AN EXERGY ANALYSIS OF AN AIR CONDITIONING SYSTEM

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Abstract

In today's modern building construction, air-conditioning system is almost a must. Even at it smallest component – the home, having the air condition installed is a great comfort; particularly in the tropics. It is due to its significant use that the air conditioning system is being selected as the subject of this study. Currently, the recognized methods in the study of energy conservation are the energy and the exergy methods. The home air conditioning unit is used to cool a space with effective temperature for human comfort at 25°C. In this study the behaviour of home air-conditioning unit using refrigerant R-134a has been investigated through the exergy method. Local ambient temperature used is 30°C. The challenge is to obtain quantitative information that will lead to a better understanding of the air conditioning irreversibility process and their distribution amongst the system component and minimizing them for optimal air conditioning cycle. In illustrating energy and availability analysis characteristic, the compressor work is altered through a pressure of 0.2 bar and was reduced from 2 bar to 1.8, 1.6, 1.4, 1.2 and eventually 1.0 bar. The greater the pressure difference which involved in the increase of the compressor work will lead to a higher rate of irreversibility. It is found that the evaporator and compressor loss contributed to the performance of the whole plant. The plant efficiency reduces about 52% when the inlet pressure at the compressor decreased from 2 bar to 1 bar. The greater the temperature difference between the evaporator and the cold room the higher is the irreversibility rate. Thus, the changes in temperature at the evaporator and pressure at the compressor contribute significantly to the plant irreversibility. Due to these phenomena there is a need for optimising the conditions imposed upon the evaporator and compressor.

Key words: Energy analysis; Optimisation; Exergy; Thermodynamic losses

Introduction

Air conditioning is defined as the simultaneous processing of temperature, humidity, purification and distribution of air current in compliance with the requirement of space needing air conditioning [1]. Recently in addition to the four elements stated above radiation and pressure have often been added as items to be controlled. Refrigeration plant component is one of the main systems for air conditioning system. The components of the air conditioning system consist of the compressor (reciprocating type), heat exchanger (condenser), throttling valve and evaporator (plate type). The conditioning process occurs at the evaporator. Here is where cooling of the air is taking place. The air contains moisture as well as gases mainly N_2. The compressor is powered by an electric motor and the type of compressor used is positive displacement type (reciprocating compressor). All the symbols used in the equations were provided in Appendix A. From the literature survey, it is found that an exergetic method is recommended in studying the performance of energy system. Exergy method is used to offer a better understanding of the process irreversibilities and their distributions among the system components and minimizing them for optimal refrigeration cycle [2].

The purpose of this investigation is to improve our understanding of refrigeration system hence their efficiencies. The exergy values in the cycle contribute to its performance when the pressure is increased in the compressor of the system studied.

Modeling Of Each Component

Figure 1 shows the schematic of the equipment and Figure 2 shows the $T-s$ diagram for a home air conditioning system. In order to determine the heat transfer and thermodynamics properties of all the components, each component was taken as a single unit [3].

Energy Analysis

Energy balance is achieved through the used of the first law of thermodynamics. The general conservation of energy equation for the steady state flow through a control volume frequently written in format on a rate basis as

$$\dot{Q}+\dot{W}_{net}+\sum_{in}\left(h+\frac{v^2}{2}+gz\right)\dot{m}-\sum_{out}\left(h+\frac{v^2}{2}+gz\right)\dot{m}=\frac{dE_{CV}}{dt} \qquad (1)$$

and on a unit-mass basis for a control volume with one inlet (1) and one outlet (2),

$$q+w=\left(h_2-h_1\right)+\left(\frac{V_2^2}{2}-\frac{V_1^2}{2}\right)+g\left(z_2-z_1\right) \qquad (2)$$

where w usually represents any shaft work present, the specific enthalpy $h\equiv u+Pv$, and $\dot{m}=\rho AV$ is the mass flow rate at a given position.

The kinetic and potential energy are neglected. The work done by the compressor is given by:

a. Actual work done

$$w_a=h_{2a}-h_1 \qquad (3)$$

b. Adiabatic work

$$w_s=h_{2s}-h_1 \qquad (4)$$

The isentropic efficiency can be calculate by

$$\eta_{isen}=\frac{w_s}{w_a} \qquad (5)$$

As the compressor is powered by an electric motor, the motor output can be calculate as

$$w_m=\frac{w_a}{\eta_e} \qquad (6)$$

Using the state notations found on Figure 1 and Figure 2, the following are *steady-state energy relations* for the compressor, condenser, throttling valve, and the evaporator.

a) Compressor

$$q_c+w_{in}=h_{2a}-h_1 \qquad (7)$$

b) Condenser

$$q_{cond}=h_3-h_{2a} \qquad (8)$$

c) Throttling valve

$$h_3 = h_4 \tag{9}$$

d) Evaporator

$$q_{evap} = h_1 - h_4 \tag{10}$$

The COP is calculated by

$$COP = \frac{q_L}{w_m} \tag{11}$$

The first law contains a term of work, but there is no energy term, which accounts for irreversibilities. The second law contains a term of irreversibilities and in developing an expression directly relating to work and entropy generation we need to combine both laws.

