Mini Screw: the development of a high-CFM compact compressor for LGWP A1 low pressure refrigerant

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Abstract. Addressing global warming concerns is one of the largest challenges facing the HVAC industry currently. Although A2L refrigerants are important options to consider for certain sizes of HVAC units, there remain many systems that require A1 category refrigerants. Among the several available refrigerants, we looked at DR-12 as one of the attractive refrigerants. Along with it being in the A1 category, its very low GWP value (32) and high cycle performance make it an excellent long-term solution if successfully applied. A significant challenge to overcome with this refrigerant is its low density. DR-12 will require almost 7 times the volumetric flow for the same cooling capacity compared to R410A. The current scroll or rotary technologies are not realistic to be applied in such a low density refrigerant system. The development of a new compact large CFM compressor is key to realize the scope of the project successfully. The authors have explored various types of compression mechanisms and identified screw as one of the most suited technologies for this refrigerant. In order to obtain such large volume flow (19.8L/s, 42CFM), a unique screw rotor design and a high speed PM motor were employed. Innovative bearing design and new compressor layout also enabled the compressor to be compact (φ145mm) and cost effective. A prototype compressor was built and tested. This concept delivered 17.6kW (5 ton) capacity at 11,000rpm and 4.4kW (1.25 ton) at 2,500rpm, meeting its target requirements. This paper will introduce details of the compressor concept, along with the test results from a physical prototype.

1. Introduction
Global warming potential of refrigerants is one of the largest challenges that the HVAC industry is currently facing. Various low global warming potential (LGWP) refrigerants have been proposed and some of them have already been deployed in applications. However, most of these recently proposed LGWP refrigerants are categorized as A2L (mildly-flammable refrigerants). This requires special hardware for leak detection and ventilation especially when it is used in the systems with large amount of refrigerant charge. Neither of the LGWP refrigerants offers the lowest global warming potential option. For example, R32 and R452B (DR-55), which are two of the R410A substitute candidates, still have a GWP of 600-700 [1]. Although these are excellent drop-in candidates, they are considered as interim solutions since considerable GWP reduction potential remains. At this time, there have been no drop-in refrigerants developed with very-low GWP and A1 (non-flammable) property for replacement of R410A.

When a long term solution such as A1 and GWP<100 is considered, options are currently very limited. Carbon dioxide (CO₂) has been used in some applications such as water heating and refrigeration (cascade) systems. It is a vital option for the long term yet would require high strength components in the A/C system due to its extreme pressure. This would likely be more challenging when a larger size system is considered. In addition, the cycle efficiency with CO₂ as a refrigerant for typical A/C system
duty is at a disadvantage to other replacement options. Another path for long term LGWP A1 refrigerant is to move to a lower pressure refrigerant option such as R1233zd(E). This type of refrigerant has been applied in some large chillers using centrifugal compressors. Since it is a low pressure refrigerant, compressors need to be large to generate the large volumetric flow required. In fact, when cycle capacities are calculated at the industry’s cooling rating condition [2], the required volumetric flow for the same cooling capacity reaches almost ten times with R1233zd(E) compared to R410A. The current compression technologies used in R410A systems such as scroll and rotary are not realistic to apply in such low pressure systems.

DR-12 is a relatively new refrigerant that has been proposed for achieving a LGWP A1 solution [3]. It is still a low pressure refrigerant with the required volumetric flow of approximately seven times that of R410A. Although the GWP and flammability issues can be mitigated with DR-12, the challenge still remains with the compression technology of how to create seven times the volumetric flow with a single compressor. An “out-of-the-box” solution is anxiously desired when considering low pressure refrigerants.

Screw compressors are widely used in larger applications above the capacity of typical R410A products. The compressor is well-suited for large volumetric flow, therefore a practical compressor could be constructed for a low pressure system. Since typical R410A applications are very different from screw chillers, the compressor should be re-designed entirely to accommodate the requirements of product styles leveraging R410A compressors. In this paper, a new compressor concept will be introduced using a novel screw mechanism. With this concept, a very large volumetric flow can be obtained without an increase of compressor shell size versus R410A. Also, as it is critical to typical R410A applications, various design elements were put into the concept in order to produce the compressor more economically. A comprehensive design study was carried out to get the optimized screw design for 17.6kW (5 ton) cooling capacity with DR-12. A prototype compressor was made and the performance was verified. More details will be described in the main article below. It should be noted that the developed compressor has been named “Mini Screw” as it is literally a mini size compared with typical screw compressors.

