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Research Paper

REFRIGERATION CYCLES AND SYSTEMS: A REVIEW

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During the last decade, substantial research activities have been undertaken regarding refrigeration cycles and systems. Besides using eco-friendly refrigerants, thrust has been given upon devising methods to increase the efficiency of the refrigeration cycle/system, which will also contribute to reducing emission of Green House Gases (GHG). For achieving the latter objective use of Liquid Vapour Heat Exchanger (LVHE), use of ejector instead of expansion valve, combining Vapor Compression Refrigeration (VCR) with Vapor Absorption Refrigeration (VAR) has been recommended. This paper provides a review of some of the methodologies which contribute to increasing the efficiency of the refrigerating plant.

Keywords: Refrigeration system, Energy efficiency, Environmental impact, Refrigeration cycle, COP

INTRODUCTION

The increasing energy consumption is more prominent in the industrial sector and is mostly related to the heavy inductive machinery like motors, refrigeration and air-conditioning units, etc. These refrigeration and air-conditioning units contribute to the major share of the energy consumption, because of heavier compressor used in such systems. Refrigeration and Air Conditioning (RAC) play a very important role in modern human life for cooling and heating requirements. It covers a wide range of applications starting from food preservation to improving the thermal and hence living standards of people. The utilization of these equipments in homes, buildings, vehicles and industries provides for thermal comfort in living/working environment and hence plays a very important role in increased industrial production of any country.

The increasing demand of energy primarily for RAC and HP (Heat Pump) applications (around 26-30%), degrades environment, produces global warming and depletes ozone layer, etc. Therefore to overcome these aspects there is urgent need of efficient energy utilization methods. The refrigeration industry is puzzled by the two of the most pressing environmental issues, namely,

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global warming and ozone depletion. It is logical that these two seemingly distinct, albeit, intricately related challenges be addressed together not only through new working fluids, but also through innovative thermodynamic cycles. At the same time the novelty must be combined with practical viability keeping in view the current state of system practices. The Vapor Compression Refrigeration (VCR) with the positive displacement compressors (such as reciprocating, rotary, and scroll or screw compressors) continues to be the workhorse of cooling demands.

Solid adsorption, liquid absorption, thermoelectric and thermo-acoustic cycles offer limited other options. Vapor absorption has been the most tested out among them. However, it lacks the benefit of scalability to low cooling capacities and is limited by the choice of working pairs. The most investigated combination is lithium bromide – water system, which cannot be used below about 5°C, and is handicapped by operation at sub atmospheric pressures. Also, it cannot be used in a hybrid compression system with conventional compression of water vapor. Ammonia-water is the other most widely investigated pair, which has the problem of carryover of water vapor into the refrigeration circuit, need for high pressures and reservations on acceptability of ammonia in small scale refrigeration units. Thermoelectric systems are seldom scalable to large capacities and the thermo-acoustics is still in developmental stages.

In this paper detailed mathematical models and their application to different refrigeration systems are presented. Thermoeconomic models have also been developed to obtain energy efficiency at an optimum cost. They also enable performance analysis for different arrangements and the influence of some important operating and design parameters.

ANALYSIS OF DIFFERENT REFRIGERATION SYSTEMS/ CYCLES

Analysis of Refrigeration Carnot-Type Cycle Based on Isothermal Vapor Compression

The principle of this Carnot-type cycle suggested by Meunier (2006) is that during this reversible cycle, the refrigerant exchanges heat with two external heat sources at constant temperatures (T_0 and T_1) and exchanges work with the environment during two reversible expansions and during an isothermal compression. An ideal adiabatic Internal Heat Exchanger (IHX) is used to heat the Low Pressure (LP) refrigerant vapor from the low temperature to the high temperature when it cools down a fraction of the High Pressure (HP) refrigerant from the high temperature to the low temperature. In this cycle only a fraction, x, of the HP refrigerant is transferred to the IHX to allow this fraction x of refrigerant to reach the low temperature T_0. As a result, we get a reversible cycle made of two isotherms and one isentropic process, which, according to Carnot theorem, yields the Carnot COP. The equipments necessary to fire such cycles consists of (Figures 1 and 2).

- One or several isothermal reversible compressor(s) connected to the high temperature heat source at T_1,
- A condenser (for the sub-critical cycle only) connected to the heat source at T_1,

- An evaporator connected to the low temperature heat source at T_0,
- An ideal adiabatic internal heat exchanger allowing the two refrigerant streams to flow from one temperature to the other without external heat exchange,
- Two reversible expansion engines (one adiabatic and an isotherm one).