Exergy Analysis

A second law analysis calculates the system performance based on exergy. Exergy always decreases, owing to thermodynamics irreversibility. Exergy analysis is the combination of the first and second law of thermodynamics and is defined as the maximum amount of work potential of a material or a form of energy in relation to the surrounding environment. Therefore, in an exergy analysis the losses represent the real losses of work. The principle irreversibilities in a process leading to these losses are due to dissipation (friction), heat transfer under a temperature difference and unrestricted expansion [4].

The exergy content of pure substances is generally given by equation

$$\psi = (h - h_0) - T_o(s - s_0) \tag{12}$$

where the terms h_o and s_o are the enthalpy and entropy values of fluid at the environmental temperature T_o, which ultimately forms the energy (heat) sink for all irreversible and reversible processes.

The availability loss in each component is calculated based on the following equation

$$\dot{W}_{act,u} = \sum_{out} \dot{m} b - \sum_{in} \dot{m} b - \sum Q_j \left(1 - \frac{T_o}{T_j}\right) + \dot{I}_{tot} \tag{13}$$

where $b = \left(h + \frac{v^2}{2} + gz - T_o s\right)$

$$\Delta\psi = \Delta b$$

On a unit –mass basis for one inlet and one outlet the above equation becomes

$$i_{cv} = \Delta\psi_{cv} = T_o \frac{\sigma_{cv}}{\dot{m}} = q_i\left(1 - \frac{T_o}{T_i}\right) - (b_2 - b_1) + W_{act} \tag{14}$$

In general the availability loss in each component is given by

$$\Delta\psi = \sum m_i \psi_i - \sum m_E \psi_E - Q\left(1 - \frac{T_o}{T_i}\right) - W \tag{15}$$

The first term on the right hand side is the sum of the exergy input. The second is the sum of the exergy output, while the third term is the energy of heat Q, which is transferred at constant temperature T. The exergy of heat is equal to the work obtain by the Carnot engine operating between T and T_o and is therefore equal to the maximum reversible work that can be obtained from heat energy Q. The last term is the mechanical work transfer to or from the system.

The availability balances for the four processes on an input/product basis are as follows

Compressor: $$\psi_1 + w_{in} = \psi_2 - \sum q_i\left(1-\frac{T_o}{T_i}\right) + i_c \quad (16)$$

Condenser: $$\psi_2 = \psi_3 - \sum q_i\left(1-\frac{T_o}{T_i}\right) + i_{cond} \quad (17)$$

Throttling valve: $$\psi_3 = \psi_4 + i_{valve} \quad (18)$$

Evaporator: $$\psi_4 = \psi_1 - \sum q_i\left(1-\frac{T_o}{T_i}\right) + i_{evap} \quad (19)$$

The total flow availability of moist air is referred to equation

$$\psi_{tot} = \left(c_{p,a} + \omega c_{p,v}\right)\left(T - T_o - T_o \ln\frac{T}{T_o}\right) + \left(1+\varpi\right)R_a T_o \ln\frac{P}{P_o} + R_a T_o\left[(1+\varpi)\ln\frac{1+\varpi_{oo}}{1+\varpi} + \varpi \ln\frac{\varpi}{\varpi_{oo}}\right] \quad (20)$$

Where the specific humidity is defined by equation

$$\omega \equiv \frac{m_v}{m_a} = \frac{N_v M_v}{N_a M_a} = \frac{M_v P_v}{M_a P_a} = 0.622\frac{y_v}{y_a} \text{ and } \varpi = 1.608\omega \quad (21)$$

The total energy of the air conditioning cycle is the sum of the exergy destruction in each component is referred to equation

$$\Delta\psi_T = \Delta\psi_{comp} + \Delta\psi_{cond} + \Delta\psi_{tv} + \Delta\psi_{evap} \quad (22)$$

Total exergy loss is referred to equation

$$i_{loss} = i_{comp} + i_{cond} + i_{tv} + i_{evap} \quad (23)$$

The second law efficiencies measure losses in availability during a process. The second law efficiency is evaluated as referring to equation

$$\text{Rational efficiency} = \varepsilon = \frac{-\Phi_{Q,evap}}{w_m}$$

$$\text{The Plant rational efficiency is} = \frac{i_{loss}}{w_m} \quad (24)$$

Methodology: Method Of Solution

Figure 3 shows the flow chart of this case study. The structures of this analysis begin with the system modelling, then recognizing the important parameters or components for further action of analysis. It is important to identify the variables for the system. In addition, there is also a need to assume the components and environment situation involved in order to carry out the analysis. On reaching this point, energy analysis on the system, including COP can be carried out. The next important stage is the exergy analysis that consists of finding the components availability, irreversibility and performance of the system. The data is then analysed in MATLAB and EXCEL programs. Finally when the data has been analysed the research draws a conclusion regarding air conditioning irreversibility processes.

