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Exergy Flow Destruction of an Ice Thermal Energy Storage Refrigeration Cycle

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Abstract: A computational model based on exergy analysis of optimization of an ice-on coil thermal energy storage refrigeration cycle is developed in this paper. The method is based on exergy destruction analysis and optimization. As there are single and/or two phase refrigerant streams involved in the heat transfer and pressure drop in the compressor, condenser, expansion valve, evaporator, and between the ice tank and the environment, then there are irreversibilities or exergy destruction due to finite temperature difference and due to pressure losses. These two irreversibilities which represent the principles of components of the total irreversibilities are not independent and there is a trade-off between them. In this paper the effects of pressure drop ratio (*PDR*) in the evaporator and the condenser on the total number of exergy destruction units and the exergetic efficiency of a refrigeration cycle are determined. The pressure drop irreversibility to the total irreversibility for ΔP_{cond}=25 → 100 kPa and $PDR = 1$ are determined to be 7.45% → 27.08%.

Keywords: Refrigeration Cycle, Exergy Analysis, Exergy Destruction, Optimization, Ice Storage

1. Introduction

The exergy method of analysis is a technique based on both the first and second law of thermodynamics. It provides an alternative to the traditional methods of thermodynamic analysis and usually is aimed to determine the maximum performance of the system and identify where the available energy is insufficiently used. Refrigeration systems are the main contributor to the global warming problem by releasing large amounts of entropy to the environment. This entropy reflects the irreversibility of the process. In general the irreversibility rates are associated with heat transfer over a finite temperature difference $\dot{I}^{\Delta T}$ and with pressure losses in the stream $\dot{I}^{\Delta P}$ and cause the system performance to degrade. Thus the irreversibilities of the refrigeration cycle need to be evaluated considering individual thermodynamic processes that make up the cycle. The exergy analysis is presents a powerful tool in the design, optimization, and performance evaluation of energy systems [1]. The exergy analysis of refrigeration cycle has been studied by several authors [2-6]. Bejan [2] considered the exergetic efficiency of refrigeration cycle only due to heat transfer over a finite temperature difference. Exergy analysis of CFC12 refrigeration cycle was investigated by Leidenfrost [3], his analysis was based on entropy generation without considering the detailed transport phenomena inside the components of the cycle (condensation and evaporation processes). Exergy analysis was implemented to optimize the parameters of multi-stage refrigeration system by Chen, et al [4]. On basis of exergy analysis, Chen and Prasad [5] conducted a performance comparison analysis of vapor compression refrigeration systems using HFC134a and CFC12. Their results indicated that the COP for CFC12 is higher by about 3% than that for HFC134a system, caused by the total exergy loss. Bilgen and Takahashi [6] developed a simulation model for the energy and exergy analysis of a domestic heat pump-air conditioner considering the irreversibility of heat transfer and friction losses. They determined the exergy destruction in the heat pump components and found that the exergy efficiency varied from 0.25-0.37 for both operation modes heating or cooling.

Recently the air conditioning systems were coupled to thermal energy storage systems to act in accordance with calls for energy consumption. Several research studies have been conducted in recent years to evaluate the ice thermal energy storage performance characteristic for use in air conditioning applications [7, 8]. MacPhee and Dincer [9] evaluated the performance of ice storage charging and discharging processes on the basis of energy and exergy analyses. Jekel et al [10] presented a numerical model to predict the time for the solidification process in the ice-on coil storage tank. Their mathematical model was based on basic heat transfer and thermodynamic relations to determine the effectiveness of the thermal energy storage capacity. These thermal energy storage systems can be either sensible or latent heat storage. The evaporator of the refrigeration cycle is, therefore, immersed in a storage tank. In general, the refrigeration vapor compression cycle that is coupled to an ice thermal energy storage comprises a compressor, a condenser, an expansion device, an evaporator and ice storage. In the refrigeration system, the compressor energy use efficiency is normally determined by the manufacturer, which depends on the pressure ratio *PR*, the application and type of the compressor. While the traditional criterion of performance used in heat exchangers (condenser and evaporator) design is the effectiveness, which compares the actual enthalpy change of the output stream with the maximum enthalpy change attainable under ideal conditions. This criterion, however, takes no account of the price which must be paid in terms of high grade energy to compensate the irreversibilities due to pressure loss.

Since there are single and/or two phase refrigerant streams involved in the refrigeration process analysis, the principal components of total process irreversibility $\dot{I}^{\Delta P}$ and $\dot{I}^{\Delta T}$ are dependent and their relative contributions for the evaporator and condenser pressure affect the total irreversibility. The effect of changes in the pressure drop ratio ($PDR = \Delta P_{evap}/\Delta P_{cond}$) on the total cycle irreversibility can be investigated. The optimisation process in general can be formulated in terms of an objective function, which can be the minimum operating cost, maximum exergetic efficiency, minimum irreversibility and other parameters. In this paper the objective function is the number of exergy destruction units ($N_{I,t}$) which is related to exergetic efficiency. Therefore, the objective of the present work is to study the effect of changes in evaporator and condenser pressure drop ratio (*PDR*) on the number of exergy destruction units and exergetic efficiency of the refrigeration cycle components including ice thermal energy storage.

Engineering Equation Solver (EES) software [11] with built-in thermodynamic properties was used to evaluate the required refrigerant properties.

2. The Refrigeration Cycle Performance

The vapor compression cycle is the most frequently used in ice storage refrigeration system and composed from a compressor, a condenser, an expansion device, an evaporator, an ice tank plus auxiliary and the piping as shown in Figure 1.

The actual coefficient of performance of a vapor compression cycle, COP_a is defined as the rate at which heat must be removed from the refrigerated space to the electric power necessary to drive the compressor motor.

