Selection of Twin Screw Compressor Economizer Port Location to Optimize Unit Efficiency

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ABSTRACT

Twin screw compressors are widely used in many applications. The performance limits of screw compressors are continuously being pushed to meet increased demand for energy efficiency. Several design adjustments can be made to the compressor to create a more efficient unit, such as the proper design of an economizer. The economizer is a port where refrigerant vapor is (re-)injected. General compressor effects of using an economizer are increase in compressor power, increase in discharge mass flow, and reduction of isentropic efficiency. However, the unit experiences an increase in refrigeration capacity and efficiency. The use of economizers for screw compressors is a developing application. This paper examines the addition of an economizer to a water chiller system with the intent of achieving a unit efficiency improvement. The economizer port size and location impacts the compressor performance. For this study, the port size location is varied to understand how it affects the unit performance. A useful procedure is developed to optimize the unit efficiency using internal compressor and system modeling tools is discussed within this paper. The selected port location will be used to create a new screw compressor design that is anticipated to achieve a 2% to 3% improvement in the unit coefficient of performance.

1. INTRODUCTION

Refrigeration and Air-conditioning systems have been around since the 1900's implementing the basic vapor compression cycle. The incorporation of these systems have become vital to daily life. However, there are several challenges that are met with the use of refrigeration and air-conditioning systems. One major challenge is global warming which stems from poor refrigerants and a high electrical demand. Over the years, many adjustments have been made to these systems to address these challenges. In order to improve environmental conditions, enhancements can be made to the vapor compression cycle or the components within the vapor compression cycle (Moon et.al, 2008). The vapor compression cycle consists of compressor, condenser, expansion valve, and evaporator. The compressor component has a large impact on the systems overall performance thusly the compressor will be the component of focus within this paper specifically the screw compressor. Screw compressors have been used for several years for a wide variety of applications. In addition to altering the compressor component, the cycle is also being modified to incorporate vapor injection. In the recent years, the advantages of vapor injection or economizers in compressor types has been explored and is known to have a positive impact on the overall performance of the cycle. The addition of an economizer within a twin screw compressor has two key benefits; increase in coefficient of performance (COP) and capacity (Wang et al., 2009). This paper observes the impact the economizer port size and location have on a water chiller system with the intent of achieving a unit efficiency improvement.

2. VAPOR COMPRESSION CYCLE

The vapor compression cycle in Figure 1 is commonly used in refrigeration and air-conditioning systems. During the vapor compression cycle the refrigerant goes through four stages. The refrigerant is compressed, after compression the vapor is condensed to a pure liquid. The pure liquid is then expanded reducing in pressure and temperature. The quality of the refrigerant changes as it goes through the expansion process and consist of two phase flow. Lastly, the refrigerant goes through the evaporator and leaves as a gas. Useful cooling and boiling is obtained using the liquid within the evaporator however the vapor is energy wasted. This is referred to as the lost refrigeration effect. The total loss varies with system pressure differential and refrigerant being used. However, this inefficiency has been known to contribute to over 30% of loss using R134a (Cambio, 2015).

