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Two-Stage Linear Compressor with Economizer Cycle Where Piston(s) Stroke(s) are Varied to Optimize Energy Efficiency

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ABSTRACT

The energy efficiency of a dual piston linear compressor with two-stage compression and an economizer cycle can be improved by controlling the relative displacement of the pistons. Existing scroll and screw compressors can take advantage of the economizer cycle which increases cooling capacity and improves energy efficiency. A dual piston linear compressor with two-stage compression can take advantage of an economizer cycle in a similar way. The efficiency gain with an economizer cycle depends on many factors, but in particular on the intermediate pressure. By varying the relative displacements of the two pistons, the energy efficiency of the compressor can be maximized for given evaporating and condensing temperatures.

1. INTRODUCTION

The economizer cycle is a commonly used and well-known method of increasing the cooling capacity and efficiency of screw, scroll, and in some instances, multi-piston reciprocating compressors. For screw and scroll compressors, a region of intermediate pressure is created inside the compressor as the mechanical elements move through the compression process. For reciprocating compressors with multiple pistons, a first set of pistons compresses gas from the primary suction pressure to an intermediate pressure, while a second set of pistons compresses gas from the intermediate pressure to the final condensing pressure. In any case, the economizer suction occurs at an intermediate pressure, and liquid refrigerant from the condenser is subcooled to a temperature roughly 5°C above the intermediate saturation temperature. Figure 1 shows a typical economizer cycle.

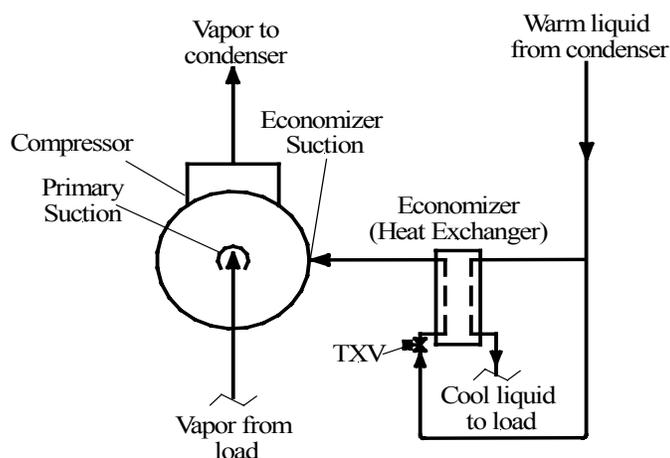


Figure 1: Typical Economizer Cycle

2. LIMITATIONS OF EXISITING APPROACH

Existing scroll and screw compressors see significant performance improvements with the economizer cycle. However, because the intermediate vapor injection point is fixed, the efficiency may not be optimized for various operating conditions. Likewise, multi-piston compressors with, for example, six pistons may have four pistons dedicated to the primary suction while the remaining two pistons are dedicated to the intermediate suction. Thus, the volume ratio of the first stage suction to the second stage suction is fixed at 2:1. Again, this volume ratio will provide significant performance improvements over a non-economized compressor, but may not be ideal for all operating conditions.

3. PROPOSED NEW APPROACH

Given the above discussion, it is possible to optimize the efficiency of a two-stage linear compressor if the volume ratio between the first stage suction and second stage suction is adjusted to match the operating conditions. Figure 2 shows a dual piston linear compressor with an economizer cycle where one piston acts as the first stage suction and a second piston acts as the second stage suction.