$$COP_a = \frac{\dot{Q}_{evap}}{\dot{W}_{el}} \tag{1}$$

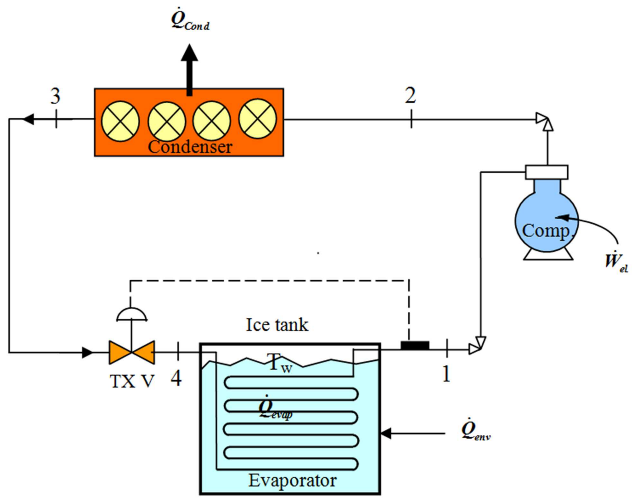

Figure 1. *A simple schematic of vapor compression cycle of an ice storage.*

The theoretical energy use by the compressor is the theoretical amount of energy that must be imparted to the refrigerant vapor as it follows a constant entropy line on the ph diagram. In practice, more power is required due to mechanical inefficiency in the compressor itself, inefficiency in the drive motor converting electrical energy to mechanical energy and the volumetric efficiency [12]. Assuming the refrigerant compressor to operate adiabatically, the electric power necessary to drive it is,

$$\dot{W}_{el} = \frac{\dot{m}_r\left(h_2 - h_1\right)}{\eta_m\, \eta_{el}\, \eta_v} = \frac{\dot{m}_r\left(h_2 - h_1\right)}{\eta_{eu}} \tag{2}$$

where η_{eu} is the compressor energy use efficiency which can be expressed in the form of a polynomial as follows;

$$\eta_{eu} = 9\times10^{-8}\times PR^6 - 6\times10^{-6}\times PR^5 + 1\times10^{-4}\times PR^4 - 1\times10^{-3}\times PR^3 - \\ -3.5\times10^{-3}\times PR^2 + 0.0618\times PR + 0.5364 \tag{3}$$

The compressor energy use efficiency is normally determined by the manufacturer. Compressor energy use efficiency depends on the pressure ratio *PR* and the application and type of the compressor.

The power input to the refrigeration cycle auxiliary equipment namely condenser fans, chilled water pumps, and control system. is estimated to be between (10-15%) of the compressor power use for water cooled or evaporative condensers but can be higher for air cooled condensers [12]. In the present study, an air- cooled condenser is used, then a 20% of the power required to drive the compressor motor is estimated for the cycle auxiliary. This power input must be regarded as the exergy input $\dot{E}_{in}$ to the cycle. The actual coefficient of performance can then be expressed as,

$$COP_a = \frac{\dot{Q}_{evap}}{1.2\,\dot{W}_{el}} \tag{4}$$

Another useful *COP* is the maximum possible *COP* which is that of the perfect system with the perfect compressor. The perfect refrigeration cycle is the reverse Carnot cycle. The *COP* of the reverse Carnot cycle is designated COP_c and is given by [12],

$$COP_c = \frac{T_e + 273.15}{T_c - T_e} \tag{5}$$

where T_e is the evaporation temperature °C, and
T_c is the condensing temperature °C
Hence, the actual refrigerating efficiency is,

$$\eta_{r,a} = \frac{COP_a}{COP_c} \tag{6}$$

An alternative approximate method is proposed by Cleland *et al* [12] for calculating the electric power usage rate for all refrigerants as follows:

$$\dot{W}_{el} = \frac{\dot{Q}_{evap}}{COP_c\left(1-\alpha x\right)^n \eta_{eu}} = \frac{\dot{Q}_{evap}\left(T_c - T_e\right)}{\left(T_e + 273.15\right)\left(1-\alpha x\right)^n \eta_{eu}} \tag{7}$$

where
α empirical constant (see Table 1)
x quality, can be evaluated at (P4) and (h4) with the help of the engineering equation solver software EES with built in thermodynamic functions [12].
n empirical constant depends on the number of compression stages and number of expansion stages. In this study is equal to 1 because a simple refrigeration cycle is considered and the number of compression and expansion stages is 1 each.

Table 1. *Values of the empirical constant [12].*

Refrigerant	α
12	0.67
22	0.77
134 a	0.69
404 A	0.82
717	1.11

Table 2. *Variation of irreversibility verses pressure drop ratio for PDR=1.*

ΔP_{cond}, kPa	$\dot{Q}_{evap}$, kW	$\dot{I}_{evap}$, kW	$\dot{I}_{evap}/\dot{Q}_{evap}$	$\dot{W}_{el}$, kW	$\dot{I}_{comp}$, kW	$\dot{I}_{comp}/\dot{W}_{el}$
0	137.3	7.56	0.0551	53.69	7.10	0.1322
25	136.4	9.74	0.0714	53.34	7.41	0.1389
50	135.4	12.11	0.0894	52.96	7.74	0.1461
75	134.4	14.71	0.1094	52.55	8.10	0.1541
100	133.2	17.59	0.1321	52.1	8.50	0.1631

Table 2. *Continued.*

ΔP_{cond}, kPa	$\dot{Q}_{cond}$ kW	$\dot{I}_{cond}$, kW	$\dot{I}_{cond}/\dot{Q}_{cond}$	$\dot{I}_{ex}$, kW	$\dot{Q}_{cond}$ kW	$\dot{I}_{cond}$, kW	$\dot{I}_{cond}/\dot{Q}_{cond}$
0	191.3	7.84	0.0410	11.4	33.9	0.0	0.0
25	193.0	8.10	0.0420	11.38	36.63	2.73	0.0745
50	194.9	8.40	0.0431	11.36	39.61	5.71	0.1442
75	196.9	8.71	0.0442	11.33	42.85	8.95	0.2089
100	199.2	9.10	0.0457	11.30	46.49	12.59	0.2708

3. Irreversibility Rate Expressions

The irreversibility rate is a measure of thermodynamic imperfection of process and is expressed in terms of lost work potential. There are irreversibilities that occur within the region (internal irreversibilities) which are due to entropy production within the control region. Also, there are those that occur outside the region (External irreversibilities), these include the degradation of the thermal energy (heat loss), dissipation of kinetic energy (pressure loss) and those due to mixing with the atmospheric air and uncontrolled chemical reaction. In refrigeration cycle, the heat loss and pressure loss are the major source of irreversibilities. In general, the expression for irreversibility rate can be expressed as [13]

$$\dot{I} = T_o\left[(\dot{S}_{out} - \dot{S}_{in}) - \sum_{i=1}^{n}\frac{\dot{Q}_i}{T_i}\right] = T_o\dot{\Pi} \geq 0 \tag{8}$$

where $\dot{S} = \dot{m}_r\, s$ entropy production rate (W/K)

$\dot{Q}_i = \dot{m}_r\, q_i$ heat transfer rate (W)

$\dot{\Pi}$ entropy production rate in the control region

4. Adiabatic Compressor and Energy Calculations

The expression for the irreversibility rate when the adiabatic compressor $(q_o = 0)$ is considered is,

$$\dot{I}_{comp} = \dot{m}_r\, T_o\, (s_2 - s_1) \tag{9}$$

The dot-hatched area in Figure (2) represents the irreversibility of an adiabatic compressor.