2. Mini Screw Concept

2.1. Compression mechanism comparison

The study started with a comparison of how the screw compressor had an advantage over other compression mechanisms. Figure 1 shows the comparison with scroll. Both screw and scroll models shown in the figure have the same displacement (swept volume), same volume ratio (Vi), and the same discharge port area. Since the compression volume shrinks inward in the scroll compressor, the compressor tends to grow in the radial direction when a larger displacement is required. Increasing vane height is limited in the scroll compressor due to its own overturning moment. Large contact force (thus large friction loss) between the two mating scrolls is unavoidable in a tall scroll to prevent its overturning during operation. On the other hand, the screw volume can be extended in the axial direction, therefore the same displacement can be achieved in smaller shell diameter with screw. It should be noted that too long of a screw shaft would cause excess deflection in the screw compressor. However, the pressure differential that causes the shaft deflection is expected to be small with the low pressure refrigerant, therefore the limitation is not overly restrictive with DR-12. The overturning moment issue will still exist in the scroll compressor regardless of the pressure level of the refrigerant.

Another advantage of the screw compressor is that it is a perfectly balanced machine. Unlike other mechanisms, there are no unbalancing components in the compressor such as pistons, counter weights, discharge valves, Oldham rings, vanes, etc. This will be very helpful for constructing a high speed compressor without facing typical speed-related issues like valve dynamics, vibration, wear, etc. In fact, the optimum speed of the screw compressor is usually higher than other type of compressors due to these characteristics of the compressor. By setting the compressor speed higher than scroll, the screw compressor could output even higher volumetric flow within the limited size of the compressor.
2.2. Screw compressor design for small capacity application

R410A compressors are typically used in residential and light commercial units in the capacity range of approximately 3-100 kW (1-30 ton). With these smaller capacity applications, production volumes (number of units) are considerably large compared to typical screw products. Most of the R410A compressors are made in a welded can (hermetic) configuration for mass production. Shell diameter (footprint) is always an important factor because it tends to drive overall cost of the compressor. The development of a Mini Screw compressor needs to accommodate these requirements. Below are some of the design considerations:

2.2.1. Opposed rotor screw compressor. Figure 2 shows the opposed rotor screw compressor concept. This configuration enables an economical bearing design without the use of rolling element bearings or special bushings. The concept itself has been known since 1923 [4]. A pair of helically opposed twin screw rotors are connected on a common male and a female shaft. Refrigerant suction takes place at the mid portion of the connected rotors and the gas flow splits towards the opposite ends. Gas compression is balanced and therefore thrust bearings can be omitted. In our configuration, screw rotors are machined separately from the shafts. Each male and female rotor has an axial bored hole within the rotor. Two male rotors (the opposed helical pair) are securely attached to a drive shaft through their rotor bores (tight fit). The drive shaft will be supported by journal bearings at the upper and lower ends of the attached rotors. The female rotors are made to have small clearance between their bore diameters and its supporting shaft (loose fit) so that the rotor bore itself can be utilized as a cost effective journal bearing. This loose-fit configuration also provides a self-aligning function to the female rotors. Since each female rotor can spin independently on a stationary shaft, it can mesh to the respective male rotor regardless of angular positioning (clocking) between the two male rotors on the shaft. This design can greatly reduce the level of complexity in the screw rotor assembly and make the opposing concept practical.