Result And Discussion

The summary of availability accounting for different compressor entering pressures has been done in Pauziah (2002)[5]. When the work of the compressor increased, the irreversibility is shown to be increased and resulted in a drop in COP and a drop in the rational efficiency of the system. The results presented showed that the total amount of energy was constant (first law), while the potential for producing useful work was reduced forever (second law) has been proved. It can be concluded that the sum of the heat input at the evaporator and the work input to the compressor is equal to the sum of the heat rejected at the condenser and at the compressor. The temperature drop at the expansion valve occurs because the vapor has a higher internal energy than the liquid. Thus, some energy must be extracted from the liquid to drive the phase change. The load in the condenser was slightly higher than that in the evaporator. This was primarily due to superheating of the inlet vapor to the condenser [6]. This was a direct casual relationship between the irreversibility, COP, and rational efficiency. Sankey diagram as shown in Figure 4, highlighted the gradually reduction of the potential work in air conditioning unit. The illustration chosen is for entering pressure of 2 bar.

The pie-chart of Figure 5 shows every component efficiency defects when different input pressure was being applied to the system. For example, for 2 bar pressure, it was found that the evaporator contributed to 16% losses, compressor losses were 21% while throttling valve contributed about 13% and lastly condenser contributed about 20% losses. Therefore, it is very important to take into consideration the refrigerant pressure before it enters the compressor.

The results were then used to plot the graph in Figure 6. The graph shows that the irreversibility values will decrease when the pressure is increased for the total plant loss, evaporator and the throttling valve component. However, the reverse is true for the compressor due to difficulties to compress the refrigerant vapor when the pressure is increased at the entering point of the compressor. On the other hand, the condenser loss is constant due to the fact that the pressure has been stabilized at a fixed value. Figure 7 is the comparison graph between evaporator loss and the total loss in the system. It was found that the evaporator was one of the main components that can affect the total efficiency of the system. When the compressor work was increased, the evaporator loss increased too. This contributes to the higher value of the total loss of the system. Figure 8 and Figure 9 show how the drops of the temperature in the evaporator affect the total irreversibility of the plant. It was also found that the plant efficiency decreased about 52 % when the pressures entering the compressor were reduced from 2 bar to 1 bar. The same type of graph has also been plotted by C. Nikolaidis and D. Probert (1998) [7]. The effect of the temperature changes in the evaporator to the plant's irreversibility rate is determined in this studied. The greater the temperature difference between evaporator and the cold room, the higher was the irreversibility rate. However, the larger the temperature difference in the heat exchanger, the greater will be the exergy loss during heat transfer.

Figure 10 shows the relationship between the availability value for the moisture of the evaporator and the cold room. It seems that the availability decreases when the pressure is increased. Lastly Figure 11 shows the values of the COP and rational efficiency when the pressure is increased. It shows that the COP and the rational efficiency had a linear relationship to the pressure change. From all these results, it is clear that the compressor and evaporator need to be optimised because heat transfer area of heat exchangers in both components is inversely proportional to the pressure difference.

However it is seen that the smallest loss is attributed to the condenser. This component needs to be investigated further by not stabilizing its inlet pressure. In order to improve the performance of the cycle, special attention must be made to reduce the irreversibilities that exist in this component in the overall design. The exergy loss in the evaporator results mainly from the temperature difference between the environment and the evaporating refrigerant. Exergy destruction in the compressor is attributed to fluid throttling, irreversible fluid mixing, friction and irreversible heat transfer [8].

A thermo-economic optimisation method needs to be carried out in order to find the optimal pressure in operating the system and the optimal heat transfer over each component. The method of the thermo-economics optimisation can be used based on minimizing the sum of the capital cost plus the net present values of the annual operating costs over the expected lifetime of the plant, for each considered plant output.

Conclusion

The result of this study has given a clearer understanding of the operation of the home air conditioning system. It is found that there is a drawback in the conventional heat balance processes. This is due to the fact that many aspects have not been accounted for. For each component, the exergy input to the process was equal to the sum of exergy output and the irreversibility rate related to each process. The individual irreversibility values must be non-negative. Two computer programs comprised of MATLAB and EXCEL have been used to help in the prediction of the performance of the air conditioning unit.