5. Condenser

The condenser model used in this study is an air cooled cross flow with continuous fins on an array of tubes as shown in Figure 3a. The function of a condenser in a refrigeration system is to reject thermal energy to the environment ($q_{cond} = -q_o$) . Thus, the expression for the irreversibility rate due to heat transfer over a finite temperature difference is given respectively by (see Figure 3b),

$$\dot{I}^{\Delta T}_{cond} = \dot{m}_r\, T_o \left[(s_3 - s_2) + \frac{q_{cond}}{T_o} \right] =$$

$$\dot{m}_r\, T_o \left[(s_3 - s_2) + \frac{(h_2 - h_3)}{T_o} \right] \tag{10}$$

The irreversibility due to pressure consideration in this study is based on the refrigerant status inside the condenser tube. The condenser is divided into three regions: superheated vapor region (Z_1), two phase (saturation) region (Z_2), and subcooled liquid region (Z_3) as shown in Figure 3a.

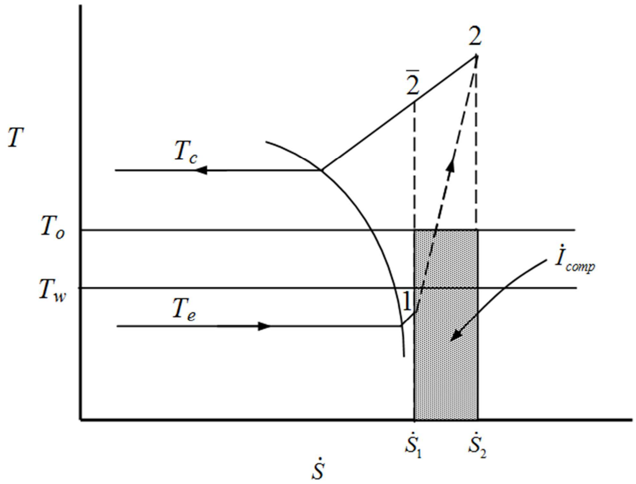

Figure 2. *Schematic presentation of compressor irreversibilities.*

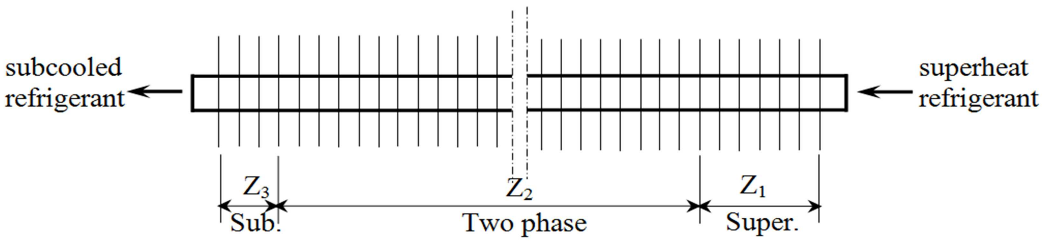

Figure 3a. *Condenser model.*

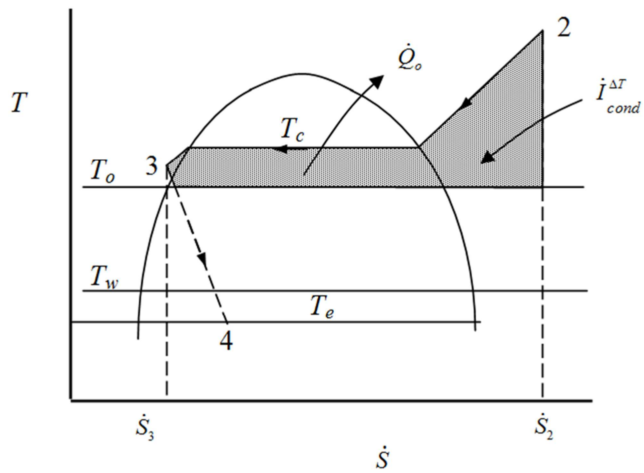

Figure 3b. *Schematic presentation of the temperature irreversibility rate of a condenser.*

5.1. Superheated Vapor Region

In this section heat exchange between two gaseous streams (superheated vapor refrigerant and the outside air), the pressure contributes significantly to the overall irreversibilities [13] and cannot be neglected in the analysis of the process. Figure 4a shows only the superheated region pressure irreversibilities which can be expressed as,

$$\dot{I}^{\Delta P}_{cond,\sup} = \dot{m}_r\, T_o \left(s_{\bar{\bar{2}}} - s_{\bar{2}} \right) \tag{11}$$

5.2. Two Phase (Saturation) Region

The two- phase flow irreversibilities have been discussed by several authors such as [14, 15]. The entropy generation due to pressure losses through the condenser can be expressed assuming that the inlet state ($\bar{2}$) and outlet state ($\bar{3}$) are both in the two- phase domain and the auxiliary state ($\bar{3}$) is defined by the two properties $T_{\bar{3}} = T_{\bar{2}}$ and $h_{\bar{3}} = h_{\bar{\bar{3}}}$ as shown in Figure 4b. Bejan [14] presented the two- phase pressure drop entropy production by

$$\dot{S}_{gen} = \dot{m}_r \left(s_{\bar{\bar{3}}} - s_{\bar{3}} \right) \tag{12}$$

The two- phase flow irreversibilities can then be expressed as,

$$\dot{I}^{\Delta P}_{cond,sat} = \dot{m}_r\, T_o \left(s_{\bar{\bar{3}}} - s_{\bar{3}} \right) \tag{13}$$