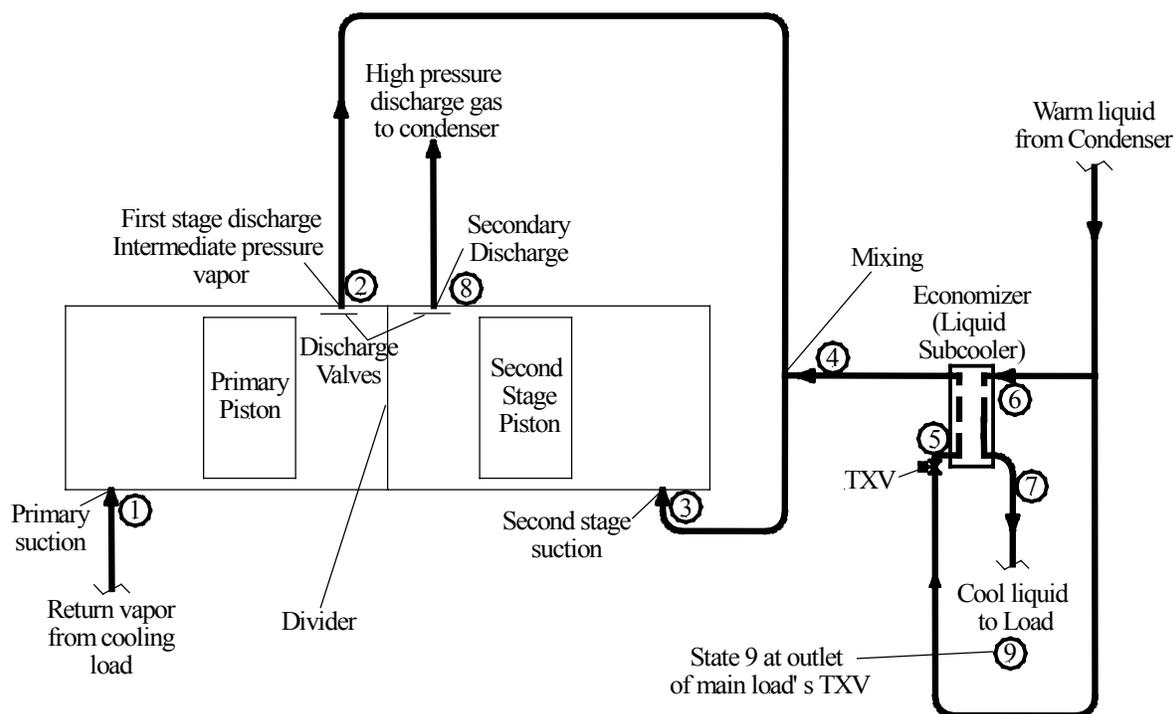


Figure 2: Two-Stage Linear Compressor with Economizer Cycle

As shown, gas is discharged from the first stage (left side) piston. Gas from the economizer (a heat exchanger) is mixed with discharge gas from the first stage, and this mixed flow enters the second stage (right side). Discharge valves are shown, and suction valves are assumed in the face of each piston. Figure 3 shows a pressure-enthalpy diagram of this approach.