The exergy flow in the condenser and in the evaporator is zero when the flow is passing through the environment temperature. In refrigeration system, the losses can be grouped into two primary categories, which are, the frictional losses referred to as ΔP losses, and heat transfer losses referred to as ΔT losses. ΔP loss actually occurs because the refrigerant mass flow rate through the compressor is larger than the expected refrigerant mass flow rate, thus leads to increase in ΔP loss [9]. The air conditioning loss increases, with increasing compressor work. The analysis showed that the performance of the system decreases with increasing compressor work or decreasing in entering pressure of the compressor. The ideal performance of the entropy compressor is for it to transfer the entropy load from the low temperature thermal reservoir to the high thermal reservoir without generating entropy as a result of dissipative effects. The isentropic efficiency of the compressor decreases as the entering pressure at the compressor is increased. This was because the average operating compression pressure ratio decreased as the compressor work decreased.

The refrigeration system can be thought of as an entropy pump which task is to remove entropy from the low temperature thermal reservoir (the cold room) and transfer it into the high temperature thermal reservoir (the ambient). The exergy loss in the evaporator results mainly from the temperature difference between the environment and the evaporating refrigerant. The loss of exergy in air-conditioning system is contributed mainly by the heat exchangers.

As this study mainly focuses on the components of the compressor and the evaporator, in future research it is important to consider the condenser as one of the main item that contributed to the loss of the system. The change in pressure at the entrance of the compressor and the change in temperature in the evaporator contributed significantly to the plant overall irreversibility. There is a considerable scope for optimizing the conditions imposed upon the compressor and evaporator. The technique has demonstrated that a change in any component variable in a plant component significantly influences other plant components, thus give a greater reduction in the irreversibility rate of the plant as a whole. An increase in the irreversibility of the two heat transfer processes requires a smaller temperature difference in each device. The thermodynamic losses in the heat exchanger can be reduced by methods such as the redesigning of the refrigerant circuitry, the changing of heat exchanger from cross flow configuration to a parallel flow configuration, the changing of the heat transfer fluids or by utilizing a refrigerant mixture with a temperature glide that closely matches the temperature glide on the external side of the heat exchangers [9].

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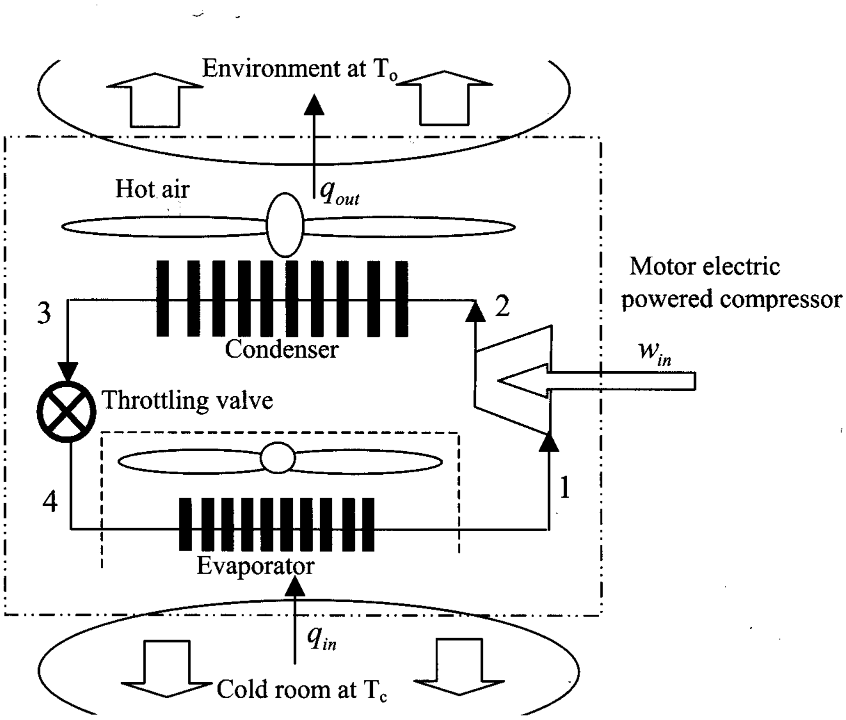


Figure 1 A schematic of the equipment for the considered home air conditioning system

