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ECONOMIZER CYCLE IN AIR CONDITIONING SYSTEMS WITH ROTARY VANE COMPRESSORS

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This paper presents the results of mathematical investigation of a complete R134a air conditioning system with an added economizer circuit. Principles of operation of air conditioning system with the economizer are described. Modelling techniques of the economizer thermodynamic cycle and rotary vane compressors with the economizer induction are presented. Potential economizer effect, system performance, operating pressures, and exergy analysis results are discussed. Conclusions are given.

INTRODUCTION

It is well-known that introduction of a regenerative heat exchanger into a refrigeration cycle does not necessarily provide an increase in cooling capacity. It does increase specific cooling capacity, but at the same time it reduces density of superheated refrigerant vapor. The reduced density causes a degradation of refrigerant mass flow rate pumped by a compressor. As a result, cooling capacity, which is as a product of specific cooling capacity and mass flow rate, is not as improved as specific cooling capacity, if at all.

An economizer cycle is applied in order to definitely increase capacity of air conditioning refrigeration and heat pump systems due to vapor injection over an economizer port /1/. The economizer port is located between the discharge outlet and the suction inlet at a point where the suction process is accomplished and the compression is to start. A part of liquid refrigerant after a condenser is used to additionally subcool a main part of the liquid refrigerant directed to an evaporator. The first part boiled out after the subcooling action at economizer pressure, which is higher than suction pressure, is introduced into the compression process without any impact on refrigerant vapor at the suction side. In the economizer cycle there is no degradation of refrigerant mass flow rate associated with the additional superheat and the vapor density reduction.

Extensive analysis of regenerative heat exchangers was produced by P. A. Domansky. In paper /2/ *COP* was calculated for various refrigerants and it was concluded that the benefit of application of the regenerative heat exchange between the liquid line and suction line depends on the combination of the operating conditions and fluid properties. Impact on *COP* and volumetric capacity may be either negative or positive. In the paper /3/, refrigeration cycles with the regenerative heat exchanger, economizer refrigeration cycle, and refrigeration ejector cycle were analyzed and compared. The first one showed the smallest *COP* improvement potential. The other two increase *COP* for all fluids.

J. Sheridan /4/ investigated a complete air conditioning system having a scroll compressor with the economizer port. The achieved goal of the investigation was to improve the system performance by reducing restrictions associated with capillary tube applied as an expansion device.

When the economizer applied, the compressor induces an additional portion of refrigerant over the economizer port and demands more power for compression. Since the compressor pumps more refrigerant, higher mass circulation of refrigerant over the condenser takes place. If the condenser is sized for operation without the economizer, high operating discharge pressure is built up, which also requires additional power for compression. The evaporator handles increased cooling capacity and suction pressure is reduced, which again requires additional power for compression. The goal of the paper is to investigate performance of a complete R134a air conditioning system with an added economizer circuit.

PRINCIPLES OF OPERATION

An air conditioning system realizing the economizer cycle is presented on Figure 1 and includes two circuits: a main circuit and an economizer circuit. The main circuit consists of in a closed loop a rotary vane compressor, a condenser unit, a high-pressure side of an economizing heat exchanger, an expansion device, and an evaporator unit. The economizer circuit connects the liquid line after the condenser unit with an economizer port of the compressor and comprises an additional expansion device, a low-pressure side of the economizing heat exchanger, and the economizer port itself. The rotary vane compressor consists of a suction inlet, a discharge outlet, and the economizer port between them. A rotor, stator and two adjacent vanes of the rotary vane compressor form compression chambers. The economizer port is located at a point after the compression chambers have been closed for the compression process.

