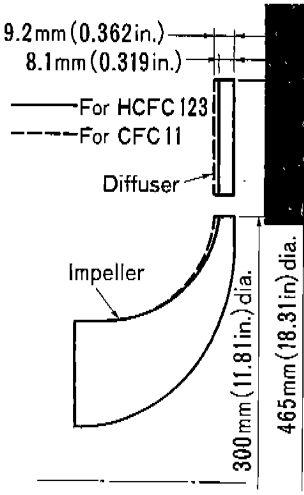

Fig. 6 Impeller modification

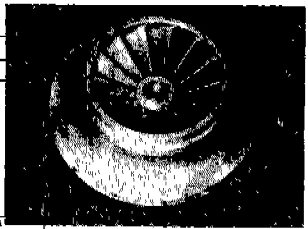

Fig. 7 Impeller and diffuser

Table 3 Mach number and diffusion ratio (modified impeller)

Gas	M_1 s	w_1 s/w_2 (CFC 11-based)
CFC 11	1.07	1.0
HCFC 123	1.15	1.0

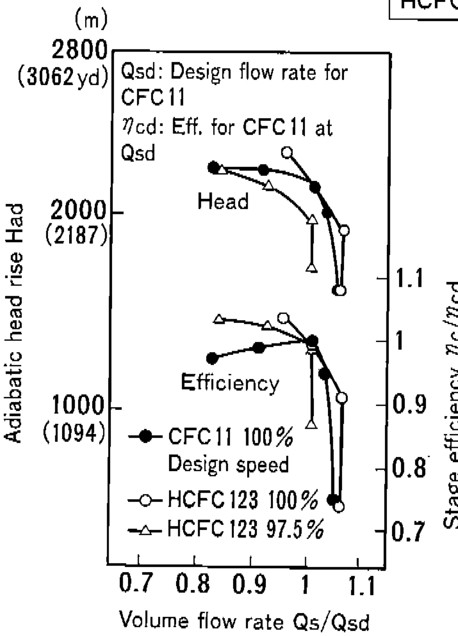

Fig. 8 Comparison of head rise and (modified impeller)

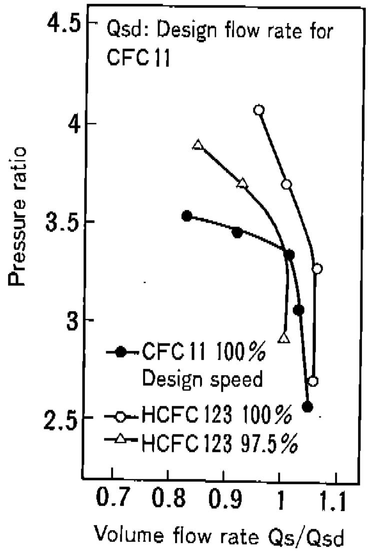

Fig. 9 Comparison of pressure ratio

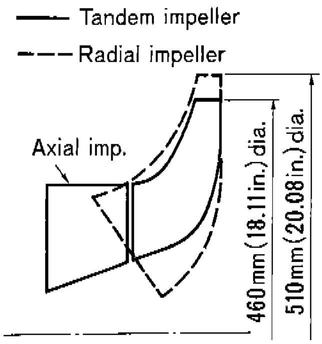

Fig. 10 Comparison of tandem and radial impellers

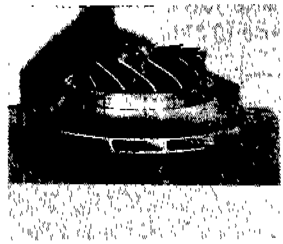

Fig. 11 Tandem impeller

Table 4 Mach number

Impeller	M_1 s
Radial Imp.	1.01
Tandem Imp.	1.17

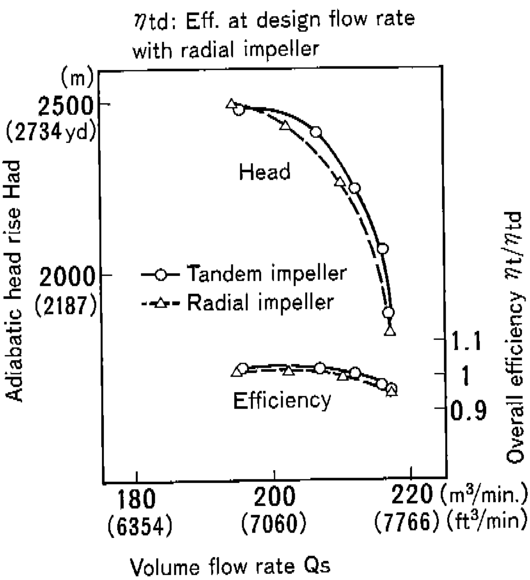

Fig. 12 Performance of tandem impeller