Figure 1: Schematic of a Unit Operating A Carnot-type Cycle With Condensation Meunier [1]

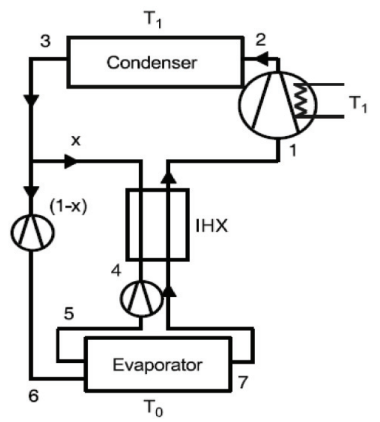
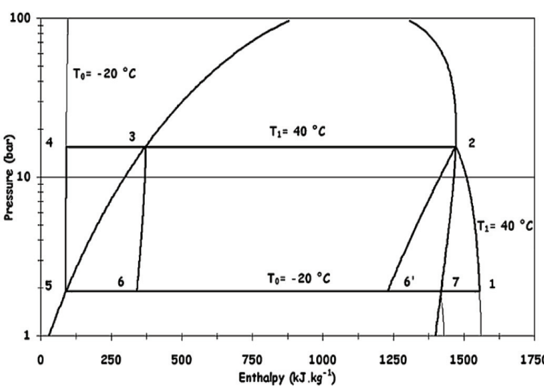


Figure 2: Carnot Type Cycle in a (h, Log P) NH_3 Diagram Meunier [1]



At point 1, superheated vapor enters the compressor (temperature T_1 equal to the HP saturated temperature). The compression being isothermal, the vapor leaves the compressor at temperature T_1 in the saturated state, it is then condensed at T_1 in the condenser. Then, the flow of saturated liquid leaving the condenser is divided into two flows (point 3). A fraction x of liquid is sent towards a liquid vapor heat exchanger (IHX) where it exchanges internally heat with the cross-current flow of vapor leaving the evaporator at T_0 equal to the LP saturating temperature. The fraction x is selected in consonance with the following relation:

$$x[h_{(HP,}T_1)-h_{(HP,}T_0)]=h_{(LP,}T_1)-h_{(LP,}T_0) \quad ...(1)$$

where h = specific enthalpy in kJ/kg.

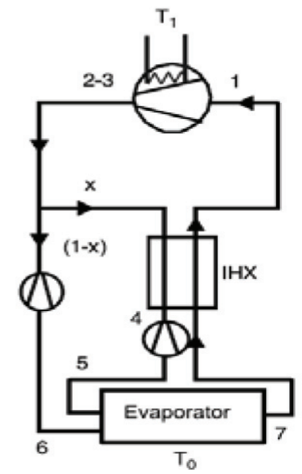
As a result, in the IHX, the outlet temperature of the HP liquid and that of the LP vapor are respectively equal to T_0 and T_1.

At the outlet of the IHX, the HP liquid at T_0 is expanded through an isotherm and reversible engine into the evaporator so that line 4 and 5 on Figure 2 is an isotherm line. The work output of this expansion is theoretically used for the compressor. The other fraction (1 – x) of HP liquid leaving the condenser is directly expanded through an adiabatic and reversible expansion engine so that line 3 to 6 on Figure 2 is an isentropic line. The compressor again uses the work output of this expansion .The cold production results from the vaporization in the evaporator. The COP of the cycle is equal to the Carnot COP:

$$COP=T_0/T_1–T_0 \quad ...(2)$$

Another option to that cycle is to perform the condensation in the IHX (Figure 3), the cycle is 1-2-3-4-5-6/-7-1. Now the condenser is no more needed but the fractionation of the HP refrigerant must be realized in point 2. In Equation (1), point 2 rather than point 3 corresponds to (HP, T1) to select x which is smaller. Expansion 2–6' results in larger work output than out of expansion 3–6

Figure 3: Schematic of a Unit Operating A Carnot-type Cycle Without Condensation Meunier (2006)



but the COP (assuming the IHX to be ideal) is the same since the cooling rate and the external power (due to the additional expansion power) are both reduced.

The above mentioned isothermal compression coupled with the IHX is all the more important for fluids with high compressor discharged temperature (ammonia, CO_2, etc.) and this could result in an interesting opportunity for CO_2 trans-critical cycles development as well as for spreading the field of application of ammonia.