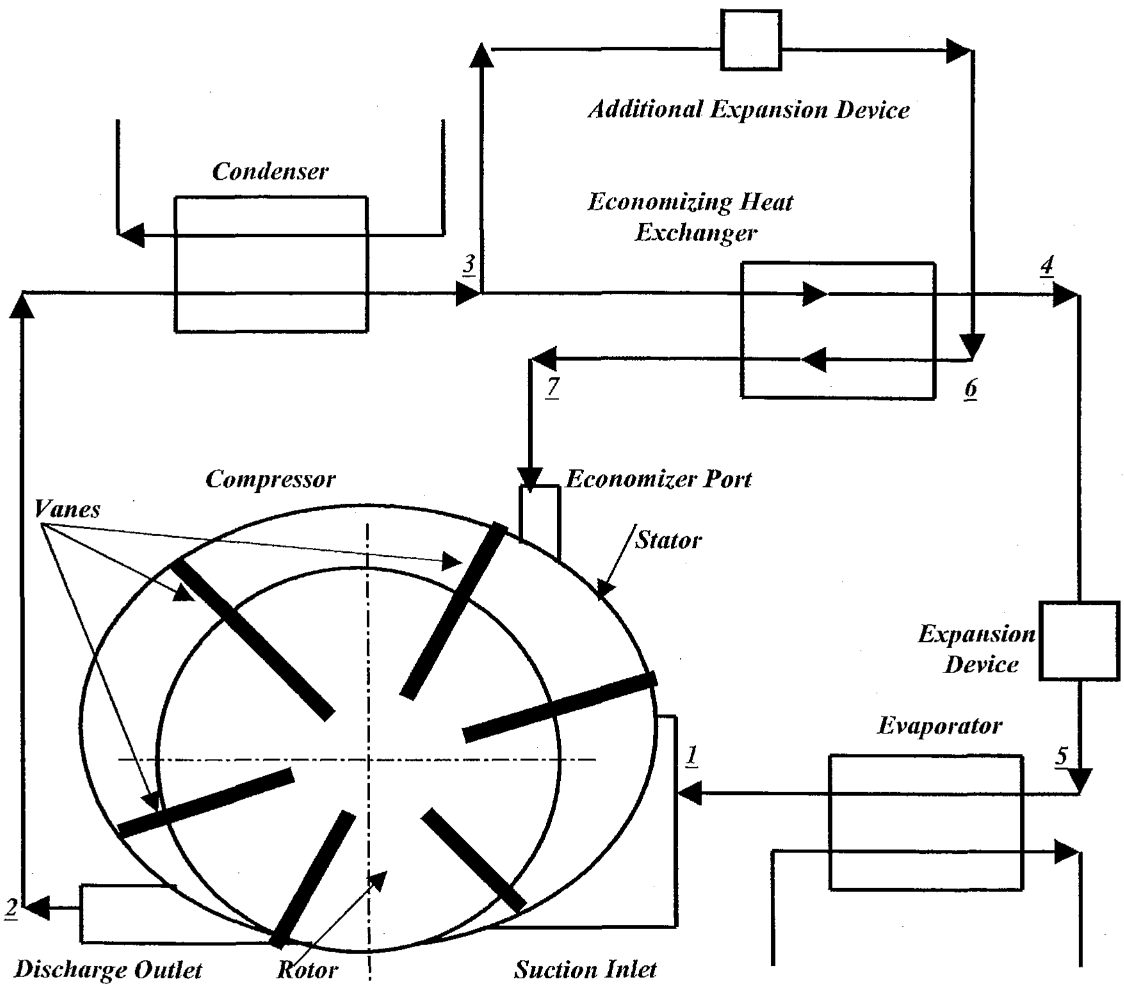

Figure 1: Schematic of a Refrigeration System with an Economizer Port

The air conditioning system operates as follows. The compressor induces a portion of superheated vapor from the evaporator, which is a main part of total refrigerant flow pumped by the compressor. When the suction process is accomplished and the compression chamber has been closed for compression, an additional portion of vapor is induced over the economizer port. This portion is an economizer portion. The induction of the economizer portion is accompanied by compression of the main portion and mixing of the both portions. When the induction of the additional portion of refrigerant over the economizer port is accomplished, vapor compression is accomplished and discharged to the condenser unit. In the condenser unit the compressed vapor is desuperheated, condensed and subcooled. The main part of the liquid is additionally subcooled in the high-pressure side of the economizing heat exchanger, expanded in the expansion valve, turned into a liquid vapor mixture, and the liquid phase of the mixture is boiled out in the evaporator producing a cooling action on objects to be cooled. The compressor induces the boiled out vapor over the suction port. The economizer portion of liquid after the condenser unit is expanded in the additional expansion valve, is turned into a liquid vapor mixture, and the liquid phase of the mixture is boiled out in the low-pressure side of the economizing heat exchanger producing subcooling action. The boiled out vapor is introduced into the compression process over the economizer port at economizer pressure, which is higher than suction pressure, is introduced into compression process without any impact on refrigerant vapor at suction side. In the economizer cycle there is no degradation of refrigerant mass flow rate associated with additional superheat and vapor density lessening.