5.3. Subcooling Liquid Region

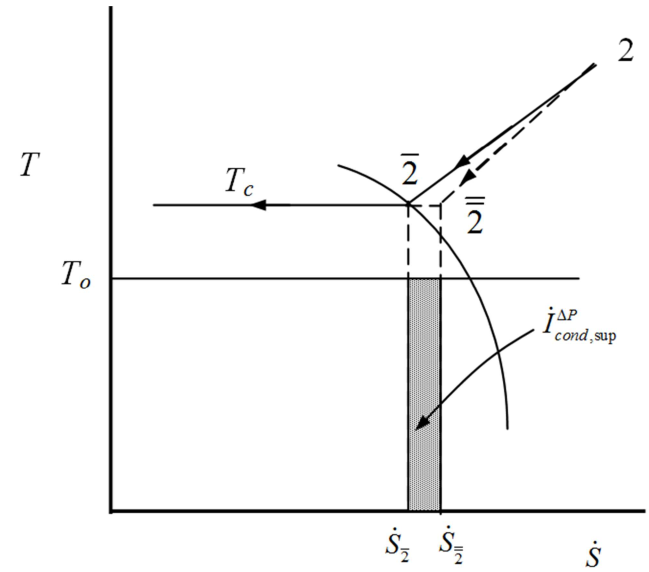

Figure 4a. *Condenser superheated region pressure drop irreversibility.*

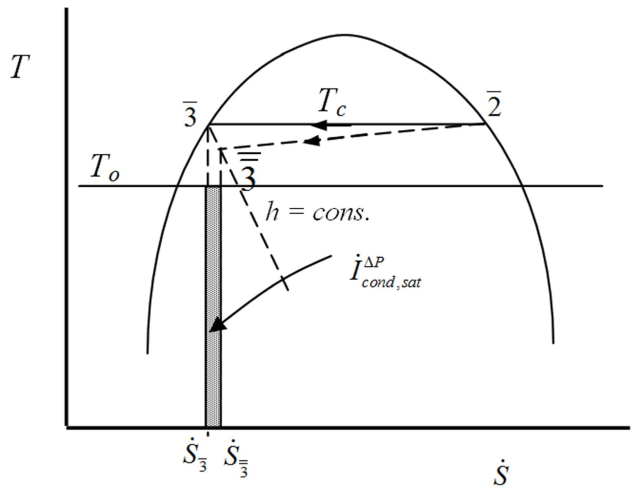

Figure 4b. *Condenser two phase region pressure drop irreversibility.*

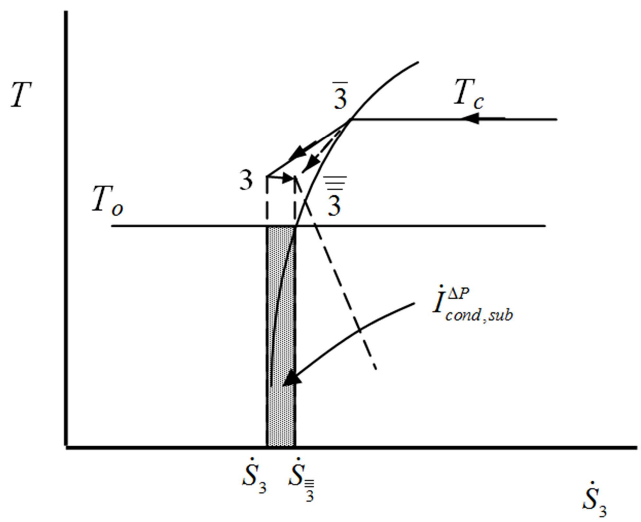

Figure 4c. *Condenser subcooling region pressure drop irreversibility.*

The irreversibility rate due to pressure losses through the condenser subcooling region can be expressed in the same manner of the superheated region (see Figure 4c).

$$\dot{I}^{\Delta P}_{cond,sub} = \dot{m}_r\ T_o\left(s_{\overline{\overline{3}}} - s_3\right) \tag{14}$$

The total condenser pressure irreversibilities rate then is

$$\dot{I}^{\Delta P}_{cond} = \dot{I}^{\Delta P}_{cond,\sup} + \dot{I}^{\Delta P}_{cond,sat} + \dot{I}^{\Delta P}_{cond,sub} \tag{15}$$

The total condenser irreversibilities are the sum of the irreversibilities due to finite temperature difference and the irreversibilities due to pressure losses.

$$\dot{I}_{cond} = \dot{I}^{\Delta T}_{cond} + \dot{I}^{\Delta P}_{cond} \tag{16}$$

6. Expansion Valve

The expansion process of a refrigeration system takes place when the flow of refrigerant passes through a restricted passage such as when partially opened. The irreversibility rate can be expressed when the kinetic and potential energies are taken to be negligible as shown in Figure 5.

$$\dot{I}_{ex} = \dot{m}_r\ T_o\left[(s_4 - s_3)\right] \tag{17}$$

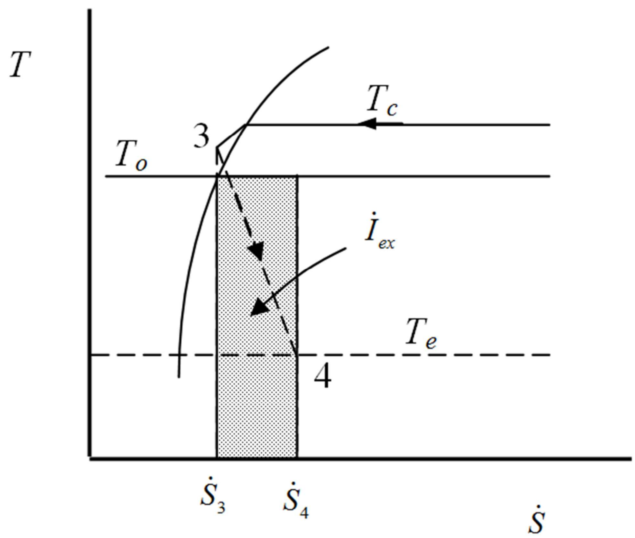

Figure 5. *Expansion process irreversibility.*

7. Evaporator

The evaporator model considered in this paper is copper tubing submerged in water which serves as a heat exchanger. Water/refrigerant circulates inside the tubing and extracts heat from the water to create ice on the outside of the tubing as shown in Figure 1. The evaporator operates over a range of temperatures below that of the environment. Heat from the lower temperature reservoir (ice storage) at temperature T_w is transferred to the evaporating refrigerant $(q_{evap} = q_w)$. The irreversibility rate due to heat transfer over a finite temperature difference and due to pressure losses are given respectively see Figure 6b,

$$\dot{I}^{\Delta T}_{evap} = \dot{m}_r\, T_o \left[(s_1 - s_4) - \frac{q_{evap}}{T_w} \right] =$$

$$\dot{I}_{evap} = \dot{m}_r\, T_o \left[(s_1 - s_4) - \frac{(h_1 - h_4)}{T_w} \right] \tag{18}$$

The irreversibility due to pressure analysis is similar to that for the condenser which is divided into two phase (saturation) region (Z_4) and super heat vapor region (Z_5) as shown in Figure 6a.