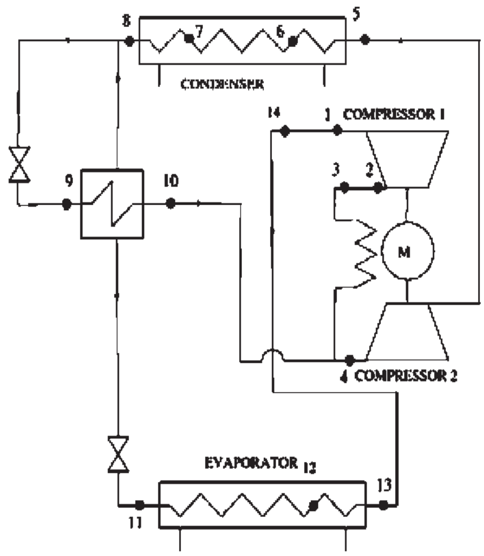
Optimizing the Refrigeration Cycle with a Two-stage Centrifugal Compressor and a Flash Intercooler

The motivation behind the study of Röyttä*et al.* (2009) was an interest in examining the possibilities of using a centrifugal compressor in a relatively small refrigeration unit to reduce the unit's weight and power consumption. At a wider operational range, the centrifugal compressor provides better efficiency (Dixon, 1989) and better power-to-volume ratio (Wilson, 1988), when compared to a displacement compressor. Therefore, they should be used in refrigeration cycle, when possible, as low power consumption and weight are desired. The performance of the centrifugal compressor has a strong dependence on the working fluid and the operational range (Brown, 2007; Calm, 2006). If the cooling load of a single compressor does not vary greatly, a centrifugal compressor is feasible. Thus, for a wide range of power, parallel compressors are recommended.

In the present design, the weight of the cycle was also essential and therefore small heat exchangers were selected. This results in a higher condensation temperature as well as a rather high pressure ratio for these fluids. As such, a one-stage compressor is infeasible.

The flash intercooler is located between the condenser and the throttle valve. A small portion of the flow returning from the condenser is expanded to intermediate pressure and then evaporated in a heat exchanger cooling the main

Figure 4: Flow Chart of the Refrigeration Process Röyttä *et al.*(2009)



flow (Figure 4). Since the flash intercooler has a favorable power-to-weight ratio when compared to other intercooling options, it was selected.

A non-flammable fluid with a relatively low Global Warming Potential (GWP) and zero Ozone Depletion Potential (ODP) was required while deciding for the working fluid. Because of their flammability, many hydrocarbons that would have offered environmentally sustainable, high efficiency options (Saleh, 2006) were discarded as possible working fluids. As far as other natural refrigerants are concerned, they were not particularly favorable, e.g., CO_2 would have very high pressure levels and thus very heavy heat exchanger and piping. Very low sub-atmospheric pressures, e.g., water, would also lead to very heavy machinery. Finally R134a and R245fa were selected, the former being the most widely used fluid in today's centrifugal chillers. The fluids have GWP of 1320 and 1020 respectively (Calm, 2006). The developed model was found to be robust and relatively fast.

The refrigeration process is optimized by maximizing the Coefficient Of the Performance (COP). The COP is determined by

$$COP = \Phi_{cool}/P_e \qquad ...(3)$$

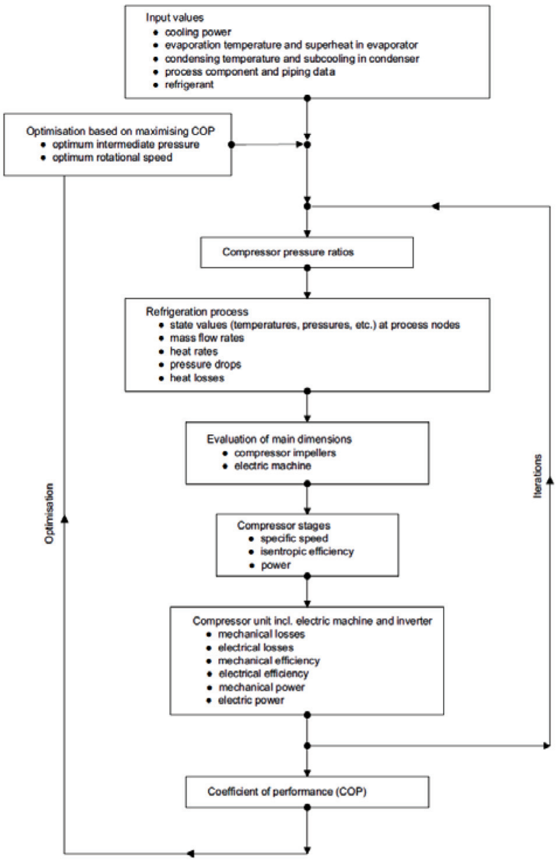
where Φ_{cool} is the required cooling power and P_e is the electrical power. The electrical power is calculated by

$$P_e = Pc_1 + Pc_2 + P_{bearing} + P_{ec}/\eta_e \qquad ...(4)$$

where η_e is the electric motor and inverter efficiency, Pc_1 and Pc_2 represent the power of each stage, $P_{bearing}$ is the bearing friction power, and P_{ec} is the power consumed by mechanical losses in the air gap.

