UNIVERSITY OF GAZİANTEP
GRADUATE SCHOOL OF
NATURAL & APPLIED SCIENCES

THERMODYNAMIC ANALYSIS OF A CASCADE REFRIGERATION SYSTEM

M. Sc. THESIS
IN
MECHANICAL ENGINEERING

BY
HATEM ISMAEEL
AUGUST 2012

Thermodynamic Analysis of a Cascade Refrigeration System

M.Sc. Thesis

in

Mechanical Engineering

University of Gaziantep

Supervisor

Prof. Dr. M. Yaşar GÜNDOĞDU

by

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AUGUST 2012

REPUBLIC OF TURKEY
UNIVERSITY OF GAZİANTEP
GRADUATE SCHOOL OF
NATURAL & APPLIED SCIENCES
MECHANICAL ENGINEERING DEPARTMENT

Name of the thesis: Thermodynamic analysis of a cascade refrigeration system

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Exam date: 27.08.2012

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Hatem ISMAEEL

ABSTRACT

THERMODYNAMIC ANALYSIS OF A CASCADE REFRIGERATION SYSTEM

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M.Sc. in Mechanical Engineering

Supervisor: Prof. Dr. MEHMET YAŞAR GÜNDOĞDU

August 2012, 82 pages

In this study, energy, exergy, and cost analyses of cascade refrigeration system is investigated. Thermodynamic analysis is conducted for the Gaziantep city environment temperature for a fixed refrigerant couple. The cooling load, refrigerated space temperature and the temperatures during evaporation and condensations in all heat exchangers are considered to be constant. The operation and design parameters considered in this study include condenser fan velocity, evaporator fan velocity, isentropic efficiency of HTS compressor and isentropic efficiency of LTS compressor. Firstly, estimation of total cost including investment cost and operation cost at Gaziantep dead state by trade-off between the input exergy cost and capital (purchase) cost then determination of the effect of each parameter on total cost are conducted. Secondly determination of operation cost per year by means of the dead state temperature variation seasons gives a practical energy saving comparison basis. Results of the present thermodynamic analysis showed that; a minimum annual cost was found for the decision parameters such as LTS and HTS compressor efficiencies, air cooled condenser and evaporator fan velocities. The total annual cost of the cascade refrigeration system can be reduced by 30.86 percent in comparison to the medium range selected. The total annual operation cost of the system can be decreased by 11 percent in comparison with operation cost of initial design and the total annual cost can be decreased by 2.36 percent where the environment temperature variation taken into account. Finally the sum of annual cost saving about 33.22 percent according to the approach in this study.

Key words: Cascade Refrigeration System, Energy, Exergy, Cost Optimization, Thermoeconomic Analysis.

ÖZET

KASKAT BİR SOĞUTMA SİSTEMİNİN TERMODİNAMİK ANALİZİ

ISMAEEL Hatem

Yüksek Lisans Tezi, Mak. Müh. Bölümü

Tez Yöneticisi: Prof. Dr. M. YAŞAR GÜNDOĞDU

Ağustos 2012, 82 sayfa

Bu çalışmada; kaskat soğutma sisteminin enerji, ekserji ve maliyet analizleri araştırıldı. Termodinamik analiz, Gaziantep şehri ve değişmeyen bir soğutucu akışkan çifti için yapıldı. Soğutma yükü, soğutulan bölge sıcaklığı ve bütün ısı eşanjörleri içindeki buharlaşma ve yoğuşma esnasındaki sıcaklıklar sabit olarak kabul edildi. Bu çalışmada dikkate alınan işletim ve tasarım parametreleri; yoğuşturucunun fan hızını, buharlaştırıcının fan hızını, HTS ve LTS kompresörlerinin izentropik verimlerini içermektedir. Birinci olarak; giren ekserji maliyeti ve Gaziantep şehri şartlarındaki işletim maliyeti arasında ödünleşim yapılarak içeriğinde ilk yatırım ve işletim maliyetlerini bulunduran toplam maliyet hesaplaması ve akabinde her bir parametrenin toplam maliyete olan etkisinin belirlenmesi yapıldı. İkinci olarak; sezonluk dış ortam sıcaklığı değişiminin dikkate alınmasıyla elde edilen yıllık işletim maliyetinin belirlenmesi pratik bir enerji tasarrufu kıyaslama zemini teşkil eder. Mevcut termodinamik analizin sonuçları gösterdi ki; Eğer LTS ve HTS kompresör izentropik verimlilikleri, hava-soğutmalı yoğuşturucu ve buharlaştırıcı fan hızları gibi karar parametreleri optimize edilirse kaskat soğutma sisteminin yıllık maliyeti yüzde 30.86 oranında azaltılabilir. Eğer dış ortam sıcaklık değişimi dikkate alınır ise sistemin yıllık toplam işletim maliyeti yüzde 11 oranında ve yıllık toplam maliyeti yüzde 2.36 oranında azaltılabilir. Nihai olarak, bu çalışmada kullanılan yaklaşıma göre yıllık maliyet tasarrufunun tamamı yaklaşık yüzde 33.22 oranındadır.

Anahtar Kelimele: Kaskad Soğutma Sistemi, Enerji, Ekserji, Maliyet Optimizasyon, Termoekonomik analiz.

To my Parents, wife, daughter, son, sisters, brothers and friends

ACKNOWLEDGEMENTS

I would like to express highest appreciation my supervisor Prof. Dr. M. YAŞAR GÜNDOĞDU for giving me the opportunity to do my dissertation in the area of thermodynamic. Without his tireless interest and continuous guidance, the completion of this thesis would not have been possible.

I would like to thank; my mother, my wife, my sisters, brothers and others for their endless supports.

I would also like to thank the University of Salahaddin / Erbil to give me the opportunity to complete the master's study

Finally, I would like to thank to those who took part in completion of this thesis.

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NOMENCLATURE

ΔT Temperature difference between the saturation temperatures of the lower and higher temperature systems in the heat exchanger (K)

COP Coefficient of performance

T_c Condenser temperature (K)

T_e Evaporator temperature (K)

HTS High temperature side

LTS Low temperature side

IT Intermediate temperature (evaporator temperature for HTS)

CRS Cascade refrigeration system

c_p Constant pressure specific heat of (Kj kg^{-1} K^{-1})

E Energy (kJ)

g Gravitational acceleration (m s^{-2})

h Specific enthalpy of refrigerant (kJ kg^{-1})

K ratio of constant specific heats

$\dot{m}$ mass flow rate (kg s^{-1})

η_{is} Isentropic efficiency of the compressor

q Specific capacity (kJ kg^{-1})

$\dot{Q}$ Heat transfer rate (kW)

P Pressure (kPa)

s Specific entropy (kJ kg^{-1} K^{-1})

S Entropy (kJ K^{-1})

$\dot{S}_{gen}$ Total entropy generated (kW K^{-1})

T Temperature (K)

w Specific compressor capacity (kj kg^{-1})

$\dot{W}$ Power (kW)

V Velocity (m s^{-1})

C_{el} Unit cost of input exergy ($\$kW^{-1}\ h^{-1}$)

PO Period of operation per year

C Cost

C_{OP} Operation cost

C_{purch} Purchase cost

C_{total} Total cost

ε Second law efficiency (exergetic efficiency)

U Overall heat transfer coefficient ($Wm^{-2}\ K^{-1}$)

A Exchanger total heat transfer area on one side (m^2)

A_{fr} Exchanger minimum free-flow area (m2)

h_i Inner heat transfer coefficient ($Wm^{-2}\ K^{-1}$)

K Thermal conductivity ($Wm^{-1}\ K^{-1}$)

R Tube radius (m)

ρ Density (kg/m^3)

x Quality

G mass flow velocity ($kg\ m^{-2}s^{-1}$)

D Tube diameter (m)

D_h Hydraulic diameter

C_p Specific heat (kj/kg k)

v Specific volume ($kg\ m^{-3}$)

Subscripts

opt Optimum

H high temperature side or sink

L low temperature side or source

i Inner

o Outer

1 liquid

tp Two phase region

m Mean specific volume

Dimensionless

Re	Reynolds number	[GD/μ]
Nu	Nusselt number	[hD/k]
Pr	Prandtl number	[$c_p\ \mu/k$]
Bo	Boiling number	[$q/(G\ h)$]

CHAPTER 1

INTRODUCTION

1.1 General

There are few phases of modern living untouched by air conditioning and refrigeration. Manufacturing processes, business operation, storage, and shipping are almost always carried out now under controlled temperature conditions. A single vapour refrigeration cycle usually cannot be used to achieve the large temperature difference. To solve this problem, a cascade refrigeration system must be used. There are several variations of the basic vapour refrigeration cycle. A cascade system is used when the temperature variation between evaporator and condenser is too large. The multistage cycle is used to reduce the required compressor power input [1]. Cascade systems are often used in industrial processes where objects must be cooled to temperatures below (-46°C) [2]. If there is an evaporator temperature need to below -18℃ the cascade refrigeration is required [3]. The cascade refrigeration system is typically utilized at temperature zone of (-70 to 0)℃ [4]. The cascade refrigeration system is helpful in industrial applications while the temperatures below -40℃ are required [5]. As long as, cascade refrigeration system are useful to gain big difference between the heat sink and heat source. Those systems are valid for temperatures domain (-70 to 100) °C [6]. The difficulty of finding one refrigerant that is ideally suited to both the high and low temperatures and pressure ranges of a system operating with a large temperature available lift guide to the conception of cascade refrigeration system [7]. A cascade system is composed of two independent refrigeration cycles, one of them at a HTS and the other one at LTS. The condenser in the LTS is cooled by the evaporator in the HTS. The classic cascade system predominating depends on R13 and R22 as working fluids. Normally use R13 in cascade system at the lower stage end and R22 for the high stage. Consequent to the intensive environmental effect of these two working fluids whilst, various natural refrigerants, e.g. carbon dioxide, ammonia and hydrocarbons and others, have lately been used in cascade refrigeration systems [8]. There are five substances that we can call “natural refrigerants”: Air, Water, Ammonia, Hydrocarbons and Carbon dioxide [9]. Cascade staging combines different individual refrigeration systems that use various refrigerants and have closed heat exchangers to achieve low operating temperatures and reasonable condensing pressures [6]. Low temperatures for air conditioning are around 0°C; for industrial refrigeration, -35 to -5°C ; and for cryogenics, approaching 0 K. Applications such as freeze-drying, as well as the pharmaceutical, chemical, and petroleum industries, use refrigeration in the temperature range designated low. The -50 to -100°C temperature range is treated separately because design and construction considerations for systems that operate in this range differ from those encountered in the two fields bracketing it, namely industrial refrigeration and cryogenics. Designers and builders of cryogenic facilities are rarely active in the low-temperature refrigeration field [10].

Cryogenics is related also with low temperatures, and defined the temperature to be lower than -100°C [2]. The applications of cryogenic engineering include various chemical processes, separation and liquefaction of gases, food freezing, medical procedures such as cryogenic surgery, and various chemical processes [11, 12]. The guidelines of cascade refrigeration system at low temperature application is can be seen in the study which presented by Stegmann [13]. A cascade refrigeration system is used in industrial applications such as sedimentation the hardening of special alloy steel, liquefaction of petroleum vapor and natural gas. The most important advantages of these cascade systems are that refrigerants can be chosen with the appropriate properties, avoiding large dimensions for the system components. Refrigerants used in each stage can be different and are selected for optimum performance at the given evaporator and condenser temperatures [10]. The major advantage of a cascade system is that different refrigerants, equipment, and oils can be used for the higher and the lower systems. The disadvantages of a cascade system are higher initial cost and a more complicated system than that for a single-stage system [14].

The reasons and other advantages for using a multistage vapour compression system instead of a single-stage system are as follows:

• Improved energy efficiency at the intermediate pressure and de-superheating of the discharge gas from the low-stage compressor before it enters the high-stage compressor.

• Improved energy efficiency because the two-stage compressors are more efficient operating against discharge to suction pressure ratios that are lower than for a single-stage compressor.

• Avoidance of high discharge temperatures typical of single-stage compression. This is important in reciprocating compressors but of less concern with oil-injected screw compressors.

• Possibility of a lower flow rate of liquid refrigerant to the evaporator because the liquid is at the saturation temperature of the intermediate pressure rather than the condensing pressure, as is true of single-stage operation. And smaller size of suction line from the low-temperature evaporator [10].

1.2 Thesis Structure

In this research we analyze a thermodynamic analysis by means of energy, exergy, and cost accounting. The soft program which used for this aim is EES (Engineering Equation Solver). The outline of the study with respect to chapters is as follows:

Chapter 1 gives general information about cascade refrigeration system.

Chapter 2 presents a detailed literature survey about cascade refrigeration system. Previous work on energy and exergy analyses and the current status of thermoeconomic and operation cost analyses with variable environment temperature are provided.

In Chapter 3, general formulations of thermodynamic analysis of energy, exergy and general principles, terminology, and formulation of cost analysis are presented. The formulations given in this chapter are applicable to cascade refrigeration system.

In Chapter 4, programing details of analysis is performed using energy, exergy and cost analysis. The input data, system description and cost analysis optimization flowchart are mentioned.

In Chapter 5, results and discussion are performed. Cost data is obtained for initial design. The investment capital cost is obtained and operation cost, then total annual cost which is the minimum value depends on used ranges of decision variables and optimization of them. Also in this study, the modification of operation cost is done.