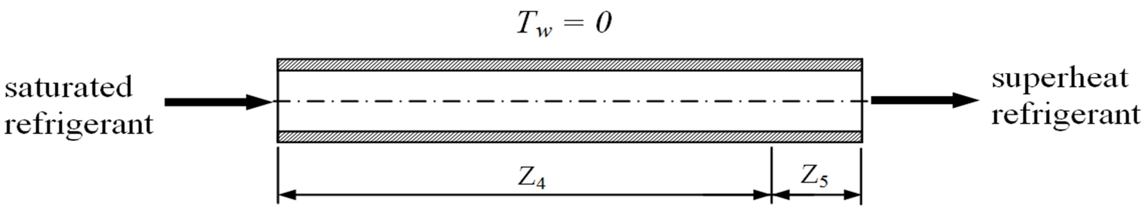

Figure 6a. *Evaporator model.*

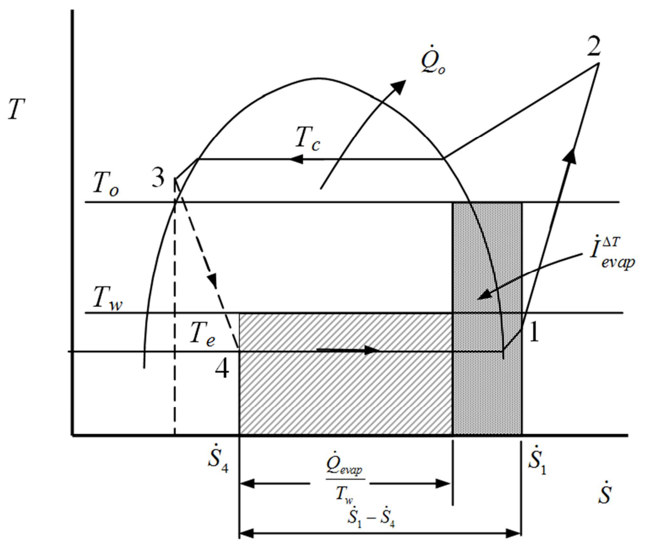

Figure 6b. *Schematic presentation of the temperature irreversibility rate of an evaporator.*

7.1. Two- Phase (Saturation) Region

The two- phase pressure drop irreversibility through the evaporator is shown in Figure 7a and can also be expressed assuming that the outlet state ($\bar{1}$) is situated on the saturated vapor line and $h_{\bar{1}} = h_{\bar{\bar{1}}}$. The irreversibility can be expressed as,

$$\dot{I}^{\Delta P}_{evap,sat} = \dot{m}_r\, T_o \left(s_{\bar{\bar{1}}} - s_{\bar{1}} \right) \tag{19}$$

$$\dot{I}_{evap} = \dot{I}^{\Delta T}_{evap} + \dot{I}^{\Delta P}_{evap} \tag{22}$$

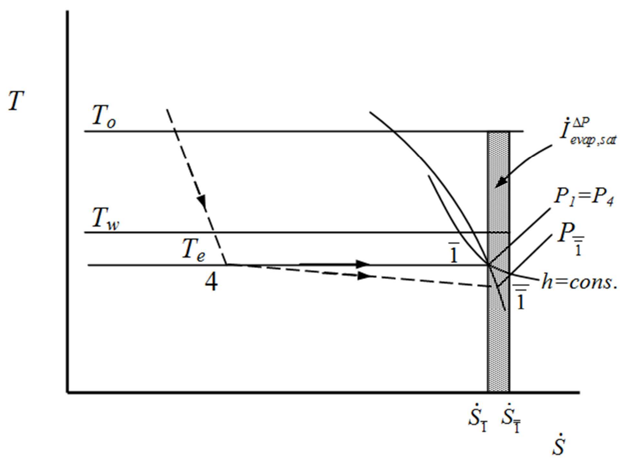

Figure 7a. *Evaporator two- phase region pressure drop irreversibility.*

7.2. Superheat Vapor Region

The pressure irreversibility of the superheat vapor region is given by, see Figure 7b

$$\dot{I}^{\Delta P}_{evap,sup} = \dot{m}_r\, T_o \left(s_{\bar{\bar{\bar{1}}}} - s_1 \right) \tag{20}$$

The total evaporator pressure irreversibilities rate then is

$$\dot{I}^{\Delta P}_{evap} = \dot{I}^{\Delta P}_{evap,sat} + \dot{I}^{\Delta P}_{evap,\sup} \tag{21}$$

The total evaporator irreversibilities is the sum of the irreversibilities due to finite temperature difference and the irreversibilities due to pressure losses.

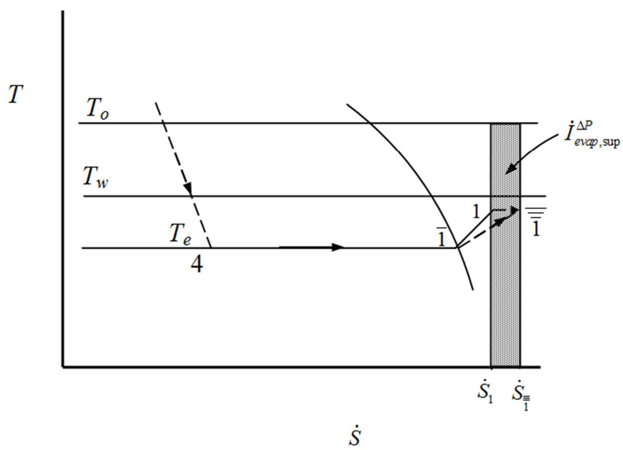

Figure 7b. *Evaporator superheated region pressure drop irreversibility.*

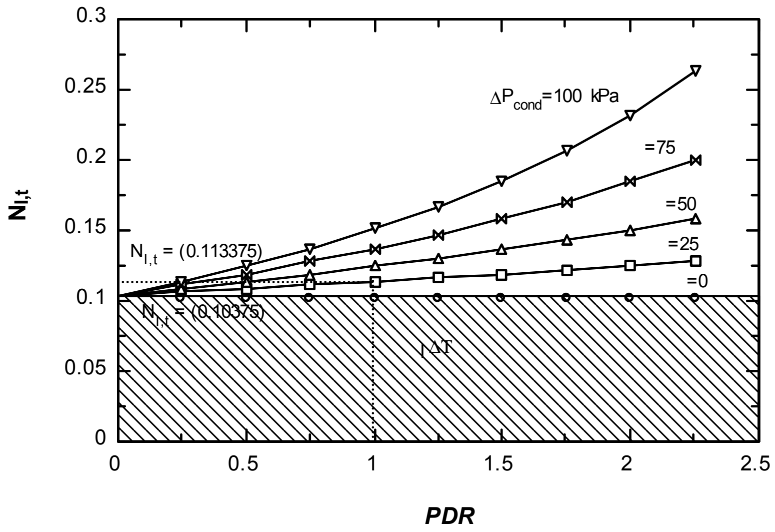

Figure 8. *Variation of the number of exergy destruction units ($N_{I,t}$) verses pressure drop ratio.*