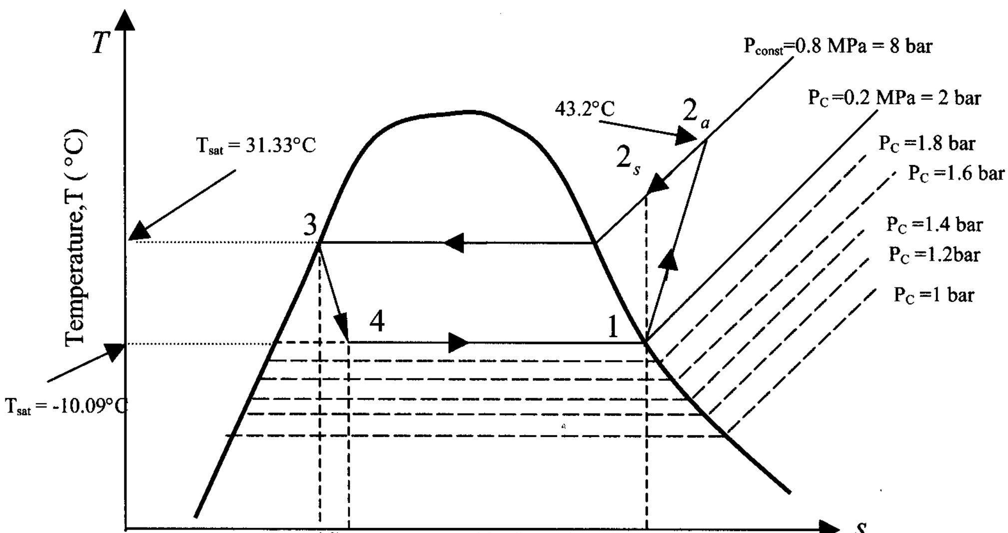


Figure 2 Schematic temperature versus entropy diagram for the considered air conditioning cycle