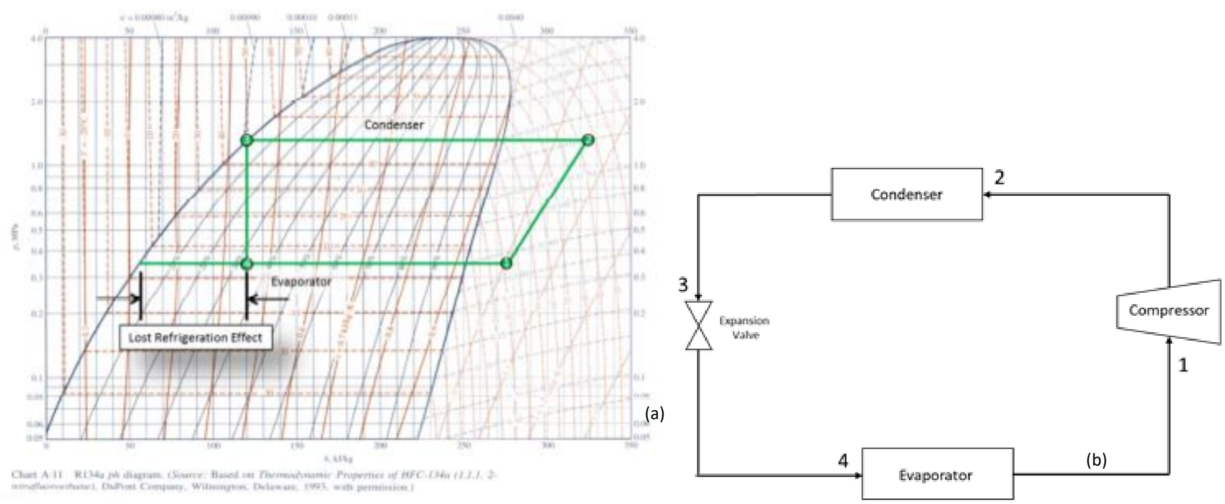

Figure 1: (a) p-h chart of vapor compression cycle (b) schematic diagram of vapor compression

One modification that can be implemented to improve the lost refrigeration effect within the vapor compression cycle is the inclusion of vapor injection also known as an economizer. The vapor compression with vapor injection cycle includes basic components with the addition of a flash-tank and modified compressor that contains an economizer port to allow for vapor injection. The vapor compression cycle with vapor injection (seen in Figure 2) is slightly different from the basic cycle. The refrigerant vapor enters the condenser (point 4). From the condenser (point 5) the refrigerant is expanded to an intermediate pressure and enters the flash-tank (point 6). The liquid (point 7) and vapor (point 9) are separated from each other within the flash-tank. This is done by using both velocity reduction and gravity effect which allows the liquid portion to exit the bottom of the tank. The liquid portion is further expanded and continues on to the evaporator (point 8). The refrigerant enters at suction of the compressor (point 1) from the evaporator which starts the 1st stage of compression. The vapor portion in the flash tank is injected into the compressor through the economizer port (point 9) and mixes with the refrigerant in the compression chamber (point 2) and this starts the 2nd stage of compression (point 3). This is often referred to as flash-tank vapor injection. There are several benefits of using a flash-tank cycle, namely; lower compressor discharge temperature, increases capacity, and reduces power consumption (Xu et al., 2010). In order to obtain the efficiency benefits, the economizer port size and location are important parameters that should be optimized.