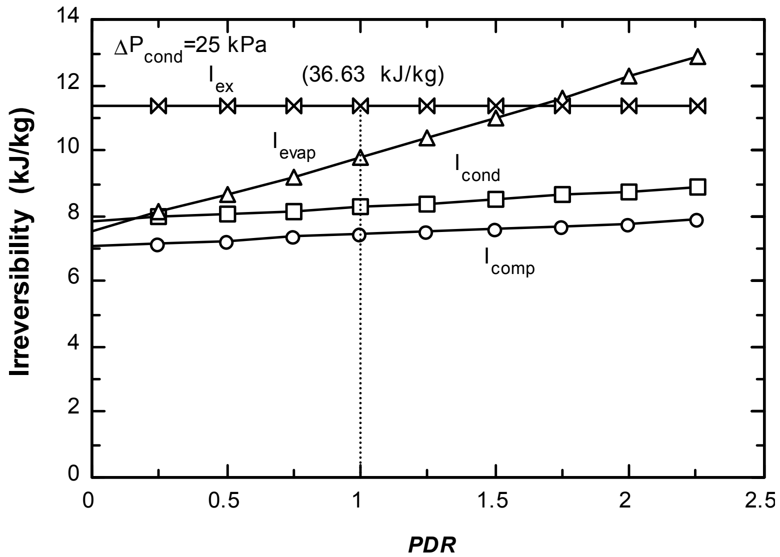

Figure 9. *Variation of irreversibility verses pressure drop ratio for ΔP_{cond} =25 kPa.*

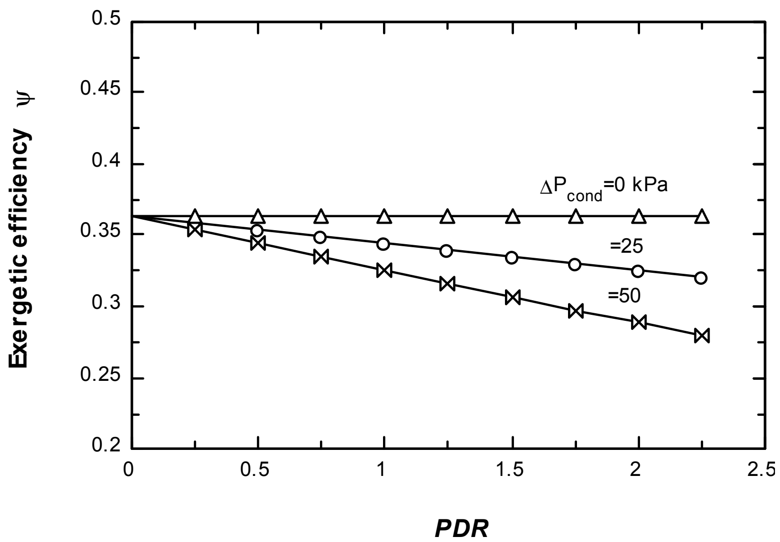

Figure 10. *Variation of exergetic efficiency versus pressure drop ratio.*

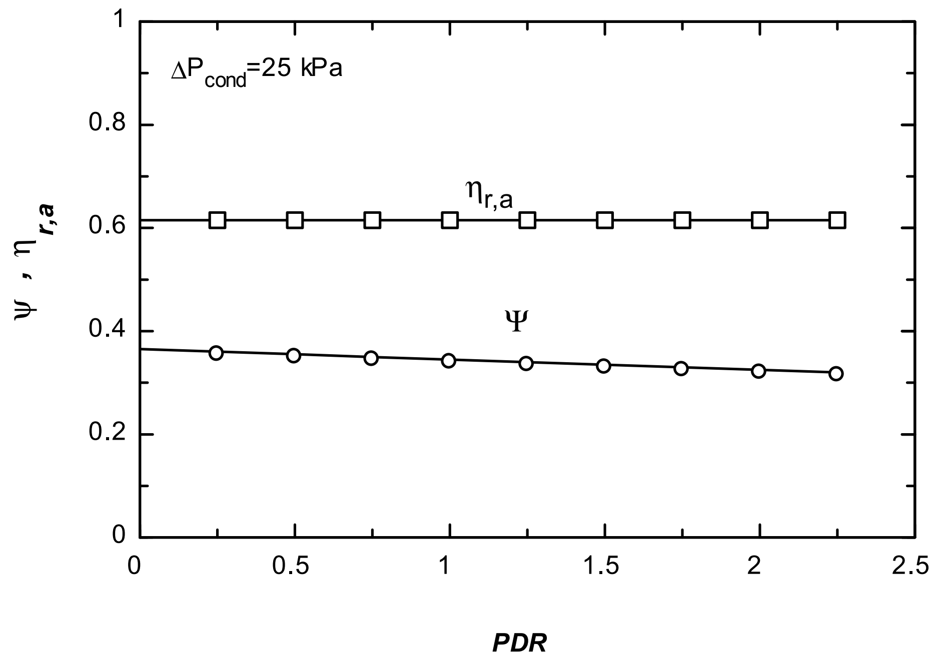

Figure 11. *Comparison between the exergetic efficiency and the actual refrigeration efficiency for ΔP_{cond} = 25 kPa.*

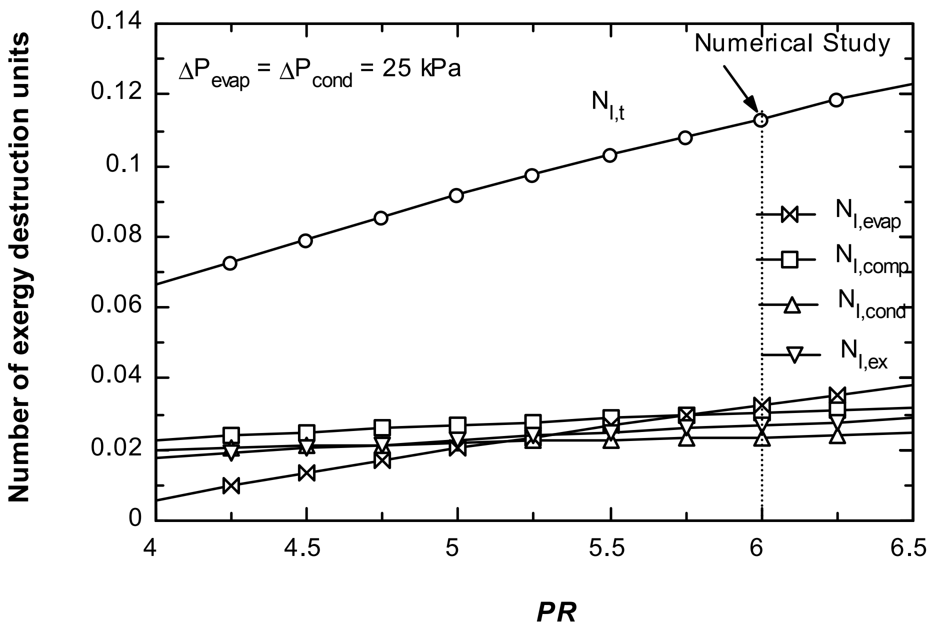

Figure 12. *Variation of the refrigeration cycle component number of exergy destruction units ($N_{I,t}$) verses pressure drop ratio for ΔP_{cond} =25 kPa.*