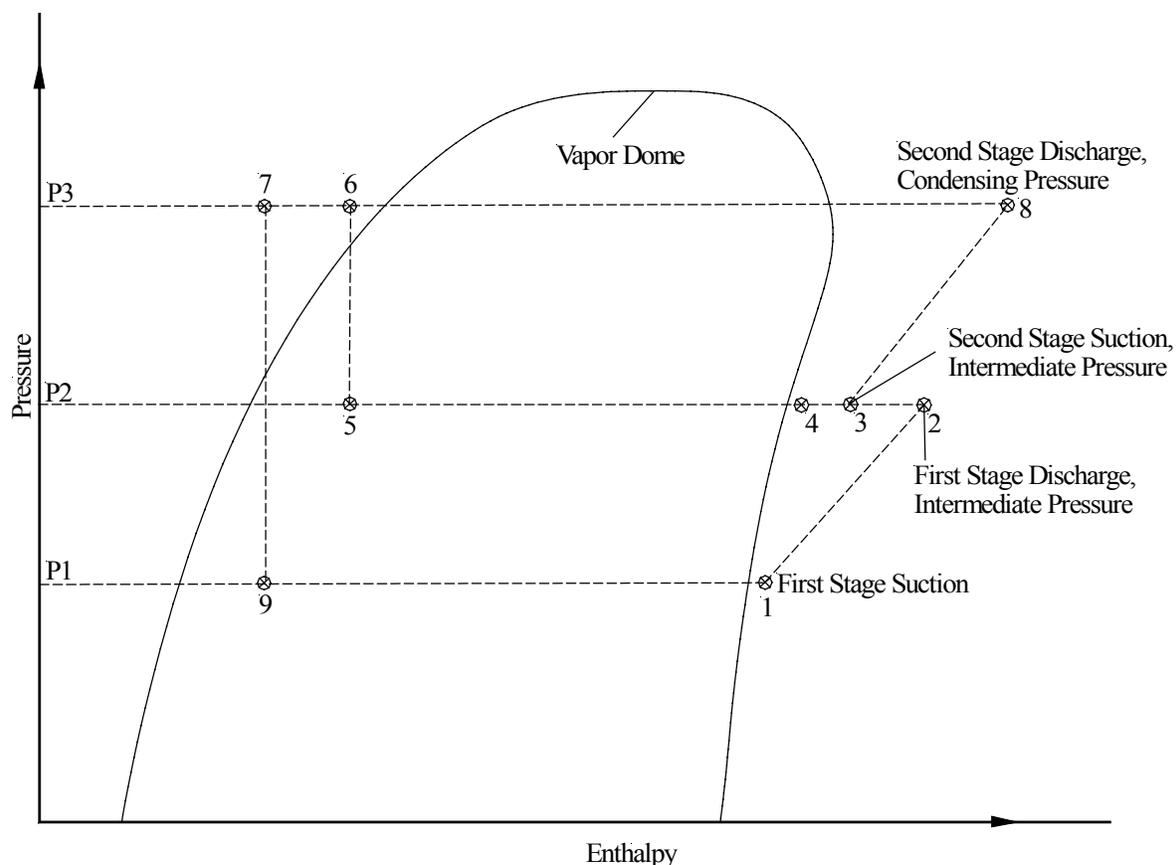


Figure 3: Pressure-Enthalpy Diagram of Two-Stage Compression with Economizer Cycle

The overall Coefficient of Performance (COP) of this cycle is the cooling effect divided by the total work input to both compression stages:

$$\text{COP} = Q_1 / (W_1 + W_2) \quad (1)$$

Where:

Q_1 = Cooling effect at main evaporator
 W_1 = Work of compression, first stage
 W_2 = Work of compression, second stage

Further:

$$Q_1 = m_1(h_1 - h_9) \quad (2)$$

$$W_1 = m_1(h_2 - h_1) \quad (3)$$

$$W_2 = m_2(h_8 - h_3) \quad (4)$$

Where:

m_1 = mass flow rate through first stage

m_2 = mass flow rate through second stage

$h_1, h_2, \text{ etc.}$ are enthalpy values at the corresponding points on the pressure-enthalpy diagram

Let's also define the volumetric flow ratio as follows:

$$\text{Volumetric Flow Ratio} = \text{First Stage Displacement} / \text{Second Stage Displacement} \quad (5)$$

When both pistons are moving at the same frequency, there is a clear relationship between the relative displacements of the two pistons and the intermediate pressure. For example, if the first stage piston is fixed (zero stroke), the volumetric flow ratio is zero and the intermediate pressure will be essentially equal to the first stage suction pressure. At the other extreme, if the second stage piston is fixed (zero stroke) then the volumetric flow ratio is infinite, and the intermediate pressure will be essentially equal to the condensing pressure. Accordingly, if the volumetric flow ratio is set between these two extremes by controlling the stroke of one or both pistons, the intermediate pressure will fall between the first stage suction pressure and the condensing pressure.

A full discussion of COP calculations is beyond the scope of this manuscript. However, the COP values at various volumetric flow ratios and operating conditions may be calculated by hand or by using refrigerant property computer programs which are readily available. Some assumptions used in the COP calculations were:

- 80% Isentropic efficiency for each stage of compression
- 5°C approach temperature for the economizer
- No ambient liquid subcooling for the liquid leaving the condenser
- Refrigerant is R410A
- 18°C return gas temperature to the first stage suction
- No pressure drop or heat gain/loss in any connecting lines

4. RESULTS

The results of COP calculations are shown in Figure 4. Calculated COP values are on the Y-axis while the X-axis has two scales: 1) intermediate saturation temperature, and 2) volumetric flow ratio. Figure 4 shows one operating condition, -40°C evaporating and 50°C condensing. As shown, the COP is highest when the volumetric flow ratio is approximately 3.14, and this volumetric flow ratio corresponds to approximately 5.6°C intermediate saturation temperature on the bottom X-axis. Further, the optimized COP value of 1.42 is approximately 2% greater than the COP of 1.39 at the 2:1 volume ratio. From this figure, it can be seen that the system can be optimized by controlling the volumetric flow ratio, which also controls the intermediate saturation temperature.