Figure 5 shows the calculation steps as well as the optimization scheme. The cooling power,

Figure 5: Optimisation and Calculation Procedure Röyttä *et al.*(2009)



condensing temperature, evaporating temperature, bearing power loss, superheating in the evaporator and vapor content after the flash intercooler are known. Modelling was done for two different refrigerants. From a study with several refrigerants, these two were chosen because they are non-flammable, non-toxic, they have relatively low GWP and zero ODP, and the COP value was good with the chosen initial values. With the help of COP value, the overall performance of the refrigeration process was monitored. Optimization was performed by changing the rotational speed and the intermediate pressure of the compressor. The best performance was achieved at the rotational speed of N_{opt} and at a first stage pressure ratio of $\pi_{c1,opt}$ with R245fa and at a rotational speed of 1.36 N_{opt} and at a first stage pressure ratio of 0.67 $\pi_{c1,opt}$ with R134a, respectively. The COP value was higher with R245fa than the values for the performance of R134a. These values were COP_{opt} and $0.77COP_{opt}$. Figure 6 shows the values for the performance of the refrigeration process for R245fa and Figure 7 for R134a.

Figure 6: COP with Different Rotational Speeds and Intermediate Pressures with R245fa Röyttä *et al.*(2009)

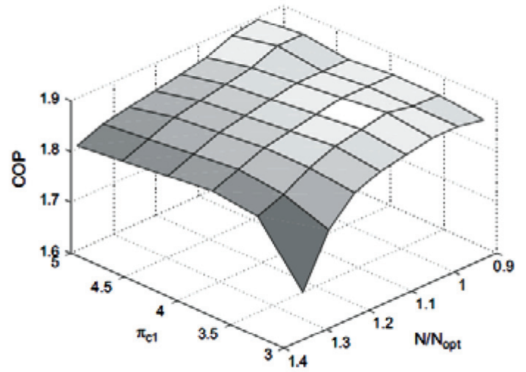
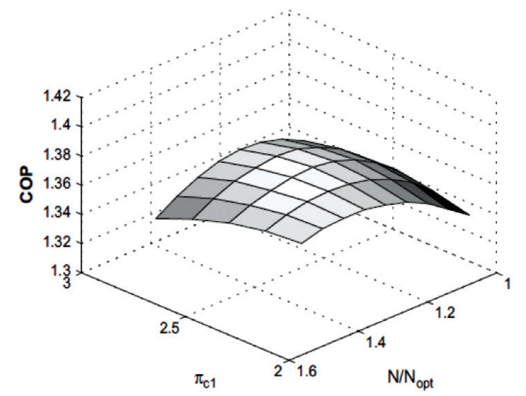


Figure 7: COP with Different Rotational Speeds and Intermediate Pressures with R134a Röyttä *et al.*(2009)



An optimum specific speed cannot be selected for both compressor stages because both wheels are on the same shaft in this particular compressor construction. Therefore, the pressure ratio and rotational speed need to be optimized in a way that both compressor stages achieve reasonable efficiency leading to the best possible compressor efficiency on the whole. The isentropic efficiency of the first stage increases at the lower rotational speed but the isentropic efficiency of the second stage decreases.

ANALYSIS OF MECHANICAL AND ADSORPTION HYBRID COMPRESSION REFRIGERATION CYCLES

Solid adsorption based refrigeration systems have the advantage of scalability to all capacities, ranging from a few watts to several kilowatts (Gordan *et al.,* 2000). In the last couple of decades, a considerable amount of work on solid sorption systems has been done. The adsorbents most investigated are silica gel (e.g. Ng, 2003; Chua, 1988; 1999; Saha, 1999), zeolite (e.g. Meunier and Douss, 1990; Guillemin *et al.,* 1980; Ramos *et al.,* 2003) and activated carbons (Critoph, 1989; Hamamotoh *et al.,* 2006; Cacciola, 1995; El-Sharkawy *et al.,* 2006) while water, ammonia, methanol and ethanol are proposed as refrigerants. Activated carbon + HFC 134a systems have been explored (Banker, 2003; 2004a; 2006; Akkimaradi *et al.,* 2002; Srinivasan, 2006) only recently.