In Chapter 6, the major conclusions from this research are drawn. The results of the energy, exergy and cost analyses can be used with other couples of refrigerants, also can be used to estimate the investment cost, operation cost and total annual cost for any new system.

CHAPTER 2

LITERATURE SURVEY

2.1 Introduction

A cascade refrigeration system has a two separate refrigeration cycles: a lower system that can better maintain lower evaporating temperatures and a higher system that performs better at higher evaporating temperatures. These two stages are connected by a cascade condenser in which the condenser of the lower system becomes the evaporator of the higher system as the higher system's evaporator takes on the heat released from the lower system's condenser. By study in literature survey we can found a large numbers of applications on cascade refrigeration systems, also the modifications available in difference way. After searching in the literature it can be observed multi-application on the concept of cascade refrigeration system, such as effect of refrigerant couples in cascade refrigeration system, cost analysis or thermoeconomic analysis focused studies, energy analysis, exergy analysis and effect of cooling load and refrigerant mass flow rates on COPs, effect of T_E,T_C and ΔT variations on COPs and deviation of actual cycles from ideal one on COP irreversibility and other critical parameters. Generally it can be classify the publications which related with cascade system to some ways and clarify the effects of design parameters in cascade refrigeration system applications as show in following:

2.2 Multi refrigerant couples used with cascade refrigeration system

These references [16, 17, 18 and 19] are studied on two stages cascade refrigeration system by used multi refrigerant couples. Kilicarslan and Hosoz [16] studied the energy and exergy destruction by use different refrigerant couples. In this study the effect of variable parameters T_e, T_c, ΔT and the polytropic efficiency for the all refrigerant couples on COP and irreversibility. Refrigeration load, refrigerated space temperature and the environment temperature are considered unchanged, while the degrees of condenser subcooling and evaporator superheat are taken constant for all types of refrigerants couples. By varied in a range for each variable parameter, COP of the cascade refrigeration system increases and the irreversibility decreases with increasing evaporator temperature and polytropic efficiency for all studied refrigerant couples. The irreversibility increases and the COP of the cascade refrigeration system decreases by condenser temperature and ΔT increasing. [17 and 18] investigated on these refrigerant couples R22–R23, R12–R23, R717–R23, R22–R23 and R12–R23 and R717–R23 which used in cascade refrigeration system. Gupta and Parasad [17] investigated the effect of subcooling, superheating and the ΔT on the COP to obtain the best refrigerant couple among them. The optimum intermediate temperature of a cascade refrigeration system is obtained by Agrawal [18]. The thermal design of heat exchangers for two refrigerants couples in the cascade refrigeration system is numerically studied by Parekh and Tailor [19]. The types of refrigerants couples are R410A-R23 and R404A-R508B. The effect of condensing and evaporating temperatures for both systems on condenser heat transfer, evaporator heat transfer and cascade condenser heat transfer areas determined and compared. It has been observed by use the refrigerant couple R410A-R23 the cascade refrigeration system need to less areas of heat exchangers and cheaper compared with R404A-R508B. It's clear from this study to observe the varied parameters considered are condensing and evaporating temperatures and the others are fixed during the analysis.

2.3 Cascade refrigeration system used for additional heating objective

The references [20 to 26] studied on cascade refrigeration system for refrigeration and heating together. Bhattacharyya et al. [20] investigated an optimization of both refrigeration and heating system by used CO_2-C_3H_8 as a refrigerant couple. The CO_2 used in HTS and the C_3H_8 cycle used in LTS is used for low temperature (0 to -30 °C) with used internal heat exchanger in both cycles. It can operated this system in the same time between refrigerating space temperature of -40°C and a heating output temperature of about 120°C. It has been observed the COP_{system} remains unchanged with the effectiveness (for internal heat exchanger) of the LTS and increases with increase in effectiveness of the HTS. Bhattacharyya et al. [21] presented an analytical study on heating and refrigeration together incorporating both external and internal irreversibilities. The variables parameters in this system have been non-dimensionalised. The variation only in heat source and between 303 K and 353 K while the heat sink consider constant at 233 K. Souvik Bhattacharyya et al. [22] studied theoretically thermodynamic analysis of cascaded system, by use refrigerant N_2O in LTS and CO_2 in HTS, using internal heat exchangers are used and analysed for simultaneous cooling and heating system. It can be seen the effects of gas cooler, evaporator and cascade condenser on COP_{system} are similar, and there is a similar in thermophysical properties of working fluids. The COP_{sys} is not effected by used internal heat exchanger in LTS and HTS. Kaushik at el. [23] using the lagrangian multipliers method for cooling and heating in the same time. The study focuses on parameters effect on COP and listed in tables. It's showed the input power decreases by increase the inlet temperature of the external side. The temperature of refrigerant in HTS decreases and for refrigerant in LTS increases by any increase in inlet temperature of external sink fluid, the COP of the system decreases, whereas the temperature of the refrigerant on the LTS remains constant. The effect of internal irreversibility is more obvious as compared to the external irreversibility on the cooling/heating performance of the cascaded refrigeration/heat pump system.

Gupta [24,25] studied on cascade refrigeration-heat pump simultaneously by used difference couples with consider difference parameters and the results also differ in each investigation. Gupta [24] investigated an optimization cascade refrigeration and heating system in terms of COP_{system} and total operating costs by using R23 for the refrigeration system and R12 for the heat pump. The results showed COP ranged was between 2.5 and 7.4. The variable parameters used in the analysis are superheating, subcooling, exergy destruction, overlap temperature and ambient temperature, and water and electricity costs are incorporated therein. Gupta [25] investigated on cascade refrigeration-heat pump system and numerically for preliminary design of the system optimized. The refrigerant R-22 is used for heating and R-13 for refrigeration. The evaporator temperature used between (-60 to -30) °C and environment temperature between (10 to 60)°C. The results show the COP_{system} to be (2.8 to 7.4). Srinivasa Murthy and Krishna Murthy [26] performed experimentally R11-R12 couple for cooling and heating effect simultaneously. It has been experimentally found R11 condensing temperatures in the range of 65-95℃ could be achived corresponding to which the R12 evaporating temperatures varied from -2 to 8. It has been observed that COPs and second law efficiencies ranged between 1.2–1.7 and 10–15%, respectively.

2.4 Use of alternative refrigerant mixtures or blends (azetropic and zeotropic) in cascade refrigeration system

The researchers [27, 28, 29, 30, 31, 32 and 33] used alternative refrigerant mixtures in cascade refrigeration system. Niu and Zhang [27] studied experimentally on mixture R744 and R290 (71/29, mole fraction). It has been considered in this study two types of refrigerants, by use of R290 in HTS and (R744/R290) in LTS compared with R13. The mixture R744 and R290 has a good alternative refrigerant for R13 at evaporator temperature higher than 201K. Parekh and Tailor [28] investigated on azetropic mixture R507A composed of R125/R143a (50%/50%) R23. R507A is azeotropic mixture composed of HFC refrigerants R125/R143a (50%/50% wt.) and R23 used as replacement to R13 in low temperature applications. The variable parameters used in this research were evaporating, condensing, superheating and subcooling temperatures in both HTS and LTS, and temperature difference in cascade condenser. Computational model has been developed in engineering equation solver to evaluate the performance of the system. Nicola et al. [29] consideration R44 blend with R170, R290, R1150, R1270 and RE170 used as a working fluid in LTS while R717 as a working fluid in HTS in the analysis of cascade refrigeration system. It has been observed the mixture of carbon dioxide or R44 blend can be used in the applications when the temperatures below its triple point (216.59 K). Santis [30] developed software for the analysis by using binary interaction parameters data concluded experimentally from the literature, constructing a model for predicting the R744 blend’s behaviour at temperatures below 216.59 K in three chemical steps. It is observed generality of the cascade systems studied, the variation in COP within range of (085-0.93), and it was lower for blends containing CO_2 than for the pure fluids here consider. A higher of content of CO_2, the lower COP, except for the mixtures of R1150 in “no HX” configuration, where the COP is independent of the fluid’s composition. The performance difference between the system working with blends and pure fluids. The difference in COP between pure fluid and blend is up to the maximum for propane (ΔCOP=-14%) and minimum for ethylene ((ΔCOP=-2%). Comparing these results with those obtained previously by the researcher in others study with the same model for R744 HFCs systems Nicola et al. [31] reveals very similar COP values and trends, both in composition and intermediate temperature. Its noted the COP depends on intermediate temperature for all used refrigerants. Without heat exchanger the maximum COP. It is possible to note that for all refrigerants, the COP clearly depends on the intermediate temperature T_{int}. For all systems with “no HX”, the maximum COP was observed for temperatures between 240 and 260 K. These values are close to T = 258 K. For the systems with “no HX”, the maximum COP was obtained for RE170, and with heat exchanger, the maximum COP is observed for R170 and R1150 only. While for other fluids, the COP increased gradually with increase in intermediate temperature.

Another study on refrigerant blend used in cascade refrigeration system investigated by Gong et al. [32]. The blends include two binary azetropic mixture (R170 with R23, R170 with R116), and a ternary azeotropic mixture (R170 with R23 & R116) and used as working fluid in LTS while R404A was used as working fluid in HTS. The study done ate same conditions. The condenser temperatures to be constant, while the evaporator temperature to be varied. The performance of R508B and three azeotropic R170 mixtures were evaluated at same conditions. COP, $\dot{Q}_L$, compression ratio and discharge temperature of LTS using these different blends were evaluated at different evaporating and condensing temperatures. In addition to that the performance of R508B refrigerant was also evaluated at same conditions for comparison, while R404A was used as working fluid in HTS. The results showed the R170 mixtures has quite similar COP values to that of R508B, higher cooling capacities than that of R508B, low compression ratio than R508B and the (R170 with R23) has highest discharge temperature, which is about 20 K higher than that of R508B; while the (R170 with R116) mixture has the lowest discharge temperature, which is about 10 K lower than that of R508B. Finally the R170+R116 mixture may be the best candidate for substitute to R503 among the three R170 mixtures by the comprehensive the consideration.

Agnew and Ameli [33] compared the traditional refrigerant couple R12–R23 refrigerant couple R717–R508b in a three stage cascade refrigeration system using the finite time temperature approach, and found that refrigerant

couple R717–R508b shows a better performance compared with the couple R12–R23, The refrigeration system examined employed two stages on the high-pressure side and a single stage on the low-pressure side. It can be seen that the R717–R508b combination requires the minimum flow rates on both the HTS and LTS but a larger compressor is required on the HTS. This pair also exhibits the largest refrigeration efficiency. The higher volumetric efficiency compare with R12. The discharge temperatures of compressor in case R508b are significantly lower than that of R13 which equates to higher efficiency, longer compressor life and better lubricant stability. The mass flow rate required rate for R508b is less than for R13. The data presented graphically in the results will be a useful aid for design engineers to produce optimum operating conditions.

2.5 Use of CO_2/NH_3 couple

The references [34-44] investigated on CO_2/NH_3 couple, as use CO_2 in LTS and NH_3 in HTS in different situations. Yabusita and Kitaura [34] used the refrigerant couple CO_2/NH_3 as working fluids in a cascade refrigeration system. The design parameters are evaporator, condenser and cascade condenser temperatures. Cooling capacity, evaporator temperature, cascade condenser temperature and condenser temperature are 175 KW, -54°C, 5°C and 35°C respectively. Power input to each compressor 90 KW for HTS and LTS. This study focused on optimal condensing temperatures under same evaporating temperature with condenser temperature variation to evaluate maximum COP. Lee et al. [35] employs the same design parameters and refrigerant couple which used by Yabusita and Kitaura [34] to analyze the cascade refrigeration system. But here the isentropic and volumetric efficiencies considered as a function of compressor pressure ratio. The irreversibility of each component, to obtain the contribution of each component to that COP of the system is presented. The results compared with evaluated results by Yabusita and Kitaura and two correlations to determine the optimal condensing temperature of CO_2/NH_3 cascade refrigeration system are developed and they are suitable to use in the design case. Bingming et al. [36] investigated the performance of the cascade refrigeration system with NH_3/CO_2 and compared with that of cascade refrigeration system use NH_3 in both HTS and LTS and single-stage NH_3 system with or without economizer. The effect of cascade condenser, evaporator and condenser temperatures with superheat degree presented. This study done experimentally under laboratory environment. The effect of isentropic efficiencies of the compressors, condensing temperature, evaporator temperature and temperature difference on COP and second law efficiency studied by Dopazo et al. [37] by use the refrigerant couple CO_2/NH_3 in cascade refrigeration system. The concluded results, two correlations established, the optimum values of CO_2 condenser temperature and COP as a function of evaporator temperature, condenser temperature and temperature difference. Dopazo et al. [38] also investigated on a cascade refrigeration system prototype in experimental study by use $CO_{2/}NH_3$ as working fluids, for freezing application. The real conditions are used in the experimental evaluation with design parameters. Different tests were examined with select four constant evaporating temperatures at (-35, -40, -45 and -50) in LTS. At each evaporator temperature computed the condensing temperature for LTS (which used CO_2 as a working fluid) was varied from (-17.5 to -7.5)℃ and evaluated the optimum value of LTS experimentally. The results of experimental study are comparing with two stages system with using economiser and with intercooling by using refrigerant NH_3 as a working fluid. It has been showed by use 9 KW as a nominal cooling capacity at evaporating temperature -50℃, the cooling capacity measured experimentally 9.45 KW, meaning 5% above cooling capacity designed of plate freezer. In this study also the effect of condensing and evaporating temperatures of LTS on COP_{system} experimentally evaluated.