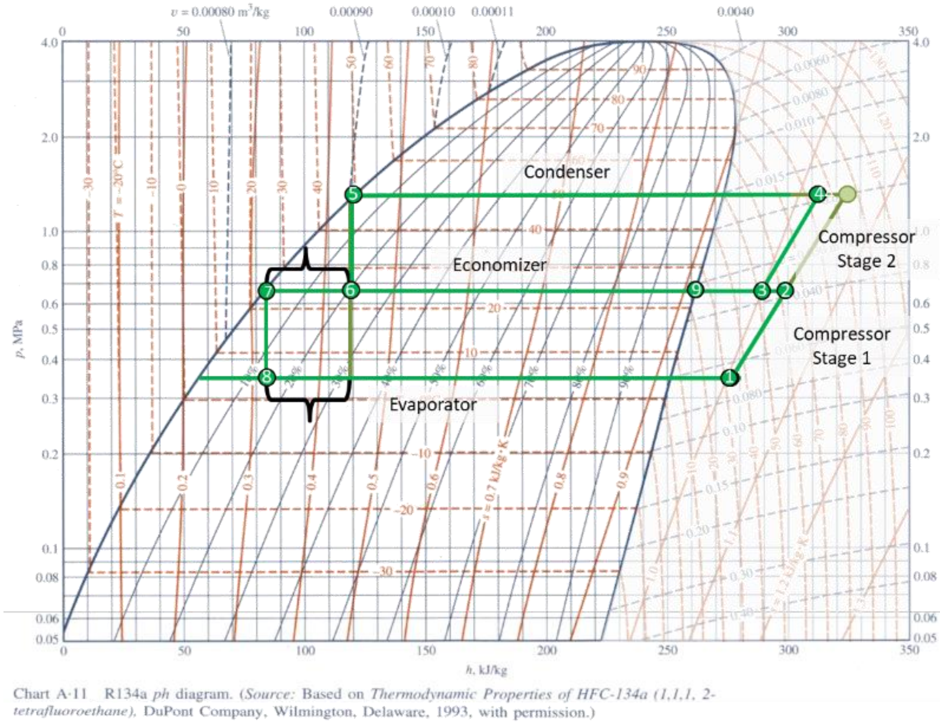

Figure 2: Vapor compression cycle with vapor injection

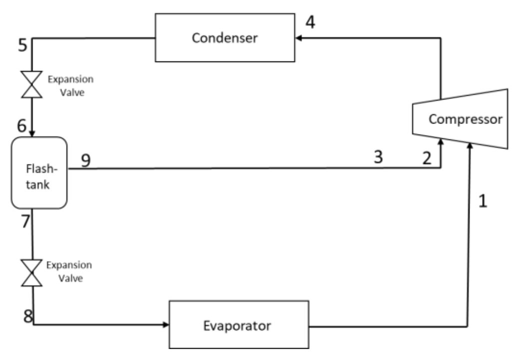

Figure 3: Schematic diagram of vapor compression cycle with vapor injection

3. ECONOMIZER PORT OPTIMIZATION

The inclusion of an economizer increases the overall unit performance. Two critical design parameters of compressors with economizer are port size and location. In-house analytical tools were used to determine a location and size that maximizes the overall unit performance. Each parameter affects the performance differently and the prime combination is desired. The final results are compared to the unit without an economizer port with COP, power, and capacity at 5.89, 317.43 kW, and 413.14 tons respectfully. R134a is the working fluid for this study. Theoretically the flash-tank cycle was modeled to understand how COP changes with respect to the economizer injection pressure while assuming a 100% compressor isentropic efficiency. The economizer injection pressure is calculated using Equation (1) where evaporator and condenser saturation temperature is 40°F and 100°F respectfully. The pressure split, r, is defined as the percentage of pressure between condenser and evaporator and is varied from 10% to 80% with 10% being closer to the pressure of the evaporator and 80% being closer to the condenser. As seen in Figure 4, the maximum COP is found around an injection pressure of 40% of the compressor total pressure rise. Furthermore, this shows us that the COP has a strong dependence on the economizer pressure in the vapor compression cycle.

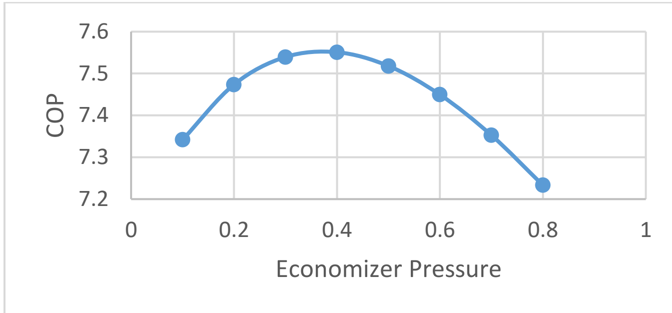

Figure 4: COP vs. Economizer Pressure

Applying this understanding, the economizer port size and location parameters are evaluated using a screw compressor model with the addition of an economizer port. The analysis was completed with an in-house 1-D compressor model along with a unit model. The 1-D screw compressors model is a Trane internal design code. The model follows conservation of mass and energy by modeling a single compression chamber. Leakages from the upstream and downstream chambers are accounted for. The model is comprised of an inner loop which use commercial differential equation solver for the compression process and an outer loop which takes care of parasitic losses such as motor and bearing losses, heat transfer, etc. Inputs to the compressor model include chamber volume and port areas vs. crank angle, as well as motor efficiency and mechanical loss models. In this case a flash tank economizer was used to simplify the analysis. Other inputs include suction, discharge and economizer saturation conditions. However, when conducting a study of economizer port geometry and attempting to capture the unit effects it was necessary to link the compressor model to a Trane internal unit modeling tool. The unit model has detailed models of the heat exchangers and economizer so that the effects on the unit of changing mass flow are accounted for. This was done by transferring mass flows and power from the compressor model to the unit model, solving the unit model and transferring saturation conditions back to the compressor model in an iterative method until mass flows and power was converged. This modeling tool produced unit capacity, coefficient of performance (COP), and other unit values. Suction flow, economizer flow, and power, are calculated by the compressor model using Equations (2) - (6).

