A Study to Improve Efficiency of a Variable Speed Scroll Compressor

Mathew Pazhathara James, Alex Schmig, and Joe Ziolkowski

Abstract A study using a 1D chamber model was done to improve isentropic efficiency of a variable speed scroll compressor and then testing was conducted to validate. It was noticed in the initial analysis that the economizer location, sizing, and timing optimization gave an efficiency improvement of 0.5–2.5% depending on the operating point. Since the economizer was optimized for location, area and timing or activeness, the back flow was reduced, hence the mass flow through the port and efficiency improved. A study with focus on speed resulted in an improvement in weighted efficiency to the tune of 4–6%. Investigating the effect on port delay showed, improvement was on a rising trend as the discharge port delay was increased until around 110° and then efficiency started to decline. The improvements for different operating conditions were in the range of 2–4%. An improvement in involute start angle also resulted in improvement in efficiency to the tune of 0.7–1.3% depending on the operating condition. Changing this parameter affected the structural integrity, hence not adapted. A new compressor was built with new economizer port geometry to understand the effect. The results were promising with variation in efficiency from − 0.6 to 5.1% for different operating conditions investigated. Speed sweeps showed there are peak efficiency points, which can be used to decide the speed for operating condition.

Keywords Variable speed scroll · Efficiency · Economizer · Speed

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1 Introduction

A hermetically-sealed, variable-speed, asymmetric-involute scroll compressor, for transport application, was under consideration for this work. The main intent was to improve efficiency of an existing R452A compressor (baseline compressor), which was manufactured and tested with an elliptical profile injection port. A GTsuite chamber model was used for predicting the behaviour of the manufactured compressor. Five different operating conditions were considered for targeting efficiency for the compressor. These are provided in Table 1. The geometric parameters of the scroll were proprietary, hence were not included in this paper.

The isentropic efficiency was calculated using ASHRAE standard 23–2022 [1]. The equation based on above standard is shown below.

$$\eta_{isen} = \frac{\left[\dot{m}_1\left(h_{3_{1s}} - h_{2_{1s}}\right) + \dot{m}_2\left(h_{3_{2s}} - h_{2_{2s}}\right)\right]}{P} \tag{1}$$

- P Total power input to the unit under test (hence motor power) (kW).
- m_1 total mass flowrate entering the compressor (kg/s).
- m_2 refrigerant mass flow rate after mixing the injection and inlet flow (kg/s).
- h_{21s} Specific enthalpy of refrigerant vapor at suction pressure and temperature entering compressor (kJ/kg).
- h_{22s} specific enthalpy of refrigerant vapor after mixing the intermediate pressure flow at State Point 5 with the flow at State Point 3_{1s} (kJ/kg).
- h_{31s} Specific enthalpy of refrigerant vapor at intermediate pressure following an isentropic compression of the refrigerant from compressor suction pressure (kJ/kg).
- h_{32s} Specific enthalpy of refrigerant vapor at compressor discharge pressure following an isentropic compression of the refrigerant from state point 2_{2s} (kJ/kg).

Table 1 Operating points considered for comparison

Refrigerant: R452A					
Point name	P1	P2	P3	P4	P5
Compressor speed (Hz)	120	75	75	45	15
Suction pressure (PSIA)	18	22	55	23	71
Discharge pressure (PSIA)	300	230	270	226	168
Economizer pressure (PSIA)	50	80	N/A	60	N/A
Suction temperature (°F)	− 19	− 4	29	14	44
Economizer temperature (°F)	26	52	N/A	41	N/A
Suction mass flow normalized	0.5	0.4	1.0	0.2	0.1
Economizer mass flow normalized	0.7	1.0	0.0	0.3	0.0

