On the Feasibility of Compressing CO2 as Working Fluid in Scroll Compressors

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ABSTRACT

A theoretical analysis of compressing CO_2 gas (R-744) in an open scroll machine for automtive air conditioner has been carried out. A R-134a scroll compressor with swept volume of 80 cm^3 i.e. for mediumed-size car was applied as a baseline compressor. The focus has been on overall geometrical dimensions, weight of moving parts and to quantify the gas leakage. The objective was to achieve competitive parameter values for the R-744. It was applied two different set of R-744 scroll profiles. One scroll set with maximum 3 pairs of compression pockets and one set with maximum 2 pairs. Maximum 2 pairs was also the case for the R-134a compressor.

The total weight of the moving parts and overall dimensions was fulfilled compared to the R-134a scroll for the R-744 scroll set with maximum 2 pairs of compression pockets but not for the R-744 scroll set with 3 pairs. The volumetric- and the isentropic efficiency based only on gas leakage loss without influence of lubricant was lower for both R-744 scroll set at equal clearance gap. Equal volumetric- and isentropic efficiency could be obtained when the clearance gap was 56 and 67% respectively compared to the R-134a compressor. This was the case at both 1000 rpm (idle condition) and 2500 (driving condition).

A self-adjusting back pressure mechanism must be applied in order counteract the larger thrust/axial force in the R-744 compressor.

NOMENCLATURE

C	[-]	flow coefficient	p	[N/m^2]	gas pressure	A	[m^2]	flow area
ρ	[kg/m^3]	gas density	R	[-]	pressure ratio	θ	[rad]	crank angle
Δp	[N/m^2]	pressure difference	D	[m]	hydraulic diameter	f	[-]	friction factor
l	[m]	leakage length (radial)	P_s	[bar]	suction pressure	m	[kg/s]	gas mass discharge
P_d	[bar]	discharge pressure	ω	[rad/s]	crankshaft velocity	$P_{ad.}$	[W]	average torque power
T	[Nm]	crankshaft torque	$\dot{V}$	[m^3/s]	swept volume	ρ_s	[kg/m^3]	inlet gas density
$\dot{m}$	[kg/s]	discharge mass rate	Φ	[rad]	involute angle	a	[m]	generating radius
$P_{tor.}$	[W]	reversible adiabatic compression power						
κ	[-]	reversible adiabatic exponent						

Subscripts: t tip f flank u upstream

INTRODUCTION

Public literature on CO_2 compressor up to now as been on reciprocating types. The question has been arised, is it possible to compress CO_2 in other type of compressors? Scroll compressor is used in many applications due to the perception of high durability, reliability, and efficiencies in additional to low noise and vibration. The scroll compressor can also compete on price, weight and physical size with respect to a reciprocating compressor. A feasibility analysis of the possibilities of compressing CO_2 in a scroll machine should therefore be carried out. The focus should be on gas forces and gas leakage because an R-744 scroll compressor must work with higher-pressure differential.

Section 1 establishes an assumed design condition for the R-134a compressor and a comparable temperature condition for the R-744 compressor. A short description of the scroll profiles with the definitions is shown in section 2. Scroll profile constraints for this feasibility analysis are described in section 3. Scroll design- and gas leakage results are shown in respectively section 4 and 5. The discussion of the results is briefly discussed in section 6.

1. OPERATING CONDITION FOR THE R-134A AND THE R-744 COMPRESSOR

There are many combinations of temperature, pressures, shaft velocity, air velocity etc. that constitute an operating condition. Available literature on testing automotive A/C system describes wind tunnel air velocity, humidity, air temperature etc. and not the system pressures for a specific ambient temperature condition. The applied condition for the R-134a system is assumed to be common A/C system operation values. A comparable operating condition for the R-744 system (with internal heat exchanger) was thereafter established.