$$p_{econ} = r\left(p_{cond} - p_{evap}\right) + p_{evap} \tag{1}$$

$$\dot{m}_{suc} = \frac{V * N * \omega}{v_{suc} * 2\pi} \tag{2}$$

$$\dot{m}_{econ} = \int_{t_1}^{t_2} \frac{dm_{econ}}{dt} dt \tag{3}$$

$$\frac{dm_{econ}}{dt} = A_{econ} * C_d * \left(p_{econ} \sqrt{\frac{2 * k * \rho_{econ}}{(k-1)p_{econ}}} * \sqrt{\left(\frac{p_{pocket}}{p_{econ}}\right)^{\frac{2}{k}} - \left(\frac{p_{pocket}}{p_{econ}}\right)^{\frac{k+1}{k}}} \right) \tag{4}$$

$$P_{shaft} = \dot{m}_1(h_2 - h_1) + \dot{m}_9(h_2 - h_9) \tag{5}$$

$$P_{kw} = \frac{P_{shaft}}{\eta_{motor}} \tag{6}$$

3.1 Size

For this study, the port size ranges from 0.2 in. to 1.2 in. while keeping the port location at 25 degrees of the male rotor rotation. This position is just after closure of the suction port. The evaporator and condenser saturation temperatures are 43°F (6.11° C) and 96°F (35.56° C). When the size goes below 0.2, the injection flow decreases substantially and there is no reason to reduce the port further in size. Figure 5 shows the change in performance due to port size. It is seen that as the port decreases in size, the unit performance increases. This is partly due to the decrease in leakage effect. The increase in COP is mostly due to the economizer injection pressure moving closer to the middle of the cycle. The middle of the cycle is defined using the condenser and evaporator pressures. Likewise, the economizer port size and injection pressure affect the economizer injection flow which in turn affects the pocket pressure and fixed volume ratio effects. Additionally, as the economizer port gets smaller in size the leakage effect from the high-pressure pocket to the low pressure pocket decreases. Observing the trend, there is no maxima reached and this is likely due to the placement of the economizer port being so close to suction and the large reduction in economizer flow. As the economizer flow decreases, the condenser flow and saturation temperature decreases as well which leads to a lower condenser pressure and an increase in cycle performance.

Table 1: Size vs Unit Performance

Case #	D_{econ} in (cm)	Angle (Deg)	Econ Flow CFM (m^3/s)	COP	% COP Improvement	Power (kW)	Capacity Tons
1	**1.2 (305)**	**25**	79.48 (0.038)	6.09	3.3%	264.72	458.18
2	**1 (254)**		69.57 (0.033)	6.10	3.6%	262.18	455.05
3	**0.8 (203)**		54.02 (0.026)	6.12	4.0%	258.10	449.35
4	**0.6 (152)**		33.34 (0.016)	6.14	4.3%	251.95	440.03
5	**0.4 (.02)**		15.10 (0.007)	6.17	4.7%	244.85	429.51
6	**0.2 (50.8)**		3.80 (0.002)	6.18	5.0%	240.36	422.58

Table 2: Running conditions per case with varying size

Case #	D_{econ} in (cm)	Angle (Deg)	Econ Pressure Psi (MPa)	Evap Sat Temp °F (°C)	Cond Sat Temp °F (°C)	Normalized Compressor Efficiency
1	**1.2 (305)**	**25**	76.32 (0.526)	43.40 (6.33)	96.39 (35.77)	0.957
2	**1 (254)**		79.40 (0.547)	43.40 (6.33)	96.37 (35.76)	0.959
3	**0.8 (203)**		84.80 (0.585)	43.41 (6.34)	96.23 (35.68)	0.963
4	**0.6 (152)**		92.79 (0.640)	43.43 (6.35)	96.01 (35.56)	0.957
5	**0.4 (102)**		95.68 (0.660)	43.46 (6.37)	95.75 (35.42)	0.988
6	**0.2 (50.8)**		96.66	43.47 (6.37)	95.58 (35.32)	1.000

3.2 Location

The location of the economizer port can be chosen to maximize capacity, performance, or something in between. The scope of improvement has a huge impact on the placement of the economizer port. Higher COP is found when the vapor is injected at an optimum pressure between the suction and discharge pressures of the compressor. There are several things to consider when placing the injection port. Injecting the vapor too close to suction can cause the vapor to enter the suction. The effects of vapor entering at the suction decreases the unit performance (Wang et.al, 2009). The injected vapor displaces flow from the evaporator and reduces suction capacity. Likewise, placing the injection port too close to the discharge will have a negative impact on economizer effectiveness due to discharge gas reversing flow through the economizer. Due to the helical nature of the rotors, the angular position of the male rotor translates to and axial distance along the axis of rotation of the rotor (seen in Figures 5 and 6).