Samer Sawalha et al. [39] developed of a laboratory test rig on cascade refrigeration system by using the refrigerant couple CO_2/NH_3 as working fluids.
As show from the Figure 2.3 the CO_2 side or LTS used for cold food and frozen food with difference evaporating temperature. The model is coded in (EES) program to evaluated the performance in each component. It's observed the COP_{system} about 2.1, COP_{LTS} is about 4 while the COP_{HTS} is about 2.6. Also it has been developed in

model mathematically to simulate the system performance and facilitate detailed system design calculations. The results of the tests refer to possible identify strong and weak points of the system and give the recommendations for future designs and future study. Another study on cascade refrigeration system used the refrigerant couple CO_2/NH_3 as working fluids is done by Rezayan and Behbahanina [40] with consider the cost or thermoeconomic optimization in the analysis. The variables parameters were evaporating and condensing temperatures of LTS which used the CO_2 as a working fluid, condensing temperature of HTS and temperature difference in cascade condenser with environment temperature, cooling capacity and refrigerated space temperature are constants during the optimization procedure. The results showed the optimum values of each variable parameter, the minimum and maximum exergy destruction values for optimize case are 5.2% and 33.49%, and the total annual cost reduced by 9.34% compare the base case assumed in the study. Prashant Vyas et al. [41] used the eight variable parameters and their effects on COP, refrigeration effect and second law efficiency in thermodynamic analysis of cascade refrigeration system by use the refrigerant couple R744/R717 as working fluids. The observed results from this study to obtain highest refrigerating effect of the system, the system should operate at low condenser temperature, high evaporator temperature, low temperature difference in cascade condenser, low side condenser temperature, with lower degree of subcooling and superheating in high side higher degree of subcooling and superheating in low side. To evaluate maximum second law efficiency the system should work with low condenser temperature, high evaporator temperature, low ΔT, high low side condenser temperature, lower degree of superheating in both sides and higher degree of subcooling in both sides. Also analysis for maximum system COP, system should operate at low condenser temperature, with high evaporator temperature, low ΔT, high low side condenser temperature, lower degree of superheating in both sides and higher degree of subcooling in both sides. Getu and Bansal [44] investigated on cascade refrigeration system in thermodynamic analysis to obtain optimize variable parameters. It can be seen an increase in superheat the mass flow ratios of refrigerant increase but COP will be reduced, increase in subcooling the COP and mass flow ratio will be increased.. The mass flow ratios COP reduced by increase in temperature difference in cascade condenser. An increase in condensing temperature resulted in a decrease in COP and an increase in refrigerant mass flow ratios. An increase in temperature difference in cascade condenser reduced both COP and mass flow ratios.

There are another studies in the literature used also CO_2 but with another refrigerant instead ammonia. Nicola et al. [42] studied an experimental investigation and optimization of a cascade cycle with using CO_2 in the LTS and R404a in the HTS. Refrigerating units operating with cascade cycles are considered a convenient solution in terms of safety and cost for applications demanding very low temperatures. The aim of this work was to quantify the performance of a cascade refrigeration cycle using CO_2 in the LTS and R404a in the HTS cycle. The setting values of the temperature were -20 °C, -24°C, -25°C, -30°C, -40°C varying also the thermal load from 1.37 kW to 3.35 kW. The system charged by 2.00 kg of R744 in the LTS and 4.00 kg of R404A in the HTS. The results observed from this work, when the system starts, normally the temperature inside the chamber was higher than the checked one by a controller. Only when the temperature reaches this value, the thermal load started in order to balance the cooling power of the low temperature cycle and to stabilize the average value of the temperature. Problems relating to the achievement of high working and transient pressures obliged us to close partially the compressor's intake valve in LTS. Progressively reducing the evaporating pressure, it gradually reopened the valve. The average deviations for each response are very low and below the 0.8 % for the COP and below the 0.1 % for the $P_{absorbed}$. Starting from these two surfaces is possible to start the optimization procedure, using the evolutionary algorithm NSGA-II. Based on a series of experimental test performed on a cascade cycle and optimization procedure is been proposed. The optimization procedure was based on the identification of two different response surface methodologies, required to describe the relationship among the variables, relative to the cascade cycle, and the coefficient of performance and the electrical energy absorbed by the compressors. From these two surfaces, using NSGA-II algorithm, a multi-objective optimization is performed. From the optimization is possible to identify configurations of the variables that guarantee the minimization of the energy absorbed and the maximization of the coefficient of performance of the cycle. Bhattacharyya et al. [43] studied on cascade refrigeration system by use the CO_2 and propane in the analysis, and used two internal heat exchangers for the system. Evaluate of the optimum operation temperatures have been done. The cascade refrigeration system also used with cogeneration and absorption system [45, 46]. Petrenko et al. [45] studied on cogeneration sysem and Seara et al. [46] analyse a cascade refrigeration system with a compression system at the low temperature stage and an absorption system at the high temperature stage to generate cooling at low temperatures.

2.6 Cascade refrigeration system with used water loop between HTS and LTS

Kilicarslan [47] studied a cascade refrigeration system and single stage by used the refrigerant R134a as a working fluid in both systems, and comparison between them by used the compression work, refrigerant mass flow rate, discharge pressure and COP are parameters. In this study used the water circulation between the LTS condenser and HTS evaporator.

2.7 Cascade refrigeration system used for natural gas liquefaction in three stages

Kanoglu [14] studied exergy analysis of a three stage cascade refrigeration cycle used for natural gas liquefaction and derived an expression for the minimum work. The minimum work for a typical natural gas inlet and exit state is determined to be 456.8 kJ kg^{-1} of liquefied natural gas, which corresponds to COP of 1.8 by used Carnot refrigerator. Using a typical actual work input value; the exergetic efficiency of the three stages cascade refrigeration cycle is determined to be 38.5% indicating a great potential for improvements.

2.8 The cascade heat pump used with hydro geothermal system

The others study related with cascade refrigeration system available in literature but not related with it essentially . Geothermal energy is primarily used for heating buildings, greenhouses, fish-farms, in balneology and in industry. Goricanec et al. [49] studied (hydrogeothermal cascade heat pump economic and ecologic appropriacy) by consider economy of exploiting heat from low-temperature geothermal sources for high-temperature heating of buildings using a two stage heat pump. It can be observed from this study the COP of a heat pump is between 3.5 and 4.4, at an outlet temperature of geothermal water 10°C. An economic analysis of the investment was also carried out.

2.9 Cascade refrigeration system with mixed two stages

Nikolaidis and Probert [51] investigated behaviour of two-stage compression-cycle, with use flash intercooling instead of cascade condenser by using refrigerant R22 as a working fluid. Condition of this study for storing frozen meat considered requiring a temperature about -30℃. The values of evaporation and condensation temperatures variation are evaluated by computer program. The effects of condenser and evaporator temperature variation on total exergy of the system were evaluated. Increase in the temperature difference between environment and condenser or cold space room and evaporator give highest exergy destruction. Any decrease in exergy destruction for condenser gives roughly 2.4 times greater than decreases in the exergy destruction of whole plant and any decrease in exergy destruction for evaporator gives 2.87 times greater than decreases in the exergy destruction of whole plant. The greater the temperature difference between either the condenser and the environment, or the evaporator and the cold room, the higher the irreversibility rate. Any reduction in the irreversibility rate of the condenser gives approximately 2.40 times greater reduction in the irreversibility rate for the whole plant, and any reduction in the evaporator's irreversibility rate gives a 2.87 times greater mean reduction in the irreversibility rate of the whole plant. Prasad [52] used single fluid in cascade refrigeration system by means of flash intercooling. It has been evaluated an optimum intermediate pressure for refrigerant R12 depending on coefficient of performance maximizing.

Agrawal and Bhattacharyya [55], studies on a two-stage cascade refrigeration system with flash intercooling by used the refrigerant CO_2 as a working fluid. Results showed that; the flash intercooling technique is not economical with use

CO_2 refrigerant as a working fluid, unlike NH_3 refrigerant used as working fluid. COP of the cascade system is lower than that; of the single cycle for a given gas cooler and evaporator temperature. It cannot evaluate the intermediate pressure as well. Any increase in COP occurs as inter-stage pressure decreases from the classical estimate of geometric mean of gas cooler and evaporator pressure. Thermodynamic analysis of two stage and mechanical subcooling is done by Zubair et al. [56]. The analysis is tested on each component of the system to evaluate the effect of each one on exergy destruction. It's observed that; the more losses due to a low compressor efficiency, exergy destruction of expansion valves and condenser also significant. Thermodynamic analyses and optimization studies of two-stage transcritical N_2O and CO_2 cycles investigated by Agrawal et al. [57] incorporating compressor intercooling, are presented based on cycle simulation employing simultaneous optimization of intercooler pressure and gas cooler pressure. Further, performance comparisons with the basic single-stage cycles are also presented. The N_2O cycle exhibits higher cooling COP, lower optimum gas cooler pressure and discharge temperature and higher second law efficiency as compared to an equivalent CO_2 cycle. However, two-stage compression with intercooling yields lesser COP improvement for N_2O compared to CO_2. Based on the cycle simulations, correlations of optimum gas cooler pressure and inter-stage pressure in terms of gas cooler exit temperature and evaporator temperature are obtained.

2.10 Other different researches

Jameel et al. [58] investigated the two stage vapour compression refrigeration system. It is observed that maximum COP_{system} occurs at relatively large fraction of the total area is allocated to the condenser. The optimum area of heat exchanger available at minimum exergy destruction due to heat transfer rates. Mafi et al. [59] studied on cascade refrigeration system by mean of exergy analysis for the system components and using propene in HTS and ethane in LTS. The development of used equations of the analysis is given. The overall second law efficiency of the system determined to be 30.88%. Bilal and Zubair [60] investigated a cascade refrigeration system by use the combination of different refrigerant with dedicated mechanical subcooling. Either the same or different refrigerant use for both cycles. The system can have either the same refrigerant or different refrigerants flowing through the two cycles. Complete design and retrofitting cases were considered using six refrigerants. It has seen the subcooling more suitable at using R134a in the main cycle. The performance not affected by changing the refrigerant of subcooler.

2.11 Conclusions

It is clear that there are a number of studies on using cascade refrigeration system. A limited number of studies are found on cost analysis of cascade refrigeration system. However, there is no study in open literature on initial design cost estimation with operation cost estimation by consider the variable environment temperatures. In this study, thermodynamic of cascade refrigeration system by chosen decision variables to estimate the initial cost of the system and operation cost estimation with variable environment temperatures will be performed.

CHAPTER 3

THERMODYNAMIC ANALYSIS OF A CASCADE REFRIGERATION SYSTEM

3.1 Introduction

Normally thermodynamic analysis can be classified in three categories, energy analysis, exergy (availability) analysis and cost analysis which is also called exergoeconomic (thermoeconomic) analysis. In this chapter general formulations of thermodynamic analysis including both energy, exergy methods and general principles and correlations of thermoeconomic analysis are presented. The equations and correlations given in this chapter are applicable to applying on cascade refrigeration system. The following sections include the formulations which used in thermodynamic analysis applied on cascade refrigeration system.

3.2 Energy Analysis of Cascade Refrigeration System

A cascade refrigeration system consists of two vapor-compression refrigeration cycles and of a number of processes that can be analyzed by applying the first law of thermodynamics for a steady-state flow process, as applied to each of the seven components individually (Figure 3.1.a), since energy must be conserved by each component and also by the whole system. Therefore, the energy balance equation for each component of the system is as follows (with the assumption that the changes in kinetic and potential energies are negligible).