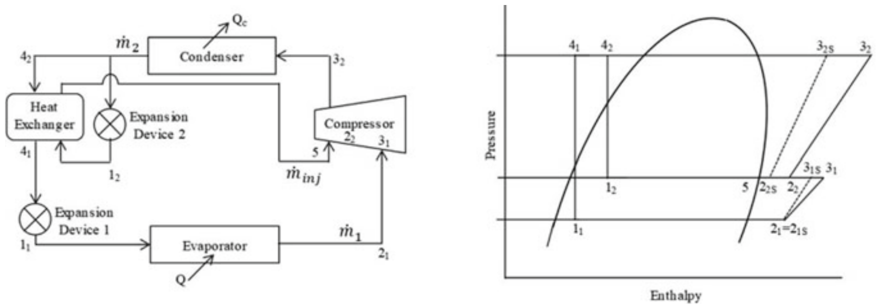

Fig. 1 Cycle and pressure-enthalpy diagram

The cycle schematic and pressure-enthalpy diagram [1] for the system under consideration is shown in Fig. 1. The system makes use of a heat exchanger-based economizer. This addition imparts a subcooling to main cycle defined by state points 4_2 to 4_1. This sub-cooling improves the capacity and hence the COP of the cycle. The data presented in this study is mainly from compressor alone testing, with test points (P1–P5) obtained from cycle's balance points. The cycle balance point determination is not part of current study.

2 Optimization

In this application, the compressor had to meet capacity and efficiency requirements at multiple operating conditions, hence efficient use of design parameters was necessary. While conducting initial assessment [2–6] it was understood that economizer sizing, compressor speed, discharge port geometry and timing had a significant influence on the efficiency of the compressor. A detailed study using a chamber model is presented below in conjunction with test results.

2.1 Economizer

Vapor injection and refrigerant selection was necessary for this design in order to improve system capacity, efficiency, and operating envelope. The different geometries considered are shown in Fig. 2.

Initially, a circular vapor injection port was used. It was intuitive in the design process that, to increase the area, the diameter of the circle had to increase. The consequence of this was that the economizer port initialized while the suction chamber was still active. In addition to this the cycle duration or crank angle for which the economizer is active, started increasing. Hence as the area increased, the cycle duration of the economizer increased and the start angle advanced more. Both resulted

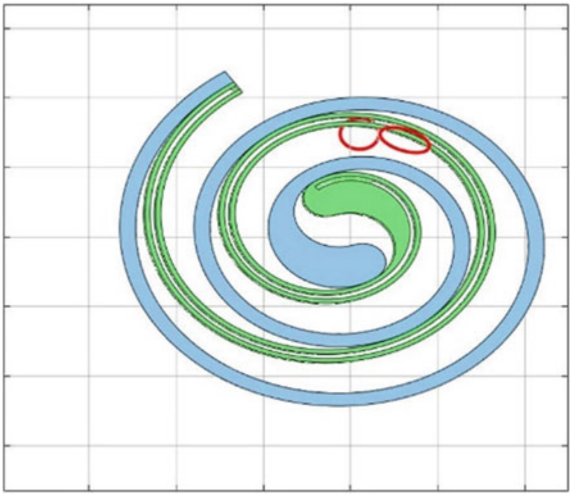

Fig. 2 Circular and elliptical injection ports in red color

in increasing the injection pressure. This posed a challenge since the target injection pressure was held constant. Increased backflow was also noted (Fig. 4). For some of the operating points, the additional area acted as a clearance volume which negatively impacted efficiency.

Another candidate geometry was an ellipse. This reduced cycle duration of the economizer and improved control of the area. Still, the dependency of area and cycle duration was higher. As area was increased to achieve more flow through the economizer, the start angle was changed and the port was moved to a more inefficient zone.

A novel method (patent pending) was devised to break the area and cycle duration relationship by making use of involute geometry itself (Fig. 5). This method ensured better control of start of economizer and could make the start angle constant, unlike other profiles discussed earlier. This novel profile can be defined such that it can be used to control end of injection of the economizer in the direct and indirect chambers.

A plot of different geometric candidates with crank angle is shown in Fig. 3. The novel design better utilized injection pressure ensuring higher flow, hence higher capacity for smaller economizer. Reduction in reverse flow through the injection port for different port geometries is evident in Fig. 4. Economizer mass flow is plotted for four operating conditions in Fig. 6. It should also be noted here that, the port area was lowest for novel design.