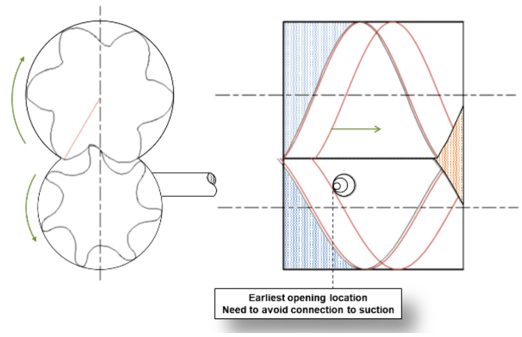

Figure 5: Economizer port location representation at the low-pressure end of rotors

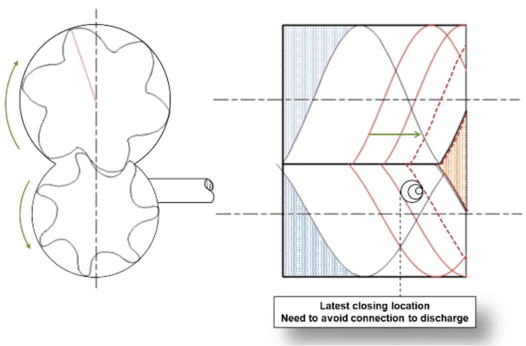

Figure 6: Economizer port location representation at the high-pressure end of rotors

For this study, the location ranges from 25 to 120 degrees of the male rotor rotation with the evaporator and condenser saturation temperatures roughly at 43°F (6.11° C) and 96°F (35.56° C). The maximum location was determined by the opening of the discharge port. Once the economizer port overlapped the discharge process, the compressor model would no longer run. Table 2 shows the change in performance and capacity due to port location. As the location moves further from suction the unit performance increases. When the port was placed at an intermediate pressure with the economizer pressure at roughly 95.52 psi (0.659MPa), the positive improvement on COP began to decrease. The intermediate pressure is approximately 55% of the difference of the evaporator and condenser pressures. From this information, it is seen that the unit performance has a strong dependence on the intermediate pressure. As the location of the port moves to a higher pressure in the compression process, the performance increases. In Figure 5, it is seen that the location has a greater influence on the COP than size does. It is seen in Tables 1 and 2 that the size and location of the port has a large influence on economizer pressure. The COP trend agrees with the original theoretical analysis shown in Figure 4 with a variation in the location of the peak. In theory, for the ideal case the maximum COP would be located around 40% while in this study it was located around 50%. Most likely the discrepancy between the theoretical analysis and the compressor model are changes in efficiency due to volume ratio effects. From this analysis, it was seen that when decreasing the size and moving the location further from suction an optimum improvement can be found. Both alterations affect the economizer injection pressure and overall system performance. Additionally, the compressor evaporator to condenser saturation temperatures ratio is designed for 40°F (4.44° C) to 100°F (37.78° C) while the conditions for this analysis are a lower pressure ratio which causes over compression. Over compression is increased with economizer injection which explains why the maximum compressor efficiency occurs with the smallest economizer flow.

Table 3: Location (angle) vs. Unit Performance

Case #	D_{econ} in (cm)	Angle (Deg)	Econ Flow CFM (m^3/s)	COP	% COP Improvement	Power (kW)	Capacity Tons
1	0.8 (203.2)	25	54.22 (0.026)	6.12	3.90%	258.23	449.42
2		60	42.17 (0.02)	6.17	4.82%	253.62	445.24
3		70	37.69 (0.018)	6.19	5.07%	251.88	443.21
4		85	30.75 (0.015)	6.21	5.39%	249.10	439.67
5		95	25.61 (0.012)	6.21	5.51%	247.13	436.68
6		109	17.92 (0.008)	6.22	5.52%	244.39	431.90
7		120	11.82 (0.006)	6.21	5.49%	242.25	427.99

Table 4: Running conditions per case with varying location (angle)