Figure 1. Compression mechanism comparison between screw and scroll.
2.2.2. High side motor shell. The high-side motor shell design is widely used in small capacity compressors. It has an advantage of simplifying the lubrication system, especially for compressors using variable frequency drives. Pressure differential is utilized to supply oil to the bearings eliminating the need for a dedicated oil pump. Figures 3 and 4 show the concept of the high-side design. Suction gas is directly introduced to the compression mechanism through the inlet pipe at the bottom side. Discharge gases from the opposed screw rotors are routed along the channel provided within the rotor housing, and flows out of the slots at the top of the rotor housing. The gas flows up through the motor/stator and provides the cooling effect to the motor. Gas pulsation could be dampened and reduced within the compressor shell before it is discharged to the outlet. Oil that mixed with the refrigerant gas is separated in the space above the motor, which then drips down to the bottom sump by gravity. The oil is supplied back to the bearings via the holes within the male/female shafts. Internal tight screw rotor discharge end clearances are utilized to regulate oil flow and therefore excessive oil circulation is prevented. A discharge outlet is provided at the top of the compressor shell. Inlet/outlet pipe sizes are chosen to accommodate the large volumetric flow from the compressor. Expected overall compressor dimensions (approximate) are also shown in the figure 3. It can be seen that the compressor size is maintained compact (ϕ145mm) even though it provides seven times the volumetric flow (19.8L/s, 42CFM) of the equivalent size R410A compressor.
2.3. Design optimization

In positive displacement machines, compressor output (capacity) is determined by its displacement and shaft speed. Variable frequency drives (VFD) are vital options, not only for adding capacity modulation capability, but also to help reduce the compressor size by operation at a higher frequency than utility supply power. Although modern VFDs are capable of handling frequencies as high as 1 kHz, the associated small displacement compressor may not be efficient due to increase in the various internal losses. In order to design the right size compressor with a VFD, it is necessary to understand the effect of speed on the compressor efficiency.

In this study, an in-house screw compressor performance simulation program (1-D model) was used to determine the optimum speed of the compressor. This simulation tool was validated against test data for larger compressors. The 1-D program includes comprehensive models to account for most of the physical effects such as leakages, porting losses, bearing losses, deflection, oil dilution, motor cooling, etc. Modifications were made to model the Mini Screw concept for its unique design considerations. A hydrodynamic fluid film loss model based on Reynolds equation was added to handle the journal bearings. Thrust bearing loss was removed since it is absent in the compressor. The oil dilution (PVT characteristics) was calculated based on the discharge condition of the compressor. The motor loss was also calculated based on the discharge temperature due to the high-side design of the Mini Screw compressor.

Speed sweep analyses were performed on five different screw displacement designs (14, 28, 57, 113 and 227 cm$^3$/rev). Table 1 shows basic screw rotor information used in the simulation. Screw designs were kept similar among all five displacements. Clearance was increased proportionally with the rotor diameter (i.e. constant clearance ratio). Figure 5 shows the effect of speed on the compressor efficiency (vertical axis) and the capacity output (horizontal axis). With each displacement, the compressor efficiency peaked during the speed sweep. For example, the 14 cm$^3$ compressor had an efficiency peak at 29,000 rpm for generating 5.3 kW (1.5 ton) cooling capacity, and 28 cm$^3$ had it at 24,000 rpm for 8.4 kW (2.4 ton) capacity. There are two trends worth noting that peak efficiency reduces with size and large efficiency change with speed in smaller sizes. Efficiency levels of these small displacement compressors were low since the size of the screw rotors were too small, resulting in relatively high internal leakage. On the other hand, larger size compressors had more favorable simulation results. Due to less leakage per volume, the compressor efficiency improves with the displacement. According to the simulation, 113 cm$^3$ and 227 cm$^3$ were expected to be superior by 18% and 21%, respectively, compared to the smallest 14 cm$^3$ compressor. Table 2 shows the summary of the analysis.