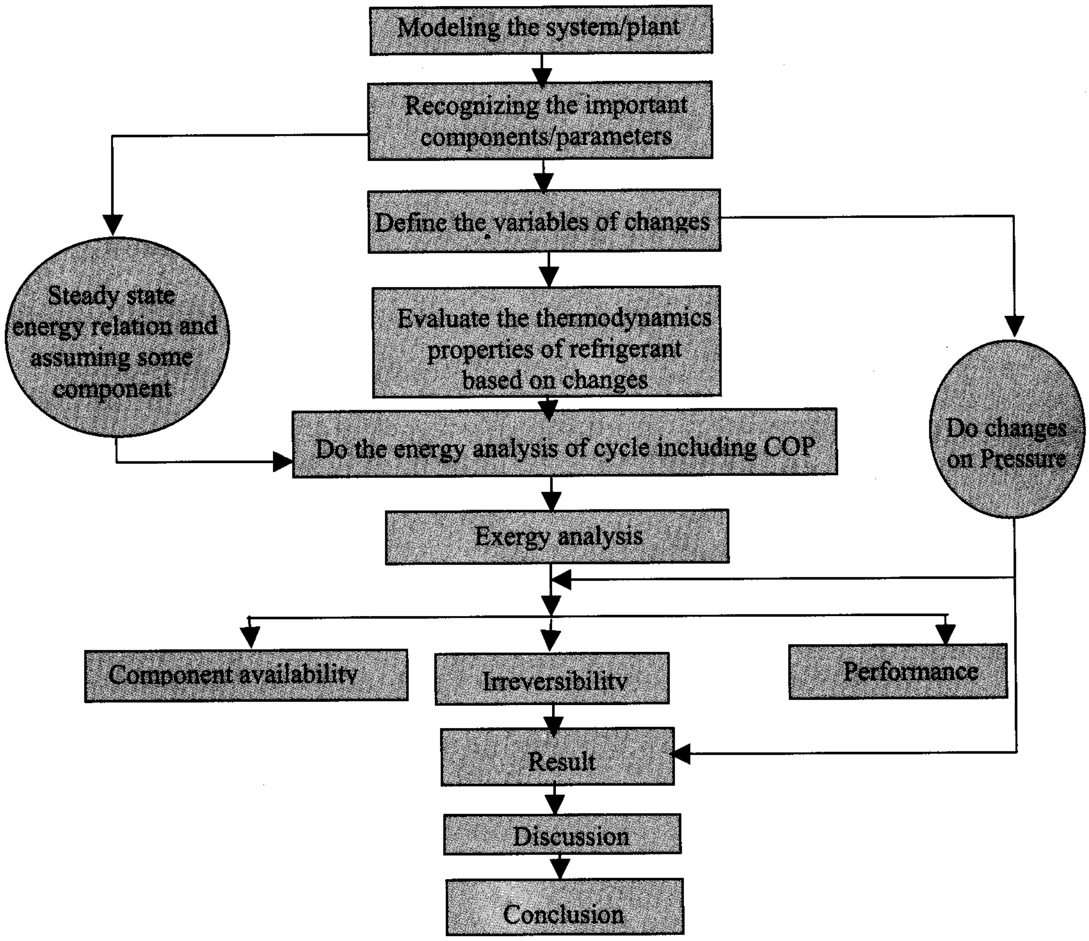


Figure 3 Flow chart for the method of solution

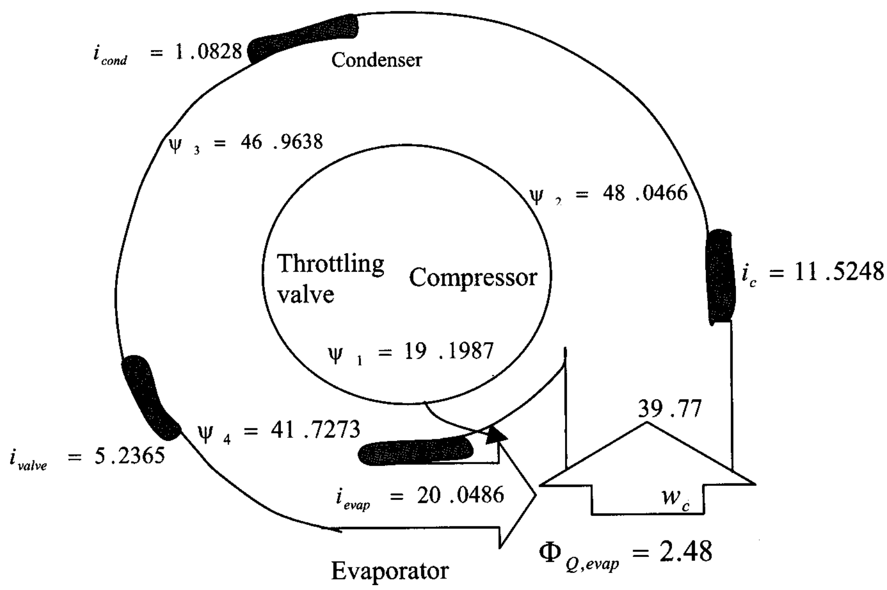
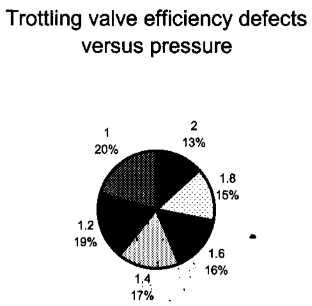
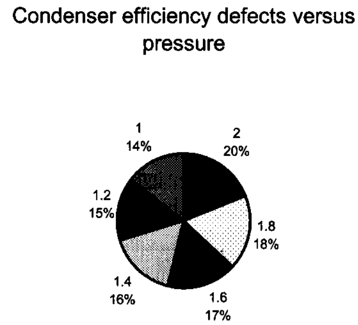


Figure 4 Availability flow diagram for a simple air conditioning cycle showing availability transfer and losses as calculated at pressure 2 bar