Table 1 *Operating conditions*

Parameter:	R-134a	R-744	Units
Evaporating temperature:	5	5	°C
Evaporating pressure:	3.5	40	bar
Compressor inlet temperature:	15	25	°C
High-side pressure:	15	100	bar
Condensing temperature:	55		°C
Pressure difference:	11.5	60	bar
Pressure ratio:	4.3	2.5	-

ambient temperature was assumed to 45°C

Table 2 *Compressor parameters*

Parameter:	R-134a	R-744	Units
Built in volume ratio:	3.6	2.0	-
Swept volume:	80	17	cm^3
Shaft velocity (idle condition):	1000		rpm
Shaft velocity (driving cond.):	2500		rpm

The R-134a baseline scroll compressor was designed for a medium-sized car A/C system and so would the size of the CO_2 scroll compressor. The R-744 swept volume was based on equal cooling capacity, shaft velocity and volumetric efficiency as for the R-134a scroll compressor.

2. SCROLL PROFILES AND SCROLL PARAMETERS DEFINITIONS

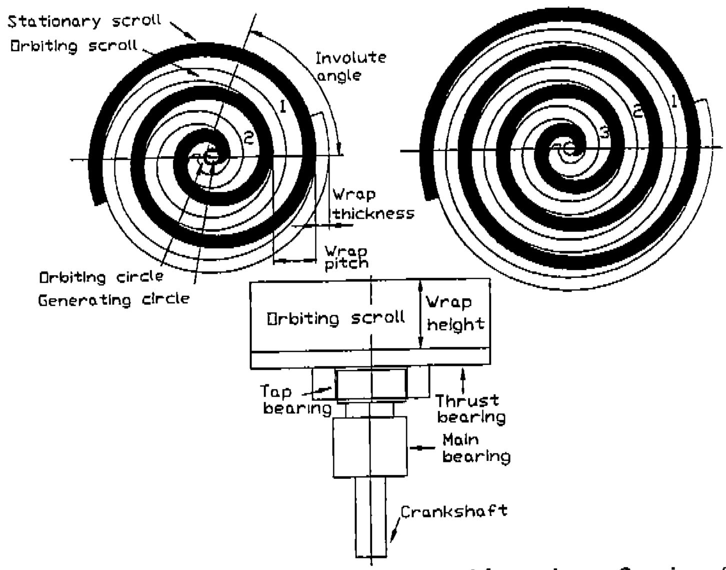

Figure 1 *Upper left figure shows a set with maximum 2 pairs of compression pockets* (Φ_e= 17.55 rad) *and the right maximum 3 pairs* (Φ_e= 23.83 rad). *The lower figure shows some definitions.*

The scroll profile might be described in several ways e.g. half circles, involute spiral etc. The profile applied in this analysis is involute spiral described by Cartesian co-ordinates.

$$x= a \cos \Phi + a \Phi \sin \Phi \quad (1)$$

$$y= a \sin \Phi - a \Phi \cos \Phi \quad (2)$$

The scroll profile was calculated with uniform wrap thickness along the whole scroll length and height. The design of the inner scroll profile was not given any particular attention regarding material strength and of eliminating "clearance volume". These aspects were not included in this analysis and would have a minor impact on the results presented in this paper.

The baseline compressor was designed for maximum 2 pairs of compression pockets i.e. an ending involute angle of 17.55 rad. A set of R-744 scroll profiles with maximum 3 pairs of compression pockets i.e. an involute angle of 23.83 rad was of interst with respect to the gas leakage. The pressure difference between two adjacent compression pockets would be less for 3 pairs rather than 2 pairs of compression pockets.

3. SCROLL PROFILE CONSTRAINTS

Four constraints were established for the R-744 scroll profiles. These were assumed appropriate at this stage and for this simple analysis. The chosen parameters should either have (at least) equal or preferably lower magnitude for the R-744.