Figure 4. Mini Screw concept showing the internal gas passage

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Table 1. Screw rotor information

| Lobe counts (male/female) | 4/5  |
|---------------------------|------|
| Wrap angle                | 330deg|
| L/D (male rotor)          | 0.96 |

![Figure 5. Compressor speed sweep analysis using 1-D simulation tool.](image)

Table 2. Speed sweep analysis summary.

| Displacement (cm³/rev) | Optimum Speed (rpm) | Optimum Capacity (kW) | Peak efficiency (normalized to 14 cm³) |
|------------------------|----------------------|------------------------|---------------------------------------|
| 14                     | 29,000               | 5.3                    | 1.00                                  |
| 28                     | 24,000               | 8.4                    | 1.09                                  |
| 57                     | 14,000               | 12.7                   | 1.15                                  |
| 113                    | 11,000               | 18.3                   | 1.18                                  |
| 227                    | 9,000                | 28.1                   | 1.21                                  |

This result can be used to determine the compressor speed and displacement for a given cooling capacity. For example, 227 cm³ is indicated to be the best for 25-30 kW (7-9 ton) capacity and 113 cm³ is shown as optimum for 15-20 kW (4-6 ton). Although 10 kW (3 ton) and below capacities can be possibly pursued with 57 cm³ and lower displacement, efficiency levels may not be as attractive as the larger models. In this study, 113 cm³/rev was selected for 17.6 kW (5 ton) target full load capacity. The compressor is planned to be operated at 11,000 rpm for this capacity.

2.4. Prototype compressor

Based on the above optimization, the screw rotor design was finalized, and associated motor torque and bearing load calculations were made. Volume index (Vi) was further optimized based on the part load conditions of an expected application. A permanent magnet motor and VFD were selected which could accommodate the required range of speed and torque of the compressor. Figure 6 shows pictures of the
actual prototype built. Instead of a welded canned shell, a bolted construction with increased thickness was used. The prototype design allows adding instrumentation connectors and facilitates periodic inspections of the rotor and bearings. The critical design elements of the compressor such as screw rotors, shafts, suction/discharge ports, clearances, etc. were maintained as proposed in the original hermetic concept, therefore there should be no influence of the modifications on the compressor performance. In addition, Coriolis flow meter was connected to measure the amount of oil flow to the bearings during the test.

3. Prototype Performance
Compressor tests were carried out on an automatic calorimeter and tests were performed in accordance with ASHRAE Standard 23.1. Electric power was measured before the VFD, therefore power loss of the drive is included in the efficiency data. Oil circulation rate (OCR) was measured by sampling a small amount of condensing liquid from the calorimeter and then measuring the weight of non-evaporated liquid (oil) from the sample. DR-12 was used for refrigerant. VG32 POE was used for oil. Figure 7 shows the schematic diagram of the calorimeter setup.
Compressor performance was measured at four different rating conditions (named here as A, B, C and D). Condition A is for the full load operation, where the compressor should output 100% of maximum capacity i.e. 17.6 kW (5 ton) in our case. The other conditions, B, C and D, are part load at capacities corresponding to 75%, 50% and 25%. In the present case, the compressor speed was reduced from 11,000 rpm to 2,500 rpm to achieve the 25% (4.4 kW, 1.25 ton) unloaded capacity. All four of these conditions constitute a system efficiency rating, therefore it is important that the compressor should achieve near optimum performance at all conditions. Table 3 shows the test conditions. It should be noted that discharge pressure decreases with reduced capacity due to low refrigerant mass flow and reduced heat rejection in an A/C system.

### Table 3. Test conditions.

| Test Condition | Target Capacity (kW) | Compressor Speed (rpm) |
|----------------|-----------------------|-------------------------|
| A              | 17.6 (100%)           | 11,000                  |
| B              | 13.2 (75%)            | 7,200                   |
| C              | 8.8 (50%)             | 4,500                   |
| D              | 4.4 (25%)             | 2,500                   |

The Mini Screw prototype compressor was run at all four conditions without any issues during the tests. The initial performance data is summarized in figure 8 (red line). It can be seen on the horizontal axis that the 17.6 kW full load capacity and respective 75% (13.2 kW), 50% (8.8 kW) and 25% (4.4 kW) capacities were all achieved per the targets. However, the compressor efficiency (EER; vertical axis) did not achieve the targeted results, especially at the reduced capacity conditions. As shown in the figure, EER was inferior by 27% at 4.4 kW unloaded capacity when compared to a current R410A compressor (shown in the black line).