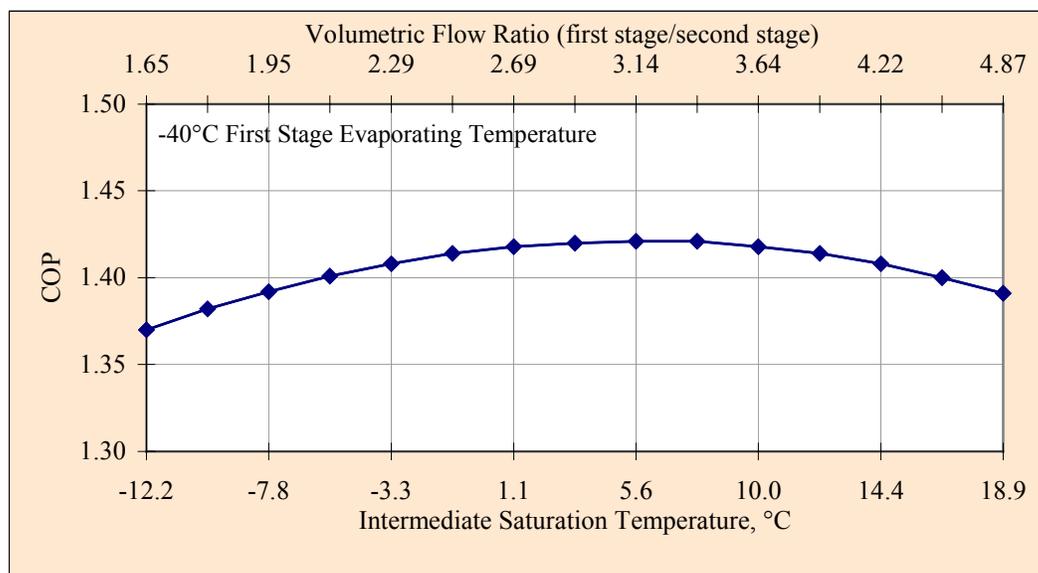


Figure 4: COP of Two-Stage Compressor with Economizer Cycle at Various Volumetric Flow Ratios and Intermediate Saturation Temperatures

Figure 5 takes this idea a step further. Condensing temperature is shown on the X-axis, and the intermediate saturation temperature and volume flow ratio are shown on the two Y-axis. The lines on the graph indicate where the volume flow ratio must be to achieve the highest efficiency. For example, at 48.9°C condensing, the intermediate saturation temperature must be approximately 6°C and the volume flow ratio must be approximately 3.2 to get the highest efficiency – this was also shown in Figure 4. On the other hand, at 21.1°C condensing, the intermediate saturation temperature must be approximately -11°C and the volume flow ratio approximately 2.2 to get the highest efficiency. For any condensing temperature, the best (highest efficiency) intermediate saturation temperature and volume flow ratio can be found by selecting the appropriate points on the graph.

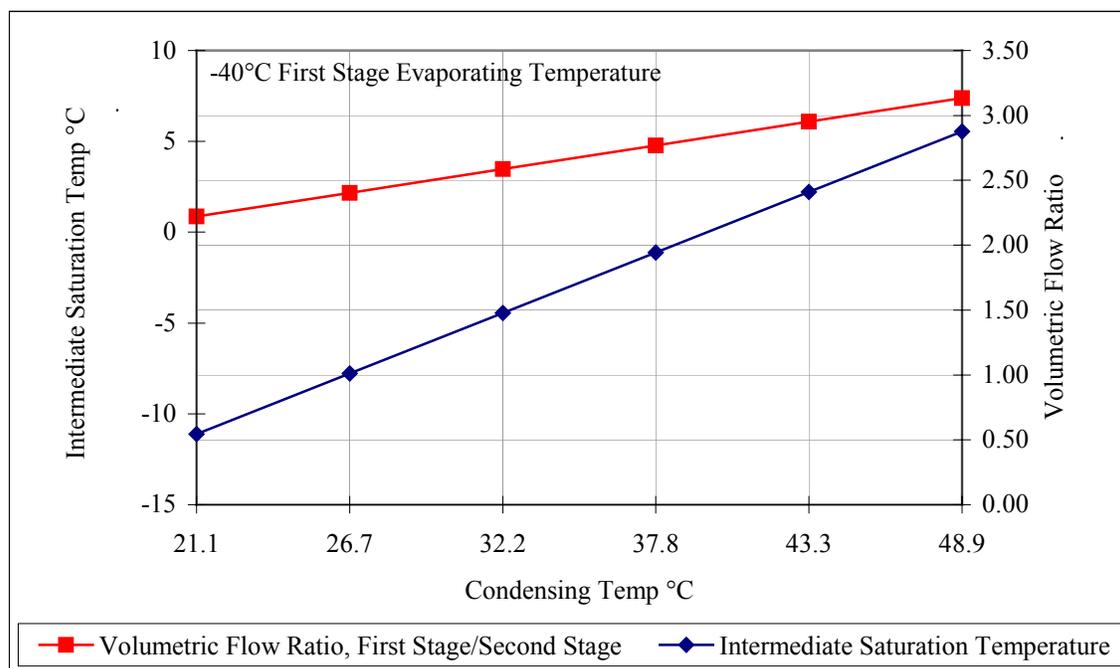


Figure 5: Intermediate Saturation Temperature and Volumetric Flow Ratio which Yield Highest COP