1. Thrust bearing pressure load
2. Tap bearing pressure load
3. Total mass of moving parts
4. Overall geometric dimensions

The intention of the bearing load constraint was twofold. The mechanical friction in the bearing for the R-744 compressor should be at least equal to the R-134a compressor. The friction has an indirect influence on the isentropic efficiency and the efficiency should be at least equal if the R-744 scroll should be regarded as an alternative. The bearings size has also an impact on the mass of moving parts, which is one of the constraint.

The total mass of moving parts puts a limit on the maximum crankshaft velocity and to some extent the vibration. A very important factor in automotive A/C is the total weight of the compressor, which should be kept as small as possible.

Overall geometric dimensions are in many cases linked to the weight of the compressor and should be held on a minimum. Space in the engine compartment is regarded as limited in order to keep the vehicle as compact as possible.

The CO_2 scroll profiles based on the four constraints are applied in the gas leakage analysis, which has also one main constraint. As it was earlier mentioned, the isentropic efficiency for the CO_2 compressor should be at least equal as for the R-134a. The gas leakage is the only (volumetric) loss parameter, which is incorporated for the calculation of the insentropic efficiency. One assumption has already been made regarding the volumetric efficiency and that is equal efficiency at design point.

4. SCROLL DESIGN RESULTS

The R-744 wrap thickness was made 6% larger and the base plate thickness was kept equal for both the R-744 and the R-134a. The maximum involute angle (see Figure 1) also called ending angle of the involute spiral was the same for one set of R-744 and the baseline R-134a compressor. The maximum involute spiral for the other R-744 scroll set was increased by 360°. Another constant parameter was the tap-bearing diameter in the R-744 compressor. The tap bearing length was varied in order to keep the tap-bearing load equal as for the R-134a compressor.

The simulation of the R-744 compressor started with suction pressure behind the orbiting scroll, which was the case for the R-134a compressor. The R-744 compressor works with higher pressure difference, which resulted in a higher thrust-/axial force. This force became considerably higher than for the R-134a compressor. The thrust bearing load constraint was therefore exceeded even if the base plate diameter was increased to the maximum diameter i.e. R-134a base plate diameter. This high thrust force was later counteracted by using a back-pressure, which is higher than the suction pressure. An appropriate system would be a so-called self-adjusting back-pressure mechanism, which is quite common in hermetic scroll compressor but not in automotive A/C.

The built in volume ratio was fixed i.e. suction- and discharge volume. The ending involute angle and the wrap thickness are also set to a fixed value. There are therefore several wrap height values, which might fulfil the geometric- and the mass constraint. Normalised length- and mass parameters i.e. referred to the R-134a parameters were plotted versus wrap heights in Figure 2 and Figure 3.

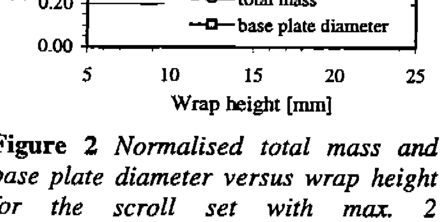

Figure 2 *Normalised total mass and base plate diameter versus wrap height for the scroll set with max. 2 compression pockets*

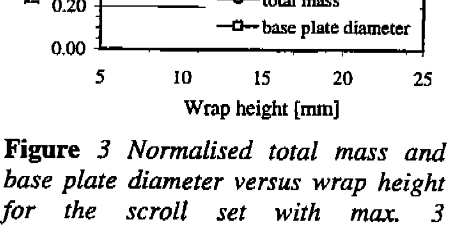

Figure *3 Normalised total mass and base plate diameter versus wrap height for the scroll set with max. 3 compression pockets*

The scroll set with the involute ending angle of 23.83 rad Figure 2 gave not a wrap height interval with respect to constraint 3 and 4. The wrap height should be less than 9.5 μm and higher than 12.5 μm if the constraint 3 or 4 should be fulfilled. Figure 2 shows the simulation results from the scroll set with the involute ending angle of 17.55 rad, which gave a wrap height interval where the constraint 3 and 4 is fulfilled. The wrap height should be between 9.5 and 18.5 μm in order to fulfil the constraints.