In one of the studies, the authors Banker *et al.* (2008) present the results of an analysis of a hybrid compression system where the mechanical compression process is the dominant means of raising the pressure while the thermal compression supplements it. Thus, the accent is on energy conservation. Conditions are identified under which such a system will be viable. Yanagi *et al.* (2002) proposed a booster pump to increase the pressure of water vapor from an adsorption bed, which provides primary means of compression, but the present approach is somewhat different from this. Here accent is on the reduction of temperature of regeneration in the adsorption bed, such that low grade heat energy can be utilized in a silica gel + water adsorption cycle. Performance comparison is made with single-stage mechanical and thermal compression and two-stage thermal compression. It is also investigated whether thermal compression should form the low or high stage.

Figure 8 shows a schematic hybrid cycle where the adsorption provides the low stage compression. T–s and p–h diagrams for the same are shown in Figure 9. The refrigerant from evaporator is taken to the adsorption compressor, where it gets adsorbed by activated carbon at near condensation temperature. It is desorbed by supplying heat (at d). The desorbed refrigerant vapor at nearly the waste heat source temperature, is passed through intercooler (d–1) and then is drawn into a mechanical compressor for high stage compression (1–2).

Figure 8: Schematic Diagram of Adsorption + Compressor Hybrid Refrigeration System Banker *et al*. (2008)

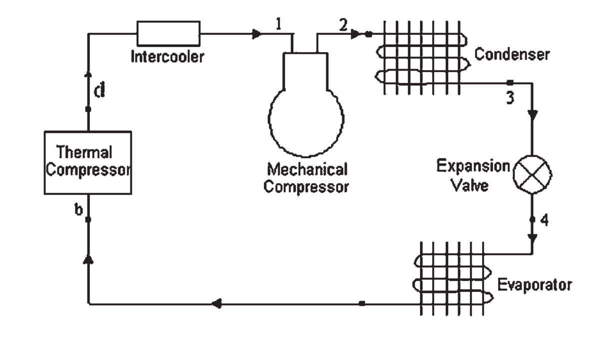
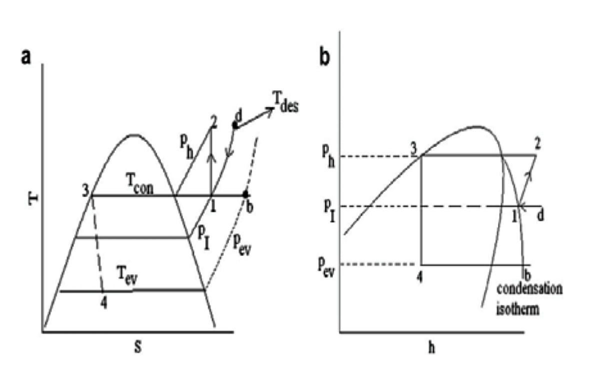


Figure 9: Hybrid Cycle with Low Stage Thermal Compression (a) T-s Plane (b) p-h Plane Banker *et al*. (2008)



The adsorption segment requires several minutes to complete one cycle of compression process whereas a mechanical compressor has a cycle time of a few milliseconds. Therefore, a string of sufficiently large adsorbers (typically four) will be required to ensure that the mass flowrate through each of the compressor segments is the same.

The equation of state for HFC 134a (Tillner *et al.,* 1994) and the adsorption characteristics of activated carbon + HFC 134a system (Akkimaradi *et al.,* 2001) are the data required for the analysis. A computer program was written on a Matlab platform.

A single and two-stage thermal compression was compared with the hybrid compression processes. The calculation procedure covered a number of condensing/adsorption, evaporating and desorption temperatures, various packing densities of activated carbons and various specimens of activated carbon and a range of intermediate pressures.

It is found that: (i) The COP of a two-stage system remains fairly uniform over the entire range of evaporating temperatures investigated, (ii) single-stage system is better than the two-stage one for $t_e > 5^oC$, and (iii) the COP of the hybrid cycle is the highest because of less energy needed for second stage which is mechanical compression. Because of more compact adsorption compressors and consequent reduction in heat inputs, two-stage adsorption and hybrid systems show a significant improvement in intrinsic COP.