Low temperature side (LTS);

$$\dot{E}_{in} - \dot{E}_{out} = \frac{dE_{system}}{dt}(steady) = 0 \tag{3.1}$$

For LTS compressor;

$$\dot{E}_{in} = \dot{E}_{out} \tag{3.2}$$

$$\dot{m}_L \times h_1 + \dot{W}_{LTS,comp} = \dot{m}_L \times h_2 \tag{3.3}$$

$$\dot{W}_{LTS,comp} = \dot{m}_L \times (h_2 - h_1) \tag{3.4}$$

For cascade condenser;

$$\dot{Q}_{c,LTS} = \dot{Q}_{L,LTS} \tag{3.5}$$

$$\dot{m}_L \times (h_2 - h_3) = \dot{m}_H \times (h_5 - h_8) \tag{3.6}$$

For LTS expansion;

$$\dot{m}_L \times h_3 = \dot{m}_L \times h_4 \tag{3.7}$$

$$h_3 = h_4 \tag{3.8}$$

For evaporator;

$$(\dot{m}_L \times h_4) + \dot{Q}_{L,LTS} = \dot{m}_L \times h_1 \tag{3.9}$$

$$\dot{Q}_{L,LTS} = \dot{m}_L \times (h_1 - h_4) \tag{3.10}$$

COP for LTS;

$$\dot{W}_{LTS,comp} + \dot{Q}_{L,LTS} = \dot{Q}_{c,LTS} \tag{3.11}$$

$$COP_L = \frac{\dot{Q}_{L,LTS}}{\dot{W}_{LTS,comp} + \dot{W}_{evap,fan}} \tag{3.12}$$

The isentropic efficiency of an adiabatic compressor is defined as

$$\eta_{LTS,comp} = \frac{\dot{W}_{LTS,comp,isent}}{\dot{W}_{LTS,comp}} = \frac{h_{2s} - h_1}{h_2 - h_1} \tag{3.13}$$

High temperature side (HTS);

$$\dot{E}_{in} - \dot{E}_{out} = \frac{dE_{system}}{dt}(steady) = 0 \tag{3.14}$$

For HTS compressor;

$$\dot{E}_{in} = \dot{E}_{out} \tag{3.15}$$

$$\dot{m}_H \times h_5 + \dot{W}_{HTS,comp} = \dot{m}_H \times h_6 \tag{3.16}$$

$$\dot{W}_{HTS,comp} = \dot{m}_H \times (h_6 - h_5) \tag{3.17}$$

For LTS expansion;

$$\dot{m}_H \times h_7 = \dot{m}_H \times h_8 \tag{3.18}$$

$$h_7 = h_8 \tag{3.19}$$

For condenser;

$$(\dot{m}_H \times h_7) + \dot{Q}_{c,HTS} = \dot{m}_H \times h_6 \tag{3.20}$$

$$\dot{Q}_{c,HTS} = \dot{m}_H \times (h_6 - h_7) \tag{3.21}$$

COP for HTS;

$$\dot{W}_{HTS,comp} + \dot{Q}_{L,HTS} = \dot{Q}_{c,HTS} \tag{3.22}$$

$$COP_H = \frac{\dot{Q}_{L,HTS}}{\dot{W}_{HTS,comp} + \dot{W}_{cond,fan}} \tag{3.23}$$

The isentropic efficiency of an adiabatic compressor is defined as

$$\eta_{HTS,comp} = \frac{\dot{W}_{HTS,comp,isent}}{\dot{W}_{HTS,comp}} = \frac{h_{6s} - h_5}{h_6 - h_5} \tag{3.24}$$

And the coefficient of performance (COP) of the cascade system becomes

$$COP_{sys} = \frac{\dot{Q}_{L,LTS}}{\left(\dot{W}_{HTS,comp} + \dot{W}_{cond,fan}\right) + \left(\dot{W}_{LTS,comp} + \dot{W}_{evap,fan}\right)} \tag{3.25}$$

3.3 Exergy analysis of a cascade refrigeration system

Figure 3.1 is a schematic of cascade refrigeration system (two vapor-compression refrigeration cycles) operating between a low temperature medium (T_L) and a high-temperature medium (T_H). The maximum COP of a refrigeration cycle operating between temperature limits of T_L and T_H based on the Carnot refrigeration cycle

$$COP_{carnot} = \frac{T_L}{T_H - T_L} \tag{3.26}$$

Practical refrigeration systems are not as efficient as ideal models like the Carnot cycle, because of the lower COP due to irreversibilities in the system. As a result of Equation 3.26, a smaller temperature difference between the heat sink and the heat source ($T_H - T_L$) provides greater refrigeration system efficiency (i.e., COP). The Carnot cycle has certain limitations, because it represents the cycle of the maximum theoretical performance. The aim in an exergy analysis is usually to determine the exergy destructions in each component of the system and to determine exergy efficiencies. The components with greater exergy destructions are also those with more potential for improvements. Exergy destruction in a component can be determined from an exergy balance on the component. It can also be determined by first calculating the entropy generation and using

$$\dot{E}x_{dest} = T_0 \times \dot{S}_{gen} \quad (3.27)$$

where T_0 is the dead-state temperature or environment temperature. In a refrigerator, T_0 is usually equal to the temperature of the high-temperature medium T_H . Exergy destructions and exergy efficiencies for major components of the cycle are as follows (state numbers refer to Figure 3.1):

Low temperature side (LTS);

For LTS compressor;

$$\dot{E}x_{in} - \dot{E}x_{out} - \dot{E}x_{D,LTS,comp} = 0 \quad (3.28)$$

$$\dot{E}x_{D,LTS,comp} = \dot{E}x_{in} - \dot{E}x_{out} \quad (3.29)$$

$$\dot{E}x_{D,LTS,comp} = \dot{W}_{LTS,comp} - \dot{m}_L \times [h_2 - h_1 - T_0 \times (s_2 - s_1)] \quad (3.30)$$

$$\varepsilon = 1 - \left(\frac{\dot{E}x_1 - \dot{E}x_2}{\dot{W}_{LTS,comp}}\right) \quad (3.31)$$

Cascade heat exchanger;

The operation of a cascade H.E, an entropy balance;

$$\dot{S}_{gen} = \dot{S}_{out} - \dot{S}_{in} \quad (3.32)$$

$$\dot{S}_{gen} = (\dot{m}_L \times s_3 + \dot{m}_H \times s_5) - (\dot{m}_L \times s_2 + \dot{m}_H \times s_8) \quad (3.33)$$

Then the exergy destruction in the evaporator becomes;

$$\dot{E}x_{D,cas,cond} = T_0 \times \dot{S}_{gen} = T_0 \times [\dot{m}_L \times (s_3 - s_2) - \dot{m}_H(s_8 - s_5)] \quad (3.34)$$

LTS Expansion valve;

$$\dot{E}x_{D,LTS,exp} = \dot{E}x_{in} - \dot{E}x_{out} \tag{3.35}$$

$$\dot{E}x_{D,LTS,exp} = \dot{E}x_3 - \dot{E}x_4 = \dot{m}_L \times [h_3 - h_4 - T_0 \times (s_3 - s_4)] \tag{3.36}$$

And,

$$h_3 = h_4 \tag{3.37}$$

So that,

$$\dot{E}x_{D,LTS,exp} = \dot{m}_L \times T_0 \times (s_4 - s_3) \tag{3.38}$$

Evaporator;

$$\dot{E}x_{D,evap} = \dot{E}x_{in} - \dot{E}x_{out} \tag{3.39}$$

$$\dot{E}x_{D,evap} = (\dot{E}x_4 - \dot{E}x_1) - \dot{E}x_{\dot{Q}_{L,LTS}} \tag{3.40}$$

$$\dot{E}x_{\dot{Q}_{L,LTS}} = \dot{Q}_{L,LTS} \times ((\frac{T_0}{T_L}) - 1) \tag{3.41}$$

$$\dot{E}x_{D,evap} = \left(\dot{E}x_4 - \dot{E}x_1\right) - \dot{E}x_{\dot{Q}_{L,LTS}} + \dot{W}_{fan,evap} \tag{3.42}$$

$$\dot{E}x_{D,evap} = \dot{m}_L \times [h_4 - h_1 - T_0 \times (s_4 - s_1)] - [\dot{Q}_{L,LTS} \times ((\tfrac{T_0}{T_L}) - 1)] + \dot{W}_{fan,evap} \tag{3.43}$$

High temperature stage (HTS);

HTS compressor;

$$\dot{E}x_{in} - \dot{E}x_{out} - \dot{E}x_{D,LTS,comp} = 0 \tag{3.44}$$

$$\dot{E}x_{D,HTS,comp} = \dot{E}x_{in} - \dot{E}x_{out} \tag{3.45}$$

$$\dot{E}x_{D,HTS,comp} = \dot{W}_{HTS,comp} - \dot{m}_H \times [h_6 - h_5 - T_0 \times (s_6 - s_5)] \tag{3.46}$$

$$\varepsilon = 1 - \left(\frac{\dot{E}x_{D,5-6}}{\dot{E}x_{D,HTS,comp}}\right) \tag{3.47}$$

By use this equation; it can be evaluate second law efficiency (exergetic efficiency)

HTS Expansion valve;

$$\dot{E}x_{D,HTS,exp} = \dot{E}x_{in} - \dot{E}x_{out} \tag{3.48}$$

$$\dot{E}x_{D,HTS,exp} = \dot{E}x_7 - \dot{E}x_8 = \dot{m}_H \times [h_7 - h_8 - T_0 \times (s_7 - s_8)] \tag{3.49}$$

And,

$$h_7 = h_8 \tag{3.50}$$

So that,

$$\dot{E}x_{D,HTS,exp} = \dot{m}_H \times T_0 \times (s_8 - s_7) \tag{3.51}$$

Condenser;

$$\dot{E}x_{D,cond} = \dot{E}x_{in} - \dot{E}x_{out} \tag{3.52}$$

$$\dot{E}x_{D,cond} = (\dot{E}x_6 - \dot{E}x_7) - \dot{E}x_{\dot{Q}_{H,HTS}} + \dot{W}_{fan,cond} \tag{3.53}$$

$$\dot{E}x_{\dot{Q}_{H,HTS}} = \dot{Q}_{H,HTS} \times ((\frac{T_0}{T_0}) - 1)$$

$$\dot{E}x_{D,cond} = [\dot{m}_H \times [h_6 - h_7 - T_0 \times (s_6 - s_7)] - [\dot{Q}_{H,HTS} \times ((\frac{T_0}{T_0}) - 1)] + \dot{W}_{fan,cond}] \tag{3.54}$$

The total exergy input to the system, total power of compressors, evaporator fan and condenser fan is equal to:

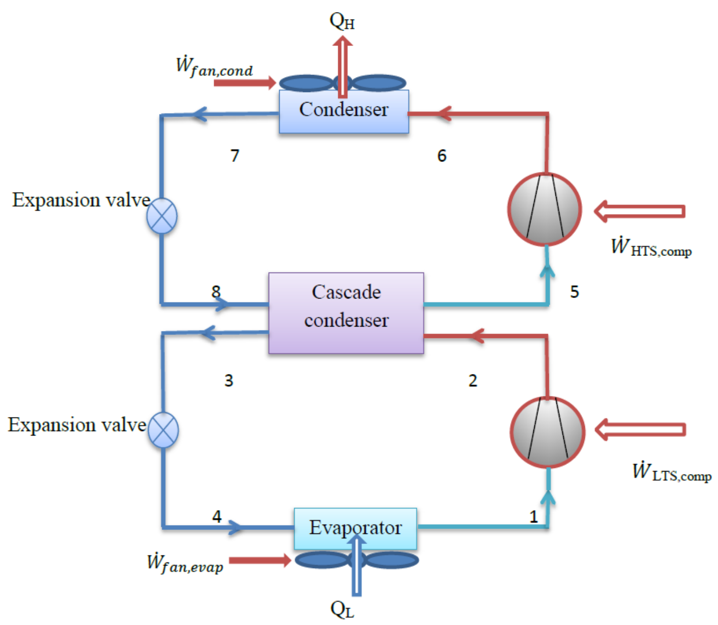
$$\dot{E}x_{in} = \dot{W}_{HTS,comp} + \dot{W}_{LTS,comp} + \dot{W}_{fan,cond} + \dot{W}_{fan,evap} \tag{3.55}$$

The exergy destruction and exergetic efficiency of the overall system can be obtained by

$$\dot{E}x_{D,total} = \dot{E}x_{in} - \dot{E}x_{out} \tag{3.56}$$

$$\varepsilon = 1 - (\frac{\dot{E}x_{D,total}}{\dot{E}x_{in}}) \tag{3.57}$$

Note: Most of the equations that have been mentioned in the preceding paragraphs have been adopted by [15] and [80].



(a)

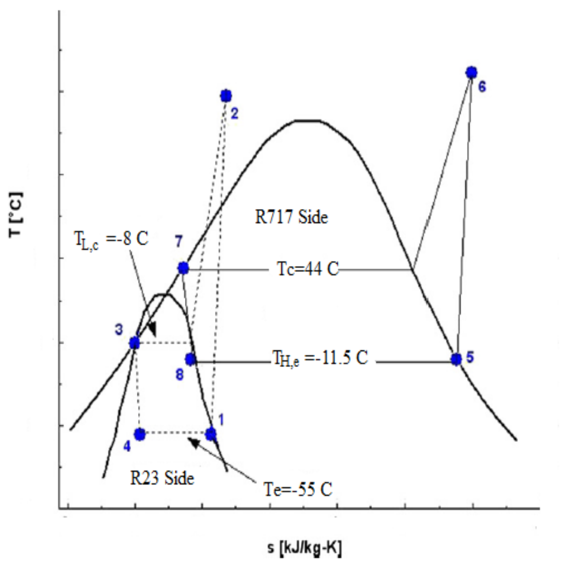


Figure 3.1 (a) Schematic of a two-stage (binary) cascade refrigeration system, (b) its *T-s* diagram.

3.4 Thermoeconomic analysis of cascade refrigeration system

The major important aspect of “combining exergy concept and cost allocation” which is known as thermoeconomics. Thermoeconomics (i.e., Exergoeconomics) is the branch of engineering that combines exergy analysis and economic principles to provide the system designer or operator with information not available through conventional energy analysis and economic evaluations but crucial to the design and operation of a cost effective system. The plant owner wants to know the true cost at which each of the utilities is generated; these costs are then charged to the appropriate final products according to the type and amount of each utility used to generate a final product. Accordingly, the objectives of the thermoeconomic analysis are to calculate separately the costs of each product generated by a system having more than one product, to understand the cost formation process and the flow of costs in the system, to optimize specific variables in a single component, or to optimize the overall system [66].