The wrap height for both R-744 scroll sets was chosen so that the overall geometric dimension in axial direction was equal to the R-134a compressor, see Figure 4. This implied nearly a minimum total leakage length i.e. perpendicular to the flow direction for the scroll set with the involute ending angle of 23.83 rad and 17.55 rad, see Figure 6. The R-744 wrap height became 55% lower than the baseline wrap height.

This might imply less impact by thermal (wrap) expansion in radial direction even if the temperature in the R-744 compression pockets might be higher than in the R-134a pockets. Another advantageous caused by reduced wrap height might be moderate wrap thickness with respect to material stress even if the gas pressure difference is larger for the R-744 compressor.

The bearings applied in the baseline compressor were needle bearing (tap), ball bearing (main) and glide bearing (thrust). The bearing pressure load transferred in the thrust bearing and tap bearing is equal for the investigated compressors. Equal R-744 thrust bearing load as for the R-134a was achieved by a back-pressure higher than the suction pressure. Equal R-744 tap bearing load as for the R-134a was established by enlarging the tap bearing length. It was applied a constant 19% larger tap bearing diameter for the R-744 compressor. The main bearing is kept unchanged but it must transfer a larger force. How the main bearing could be designed is dependent on the bearing type but also the diameter – length combination. The practical design and choice of bearing type is outside the frame of this study.

The back-pressure for the chosen wrap height was approximately 50 bar for both R-744 scroll sets, which was necessary in order to reduce the thrust force and in the end to give an equal thrust bearing load. The back-pressure in this case acts on the whole back-side of the orbiting scroll, so the suction gas inlet must be placed "above" the thrust bearing. The back-pressure calculated by the simulation program was 10 bar higher than the suction pressure.

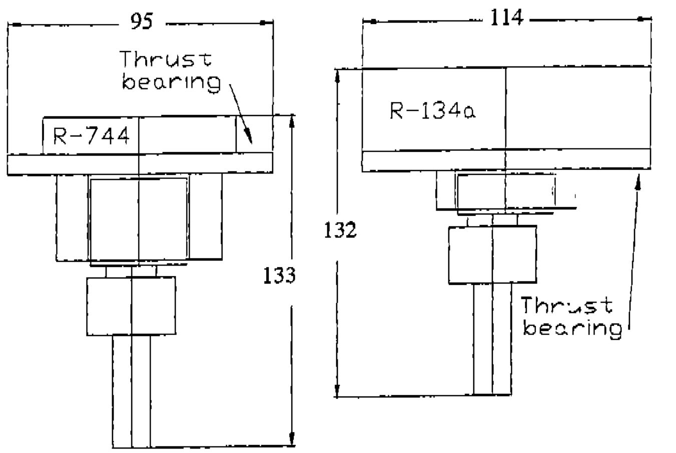

Figure 4 *Overall geometric dimensions for the R-744* (Φ_e= 17.55 rad) *and the R-134a rotating system with indication where the thrust-bearing surface is located*

5. GAS LEAKAGE RESULTS

The gas leakage types incorporated in the computer program were flank- and tip leakage also called radial- and axial leakage. The flank leakage was modelled as an isentropic steady-state compressible flow through a converging nozzle:

$$\dot{m}_{flank} = C_f\, A_f \sqrt{p_u\, \rho_u \frac{2}{\kappa-1}\left(R^{\frac{2}{\kappa}} - R^{\frac{\kappa+1}{\kappa}} \right)}, \qquad 1 > R > \left(\frac{2}{\kappa+1} \right)^{\frac{\kappa}{\kappa-1}} \tag{3}$$