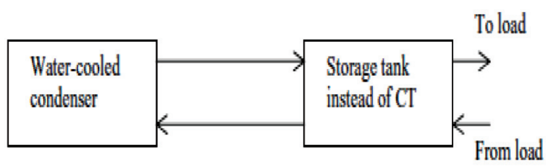
Uptake efficiency in thermal compression is similar to volumetric efficiency (in reciprocating compressor). The hybrid compression improves the uptake efficiency of the thermal stage significantly. Use of hybrid cycle substantially reduces the amount of charcoal required by the increased uptake efficiency and larger concentration differentials across the thermal compression stage. There is a saving of power >40% while comparing the shaft power requirement for mechanical stage of hybrid cycle single-stage mechanical compressor. The least energy requirement is attained when the saturation temperature differentials corresponding to suction and discharge temperatures are 20–30°C for most refrigerants (Srinivasan, 1994) for compression systems. In this case, for x = 1, the maximum saving in power per unit mass of carbon is attained at $\sim t_{ev} = -5^{o}C$ for which the ideal interstage pressure is ~5 bar corresponding to a saturation temperature of ~15°C. The operating differential saturation temperature is ~20°C, confirming that finding.

As compared to the high stage compression, the overall performance indicator (reduction in power per unit mass of carbon) of low stage compression is only marginally smaller. Although the pressure ratios across each stage remain the same, the pressure differential across which the thermal compressor operates with high stage adsorption will be larger than when it is for the low stage. Consequently, the void volume effect will be more pronounced and hence, the uptake efficiencies will be reduced in a reverse hybrid case. This leads to larger carbon inventories and hence a marginally lower saving in power per unit mass of carbon despite the reduction in mechanical power. At higher evaporating temperatures, the difference becomes insignificant which do not warrant a multistage compression in any case. By applying the analogy of minimum desorption temperatures needed for operating the adsorption system (Saha, 2007) the minimum desorption temperature needs for low stage adsorption will be ~73°C, while it will be ~77°C for the high stage adsorption.

THERMOECONOMIC ANALYSIS OF SINGLE STAGE REFRIGERATION SYSTEMS

For a refrigeration plant that employs a coupled waste heat recovery system, optimum operating temperatures can be determined by fixing, and, so eliminating all these thermal and economical parameters, except the inside and outside temperature difference, depending on the certainty of operating characteristics of applications and the most efficient operating condition of the refrigerating system as illustrated in Figure 10.

Figure 10: A Refrigeration System Utilizing Waste Heat Recovery System Soylemez (2004)



There are several studies about the thermo economic optimization for the refrigeration systems in the present literature (El-Sayed, 1999; d'Accadia, 1998; 1998; Sahin, 1991, 2001; Kodal, 2001; Salah, 1999; Tozar, 1998), but none of these are directly related to the work of Söylemez (2004). For calculating the optimum operating temperatures for the refrigeration systems, a new formula is derived at which maximum energy saving for heat recovery system occurs.

Technical life and installation cost per capacity of the heat recovery system, annual interest rate, present net prices of the recovered energy and electricity, annual energy price rate, thermo-physical properties of water, usage factor, inside volume of the refrigerated space, value of constants relating the volume of the refrigerated space to its heat transfer area due to the physical shape of the refrigerated space, overall heat transfer coefficient, maximum pressure drop through the circulating system, mass flow-rate of the circulating water per unit heat rejection capacity, evaporation design temperature, pump and motor efficiencies, evaporator and condenser design temperature difference, annual average full load operation time, resale value and the ratio of annual maintenance and operation cost to the original cost are listed as variables and parameters used in formulating the thermo economically optimum operating temperature difference.

Amount of total refrigeration load is calculated by using the usage factor for interior loads and overall heat transfer coefficient for wall gain load as follows (Dossat, 1980):

$$Q_L = FV_i\Delta T + cUV_i^{2/3}\Delta T = a_6\,\Delta T \qquad ...(5)$$

where

a_6 = Fixed parameter, $(cUV_i^{\ 2/3} + FV_i)$ (W/K)

c = Coefficient that is used in Eq. (6)

F = Usage factor (W/ (m^3/K))

U = Overall heat transfer coefficient of the walls of the refrigerated room [W/ (m^2 K)]

V_i = Inside volume of the refrigerated space (m^3)

ΔT = Temperature difference between inside design and mean ambient (K)

Area of heat transfer for a refrigerated space as a function of its volume can be determined by:

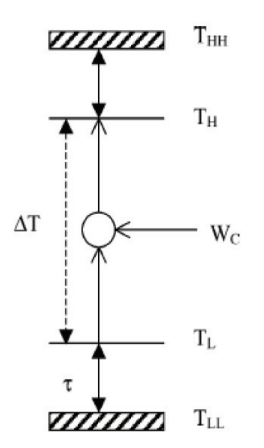
$$A = cV_i^{\ 2/3} \qquad ...(6)$$

where A is area of heat transfer surface of the outside wall of the refrigerated space (m^2) and c can be taken as 6 for a cubical room and is greater than 6 for prismatic rooms. Overall heat recovery efficiency of waste heat recovery system, taking the heat loss in the piping system into consideration can be approximated as:

$$\frac{Q_R}{Q_H} = \eta_{HR} = \frac{Q_L}{Q_H} \Rightarrow Q_R = Q_L \qquad ...(7)$$

where Q_H is the heat rejected by the water cooled condenser of the refrigerating plant (W), Q_R is the heat recovered from the condenser of the

Figure 11: Schematics for Refrigeration System Temperatures Soylemez (2004)



refrigerating plant (W) and Q_L is the refrigeration load of the refrigerating plant (W).