The correlation between product and total cost of the system as follows:

$$C_{total} = \sum c_o \, \dot{E}\, x_{out} = \sum c_i \, \dot{E}x_{in} + \sum_m z_m \tag{3.58}$$

Where $\mathbf{C_{total}}$ is the total annual cost of the system, $\mathbf{c_o}$ is the unit cost of product exergy, $\mathbf{c_i}$ is the unit cost of input exergy, $\mathbf{Ex_{out}}$ is the annual exergy rate for output products, $\mathbf{Ex_{in}}$ is the annual exergy rate from external sources, $\mathbf{z_m}$ is the annual cost of capital expenditures and other associated cost for the system (subscript m represents the number of components of the system). In cascade refrigeration system, the input exergy to the system is only electrical energy. Therefore, the unit cost of exergy input will be equal to the unit cost of electricity, while the product of this system is cooling capacity. In engineering economics, the unit of time interval chosen for capital cost is usually taken as a year. The capital cost in a year is obtained using the capital recovery factor (CRF) [67]. It can use the following correlation to estimate capital costs or purchase equipment costs of the system components [68].

- The purchase cost correlation of HTS compressor is:

$$C_{HTS,comp} = 9624.2 \times \dot{W}_{HTS,comp}^{0.46} \tag{3.59}$$

- The purchase equipment cost correlation of LTS compressor is:

$$C_{LTS,comp} = 10167.5 \times \dot{W}_{LTS,comp}^{0.46} \tag{3.60}$$

- The purchase equipment cost correlation of condenser is:

$$C_{cond} = [1397 \times A_{o,cond}^{0.89}] + [629.05 \times \dot{W}_{fan,cond}^{0.76}] \tag{3.61}$$

- The purchase equipment cost correlation of evaporator is:

$$C_{evap} = [1397 \times A_{o,evap}^{0.89}] + [629.05 \times \dot{W}_{fan,cevap}^{0.76}] \tag{3.62}$$

- The purchase equipment cost correlation of cascade condenser is:

$$C_{cas,cond} = [2382.9 \times A_{o,cas,cond}^{0.68}] \tag{3.63}$$

- The capital recovery factor can be estimate by:

$$CRF = i \times \frac{(1+i)^n}{(1+i)^n - 1} \tag{3.64}$$

- Annualized capital cost = capital cost× *CRF* (3.65)

- The annual total cost correlation of the system which includes the purchase cost of all components and electricity cost is:

$$C_{total} = [C_{HTS,comp} + C_{LTS,comp} + C_{cond} + C_{evap} + C_{cas,cond}] \times CRF + C_{el,H} \times [\dot{W}_{HTS,comp} + \dot{W}_{LTS,comp} + \dot{W}_{fan,cond} + \dot{W}_{fan,evap}] \times PO \tag{3.66}$$

- In case capital costs of expansion valves are neglected.

From the previous equations which applicable to calculate the purchase cost of each part of the cascade system showing us, to complete the calculations process must find outer areas of heat transfers , and lead us to concept of heat exchanger design.

3.5 Design of heat exchangers

Because thermal area and pressure loss are among the factors which affect the annual cost of a system, thermal design of a heat exchanger is of basic importance. The cascade system includes air-cooled condenser and evaporator that are compact air-cooled heat exchanger and the cascade condenser is a shell and tube heat exchanger. In order to design heat exchangers one needs to consider parameters and geometry like tube diameter, the number of tube rows and shell thickness as constants. In addition, heat exchangers are divided into two categories of single-phase and two-phase. Since two-phase heat transfer coefficients are functions of refrigerant quality, in which two-phase heat transfer coefficients are taken as constant. Finally, the total heat transfer area is the sum of heat transfer areas of heat exchangers:

$$\dot{Q} = U_o \times A_o \times \Delta T_{lm} \tag{3.67}$$

$$A_o = \frac{\dot{Q}}{U_o \times \Delta T_{lm}} \tag{3.68}$$

To find the area of each heat exchanger firstly it must be calculate the heat transfer coefficients in both sides of heat exchangers, then evaluate overall heat transfer coefficient and heat transfer area. Therefore the heat exchanger geometry must be assumed before calculations. It can be use the following correlations to evaluate the heat transfer coefficients in both sides of heat exchangers.

3.6 Overall heat transfer coefficients estimation

The overall heat transfer coefficient is a function of refrigerant side, air side and tube conductance:

$$\frac{1}{U_o} = \frac{d_o}{d_i}\frac{1}{h_i} + \frac{d_o}{2 \times k}\ln\frac{r_o}{r_i} + \frac{1}{h_o} \tag{3.69}$$

3.6.1 Refrigerant side heat transfer correlations

The correlation used for single-phase in the subcooled and superheated regions is the Gnielinsky [69]:

$$Nu_D = \frac{\left(\frac{f}{8}\right) \cdot (Re_D - 1000) \cdot Pr}{1 + 12.7\left(\frac{f}{8}\right)^{0.5} \cdot \left(Pr^{\frac{2}{3}} - 1\right)} \tag{3.70}$$

Where the friction factor f for smooth tubes is

$$f = (0.79.\, lnRe - 1.64)^{-2} \tag{3.71}$$

This correlation is valid for $0.5 < Pr < 2000\ and\ 10^4 < Re < 5 \times 10^6$

The two phase correlation used for condensation is from Dobson et al. [70]:

$$h_{tp} = f(X_{tt}) \times \left(\frac{\rho_1 \times (\rho_1 - \rho_V) \times g \times h_{fg} \times k_1^3}{D \times \Delta T \times \mu_1}\right)^{0.25} \tag{3.72}$$

And

$$f(X_{tt}) = \frac{0.375}{X_{tt}^{0.23}} \tag{3.73}$$

Where the Lockhart and Martinelli [71] parameter X_{tt} is

$$X_{tt} = \left(\frac{\rho_V}{\rho_1}\right)^{0.5} \times \left(\frac{\mu_1}{\mu_V}\right)^{0.1} \times \left(\frac{1-x}{x}\right)^{0.9} \tag{3.74}$$

The two phase correlation is not valid as the quality approaches 0 and 1; it is approximated by only integrating between qualities of 0.05 to 0.95. The result of integrating the $\left(\frac{1-x}{x}\right)^{0.9}$ term is 1.77 which is different from the 0.138 local value used (x=0.9) by Admiraal and Bullard [72]. The refrigerant side heat transfer coefficient of the two phase region is much larger than the air side coefficient.

The two phase correlation used for evaporation from Wattelet et al. [73]:

$$h_{tp} = h_1 \times (4.3 + 0.4 \times (Bo\ 10^4)^{1.3} \tag{3.75}$$

Where

$$Bo = \frac{q}{G\ h_{fg}} \tag{3.76}$$

$$h_1 = 0.023 \times Re_1^{0.8} \times \mathrm{Pr}_1^{0.4} \frac{K_1}{D} \tag{3.77}$$

3.6.2 Air side heat transfer correlations

$$Q = U \times A \times \Delta T_{lm} \tag{3.78}$$

ΔT_{lm}: Log Mean Temperature Difference (LMTD)

$$\Delta T_{lm} = \frac{(T_{h,in} - T_{c,in}) - (T_{h,out} - T_{c,out})}{\ln\left(\frac{T_{h,in} - T_{c,in}}{T_{h,out} - T_{c,out}}\right)} \tag{3.79}$$

$$Q = \dot{m} \times c_p \times \Delta T \tag{3.80}$$

$$\dot{m} = \dot{V} \times \rho \tag{3.81}$$

$$\Delta T = T_{air,out} - T_{air,in} \tag{3.82}$$

$$Re = \frac{G \cdot D_h}{\mu} \tag{3.83}$$

$$G = \frac{\dot{m}}{A_c} \tag{3.84}$$

$$G = \frac{\dot{m}}{\sigma \cdot A_{fr}} \tag{3.85}$$

$$\dot{m} = \rho \times V \times A_c \tag{3.86}$$

$$j_H = S_t \cdot Pr^{\frac{2}{3}} \tag{3.87}$$

$$S_t = \frac{Nu}{\mathrm{Pr} \cdot \mathrm{Re}} = \frac{h_a}{G \cdot C_P} \tag{3.88}$$

j_H and f estimated from a figure adopted by [74].

j_H used to find the heat transfer coefficient of air side and f is used to find the air side pressure drop for the compact tube-fin heat exchanger correlation.

It can be estimate the air side pressure drop for the compact tube-fin heat exchanger is equal to:

$$\Delta P = G^2 \cdot \frac{v_1}{2}\left[(1+\sigma^2)\left(\frac{v_2}{v_1} - 1\right) + f \cdot \frac{A}{A_c} \cdot \frac{v_m}{v_1}\right] \tag{3.89}$$

And the power input to the fan in air cooled heat exchanger can be calculated by:

$$\dot{W}_{fan} = \Delta P \times \dot{V} \tag{3.90}$$

3.7 Conclusions

In this chapter, we provided general principles, terminology, and formulation of thermodynamic and cost analyses. The procedure and formulation are applicable to cascade refrigeration system.

CHAPTER 4

PROGRAMMING DETAILS OF THE ANALYSIS

4.1 Introduction

The low head pressure and temperature during the winter or cold temperature can be avoided by two basic control methods; refrigerant side control and air side control [77]. In this study essentially the air side control used to prevent the low pressure in air cooled condenser of the cascade refrigeration system at environment temperature variation for Gaziantep city, and the refrigerant side control is used to supply two different mass flow rates of refrigerant in HTS because it has used two speed of HTS compressor, the lowest one used at the environment temperature less than 15℃ and at condenser temperature at 25℃ , and the highest speed of HTS compressor was used at the environment temperature more than 15℃ and condenser temperature at 44 ℃. The environment temperature of Gaziantep was selected according to reference [75] and the temperature selected is the average maximum temperature for given years. The environment temperatures for each month in a year are taken as average temperatures between the maximum temperature and minimum temperature for given years as showed in Table 4.4. The condenser design temperature is depend on two things; maximum temperature of air and evaporator temperature of the cycle, TD which is defined the temperature difference between the dry-bulb temperature of the air entering the coil and the saturated condensing temperature (which corresponds to the refrigerant pressure at the condenser outlet).

For air cooled condenser the typical DT value is (8 to 11) K for medium-temperature systems at a -7°C evaporator temperature [77]. For this reason it can be consider the DT is 11℃, therefore the condenser temperature is 44℃. By used this technique in the analysis it can decrease the operation cost as we shall see in the next chapter. This analysis will be done after optimization of initial design of the system in this study. The analysis done by Engineering Equation Solver (EES) program [76]. The benefit of using the software lies to reduce the time period of analysis for insert a lot of data, and get results accurately in very short time.

4.2 Assumptions of analysis

It is assumed that the changes in potential and energies and pressure drop through the cycle are negligible. It is assumed that steady-state and uniform flow conditions exist through the components of the cascade refrigeration system, in other words, the condition of the mass at each point of the components doesn't change with time. It is also assumed that isentropic efficiencies of the compressors are equal. The compression processes have been assumed to be adiabatic but not isentropic. The heat loss from the cascade heat exchanger and expansion valves are also considered as negligible. Refrigerants at the cascade heat exchanger outlet for HTS and evaporator outlet are saturated, the refrigerant entering the compressors as saturated vapour and leaving them as superheated, the refrigerant leaving the condenser as saturated liquid and entering the evaporator as wet refrigerant. Pressure losses in connecting pipes, compressor valves and heat exchangers have been neglected, heat transfer processes in all the heat exchangers are isobaric.

4.3 Objective of analysis

In this analysis, firstly thermodynamic analysis are carried out and then, heat exchangers are designed. The decision variables such as η_{HTS}, $\eta_{LTS,}$ $V_{fan,cond}$ and $V_{fan,evap}$ are changed simultaneously using direct search method and the thermoeconomic objective function (annual cost of the system) which includes the cost of electricity consumption in compressors and fans and also the equipment purchase cost of the system, is minimized. Then, optimum values for geometrical design and thermal parameters including fan air velocity of condenser, fan air velocity of evaporator, isentropic efficiency of HTS compressor and isentropic efficiency of LTS compressor evaluated. Secondly repeat the thermodynamic analysis of condenser fan by variable speed used with environment temperature considered for Gaziantep and two speeds of HTS compressor is conducted.

4.4 EES Program Description

In the EES program there is function information which has fluid properties of real fluids, ideal gases and others. The real fluids has list of many refrigerants used in air conditioning and refrigeration applications. For each refrigerant listed can be evaluated on thermophysical properties, such as conductivity, C_P, C_V, density, enthalpy, entropy, Prandtl, pressure, quality, viscosity and other properties of working fluid (refrigerant). These properties can be utilized in thermodynamic analysis of cascade refrigeration system. As explained previously the cascade refrigeration system has two separate cycles in term of HTS and LTS. The refrigerant R23 used in LTS and R717 used as a working fluid in HTS. The EES program used in this study due to the licence agreement of Mechanical Engineering Department [76].

4.5 Programing steps

- Obvious when conducting any numerical analysis, the number of equations must be equal to the number of variables or unknowns, as well as with EES program.

- The input data is the key of analysis, the important input data for analysis available in Tables 1,2,3 and 4.

- When designing any refrigeration system there must be some initial data, these data are cooling capacity to be 35 KW or 10 Ton, refrigerated space temperature is -45℃, When selecting a condenser, determine the maximum air temperature entering the coil by referring to the weather data [77] , and it can be consider the average maximum air temperature of Gaziantep as outside design temperature of the system and to be 33 approximately as show in Turkish Metrological association Table 4.4.

- All necessary initial design data should be selected depending on literature.