The mass flow rate was restricted between no flow and choked flow, controlled by the pressure ratio. The tip leakage was modelled as incompressible viscous flow between two parallel surfaces including a friction factor:

$$\dot{m}_{tip} = C_t\, A_t \sqrt{2\, \rho_u\, \Delta p \frac{D}{f\, l}} \tag{4}$$

The friction coefficient was determined by an iterative solution since the coefficient is based on the Reynolds number, which in turn depends upon the mass flow rate. Equation 3 and 4 has been applied by (Caillat et.al. 1988) and (Afjei and Suter 1990) for gas leakage in scroll compressor. It has been reported by (Ishii et.al. 1996) that the equation for incompressible flow predicted both types of leakage quite good when calculated results were compared with measurements.

The lengths of gas leakage i.e. flank-,

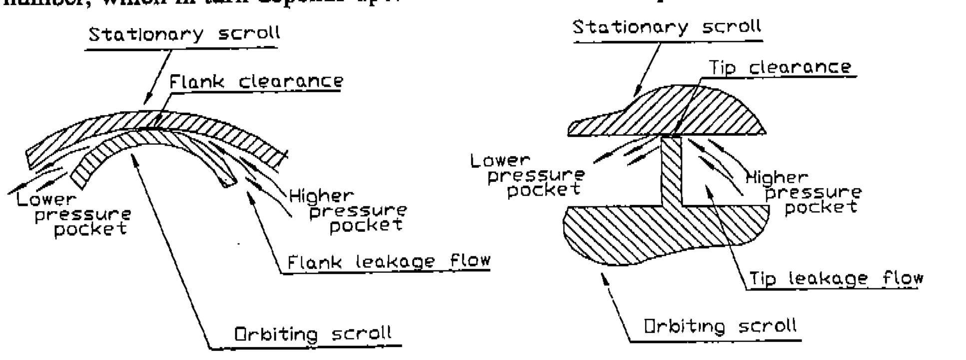

Figure 5 *Simplified sketches of the two leakage types implemented in the computer code i.e. tip- and flank leakage described by stationary- and orbiting scroll in additional to arrows showing the gas flow direction*

tip- and total length (perpendicular to the flow) were derived from a geometrical point of view at each wrap height. The orbiting scroll was in these calculation in a position described in Figure 1 i.e. end of the suction process. Figure 6 shows the normalised tip-, flank- and total clearance length referred to the R-134a parameters at varying wrap heights.

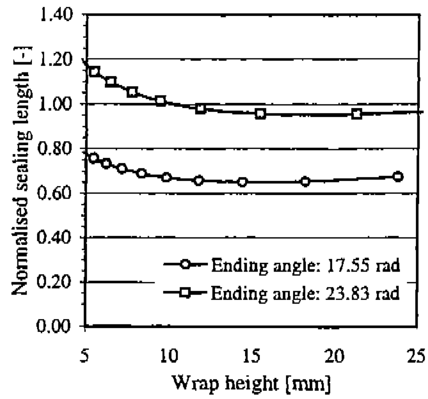

Figure 6 *Normalised sealing length perpendicular to the leakage flow at varying wrap height for both R-744 scroll sets.*

The scroll profile constraints from the previous section gave a possible wrap height range i.e. from 9.5 mm to 18.5 mm for the scroll set with 17.55 rad ending angle. The other set did not give any wrap height range based on the total mass and the base plate diameter. An intuitive action would be to choose a wrap height, which gave minimum total gas leakage length (if there was no total geometric constraint).

Another aspect arise which gas leakage type would be the dominant one. The simulation revealed that the tip leakage dominated at higher clearance gap for the R-744 compressor but the two leakage types were almost equal around a clearance gap of 5 μm. The tip- and the flank leakage for the R-134a were almost equal in magnitude for the investigated clearance gap compared to the R-744 compressor.

The total gas leakage length (both tip and flank) perpendicular to the flow direction for the chosen R-744 wrap height was 65 and 98% respectively for the involute ending angle of 17.55 and 23.83 rad compared to the baseline compressor.