![Figure 8. Mini Screw calorimeter performance (first result).](image)

![Figure 9. Mini Screw calorimeter performance (after clearance tightened).](image)

The main reason of the lower performance was the leakage within the screw rotors. In order to take a conservative approach for initial tests, the first prototype was made with large clearances between the screw rotors and housing. After the first test, the rotor housing was re-machined and components of the clearance were tightened in several steps. Figure 9 shows the performance results for the tightened clearance compressors. The clearance was reduced to half (shown in green line) and then to one-eighth (purple line). It can be seen that EER improved at all four conditions and the performance gap versus the R410A compressor had been narrowed. With one-eighth the clearance, the gap decreased to 4% at 13.5 kW and 11% at 4.5 kW unloaded capacity, respectively.
Table 4 shows a summary of the performance tests along with the OCR results. It can be seen that the OCR maintained levels less than 1% at all four conditions. The compressor was run for more than 800 hours without any issues. Figure 10 shows pictures of the internal components after the performance test.

| Condition | Clearance | R410A compressor |
|-----------|-----------|-------------------|
|           | Original  | Half              | 1/8          |                |
| A         |           |                   |              |                |
|           | Cap: 17.9 kW | Cap: 18.0 kW       | Cap: 18.3 kW | Cap: 17.6 kW |
|           | EER: 0.90  | EER: 0.92          | EER: 0.93    | EER: 1.00     |
|           | OCR: 0.6% | OCR: 0.6%          | OCR: 0.9%    |                |
| B         | Cap: 13.1 kW | Cap: 12.9 kW       | Cap: 13.4 kW | Cap: 13.2 kW |
|           | EER: 0.92  | EER: 0.94          | EER: 0.96    | EER: 1.00     |
|           | OCR: 0.2% | OCR: 0.2%          | OCR: 0.3%    |                |
| C         | Cap: 9.1 kW | Cap: 9.2 kW        | Cap: 9.1 kW  | Cap: 8.8 kW   |
|           | EER: 0.83  | EER: 0.90          | EER: 0.87    | EER: 1.00     |
|           | OCR: 0.1% | OCR: 0.1%          | OCR: 0.6%    |                |
| D         | Cap: 4.3 kW | Cap: 4.7 kW        | Cap: 4.9 kW  | Cap: 4.4 kW   |
|           | EER: 0.73  | EER: 0.83          | EER: 0.89    | EER: 1.00     |
|           | OCR: 0.2% | OCR: 0.2%          | OCR: 0.6%    |                |

*a* Capacity  
*b* EER (Energy Efficiency Ratio) normalized to R410A compressor  
*c* Oil circulation rate [weight ratio: oil / (oil + refrigerant)]

Figure 10. Screw rotors after the test.

After the calorimeter test, the test results were compared with the 1-D simulation data. Table 5 shows the comparison summary. Although the test data agreed fairly well with the 1-D tool at the high speed condition such as A (11,000rpm), the data deviated more at lower speeds, especially 2,500rpm. Also, the prediction error was more pronounced with the original larger clearance compressor. This result indicates that the 1-D tool was not capturing well the increased leakage with large clearance and/or slow speed operation. This result will be used for refining the physics model in the program. The simulation tool will be utilized for further improvement of the Mini Screw compressor.
4. Conclusion
A unique screw compressor concept has been developed for LGWP A1 low pressure refrigerant. Seven times the volumetric flow was achieved in an equivalent size to a typical R410A compressor. Various economical design aspects were introduced in the compressor to accommodate mass production in the future. A prototype was built and the concept was validated. Although the initial performance did not completely reach that of the current R410A compressor, the gap was only 4-11%. At this point in development, there are several areas that have not yet been fully optimized in the prototype such as bearings, motor, oil, gas passages, etc. Therefore, there is a high degree of confidence that the compressor will outperform the existing R410A compressors. It also should be noted that the refrigerant does not have to be limited to DR-12. Any low-to-medium pressure refrigerant that requires large volumetric flow per unit of capacity could be applied with this compressor concept. Further development is currently in progress.

5. References
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