- Ratings of air-cooled condensers are based on the temperature difference (TD) between the dry-bulb temperature of the air entering the coil and the saturated condensing temperature (which corresponds to the refrigerant pressure at the condenser outlet). Typical TD values are 5 to 8 K for low-temperature systems at a -30 to -40°C evaporator temperature, 8 to 11 K for medium-temperature systems at a -7°C evaporator temperature[77], and it can be select the condenser temperature for HTS at 44, meaning that 11°C is (TD).

- The evaporator temperature of HTS to be -11.5°C and condenser temperature to be -8°C, then ΔT will be 3.5°C

- In applications where the space humidity is of no importance, the factors governing evaporator selection: System efficiency and economy of operation, the physical space available for evaporator installation, and Initial cost, evaporator design TD at relative humidity (75-70) for forced convection the evaporator design TD is (9-10) K, for this reason the TD of evaporator for LTS to be 10°C [79] , it mean T_e to be -55°C.

- For complete preparation to make analysis there are other parameters like temperature difference between entering and leaving air in air cooled heat exchanger (condenser & evaporator), temperature differences are 10 and 9°C for condenser and evaporator respectively.

- Operating period, period of operation per year, annual interest rate and electricity cost are important parameters to annualize the total cost of the system and available in Table 4.1.

- Also the parameters of heat exchangers which necessary for analysis are available in Tables 4.2 and Table 4.3, these data important for cost analysis.

- Coil design face velocities range for evaporator from 1m/s to 1.5 m/s for low-velocity units, from 2.5 m/s to 4 m/s for medium-velocity units and from 4 m/s to 10 m/s for high-velocity units [79]

- Normally air velocities over air-cooled condensers are between 2.5 and 6 m/s. However, because of variables involved, the optimum air velocity for a given condenser design is best determined by experiment [79].

- The fan velocity selected for air cooled condenser and evaporator is 6 m/s and the fan velocity selected for air cooled condenser and evaporator is (1.5 m/s). This rang is used in optimization of initial design, also compressor isentropic efficiency in both HTS and LTS use as variable parameters during the analysis in a range (0.625-0.85).

- Now, almost all necessary parameters are available and ready to analysis.

- The analysis can be classified into three main parts; energy, exergy and cost analyses. In the energy analysis, the program finding the thermophysical properties for 8 states in cascade refrigeration system, then finds the input powers to compressors, then heat transfer rates in each heat exchanger and COPs. Exergy analysis focused on finding exergy destruction, exergy input and second law efficiency in each component and total for the system. Cost analysis is very important part of the study and all previous related with the cost analysis here.

- The initial design is done depending on variable parameters in the range mentioned. One of the parameters is varied when the others are fixed, and this process is repeated successively for each parameter. The optimum case is evaluated at minimum annual cost under the effect of each parameter. This thermodynamic optimization is the first part of this study.

- The second part of the study, dealing with re-analysis of the air cooled condenser of HTS with environment temperature variation into account for Gaziantep city, with assume two speed of HTS compressor.

- The optimization procedure is show briefly as in Figure 4.1

- The results of initial design for which environment temperature variation taken into account are available in chapter 5 in great details.

Table 4.1

Characteristics of system initial design

Cooling capacity	35 kW
HTS condensing temperature	44 ℃
HTS evaporating temperature	-11.5 ℃
LTS evaporating temperature	-55 ℃
LTS condensing temperature	-8 ℃
Cascade condenser temperature difference	3.5 ℃
Ambient temperature	33 ℃
Cold refrigerated space temperature	-45 ℃
Operating period	15 years
Period of operation per year	6570 h
Annual interest rate	8%
Electricity cost	0.08 $\$KW^{-1}h^{-1}$

Table 4.2

Input design parameters of cascade condenser

Outside diameter (m)	Tube thickness (mm)	Tube pitch (m)	Number of tube passes
0.025	2	0.032	2

Table 4.3

Input design parameters for the compact heat exchanger (circular tube continuous fins) used as a condenser and evaporator

Parameter	Value
Tube outside diameter, D_o	16.4 (mm)
Fin pitch	276 per m
flow passage hydraulic diameter, D_h	6.68 (mm)
Fin thickness, t	0.25 (mm)
Free flow area/frontal area, σ	0.449
Heat transfer area/total volume, α	269 m^2/m^3
Fin area/total area, A_f/A	0.825

Table 4.4 Maximum and Minimum Temperatures Values in the years (1970 - 2011) Gaziantep/Turkey [75]

The Months	1	2	3	4	5	6	7	8	9	10	11	12
Maximum Temperature (° C)	19	21	27.40	34	37.80	39.60	44	42	40.80	34.40	27.30	25.20
Average Maximum Temperature for given years	32.71											
Minimum Temperature (° C)	-16.80	-15.60	-11.00	-2.50	3.20	7.10	11.80	12.70	6.40	-1.30	-7.00	-13.40
Average Temperatures for given years	1.10	2.70	8.20	15.75	20.50	23.35	27.90	27.35	23.60	16.55	10.15	5.90

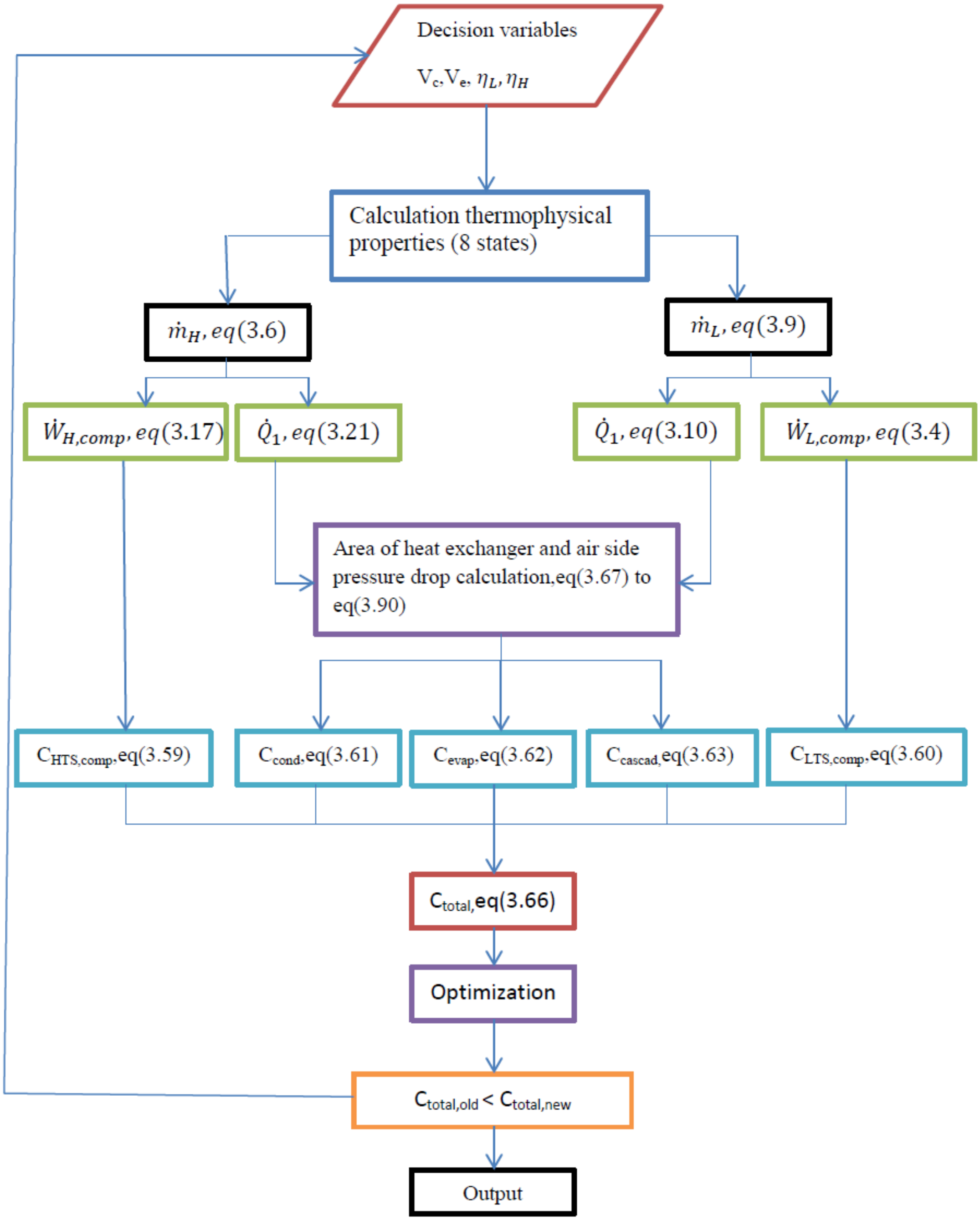


Figure 4.1. Schematic diagram of the optimization procedure.

4.6 Conclusion

The main steps of the programming and necessary input data are discussed in this chapter. The EES program description, objective of this study, assumptions, programing steps are mentioned. Also initial design optimization is presented in the flowchart. Characteristics of system initial design, input design parameters of cascade condenser, input design parameters for the compact heat exchanger used as a condenser and evaporator and maximum and minimum temperatures values in the years (1970-2011) Gaziantep/Turkey, flowchart of optimization procedure are introduced.

CHAPTER 5

RESULTS AND DISSCUSION

5.1 Introduction

In this chapter, the results of analysis on finding the optimum values of decision variables in the first part and on finding the optimum operation cost of the fan condenser and HTS compressor with environment temperature variation considered in comparison to operation cost of initial design. The results of analysis for initial design listed in Tables 5.1 to 5.4 and showed in Figures 5.1to 5.20. The results of variable environment temperature variation considered in the second part of analysis listed in Table 5.5 and show Figure 5.21.

Table 5.1 Effect of HTS compressor isentropic efficiency variation on other critical parameters

η_{HTS}	COP_{sys}	COP_{HTS}	C_{total} ($)	$\dot{E}x_{D,total}$ (KW)	$\dot{E}_{ex,total}$ (%)	$\dot{W}_{C,HTS}$ (KW)	$\dot{W}_{net}$ (KW)	$\dot{W}_{fan,cond}$ (KW)	$\dot{E}x_{in}$ (KW)
0.625	1.01	2.161	76621	22.76	34.46	20.14	32.24	1.66	34.73
0.65	1.03	2.242	75961	21.98	35.25	19.36	31.47	1.648	33.94
0.675	1.05	2.323	75347	21.25	36.03	18.65	30.75	1.637	33.21
0.7	1.08	2.403	74775	20.57	36.78	17.98	30.09	1.627	32.54
0.725	1.1	2.482	74239	19.94	37.5	17.36	29.47	1.618	31.91
0.75	1.12	2.562	73736	19.35	38.2	16.78	28.89	1.609	31.32
0.775	1.14	2.641	73264	18.81	38.89	16.24	28.35	1.601	30.77
0.8	1.16	2.719	72820	18.29	39.55	15.73	27.84	1.593	30.26
0.825	1.18	2.797	72401	17.81	40.19	15.26	27.36	1.585	29.77
0.85	1.19	2.875	72004	17.35	40.82	14.81	26.91	1.578	29.32

Table 5.2 Effect of LTS compressor isentropic efficiency variation on other critical parameter

$\boldsymbol{\eta}_{LTS}$	COP_{sys}	COP_{LTS}	C_{total} ($)	$E_{ex,total}$ (%)	$\dot{W}_{C,LTS}$ (KW)	$\dot{W}_{fan,cond}$ (KW)	$\dot{W}_{C,HTS}$(KW)	$\dot{E}x_{in}$ (KW)	$\dot{E}x_{D,total}$ (KW)
0.625	0.9946	2.024	78375	34	16.46	1.723	16.18	35.19	23.22
0.65	1.019	2.101	77458	34.85	15.83	1.702	15.98	34.34	22.37
0.675	1.043	2.178	76606	35.67	15.24	1.682	15.79	33.55	21.58
0.7	1.067	2.254	75813	36.47	14.7	1.664	15.62	32.81	20.85
0.725	1.089	2.33	75073	37.24	14.19	1.647	15.46	32.13	20.16
0.75	1.111	2.406	74380	38	13.72	1.632	15.31	31.49	19.52
0.775	1.133	2.482	73731	38.73	13.28	1.617	15.17	30.89	18.93
0.8	1.154	2.557	73121	39.45	12.86	1.603	15.04	30.34	18.37
0.825	1.174	2.632	72546	40.14	12.47	1.59	14.92	29.81	17.84
0.85	1.194	2.707	72004	40.82	12.11	1.578	14.81	29.32	17.35

Table 5.3 Effect of evaporator fan velocity variation on other critical parameters

V_{evap_fan} (m/s)	A_{evap} (m^2)	COP	COP_{LTS}	C_{OP} ($)	C_{purch} ($)	C_{total} (S)	$\dot{W}_{fan,evap}$ (KW)	$\dot{E}x_{in}$ (KW)	$\dot{E}x_{D,total}$ (KW)	$E_{ex,total}$ (%)
1.5	415.7	1.227	2.881	14998	75730	90728	0.04384	28.53	16.57	41.93
2	344.8	1.225	2.872	15017	70382	85399	0.08041	28.57	16.61	41.88
2.5	298.6	1.223	2.861	15043	66830	81872	0.1293	28.62	16.65	41.81
3	265.4	1.22	2.846	15075	64240	79315	0.1906	28.68	16.72	41.72
3.5	239.9	1.217	2.83	15113	62231	77344	0.2633	28.75	16.79	41.61
4	220.5	1.214	2.81	15158	60683	75841	0.3495	28.84	16.87	41.49
4.5	204.3	1.209	2.788	15211	59391	74602	0.4493	28.94	16.97	41.35
5	191	1.205	2.762	15271	58313	73585	0.5641	29.05	17.09	41.18
5.5	179.7	1.2	2.736	15336	57396	72732	0.6872	29.18	17.21	41.01
6	169.8	1.194	2.707	15408	56596	72004	0.8249	29.32	17.35	40.82