The economizer cycle in *P-h* diagram is presented on Figure 2 and is made of the following thermodynamic processes:

- Compression in the rotary vane compressor (process 1-2'-2"-2 and process 7-2"-2)
- Desuperheating, condensation and subcooling in the condenser unit (process 2-3)
- Subcooling in the high-pressure side of the economizing heat exchanger (process 3-4)
- Isoenthalpic expansion in the expansion valve (process 4-5)
- Evaporating and superheating in the evaporator unit (process 5-1)
- Isoenthalpic expansion in the additional expansion valve (process 3-6)
- Evaporation and superheating in the low-pressure side of the economizing heat exchanger (process 6-7)

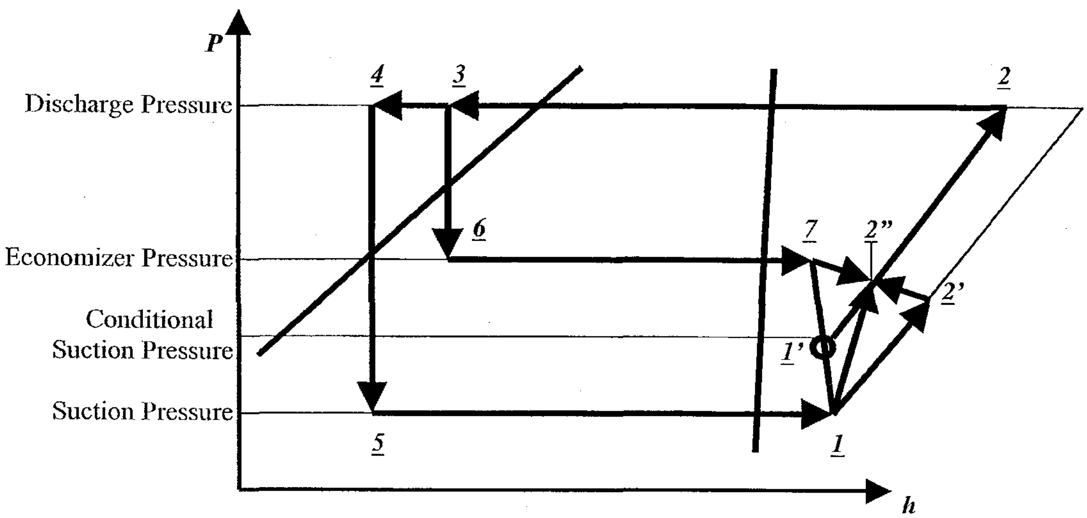

Figure 2: Economizer Cycle in P-h Diagram

Compression itself is based on the following processes:

- Suction of a portion of refrigerant vapor over the suction inlet (state 1)
- Compression of the portion arrived over the suction inlet (process 1-2')
- Induction of an additional portion of refrigerant vapor over the economizer port (state 7)
- Mixing these two portions (state 2")
- Final compression of these two portions (process 2"-2)
- Discharge of compressed portions (state 2)

ECONOMIZER CYCLE CHARACTERISTICS

Subcooling effect Q_{Ec}, kW provided in the economizing heat exchanger is defined as:

$$Q_{EC} = G \cdot q_{Ec} = G \cdot (h_3 - h_4), \qquad (1)$$

where G - is refrigerant mass flow rate over the evaporator, kg/s, q_{Ec} - specific subcoling effect, kJ/kg, h - is enthalpy of thermodynamic state pointed to by its' subscript index, kJ/kg. Mass flow rate over compressor G is derived from performance data of existing compressors.