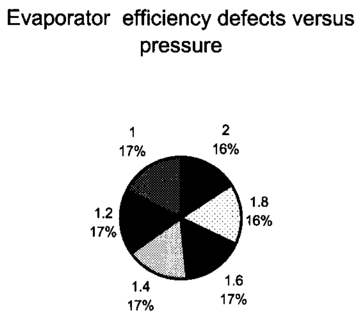
a) Efficiency defects for an Evaporator



b) Efficiency defects for a Compressor



c) Efficiency defects for a Throttling valve



d) Efficiency defects for a Condenser

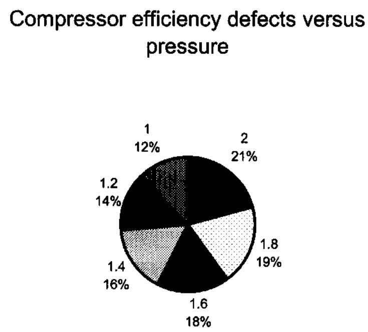


Figure 5 Efficiency defects of refrigeration components

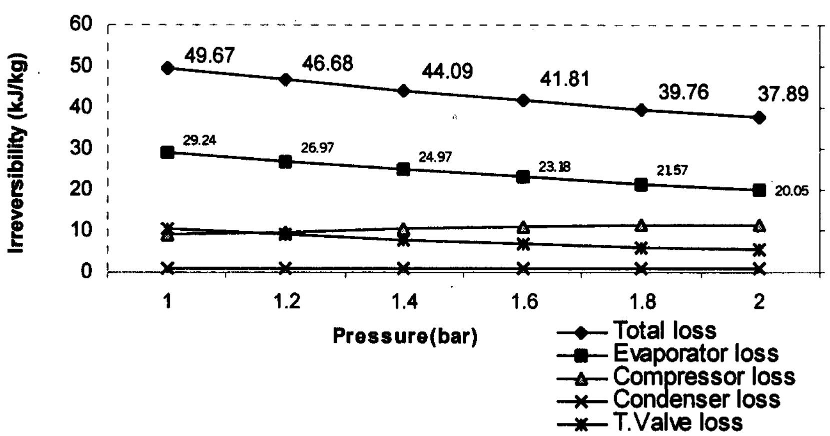


Figure 6 Plant and each of component irreversibilities for considered cycle

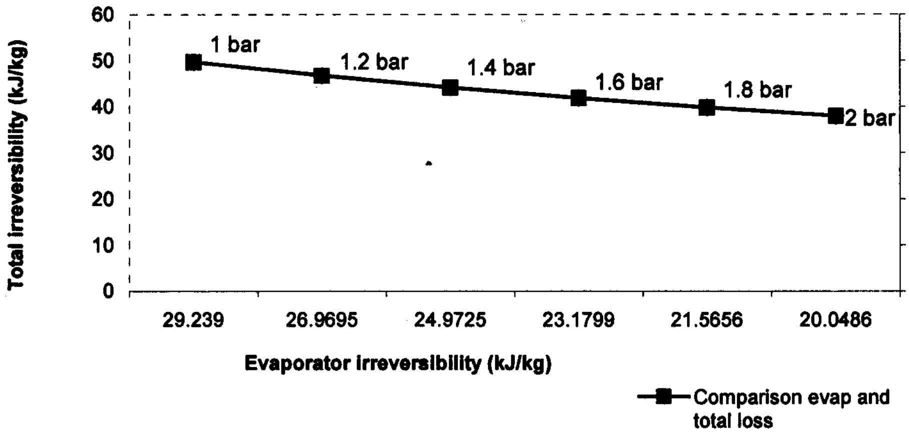


Figure 7 Plant irreversibility depends on evaporator loss

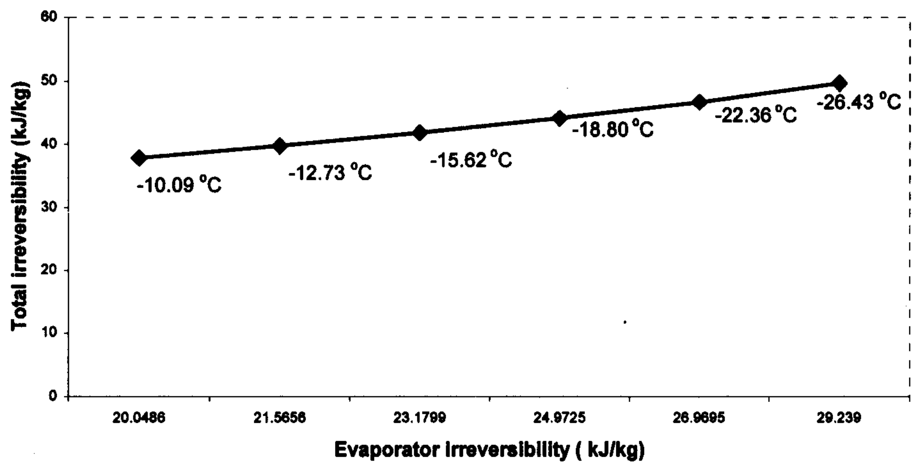


Figure 8 Temperature drop at evaporator affected the irreversibility

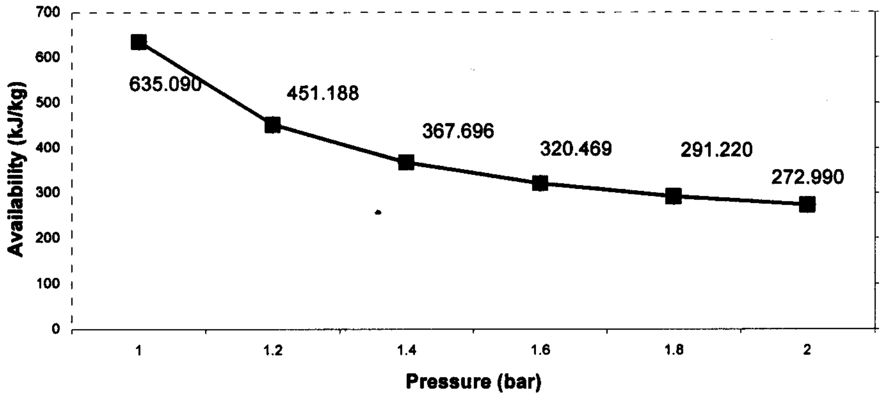


Figure 9 Plant rational efficiency compared to evaporator temperature

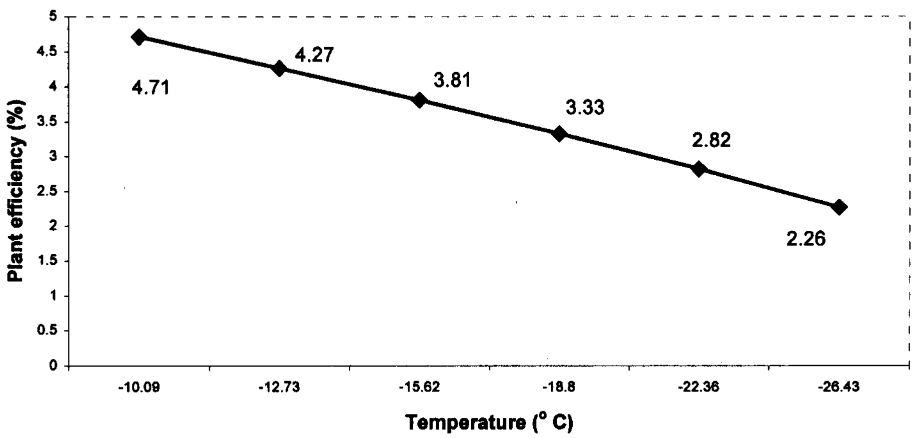


Figure 10 Moisture availability versus pressure

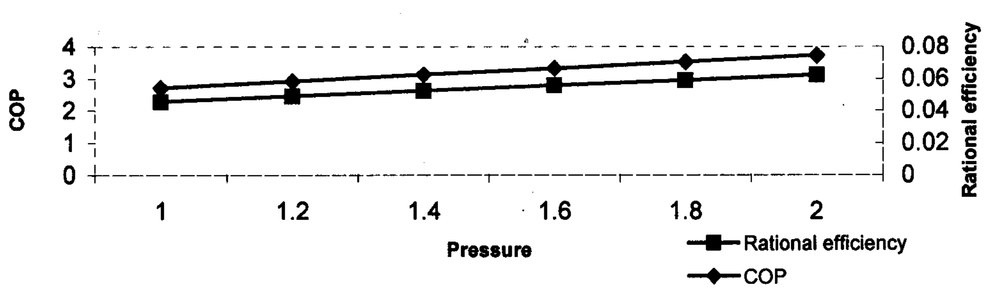


Figure 11 Rational efficiency and COP compared to Pressure

Appendix A

Notation

a	-	Dry air
b	-	Specific Darries function, $b = h + ke + pe - T_o s$ 3
C_p	-	Specific heat
COP	-	Coefficient of performance
g	-	Local acceleration of gravity
h	-	Specific enthalpy
I	-	Irreversibility
$\dot{I}, i$	-	Irreversibility per unit mass
M	-	Molar mass