8. Ice Storage Tank

The irreversibility rate of ice storage to the environment can be defined as,

$$\dot{I}_{ice\,tank} = T_{o,it}\ \dot{S}_{gen} \tag{23}$$

where $T_{o,it}$ is the ambient temperature of the air surrounding the ice tank

The generation of entropy rate is [15]

$$\dot{S}_{gen} \geq \sum\left(\frac{\dot{Q}_{env}}{T}\right)_{in} - \sum\left(\frac{\dot{Q}_{env}}{T}\right)_{out} \tag{24}$$

The heat gain from the environment

$$\dot{Q}_{env} = U\,A(T_{o.it} - T_w) \tag{25}$$

The overall heat transfer coefficient

$$U = \frac{1}{\sum R_{th}} \tag{26}$$

The overall thermal resistance

$$\sum R_{th} = \frac{1}{h_o} + \sum_{i=1}^{m} \frac{x_i}{k_i} + \frac{1}{h_w} \tag{27}$$

where m is the number of ice tank wall layers
x thickness of each layer

The term $\sum\left(\frac{\dot{Q}_{env}}{T}\right)_{in}$ is for heat transfer rate into the ice storage from the surroundings and $\sum\left(\frac{\dot{Q}_{env}}{T}\right)_{out}$ term is for the heat transfer rate out of the ice storage to the surroundings. When the equal sign is used for an idealize ice storage. Thus, the ice storage irreversibilities can be defined as

$$\dot{I}_{ice\,tank} = \dot{Q}_{env}\left(\frac{T_{o,it}}{T_w} - 1\right) \tag{28}$$

The total irreversibility rate of a vapor compression cycle and its components can be expressed as,

$$\dot{I}_t = \sum_{i=1}^{n} \dot{I}_i = \dot{I}_{comp} + \dot{I}_{cond} + \dot{I}_{ex} + \dot{I}_{evap} + \dot{I}_{ice\,tank} \tag{29}$$

9. Exergetic Efficiency and Number of Exergy Destruction Units

Rational efficiency is a criterion of performance which can be formulated for the vapor compression refrigeration cycle or its components for which the output is expressible in terms of exergy. Defining the rational efficiency ψ, as a ratio of exergy output to exergy input.

$$\psi = \frac{\dot{E}_{out}}{\dot{E}_{in}} \tag{30}$$

The exergy balance for the refrigeration cycle shown in Figure 1 can be expressed in the following form,

$$\dot{E}_{out} = \dot{E}_{in} - \dot{I}_t \tag{31}$$

From eqs. (30) and (31), the rational efficiency can be written

$$\psi = 1 - \frac{\dot{I}_t}{\dot{E}_{in}} \tag{32}$$

The exergy flux rate is associated with the power input to the refrigeration cycle or the electric power. These two quantities are simply related as (see equation 4).

$$\dot{E}_{in} = 1.2\,\dot{W}_{el} \tag{33}$$

Then

$$\psi = 1 - \frac{\dot{I}_t}{1.2\,\dot{W}_{el}} \tag{34}$$

The range of value of ψ lies within the following limits $0 \le \psi \le 1$ and is always less than unity, the difference depends upon the degree of irreversibility.

In addition to the exergetic efficiency and the common geometric dimensionless group of heat transfer processes such as Reynolds number, Prandtl number.... et cetera, an important dimensionless called number of exergy destruction units is defined by [16],

$$N_I = \frac{\dot{I}}{T_o\left(\dot{m}\,c_p\right)_{\min}} \tag{35}$$

In this paper, the only refrigerant side irreversibilities were considered and the irreversibilities are calculated per mass flow rate, hence

$$N_I = \frac{I}{T_o\,c_{p.r}} \tag{36}$$

where $c_{p,r}$ is evaluated at the average refrigerant inlet and outlet temperature for each cycle components and for each condenser and evaporator zone, with the help of the equation solver software *EES* with built in thermodynamic properties [11]. The total number of exergy destruction units ($N_{I,t}$) is defined as,

$$N_{I,t} = N_{I,comp} + N_{I,cond} + N_{I,ex} + N_{I,evap} + N_{I,icetank} \tag{37}$$

10. Numerical Study

The calculations were carried out with the help of the equation solver software *EES* [11]. An ice storage refrigeration cycle as illustrated in Figure 1 using R22 as the refrigerant is considered. Air cooled condenser and evaporator coils are considered for the heat exchangers. The details of the refrigeration cycle operation conditions are as follows:

Refrigerant mass flow rate $\dot{m}_r$ = 1 kg/s
Ambient temperature T_o = 40°C
Condenser Design Temperature Difference (TD_c) =10°C
Pressure ratio (PR) = P_{cond}/P_{evap} = 6
Subcooling = 3 K
Superheat = 5 K
Ice storage water temperature T_w = 0°C

The ice tank irreversibility was calculated for the following parameters

The ice tank material is stainless steel.
The outside air heat transfer coefficient h_o = 34.1 W/m^2°C
Internal and external ice tank material layer thickness

= 0.001 m each.

Stainless steel thermal conductivity =15 W/m K

Insulation thermal conductivity = 0.04 W/m K"

Thickness of the insulation = 0.01m

The heat transfer coefficient h_w = 50 W/m^2°C

Refrigerant heat transfer coefficient h_{ref} = 2000 W/m^2°C

Number of coil loop =12

Space between coil loop 0.05m

Coil copper tube size 3/8 in

Ambient temperature of the air surrounding the ice tank =25°C

Ice tank height = 0.5 m

11. Results and Discussion

The results of the optimization calculations are presented in Figures 8-12. Figure 8 clearly demonstrates the fact that when the pressure drop in the condenser is zero, the pressure drop in the evaporator is also zero. In this case the irreversibility will be only due to heat transfer over a finite temperature difference $\dot{I}^{\Delta T}$ and the basic number of exergy destruction units ($N_{I,t}$) is 0.10375, for the parameters given in previous section. The corresponding total irreversibility due to heat transfer over a finite temperature difference ($I_t = I_{evap} + I_{comp} + I_{cond} + I_{ex} + I_{ice\ tank}$) is 33.903 kJ/kg. For a constant value of condenser pressure drop, the exergy destruction number increases with the *PDR* due to the increasing contribution of the evaporator irreversibility $I_{evap}^{\Delta P}$.