Cooling capacity Q_0, kW is equal to

$$Q_0 = G \cdot (h_1 - h_5) = G \cdot (h_1 - h_3 + q_{Ec}). \tag{2}$$

Condenser capacity Q_{Cd}, kW is

$$Q_{Cd} = (1 + \mu) \cdot G \cdot (h_3 - h_2) \tag{3}$$

where μ - is economizer part, which is defined from a heat balance of the economizing heat exchanger

$$\mu = \frac{h_3 - h_4}{h_7 - h_3}. \tag{4}$$

The compressor power consumption P, kW is calculated as

$$P = G \cdot [h_2 - h_1 + \mu \cdot (h_2 - h_7)]. \tag{5}$$

Coefficient of performance of the economizer cycle is

$$COP = \frac{h_1 - h_3 + q_{Ec}}{h_2 - h_1 + \mu \cdot (h_2 - h_7)}. \tag{6}$$

COMPRESSOR MODELLING

Volumetric efficiency of a compressor is defined by thermodynamic parameters of suction state and the amount of refrigerant vapor left in the compression chamber when the discharge process has been completed. The remaining portion is re-expanded and brought to the suction side reducing the space intended for refrigerant vapor arriving over the suction inlet. Since the economizing cycle does not make any impact on suction state parameters, volumetric efficiency should be the same as for the compressor in a regular application at the same discharge and suction pressures:

$$\eta_v = \eta_v (p_{dis}; p_{suc}). \tag{7}$$

Isoentropic efficiency is related to isoentropic compression work $\Delta h_s, kW$ applied for pumping vapor refrigerant from the suction pressure zone to the discharge pressure zone. Total compressor power is made of an isoentropic power and friction power, formed by dynamic forces and forces produced by operating pressures. It is reasonable to calculate isoentropic efficiency for an equivalent compression process, which includes adiabatic mixing refrigerant portions at states 1 and 7 at constant volume and constant internal energy, creating a conditional state 1' and then compressing the mixture (process 1'-2). Parameters of the conditional state are pressure p'_{suc}, density $\rho_{1'}$, enthalpy $h_{1'}$, and internal energy $u_{1'}$. In that case

$$\Delta h_s = h_2 - h_{1'}, \tag{8}$$

$$\eta_s = \eta_s\left(p_{dis}; p'_{suc}\right). \quad (9)$$

The constant volume means that these two portions occupy the same volume as the portion arrived over the suction inlet and

$$\rho_{1'} = \frac{\rho_1}{1+\mu}. \quad (10)$$

The constant internal energy means that change of internal energy of one portion has the same absolute value as internal energy change of the second portion and eventually

$$\mu \cdot \left(u_7 - u_{1'}\right) + \left(u_1 - u_{1'}\right) = 0. \quad (11)$$

ANALYSIS OF CALCULATED RESULTS

Let us consider a refrigeration cycle operating between condensing temperature t_c and evaporating temperature t_0 with no subcooling and no superheat. In that case, specific cooling capacity is equal to

$$q_0 = h''(t_0) - h'(t_c), \quad (12)$$

where h'' and h' are enthalpies of saturated vapor and liquid refrigerant, kJ/kg.

A complete liquid subcooling is equal to $t_c - t_0$. This subcooling is an ideal parameter, which is impossible to achieve in reality. It provides cooling capacity, which is

$$q_0^{sub} = h''(t_0) - h'(t_0). \quad (13)$$

Let us introduce a potential economizing effect E_{Ec}, which is a relation of capacity increase gained by the complete subcooling from condensing to evaporating temperature, to the cooling capacity gained in the refrigeration cycle with no subcooling and no superheat in the following view:

$$E_{Ec} = \frac{q_0^{sub} - q_0}{q_0} = \frac{h'(t_c) - h'(t_0)}{h''(t_0) - h'(t_c)}. \quad (14)$$

Figure 3 shows the potential economizing effect at the complete subcooling for a number of refrigerants on boiling temperature at a condensing temperature of 50°C. The biggest effect is on R507A and R404A. Ammonia shows the least improvement with R22 slightly better. Economizer performance of R134a, R410A, R407C, and propane is in the middle. Real systems always generate less optimistic results, but the potential economizing effect shows which refrigerant provides the best improvements with the economizer.