Table 5.4 Effect of condenser fan velocity variation on other critical parameters

$V_{cond,fan}$ (m/s)	A_{cond} (m^2)	COP	COP_{HTS}	C_{OP} (\$)	C_{cond} (\$)	C_{Purch} (\$)	$\dot{W}_{fan,cond}$ (KW)	C_{total} (\$)	$\dot{E}x_{in}$ (KW)	$\dot{E}x_{D,total}$ (KW)	$E_{ex,total}$ %
1.5	716	1.26	3.164	14621	485434	87492	0.08091	102113	27.82	15.85	43.01
2	596.5	1.26	3.149	14659	412694	78994	0.1534	93653	27.89	15.93	42.9
2.5	516.3	1.25	3.129	14708	363045	73193	0.2463	87902	27.98	16.02	42.76
3	459.3	1.25	3.106	14769	327237	69010	0.3622	83779	28.1	16.13	42.58
3.5	416.1	1.24	3.077	14843	299819	65807	0.503	80650	28.24	16.27	42.37
4	382.1	1.23	3.044	14931	277994	63257	0.6693	78187	28.41	16.44	42.12
4.5	353.3	1.22	3.007	15030	259408	61085	0.858	76115	28.6	16.63	41.85
5	330.6	1.22	2.966	15143	244672	59364	1.073	74507	28.81	16.84	41.53
5.5	311.3	1.21	2.922	15269	232067	57891	1.313	73160	29.05	17.08	41.19
6	294.4	1.19	2.875	15408	220982	56596	1.578	72004	29.32	17.35	40.82

5.2 Effect of variable parameters on other important parameters at initial design

The effects of variations in condenser fan velocity, evaporator fan velocity, and isentropic efficiencies of HTS and LTS compressors on other critical parameters are discussed successively below.

5.2.1 Effects of variation in condenser fan velocity

Figures 5.1 through 5.6 refer to effects of condenser fan velocity variation on COPs, A_{cond}, C_{total}, $\dot{E}x_{D,total}$, costs, $\dot{W}_{fan,cond}$ and second low efficiency respectively. The Figure 5.1 described the relationship between fan speed of the condenser and the COP of HTS and its system performance. By increase of the fan speed, the coefficient of performance of the HTS will be reduced gradually with the increase in air velocity and hence system performance will be reduced slightly. Figure 5.2 show any decrease in the air velocity of the fan there will be a significant reduction in surface area of the heat exchanger. It is more than twice times value between the values of small and high range of condenser fan velocity, and this decline is substantially between (1.5 and 4 m/s), and be stable or almost constant after that. Figures 5.3 show the effect of fan speed increase on the purchase cost of system and operational cost as well as the total annual cost. Of these figures, it is noted that an increase in the speed of the fan velocity, the operating cost of the system will increase gradually, but the cost of purchasing will decrease significantly with increasing speed up to 4 m/s, then the decrease will be stable roughly in the remaining. Figure 5.4 shows the increase in the speed of the fan there will be a small increase range and then increase to speed of 4 m/s, and then settle down after that increase approximately linearly , but the total annual cost would be reduced in the end as mentioned in the previous paragraph. Figure 5.5 shows the effect of fan velocity increasing on total exergy destruction; by increasing the fan velocity there will be a second order function increase in the total exergy destruction up to 4 m/s then stabilized in the increasing of an almost linear relationship. Figure 5.6 shows the relationship between the condenser fan velocity and the second law efficiency (i.e., exergetic efficiency), and notes that increase in fan velocity can cause be a second order decrease in the second law efficiency and up to 6 m/s.

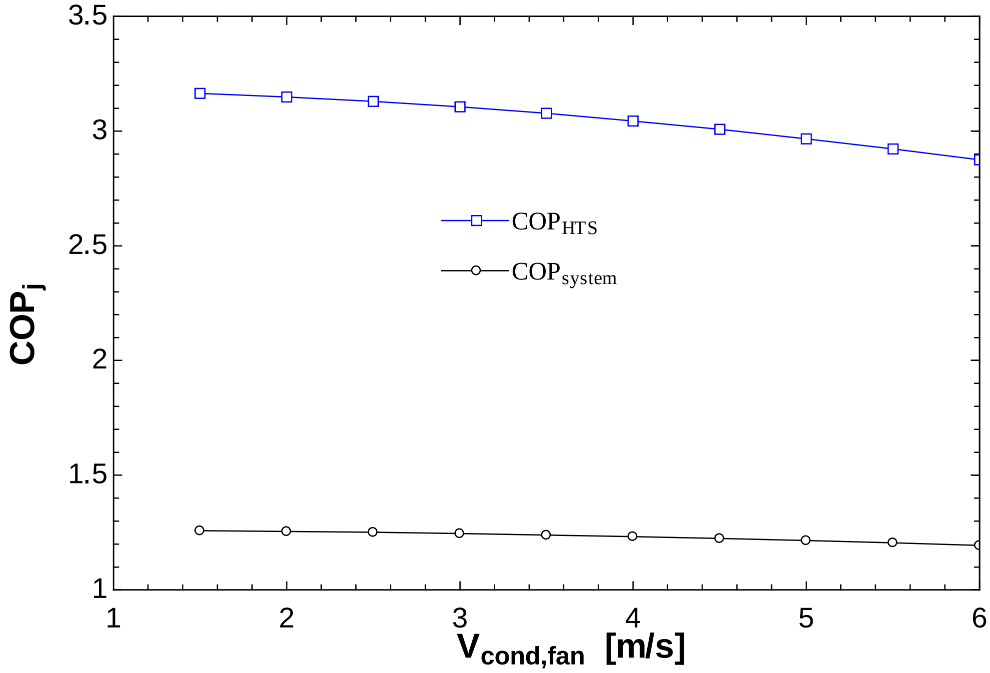


Figure 5.1 Effect of condenser fan velocity variation on COP_{HTS} and COP_{system}

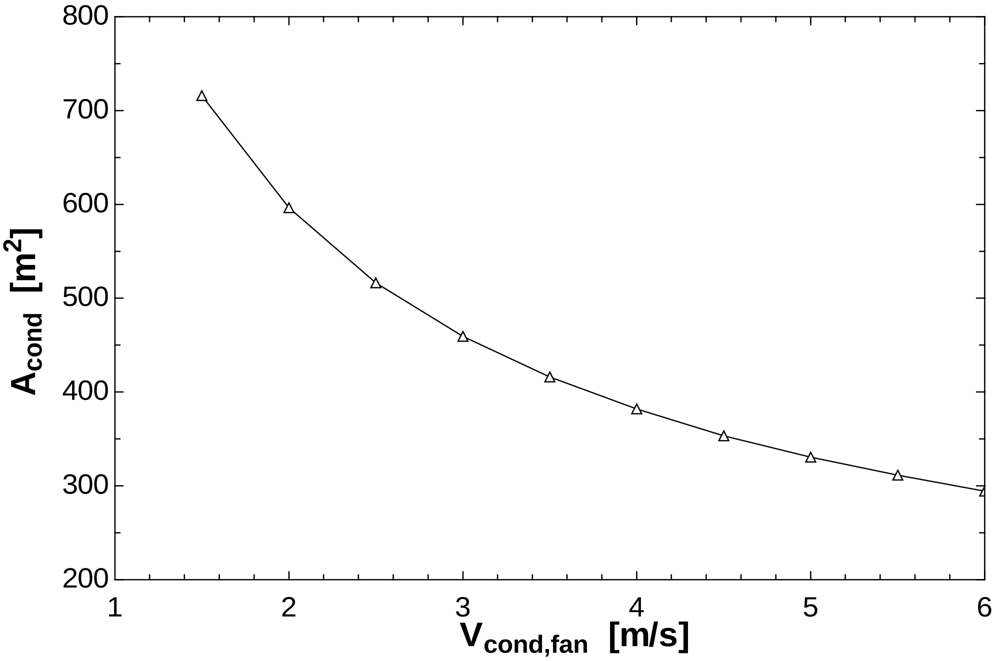


Figure 5.2 Effect of condenser fan velocity variation on condenser outer area

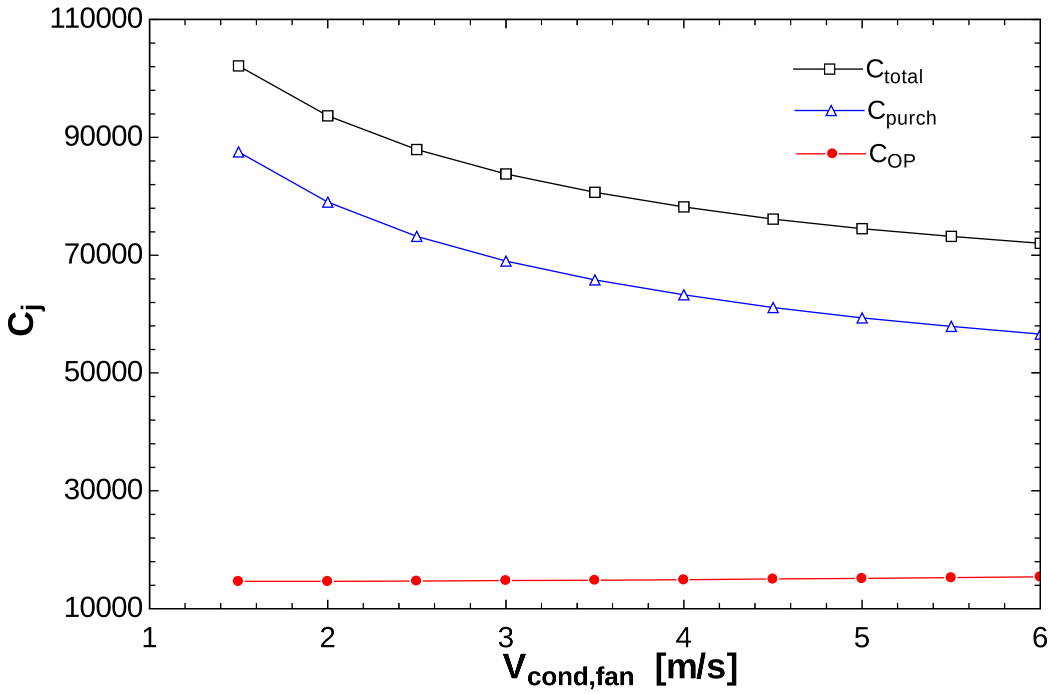


Figure 5.3 Effect of condenser fan velocity variation on total annual, purchase and operation costs of the system.

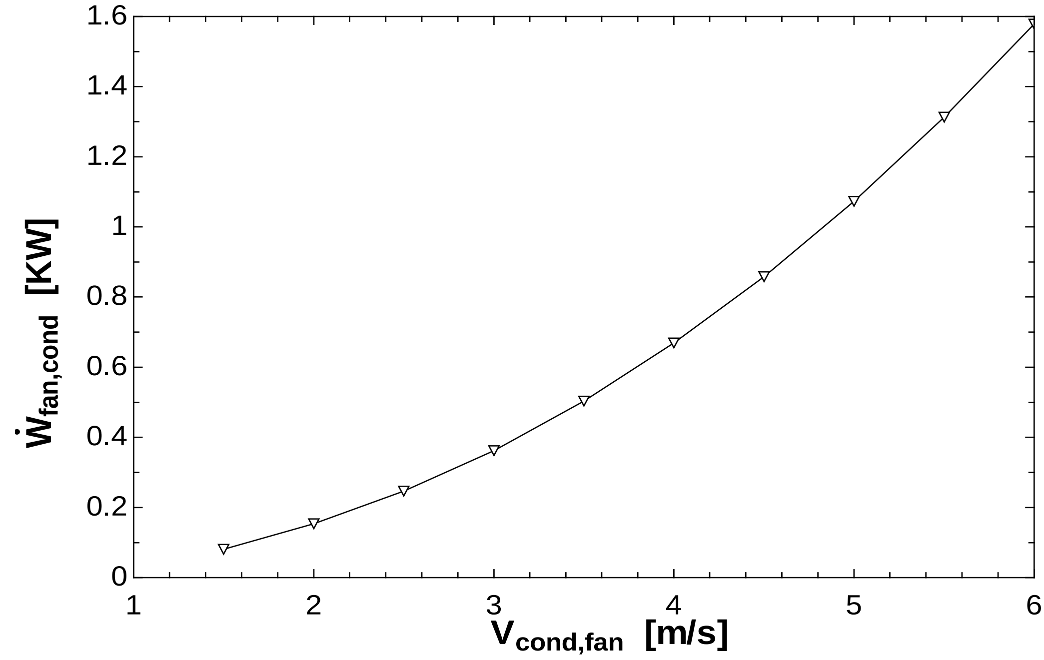


Figure 5.4 Effect of condenser fan velocity variation on power input to condenser fan