m	-	Mass of the substance
$\dot{m}$	-	Mass flow rate
m_e	-	Exit mass
m_i	-	Inlet mass
N	-	Number of moles
N_2	-	Nitrogen gas
P	-	Pressure
P_i	-	Intermediate pressure
Q	-	Heat transfer
q	-	Heat transfer per unit mass
R	-	Specific gas constant
R_u	-	Universal gas constant
S	-	Entropy
s	-	Specific entropy
T	-	Temperature
U	-	Internal energy
v	-	Vapor
V	-	Velocity
W	-	Work interaction
w	-	Work interaction per unit mass
ω	-	Specific humidity
ϖ	-	Mole fraction ratio
x	-	Fraction vapor in liquid vapor mixture
y	-	Mole fraction in vapor phase
z	-	Height, Cartesian coordinate

Greek Symbol

η	-	Efficiency
$\dot{\sigma}$	-	Entropy production
ε	-	Second law efficiency
Δ	-	A finite increase in a property
$\Phi_{Q,j}$	-	Availability transfer due to heat transfer
Φ	-	Closed system availability
ϕ	-	Specific closed system availability
δ	-	An infinitesimal increase in a path function
ψ	-	Stream exergy per unit mass

Subscripts

act,a	-	Actual
c	-	Compressor
cond	-	Condenser
CV	-	Control volume
evap	-	Evaporator
f	-	fluid state
g	-	gas state
H	-	High temperature
in	-	Inlet
L	-	Cold Temperature
o	-	Standard environment state
oo	-	Dead state (unrestricted)
refrig	-	Refrigeration
tot	-	Total
tv	-	Throttling valve
x	-	Quality, mole fraction