The total volumetric loss i.e. tip- and flank leakage is displayed in Figure 7 and 8 for both the R-744 scroll sets and the R-134a at two different crankshaft velocities.

The definition of the volumetric loss is:

$$\lambda = 1 - \frac{\dot{m}}{\dot{V}\,\rho_s} \qquad (5)$$

The volumetric loss for the two R-744 scroll sets is higher than the R-134a scroll at equal clearance gap. The R-744 loss is some higher for the scroll set with the involute ending angle of 23.83 rad i.e. maximum 3 pairs of compression pockets compared to the scroll set with the angle 17.55 rad.. Equal volumetric loss might be obtained at 5 μm for the R-744 scrolls and 15 μm for the R-134a scroll.

Crankshaft velocity has a major impact on the compressor efficiencies. This can be observed from the volumetric loss figures but also in Figure 9 and Figure 10, which show the isentropic efficiencies for the compressors at different crankshaft velocity. The isentropic efficiency was only based on the gas leakage. Mechanical friction and effect

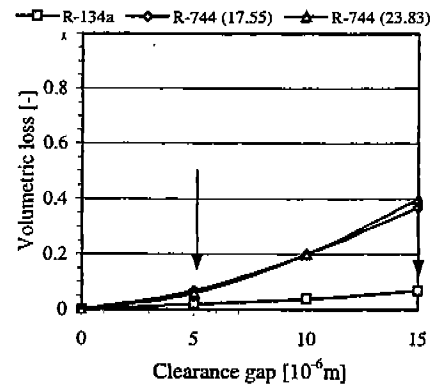

Figure 8 *Volumetric loss for both the R-744 scroll profile sets and the R-134a at varying tip- and flank clearance gap. Crankshaft velocity 2500 rpm.*

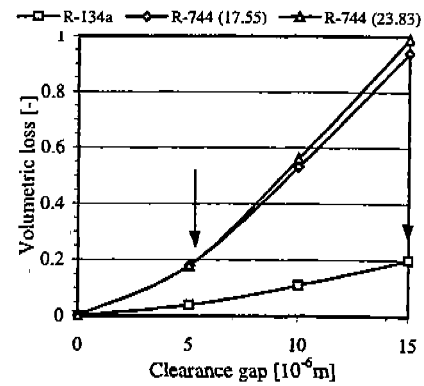

Figure 7 *Volumetric loss for both the R-744 scroll profile sets and the R-134a at varying tip- and flank clearance gap. Crankshaft velocity1000 rpm.*

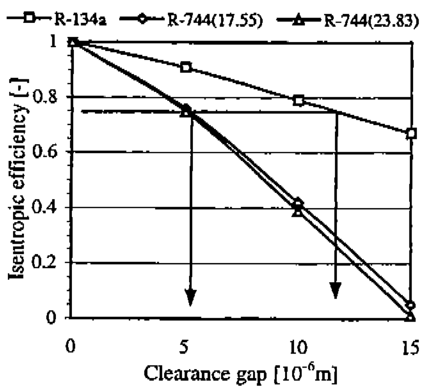

Figure 10 *Isentropic efficiency for both the R-744 scroll profile sets and the R-134a at varying tip- and flank clearance gap. Crankshaft velocity 1000 rpm.*