The object of the investigation is a R134a air conditioner operating at outdoor temperature of 35°C, indoor temperature of 27°C, and indoor relative humidity of 50%. The rotary vane compressor has a displacement rate of 80.4 m^3 at 3000*rpm*. The condenser unit consists of two equal condenser coils. Each condenser coil consists of 1"x0.866" staggered pattern, 2 cycle sine wave fins made of aluminum, 10 fins per inch, 4 rows deep, 12 rows high and 48" of finned length. The number of circuits in the condenser coil is 6. Refrigerant charge corresponds to a subcooling at the condenser outlet of 5°C. Air flow provided by the condenser fans is of 2500scfm per coil. An evaporator unit consists of two equal evaporator coils. The evaporator coil is of 1"x0.866" staggered pattern, Potential Economizer Effect of Complete Subcooling

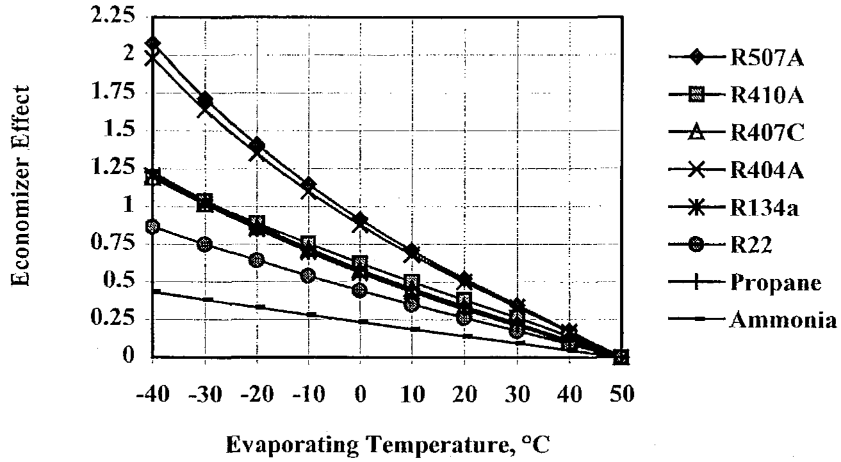

Figure 3

2 cycle sine wave fins made of aluminum, 10 fins per inch, 4 rows deep, 12 rows high and 42" of finned length. The number of circuits in the evaporator coil is 12. The expansion valve generates a superheat of 5°C at the evaporator coil outlet. Air flow rate provided by the evaporator blower is 1900csfm per coil. The economizer sub-cooling is varied and equal to 0°C, 10°C, 20°C, and 30°C. The economizer pressure corresponds to the relative economizer pressure ratio, which is a relation of discharge pressure to economizer pressure

$$\sigma_{Ec} = \frac{p_{dis}/p_{Ec}}{p_{dis}/p_{suc}} = \frac{p_{suc}}{p_{Ec}}, \qquad (15)$$

and is equal to 0.8.

Calculations are made for a complete system configuration, which includes the rotary vane compressor, a discharge piping, the condenser units with the condenser fan, a liquid piping, an expansion device, the evaporator unit with the evaporator blower, and a suction piping. The economizer circuit and all related equipment are evaluated in terms of thermodynamic performance in order to define the economizer characteristics and the influence of the economizer on the whole system. It is assumed that fluid dynamic characteristics of the economizer port allow to the required amount of refrigerant to pass from the economizing heat exchanger to the compression chamber.

Figure 4 shows that cooling capacity is increased on subcooling in the economizer, but *COP* is decreased. This happens because of increase in condenser capacity, isoentropic compressor work, isoentropic power, and compressor power. Isoentropic work is raised proportionally with cooling capacity, but isoentropic power and compressor power go up in a greater extent than the cooling capacity due to the influence of mass flow rate. Volume flow rate drawn by the compressor from the evaporator is decreased, but total mass flow rate, pumped by the compressor, is greater.