There was improvement in efficiency with the novel design as shown in Fig. 7. The last operating point (P5) corresponds to a point where economizer was inactive. But efficiency improved mainly due to the fact that the clearance volume created by the novel design was less as indicated by the area curve cycle duration or activeness in Fig. 3. The improvement from a typical circular port is in the range of 0.2–8.66 points in efficiency depending on the operating condition.

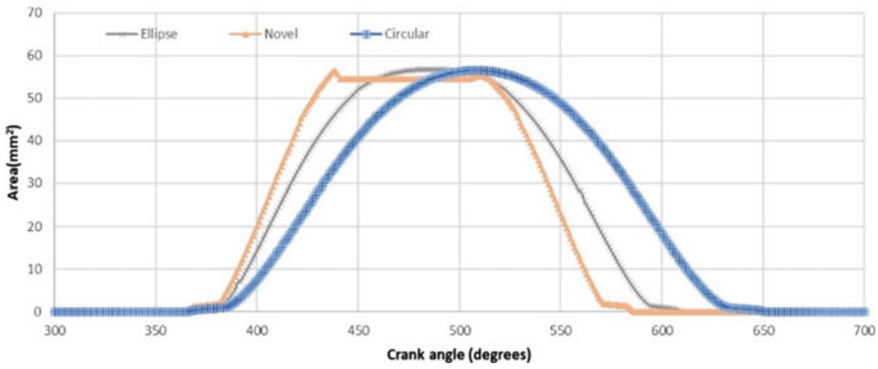

Fig. 3 Area comparison of different geometry

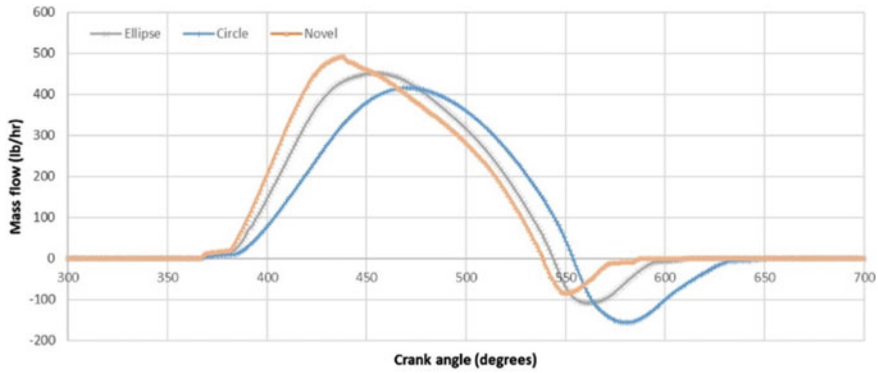

Fig. 4 Mass flow through Injection port during compression cycle

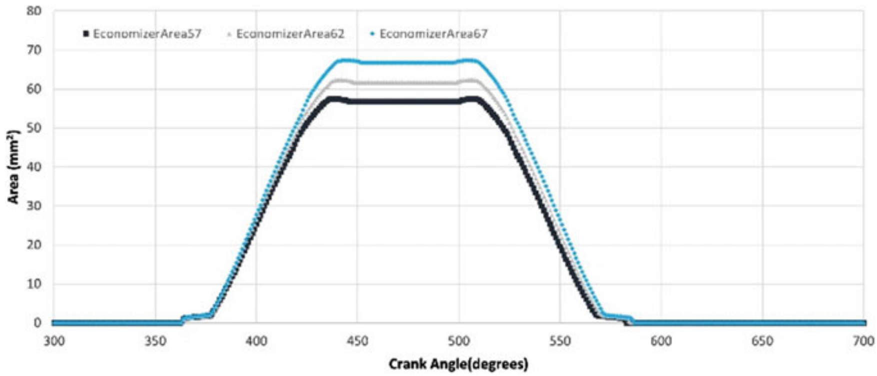

Fig. 5 Areas with same start angle

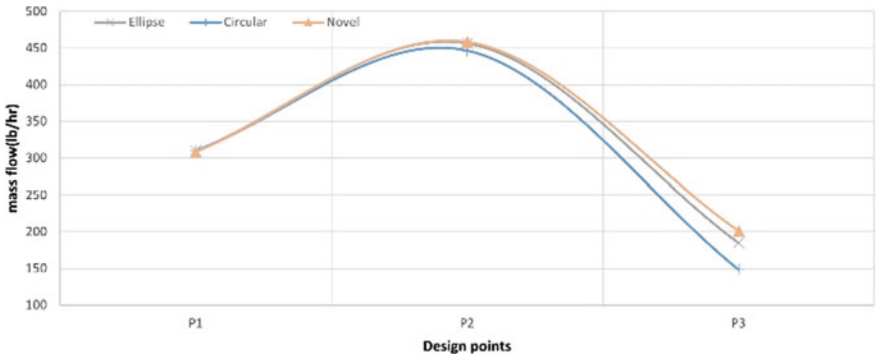

Fig. 6 Economizer mass flow

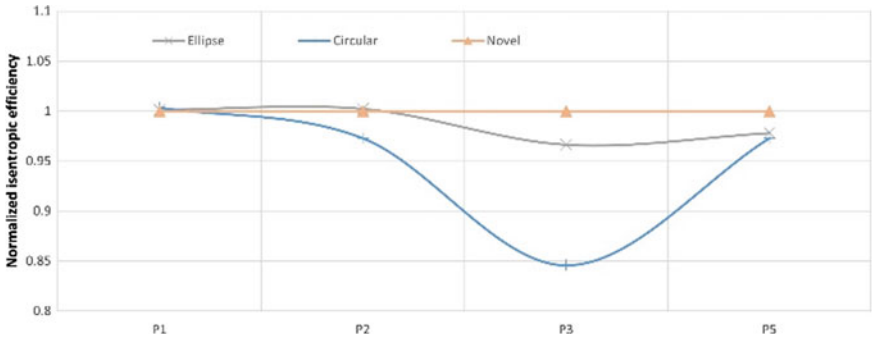

Fig. 7 Normalized concept isentropic efficiency comparison

The study was done on different geometries to improve isentropic efficiency of the baseline compressor, which was underperforming at some of the operating points. Having seen improvement with novel geometry, a new compressor was made with this profile and tested to see whether the analytical prediction aligned with test results. The efficiency comparison for two profiles are shown in Fig. 8. The comparison of mass flow at different conditions are also shown in Fig. 9. The mass flow for the elliptical port was less by 5–70%. The main reason for this change was that the location and cycle duration of the existing injection port was not optimum. When the location was corrected and changed to the novel design, the mass flow, capacity and efficiency improved.

2.2 Speed Optimization

Compressors behave differently at different speed. As speed increases, so does power monotonically (see Fig. 10). While running speed sweeps, however, it was noticed