COP of the actual refrigeration cycle can be calculated approximately by using the COP of ideal Carnot cycle as shown in Figure 11 schematically,

$$\text{COP} = \text{COP}_c\, \eta_c = \frac{T_{LL}\eta_c}{T_{HH} - T_{LL}} = \frac{Q_L}{W_C} \quad ...(8)$$

where COP is the coefficient of performance of the refrigeration system, COPc the coefficient of performance of the Carnot refrigeration cycle, T_{HH} is the condensation temperature (K), T_{LL} is the evaporation temperature (K) and W_C is the power input of the compressor (W).

η_c, the Carnot cycle efficiency varies between 0.3 and 0.5 for small systems and 0.5 and 0.7 for large and higher efficiency systems (Zogou, 1998). Eq. (8) can be written alternatively as in the following form:

$$\text{COP} = \frac{(T_L - \tau)\eta_c}{T_H - T_L + 2\tau} = \frac{(T_L - \tau)\eta_c}{\Delta T + 2\tau} \quad ...(9)$$

where T_L is the design temperature of the refrigerated room (K), τ is the design temperature difference between refrigerant and ambient medium for evaporator and condenser (K) and T_H is the design ambient air temperature (K).

The net savings function for waste heat recovery from refrigeration system can be written as:

$$S = S_E - OC - IC \quad ...(10)$$

where S is the net savings gained from waste heat recovery (\$), S_E is the savings gained from waste heat recovery (\$), OC is the operation cost of the waste heat recovery system (\$) and IC is the initial cost of the heat recovery system (\$).

This function can be rewritten explicitly by using the equipment cost estimating parameters together with the P1 – P2 method as:

$$S = P_1 C_E HQ - \frac{P_1 C_{EL} H \Delta P}{\rho \eta_p n_m}\left(\frac{m}{Q_H}\right)(Q_L + W_C)$$

$$-P_2 C_Q (Q_L)^{2/3} \quad ...(11)$$

where $P1$ is the ratio of the life cycle energy cost or savings to that of the first year (year) and $P2$ is the ratio of the life cycle expenditures incurred due to the additional capital investment to the initial investment, C_E is the cost of energy recovered (\$/(W/h)), H is the annual time of operation (h/year), C_{EL} is the cost of electricity (\$/(W/h)), ΔP is the total pressure drop through circulation pumps (kPa), η_m is the motor efficiency, η_p is the pump efficiency,

ρ is the mass density of water (kg/m^3), m/Q_H is the mass flow rate of water per unit capacity of condensing unit (kg/(s/kW)) and C_Q is the capacity dependent first cost of the heat recovery system (\$/W$^{2/3}$).

Eq. (11) is simplified to:

$$S = (a_4 - a_5 - a_7)\Delta T - a_2 \Delta T^2 - a_3 \Delta T^{2/3} \quad ...(12)$$

where a_1 = fixed parameter used in Eq. (13), [$a_4 - a_5 - a_7$] (\$/K)

a_2 = fixed parameter used in Eq. (12), {a_6 P_1 C_{EL} H ΔP (m/Q_H)/[$\rho\eta_p\eta_m\eta_c$ (T_L – τ)]} (\$/K^2)

a_3= fixed parameter used in Eq. (12), [P_2 C_Q $(a_6)^{2/3}$] (\$/K^2)

a_4 = fixed parameter, (a_6 $P_1 C_E$ H) (\$/K)

a_5= fixed parameter, {a_6 $P_1 C_{EL}$ H $\Delta P(m/Q_H)$/ [$\rho\eta_p\eta_m$]} ($/K)

a_7 = fixed parameter, {= $a_6 \cdot P_1 \cdot C_{EL}$ $\cdot m \cdot H \cdot \Delta P \cdot$ (m/ QH) $\cdot 2 \cdot \tau / [\rho \cdot \eta_p \cdot \eta_m \cdot \eta_c \cdot (T_L - \tau)]$}, ($/K)

By differentiating Eq. (12) with respect to temperature difference, ΔT, following equation arises:

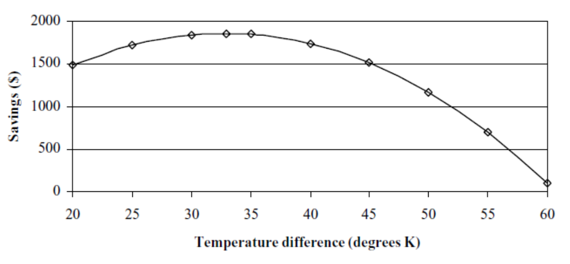
$$\frac{\partial S}{\partial(\Delta T)} = a_1 - 2a_2\Delta T_{opt} - \frac{2}{3}a_3\Delta T_{opt}^{-1/3} = 0 \quad ...(13)$$

where ΔT_{opt} is the optimum temperature difference between inside design and mean ambient (K).

By equating Eq. (13) to 0, it is possible to get the optimum temperature difference by iteration. The second derivative of the saving function with respect to ΔT, $[\partial^2 S / \partial^2 (\Delta T)^2]$, is calculable by substituting this specific optimum value, and result is found to be negative, which indicates a local maximum point.

$$\frac{\partial^2 S}{\partial \Delta T^2} = -2a_2 + \frac{2}{9}a_3\Delta T_{opt}^{-4/3} < 0 \quad ...(14)$$

Figure 12: Net Savings Vs. Temperature Difference Soylemez (2004)



As an example of a typical refrigeration plant that is interconnected to a waste heat recovery system, it is assumed that F =1 W/(m³ K), U = 1 W/(m² K), C_E = 10^{-5} $/(W h), C_{EL} = 7.5 × 10^{-5} $/(W h), H = 4000 h/year, T_L = 260 K, V_i = 500 m³, m/Q_H = 0.15 kg/(s kW), η_c = 1/3, τ = 6 K, c = 6, ΔP = 200 kPa, C_Q = 3 $/$W^{2/3}$, ρ = 1000 kg/m³, η_p = η_m = 0.7. The optimum temperature difference is calculated by using Eq. (13) as 33 K by trial and error approximately. The value of savings is plotted in Figure 12.

CONCLUSION

The present study investigates methods to increase the efficiency of the refrigeration cycle/ system, which will also contribute to reducing emission of Green House Gases (GHG). The detailed study was carried out to find out the influence of different operating parameters on the performance of refrigeration cycles and the following conclusions are drawn:

- The isothermal compression combined with the internal heat exchanger is all the more important for fluids with high compressor discharged temperature (ammonia, CO_2, etc.) and this could throw an interesting opportunity for CO_2 trans-critical cycles development as well as for spreading the field of application of ammonia. Furthermore, such cycle configuration may avoid the condenser or the gas cooler.
- The pressure ratio of the compressor was evenly distributed in both stages with R245fa at the optimum operation point of the cycle in the refrigeration process using the two-stage centrifugal compressor and the flash intercooler. The pressure ratio of the second stage was slightly higher with R134a. Moreover, the optimum operation point of the cycle is not at the point of the optimum efficiency of the either compressor stage.

- The optimum interstage pressure remains at a classical value applicable for multistage mechanical compression, with a gradual shift to lower values as the evaporating temperatures reduce in mechanical and thermal hybrid compression. An overall idea of the efficacy of the hybrid compression system is given by the reduction of mechanical compressor work per unit mass of adsorbent. Interchanging the mechanical and thermal compression systems between high and low stages has minimal effect with the low stage adsorption yielding a marginally better performance.
- A thermo economic optimization analysis is done yielding a simple algebraic formula for estimating the optimum operating temperature for refrigeration systems, which utilizes energy recovery applications.

The conventional energy analysis based on the first law of thermodynamics cannot adequately evaluate the economic trade-offs between work and heat output. As such, it is more appropriate to analyze the efficiency of refrigeration system based on the second law concept of energy. Better energy efficiency may or may not translate into lower operating costs for users and lesser environmental impact. For example future efforts to reduce compressor costs should be accompanied by a corresponding effort to improve compressor thermodynamic performance. Methods to assist engineers to feel easy in developing or operating simulation software and to use the simulation results in product development processes are required and will be developed in the future. Such techniques include: general refrigeration system simulation platform based on graph theory, model-based intelligent simulation technique, and combination of knowledge engineering methodology with simulation. In order to improve the performance of the refrigeration system, each component may be further studied from exergy usage and economics points of view. This study can be useful as it includes a review of the refrigeration cycles and the detailed thermodynamic analysis of the systems as well as the strategy for alleviating global warming.

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