While Figure 5 shows the optimized volume flow ratio only for -40°C evaporating, Figure 6 shows optimized volume flow ratios for other evaporating temperatures. Accordingly, the most efficient intermediate saturation temperature and volume flow ratio can be found for other operating conditions by locating appropriate points in Figure 6. For example, at -18°C evaporating and 32.2°C condensing, the volume flow ratio where the highest efficiency occurs is approximately 1.7, which corresponds to 7°C intermediate saturation temperature.

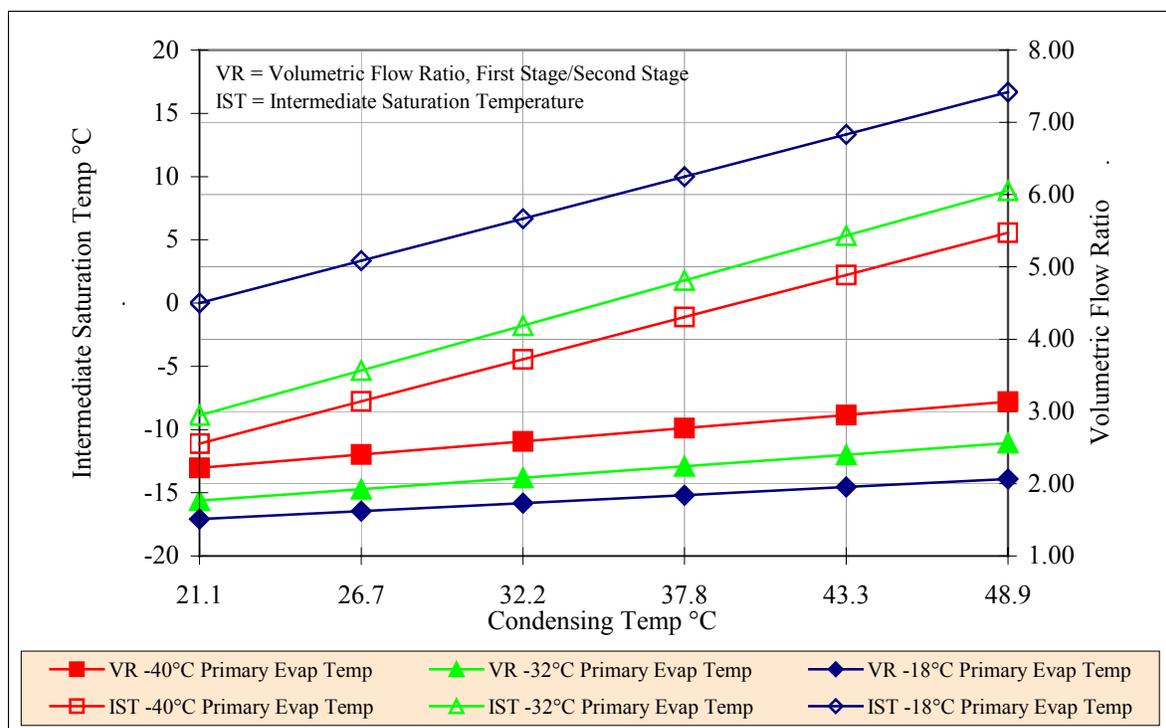


Figure 6: Intermediate Saturation Temperature and Volumetric Flow Ratio which Yield Highest COP at Various Conditions

5. CONCLUSIONS

Given the above discussion, it can be seen that the efficiency of a two-stage linear compressor can be optimized by adjusting the relative displacements of the first and second stages. Reducing the displacement of the first stage tends to decrease the intermediate pressure, while conversely increasing the displacement of the first stage tends to increase the intermediate pressure. The intermediate pressure that gives the highest COP depends on the specific operating conditions such as evaporating and condensing temperature. Thus, the intermediate pressure can be controlled such that the compressor runs at highest COP.