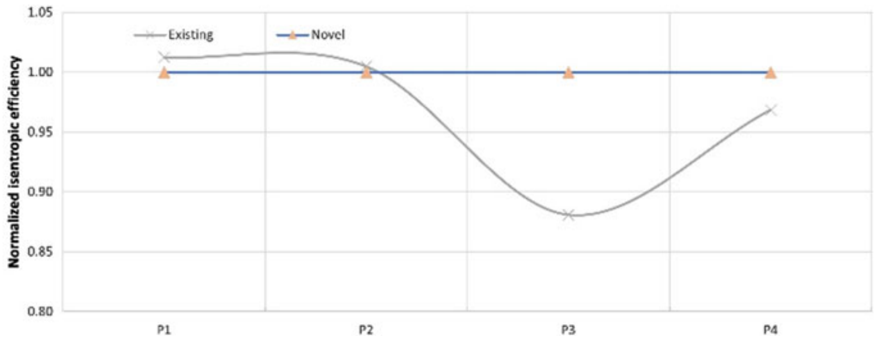

Fig. 8 Normalized tested isentropic efficiency

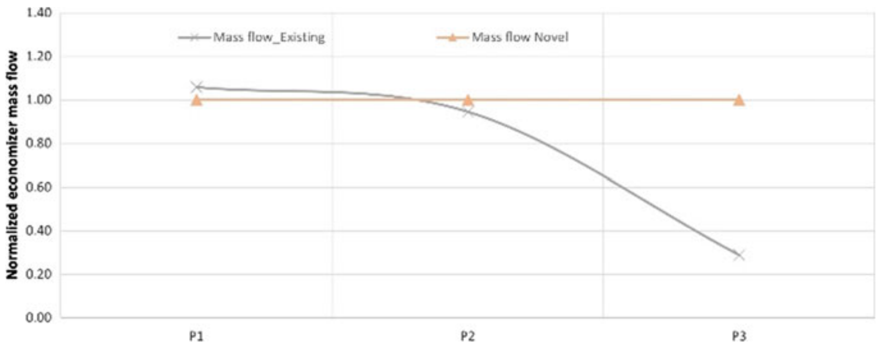

Fig. 9 Normalized mass flow comparison

that the compressor efficiency does not change monotonically with speed, but rather there was a peak efficiency. Also, the speed corresponding to peak efficiency for each operating condition was different. A plot of efficiency versus speed for different operating condition is shown in Fig. 11.

The compressor under discussion was tested at different speeds to validate this as shown in Fig. 11. From the plot it is clear that there are optimum speeds for an operating condition. Hence to extract maximum performance, it is necessary to size the compressor such that it provides design capacity and isentropic efficiency at an optimized speed. At lower speeds the inertia force (orbit scroll rotation) will decrease and hence the contact force arising from this will decrease, resulting in scrolls line contact length along the height of the flank to decrease, resulting in more leakages across chambers, motor efficiency and torque in the chamber model are based on prediction, which has error, etc. are some of the reasons for this behavior.