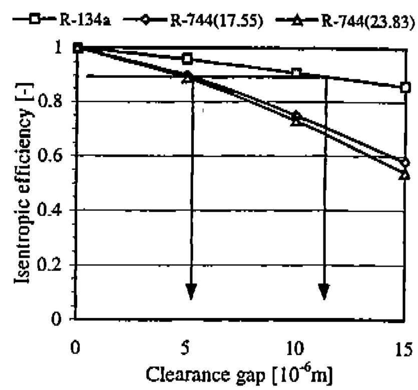

Figure 9 *Isentropic efficiency for both the R-744 scroll profile sets and the R-134a at varying tip- and flank clearance gap. Crankshaft velocity 2500 rpm.*

caused by heat transfer was not included. Gas flow loss during suction and discharge process was neither included. The definition of the isentropic efficiency:

$$\eta_{is.} = \frac{P_{ad.}}{P_{tor.}} \quad , \quad \text{where:} \quad P_{ad.} = \frac{\kappa}{\kappa - 1} P_s \left(\frac{\dot{m}}{\rho_s}\right)\left[\left(\frac{P_d}{P_s}\right)^{\frac{\kappa-1}{\kappa}} - 1\right] \quad \text{and} \quad P_{tor.} = \frac{\omega}{2\pi}\int_0^{2\pi} T\, d\theta \qquad (6), (7) \text{ and } (8)$$

The adiabatic compression power ($P_{ad.}$) decreased when the gas leakage increased i.e. increasing clearance gap. It was almost zero at the highest clearance gap at the lowest crankshaft velocity for the R-744 compressor. The average torque power ($P_{tor.}$) increased slightly at increasing clearance gap caused by steeper pressure build-up during the compression process. Steeper pressure build-up was caused by the gas leakage.

It was notice lower isentropic efficiencies for the R-744 scroll set with the ending angle of 23.83 rad compared to the scroll set with 17.55 rad. It was assumed that a clearance gap of 5 μm could be obtained for the R-744 compressor. Equal isentropic efficiency for the R-134a compressor was achieved with 11.5 μm clearance gap at both crankshaft velocities.

6. DISCUSSION

The feasibility analysis reveals a possibility to design a competitive CO_2 scroll compressor when overall geometric dimensions, bearing pressure loads, gas forces and gas leakage is taken into consideration. This is based on simple theoretical comparison to an R-134a baseline compressor.

The main challenges for the R-744 is the gas leakage, which might be equal as for the baseline compressor if the leakage clearance gap is reduced. In this study 5 μm has been used as a desired clearance gap for the R-744, which seems quite small. A clearance gap of 15 μm and 11.5 μm should be applied for the R-134a compressor when respectively equal volumetric- and isentropic efficiency is to be achieved. Leakage calculation for R-22 scroll compressors revealed from the literature has been carried out with clearance gap from 10 to 20 μm. The only possibility to achieve such a small clearance is by axial- and radial compliance. The self-adjusting back-pressure mechanism might also be applied as axial compliance in order to reduce the tip clearance. Radial compliance must also be applied in order to keep the flank leakage at an acceptable level. The radial gas force and the centrifugal force i.e. total radial force also called sealing force should probably be larger for a R-744 compressor. The magnitude of the back-pressure and the sealing force would be a compromise between friction- and volumetric loss.

If it is possible to achieve 5 μm for the R-744 compressor why do not use such clearance gap in the R-134a compressor in order to increase the efficiencies. The gas/centrifugal forces dictate the size of the gaps but also by the surface texture of the components. It might be obtained from practical experiences on R-134a compressors that such a gap (5 μm) gives not the optimum choice with respect to the mechanical - and the volumetric efficiencies. Another clearance gap might be appropriate for an R-744 compressor since the pressure levels are somewhat different compared to the R-134a compressor.

The R-744 scroll set with maximum 3 pair of compression pockets caused a lower pressure difference between the adjacent compression pockets but the efficiencies were lower compared to the scroll set with maximum 2 pair of compression pockets. Lower efficiencies were caused by higher total gas leakage length (see Figure 6) perpendicular to the flow even if the pressure difference between the pockets is smaller.

Thermal - and stress aspects were not included in the present analysis. Location of the holes regarding establishing of the back-pressure and also impact of these holes on the compressor efficiencies were not investigated. There exist several kinds of anti-rotating devices for the orbiting scroll, which might be used but this was neither included.

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