Case #	D_{econ} in (cm)	Angle (Deg)	Econ Pressure Psi (MPa)	Evap Sat Temp °F (°C)	Cond Sat Temp °F (°C)	Normalized Compressor Efficiency
1	0.8 (203.2)	25	84.72 (0.584)	43.41 (6.34)	96.23248 (35.68)	0.965
2		60	89.47 (0.617)	43.42 (6.35)	96.12066 (35.62)	0.976
3		70	91.20 (0.629)	43.43 (6.35)	96.06919 (35.59)	0.980
4		85	93.56 (0.645)	43.44 (6.36)	95.98098 (35.54)	0.986
5		95	94.62 (0.652)	43.44 (6.36)	95.90901 (35.51)	0.990
6		109	95.52 (0.659)	43.45 (6.36)	95.79605 (35.44)	0.995
7		120	95.84 (0.661)	43.46 (6.37)	95.70431 (35.39)	1.000

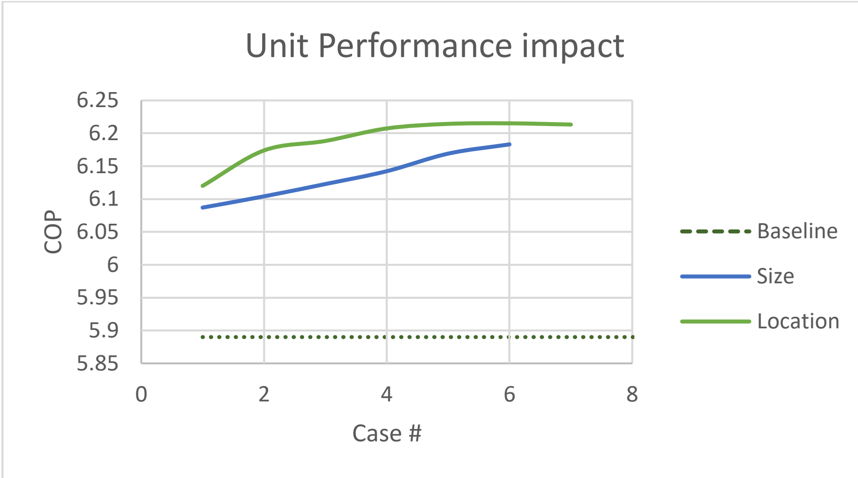

Figure 5: Graph of Unit Performance showing difference between size and location

4. CONCLUSION

The basic refrigeration cycle improvement in HVAC systems are continuing to improve. The exploration of economizers continues to increase COP within various application thusly reducing the impact on the environment. An optimized screw compressor economizer port location and size was analytically discovered in this paper to increase the unit efficiency within a water-chiller system. The design produced a predicted COP improvement of 5.52% which exceeded the 2-3% target. Ultimately, the location proved to have a larger impact on the COP in comparison to size. The change in overall performance with varying size did not follow the theoretical trend due to the influence of other parameters such as compressor efficiency, and saturation temperatures. Further investigation is required to determine size impacts when placed a significant distant past suction. Moreover, test will be conducted to determine the accuracy of these results. Based on the analytical models used, the results indicate:

- Moving the injection port location further away from suction increases the performance as a result of the intermediate pressure becoming closer to the optimum economizer injection pressure
- The size of the port caused an increase in performance due to a decreased leakage effect, reduced economizer flow, and movement of the economizer pressure towards the middle of the cycle.
- Optimal improvement can be found when adjusting both port size and location

NOMENCLATURE

The nomenclature should be located at the end of the text using the following format:

A	Area	(m^2)
C_d	Injection Port Flow Constant	(–)
COP	Coefficient of Performance	(–)
D	Diameter	(cm)
g	Gravitational Constant	(–)
h	Enthalpy	(kJ/kg)
k	Isentropic Exponent	
$\dot{m}$	Mass Flow	(kg/s)
m	Mass	(kg)
N	Number of Male Lobes	
η	Efficiency	(–)
p	Pressure	(Pa)
P	Power	(kW)
ρ	Density	(kg/m^3)

v	Specific volume	(kg/m³)
V	Suction Chamber Volume	(m³)
ω	Shaft angular velocity	(rad/sec)
X	vapor quality	(–)

Subscript

cond	Condenser
econ	Economizer port
evap	Evaporator
kW	Electirical
pocket	Compression Chamber
shaft	Shaft
suc	Suction

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