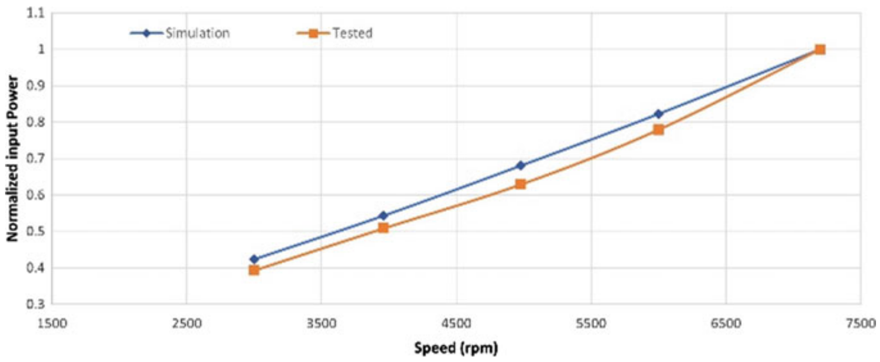

Fig. 10 Normalized power variation with speed

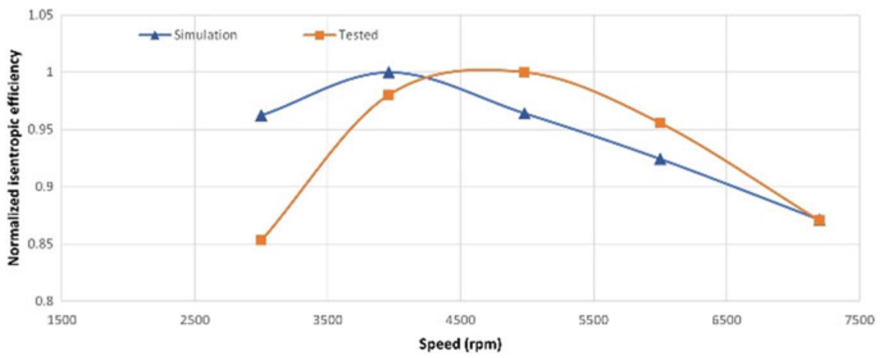

Fig. 11 Normalized efficiency variation with speed

2.3 Discharge Port Delay

During the compressor optimization process it was noticed that, when the discharge port was delayed (see Fig. 12), or volume ratio was changed, the efficiency improved. Like other parameters, this depended on operating condition as evident from Fig. 13. An optimized discharge port geometry, considering different operating condition, was selected. In this case a 110° port delay from the typical discharge timing prescribed by the theoretical maximum circular discharge port was used. Also, from Fig. 13 it was clear that operating point P5 had a decreasing efficiency, due to the low speed producing relatively higher leakage [7]. The improvement in efficiency was approximately 2–5 points depending on the operating point under consideration. This was incorporated in the baseline design.

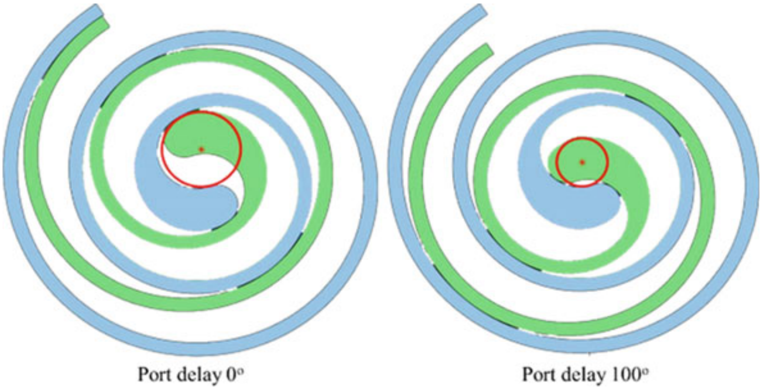

Fig. 12 Discharge port position with different port delay

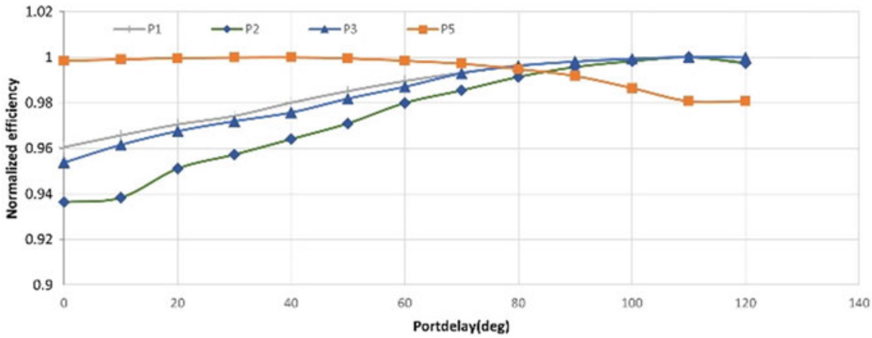

Fig. 13 Variation of efficiency with port delay

2.4 Involute Start Angle

Involute start angle along with trim details [8] determines the leading-edge thickness. Figure 14 shows start angle, trim details and the variation of leading edge of scroll as the start angle and trims are changed. It was noticed that the behavior was like port delay. The efficiency increased as start angle reduced, but as the thickness of the leading side of the scroll involute decreased, structural reliability decreased [9]. For the above reason start angle was not reduced further from the baseline design. Sensitivity of efficiency can be seen in Fig. 15. Change in start angle can bring an improvement to the tune of 0.7–1.3 points in isentropic efficiency.