For example, consider if the pressure drop ratio (*PDR*) is unity and the condenser pressure drop is 25 kPa. The number of exergy destruction units ($N_{I,t}$) in this case is 0.113375; in other words there is an increase in the irreversibility by about 7.45% which means that there are 2.73 kJ/kg irreversibilities due to pressure drop and the total compensated irreversibility should be 36.63 kJ/kg (33.903+2.74) as seen in Figure 9 and Table 2. These values should be evaluated in the design stage and compensated in the sizing up the refrigeration cycle components.

Figure 9 also shows that the evaporator irreversibility I_{evap} increases greatly with the pressure drop ratio because of the superheat effect, while changes for other components are relatively small. The ice storage irreversibilities based on the numerical study case is found to be about 0.1% from the total irreversibility of other components, which can be neglected.

The variation of the exergetic efficiency ψ, with condenser pressure drop and the ratio of *PDR* is shown in Figure 10. For constant ΔP_{cond}, the exergetic efficiency decreases with the pressure ratio because of the effect of evaporator irreversibility. The exergetic efficiency, ψ at zero condenser pressure drop is 0.3717 which corresponds to the condition for $N_{I,t}$ = 0.10375 of Figure 8.

Figure 11 clearly shows both the actual refrigerating efficiency and the exergetic efficiency variations with *PDR* for constant value of condenser pressure drop. The actual refrigerating efficiency can mislead because such efficiency weight all thermal energy equally, whilst the exergetic efficiency acknowledges the usefulness of irreversibilities on its quality and quantity. Thus, exergetic efficiency is more suitable for determining the right capacity for the refrigeration components.

The results shown in Figures 8-11 were plotted for pressure ratio (*PR*) = 6. Figure 12 shows the effect of pressure ratio (*PR*) on the number of exergy destruction units (N_I) of all components, for equal condenser and evaporator pressure drop of 25 kPa. It is seen that the significant impact of changing the *PR* is on the irreversibility of the evaporator; that is due to the constant condensing temperature. Increase in *PR* means reduction of the evaporation pressure that increases the work and hence increases N_I.

12. Conclusions

This study shows the usefulness of using the number of exergy destruction units ($N_{I,t}$) and the exergetic efficiency compared to energy efficiency for refrigeration systems that include an ice storage tank. They take into account the irreversibilities of the refrigeration cycle, and hence it reflects the thermodynamic and economic values of the combined refrigeration cycle performance. As a result, the irreversibility of the ice storing tank is small compared to those other components and can be safely ignored. As mentioned earlier, the irreversibility due to heat transfer over the finite temperature difference depends on the condensing temperature; hence, this irreversibility can be minimized by using the correctly designed heat exchangers' temperature difference. When *PDR =1*, the pressure drop irreversibility to the total irreversibility for ΔP_{cond} =25 → 100 kPa is determined to be 7.45% → 27.08% as shown in Table 2. Also, the irreversibility due to pressure losses can be minimized by selecting the optimum tube diameter and geometrical parameters.

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Nomenclature

A heat transfer area, according to subscripts, m^2

c_p specific heat capacity, kJ/ kg K

EES Engineering Equation Solver

I irreversibility, kJ/kg

N_I number of exergy destruction units, see equations (35 and 36), dimensionless

$\dot{m}$ mass flow rate, kg/s

P_o atmospheric pressure, kPa

PDR evaporator and condenser pressure drop ratio

($\Delta P_{evap}/\Delta P_{cond}$)

PR pressure ratio (P_{cond}/P_{evap})

T_o atmospheric temperature, K

T temperature, K, according to subscripts

Greek Symbols

ΔP pressure drop, kPa

η energy efficiency

ψ exergetic efficiency

Subscripts

cond condenser

comp compressor

e evaporator

ex expansion valve

el electrical

m mechanical

o,env environmental state

r refrigerant

t total

v volumetric

Superscripts

ΔP pressure component

ΔT thermal component

References

[1] R. Yumrutas, M. Kunduz, and M. Kanoglu, Exergy analysis of vapor compression refrigeration systems, Exergy, an international Journal, No. 2, pp 266-272, 2002.

[2] A. Bejan, Theory of heat transfer-irreversible refrigeration plants, J. Heat Mass Transfer 32 (9), pp 1631-1639, 1989.

[3] W. Leidenfrost, K. H. Lee, and K. H. Korenic, Conservation of energy estimated by second law analysis of power-consuming process, Energy (5), pp 47-61, 1980.

[4] J. Chen, X. Chen, and C. Wu, Optimization of the rate of exergy output of multistage endoreversible combined refrigeration system, Exergy, An International Journal, 1, (2) pp100-106, 2001.

[5] Q. Chen, and R. C. Prasad, Simulation of a vapour-compression refrigeration cycles using HFC134a and CFC12, Int. Comm. Heat Mass Transfer, Vol 26, No 4 pp 513-521, 1999.

[6] E. Bilgen and H. Takahashi, Exergy analysis and experimental study of heat pump systems, Exergy, An International Journal, 2, (4) pp259-265, 2002.

[7] B. A. Habeebullah, Economic feasibility of thermal energy storage systems, Energy and Buildings 39 (2007) 355–363.

[8] D. MacPhee, I. Dincer, Performance assessment of some ice TES systems, International Journal of Thermal Sciences 48 (2009) 2288–2299.

[9] I. Dincer, M. A. Rosen, Energetic, environmental and economic aspects of thermal energy storage systems for cooling capacity, Applied Thermal Engineering 21 (2001) 1105–1117

[10] Jekel TB, Mitchell JW, Klein SA. Modeling of ice storage tanks. ASHRAE Transactions 1993; 99: 1016-1024

[11] K. A. Klein, and F. L. Alvarado, EES, Engineering equation solver (2007), Version 7.933, F-Chart Software, WI. USA.

[12] A. C. Cleland, D. J. Cleland, and S. D. White, Cost-Effective Refrigeration, A five day Teaching Workshop. Institute of Technology and Engineering, Massey University, New Zealand, 2000.

[13] T. J. Kotas, The exergy method of thermal plant analysis". Reprinted, Krieger, Malabar, Florida, USA, 1995.

[14] A. Bejan, Advanced engineering thermodynamics" John Willey, New York, USA, 1997.

[15] A. L. London, and R. K. Shah, Cost of irreversibilities in heat exchangers design, Heat Transfer Engineering, Vol. 4, No. 2, pp 59-73, 1983.

[16] C. F. Tapia, and M. J. Moran, Computer-Aided Design and optimization of heat exchangers", Computer-Aided Engineering of Energy Systems, Vol. 1-Optimization ASME, pp 93-103, 1986.