Figure 5 shows an increase of pressure ratio and discharge pressures and an decrease of suction pressure. Heat rejected in the condenser unit is increased and the rise of discharge pressure is a reaction of the condenser unit on the increase. The economizer provides the increase of cooling capacity and at the same time, the film coefficient of boiling refrigerant is improved since vapor quality at the expansion valve inlet is improved as well. However, these improvements are not enough, and that is why evaporating and suction pressures are decreased slightly. Decrease of suction pressure causes degradation of volume flow rate drawn by the compressor from the evaporator. Changes in pressures cause a significant increase in pressure ratio, and that is why isoentropic compressor work, isoentropic power, compressor power and COP all go up. If the condenser capacity is enhanced, discharge

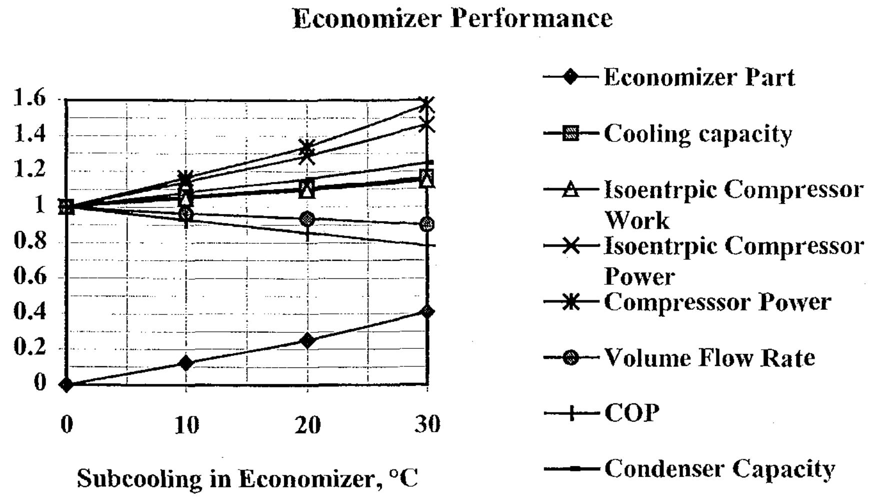

Figure 4

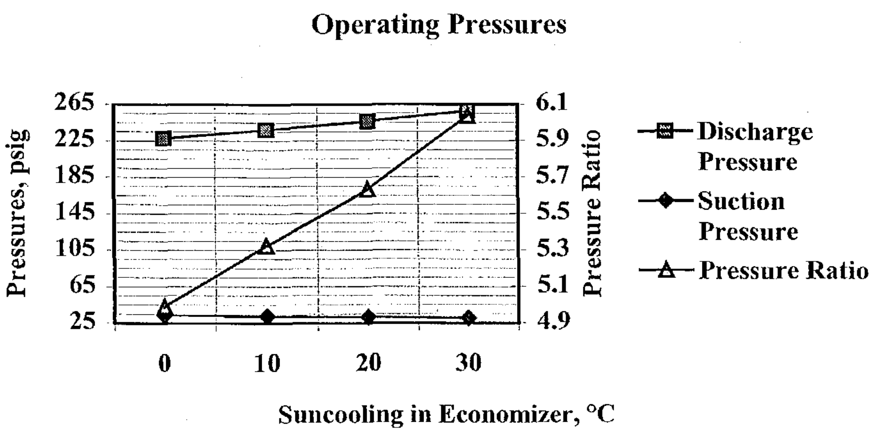

Figure 5

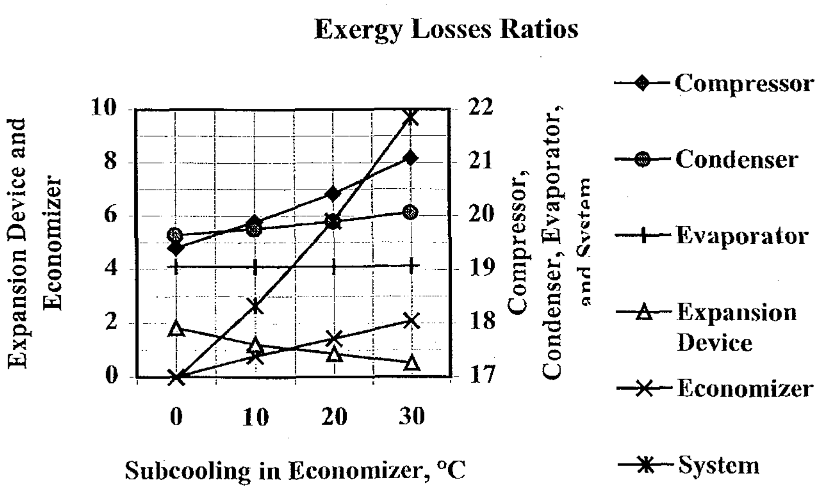

Figure 6

pressure drops off. This definitely improves the condenser performance and provides conditions for an additional increase in the evaporator capacity. In that case, the evaporator capacity should be enhanced as well.