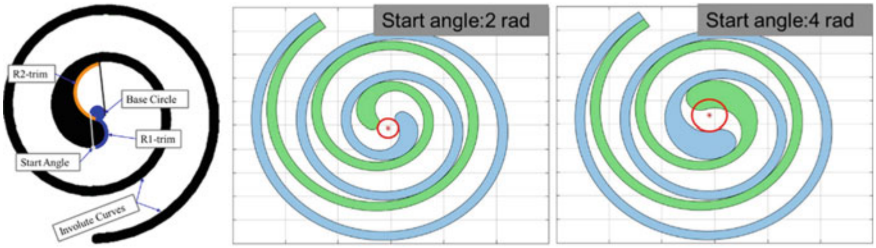

Fig. 14 Geometry nomenclature and examples with different start angle and trims

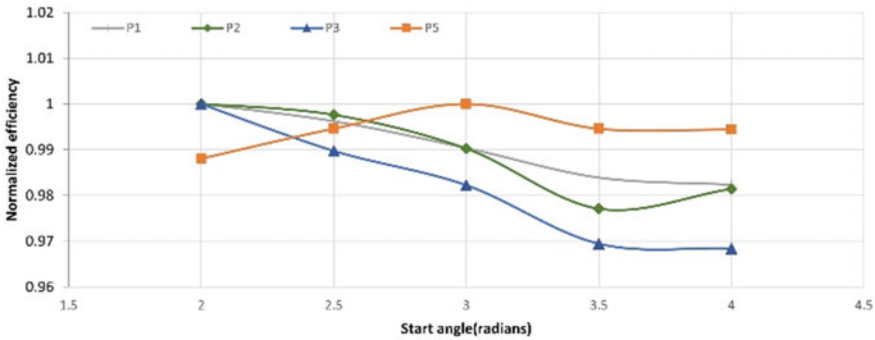

Fig. 15 Effect of start angle on efficiency

3 Conclusion

It is clear from above study that

- A suitably sized and positioned injection port has significant effect on isentropic efficiency of a compressor with injection port.
- Injection port using the novel geometry has a superior performance compared to other port geometries.
- By suitably selecting the speed and economizer port improvement can be seen in the compressor isentropic efficiency in comparison to other port geometries.
- An optimum discharge port delay or volume ratio has significant effect on the isentropic efficiency.
- Though involute start angle affect the reliability, it can still be optimized by suitably selecting the scroll material and thus can add to the compressor efficiency.

Hence different economizer port designs result in different compressor performance. It can be sized to maximize capacity and efficiency for a given type of operating point and reduce the capacity and efficiency for others. Hence this economizer can be sized and tuned based on the requirement of the operation allowing trade offs based on the target operating points.

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References

1. ASHRAE standard 23-2022, *Methods for Performance Testing Positive Displacement Refrigerant Compressors and Compressor Units*
2. J. Sauls, Optimization of *Scroll Compressor Performance with Manufacturing Capability and Reliability Constraints, International Compressor Engineering Conference.* Paper 1633 (Purdue University, 2004)
3. X. Xing, Y. Hwang et al., Refrigerant injection for heat pumping/air conditioning systems: literature review and challenges discussions. Int. J. Refrig. **34**(2), 402–415 (2011)
4. F.M. Tello Oquendo, et al., *A Methodology for Characterization of Vapor injection Compressors, International Compressor Engineering Conference.* Paper 2424 (Purdue University, 2016)
5. Y. Song, et al., *An Experimental Study of a Multi-port Vapor Injected Scroll Compressor, International Compressor Engineering Conference.* Paper 2376 (Purdue University, 2014).
6. R. Baumgart, et al., *Optimization of the Pre-Outlet and Main-Outlet Bores in Scroll Compressors, International Compressor Engineering Conference.* Paper 2763 (Purdue University, 2022)
7. D. Gross, et al., *Scroll Wrap Additional Stress in Variable Speed Application, International Compressor Engineering Conference.* Paper 2584 (Purdue University, 2018)
8. Y. Liu et al., Study on involute of circle with variable radii in a scroll compressor. Mech. Mach. Theor. **45**(11), 1520–1536 (2010)
9. C. Ancel, et al., *Fatigue Design for Scroll Compressor Wraps, International Compressor Engineering Conference.* Paper 2034 (Purdue University, 2010)