Figure 6 presents results of exergy analysis. Exergy losses ratio is a relation of exergy losses in a component or system, to the system exergy capacity. Exergy losses in the condenser and evaporator units include losses associated with the condenser fan and the evaporator blower. Subcooling in the economizer causes a significant growth of exergy losses in the compressor and in the condenser due to the increased mass flow rate over these components and loads on them. Exergy losses in the evaporator are not changed since dropping of the evaporating temperature associated with the suction pressure reduction is not significant. The economizer brings additional exergy losses. The only component that shows a decrease of exergy losses is the expansion valve. However, the decrease is approximately the same as the contribution in exergy losses made by the economizing heat exchanger. The total system exergy losses ratio is significantly increased, which also confirms that the system is overloaded.

Let us consider a modified system with the enhanced condenser and evaporator. The condenser has a row height of 16" instead of 10" with an air flow rate of 3300scfm per coil instead of 2500scfm. The evaporator has 50"long finned tubes instead of 42" and air flow rate of 2185scfm per coil instead of 1900scfm. The calculated discharge and suction pressures, cooling capacity, total refrigerant mass flow rate, and COP for the original and modified systems, with and without the economizer, are in Table 1. The cooling capacity is given in relation to the original system without the economizer. The results show that enhancement of the condenser and evaporator provides a greater increase of cooling capacity and keeps a high level of *COP* since discharge pressure is reduced and suction pressure is increased. Mass flow rates in case #2 and #4 are about the same due to a combination of refrigerant density at the compression suction and pressure ratio.

Table 1

No.	System	Discharge Pressure, Psig	Suction Pressure, Psig	Cooling Capacity Increase	Mass Flow Increase	COP
1	Original System, no Economizer	227.48	33.82	0%	0%	2.150
2	Original System & Economizer	258.00	30.47	16.7%	27.1%	1.689
3	Enhanced System, no Economizer	203.88	34.93	8.9%	4.1%	2.400
4	Enhanced System & Economizer	223.21	31.98	26.7%	26.3%	2.076

The economizer pressure does not show any significant impact on the cooling capacity, but influences *COP*. For example, the original system with the economizer subcooling of 10°C, provides almost the same cooling capacity and *COP* of 2.045, 2.019, 1.996, and 1.977 at relative economizer pressure ratios of 0.6, 0.7, 0.8, and 0.9 respectively. The economizer subcooling of 20°C also provides almost the same cooling capacity, and *COP* is reduced as well: 1.929, 1.882, 1.843, and 1.809 respectively. The *COP* degradation is associated with an increase of the economizer part of flow rate and refrigerant mass flow rate over the condenser. Also, the economizer pressure influences hydrodynamic characteristics of the economizer circuit and mass flow rate over the economizer port, sizes of the economizing heat exchanger, and also limits the maximum achievable subcooling.

CONCLUSIONS

1. The biggest potential economizing effect is on R507A and R404A. Ammonia shows the least improvement with R22 slightly better. Economizer performance of R134a, R410A, R407C, and propane is in the middle.
2. Depending on subcooling the economizer provides a significant growth of cooling capacity, condenser capacity, isoentropic compressor work, mass flow rate pumped by the compressor, isoentropic compressor power, and compressor power in R134a air conditioning systems sized for operation without the economizer. Flow rate over the evaporator is slightly reduced. The system COP shows degradation. In order to keep the same level of *COP*, the condenser and evaporator should be thermally enhanced in order to handle the increased capacities.
3. If proper hydrodynamic characteristics of the economizer circuit are provided, economizer pressures do not show any significant impact on the cooling capacity, but influence *COP*.

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