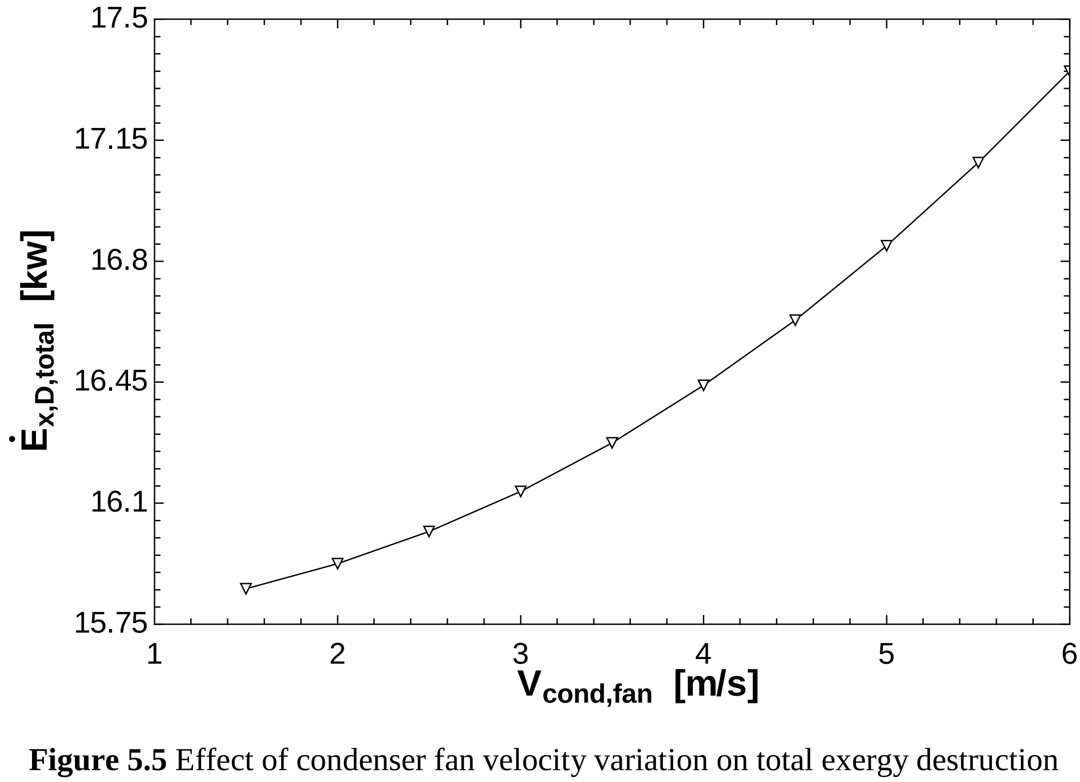


Figure 5.5 Effect of condenser fan velocity variation on total exergy destruction

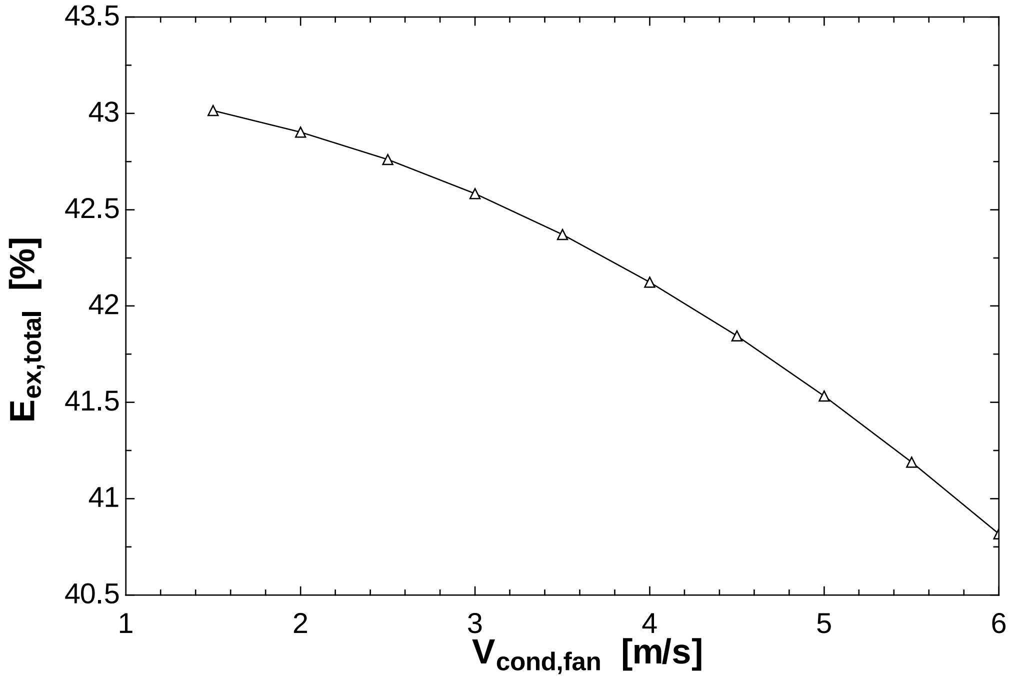


Figure 5.6 Effect of condenser fan velocity variation on total exergetic efficiency

5.2.2 Effects of variation in evaporator fan velocity

Figures 5.7 through 5.12 refer to effects of evaporator fan velocity variation on COPs, A_{evap}, C_{total}, $\dot{E}x_{D,total}$, costs, $\dot{W}_{fan,evap}$ and second low efficiency respectively. Figure 5.7 shows the effects of evaporator fan velocity variation on COP_{LTS} and COP_{system}. As an increase in fan velocity the COP_{LTS} decreases almost gradually in a linear relation approximately, and also lead to slightly low efficiency of the COP_{system}. Figure 5.8 shows the effect of evaporator fan velocity increasing on the surface area of the evaporator, where if velocity increases the surface area will be reduced dramatically and then decline slightly up 4 m/s, then settle down later in the form of an almost linear relationship. It can be observed from Figure 5.9; by an increase in evaporator fan velocity up to 4 m/s, the operating cost increased slightly, but the cost of purchasing of the system and the total annual cost decreases importantly, then constitute straight lines. Figure 5.10 shows that any increase in evaporator fan velocity will lead to an increase in power input to the fan and thus operating costs increase, but this increase is small compared to a decline in the cost of components purchase and total annual cost. Figure 5.11 shows the relationship between the effect of fan velocity increase on the total exergy destruction of the system, where it is any increase in fan velocity leads to second order increase in the total exergy destruction, then increase up to 4 m/s and approximately stabilized after that of an almost linear relationship. Figure 5.12 Shows the effect of evaporator fan velocity on the second law efficiency, an increase in the fan velocity there will be a slowly decrease at first, and the slop of curve increase up to 4 m/s then it will be stable as a straight line approximately .

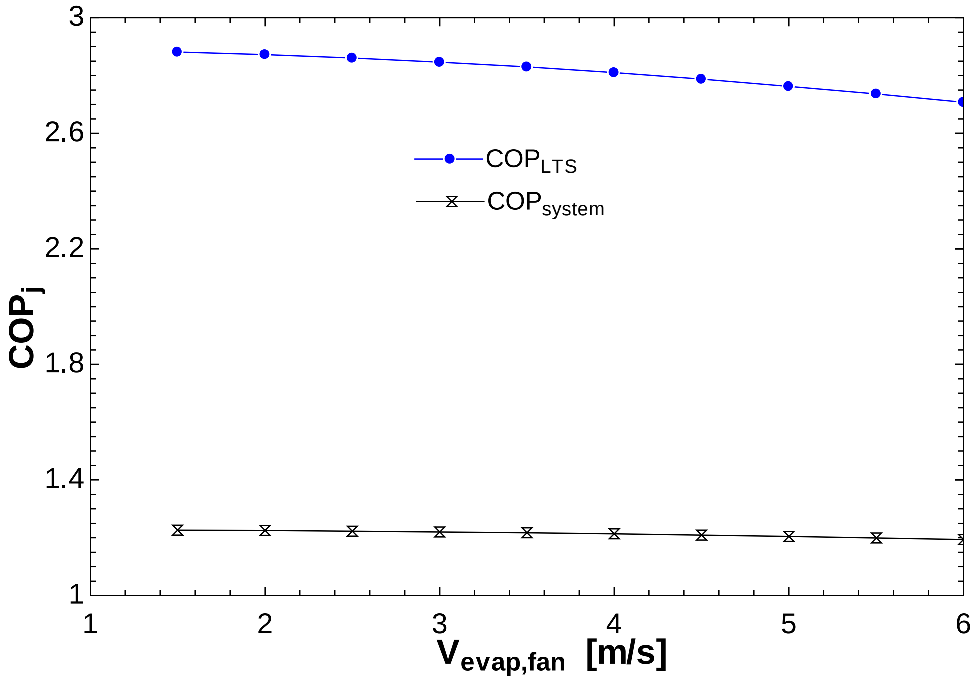


Figure 5.7 Effect of evaporator fan velocity variation on COP_{LTS} & COP_{system}

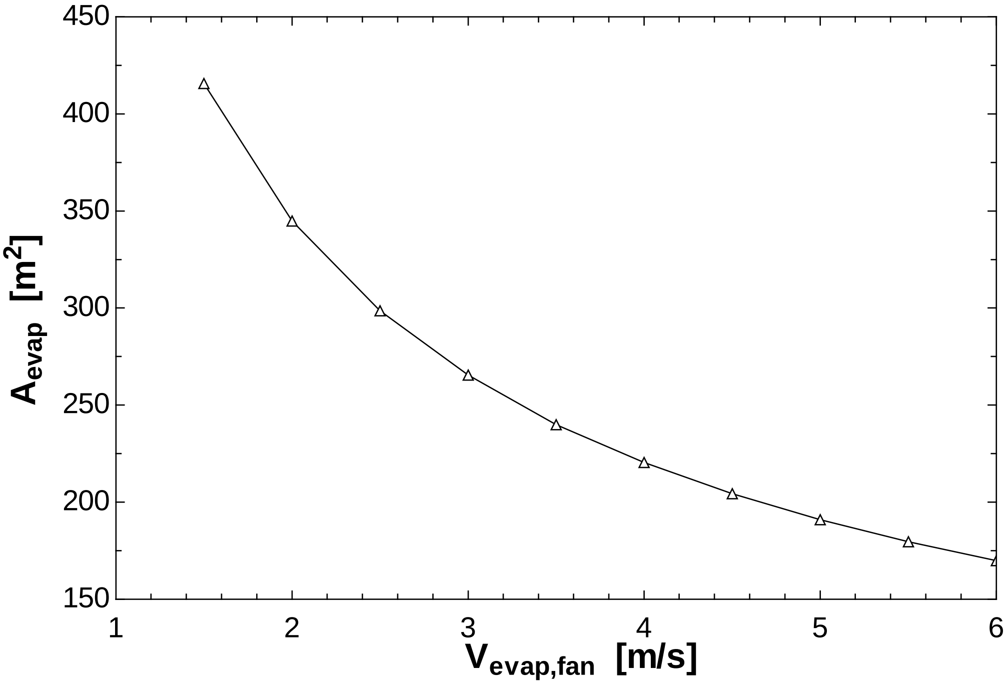


Figure 5.8 Effect of evaporator fan velocity variation on evaporator outer area

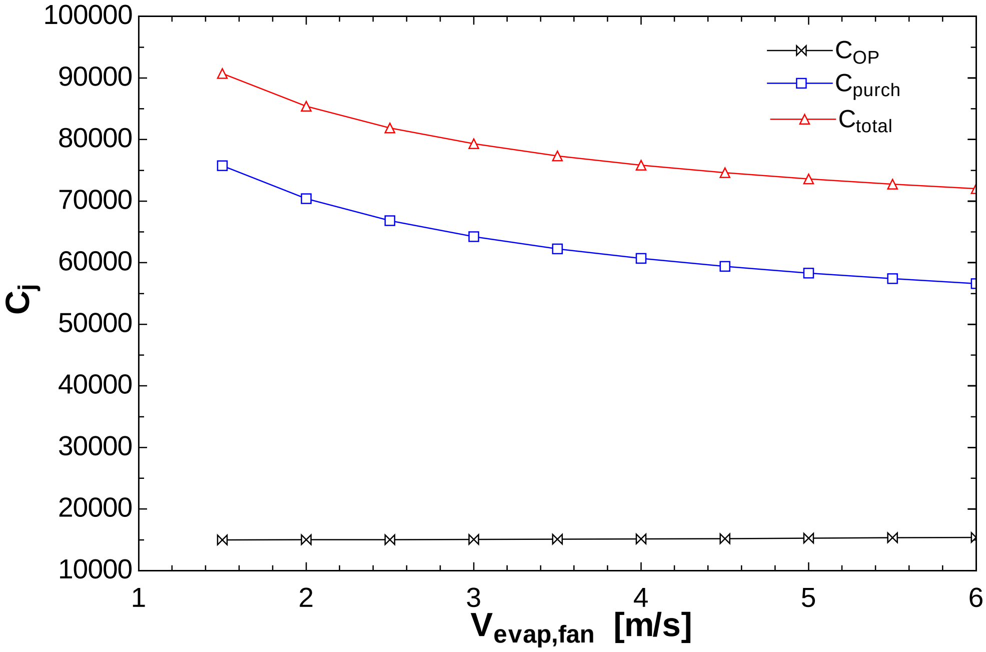


Figure 5.9 Effect of evaporator fan velocity variation on operation, purchase and total costs

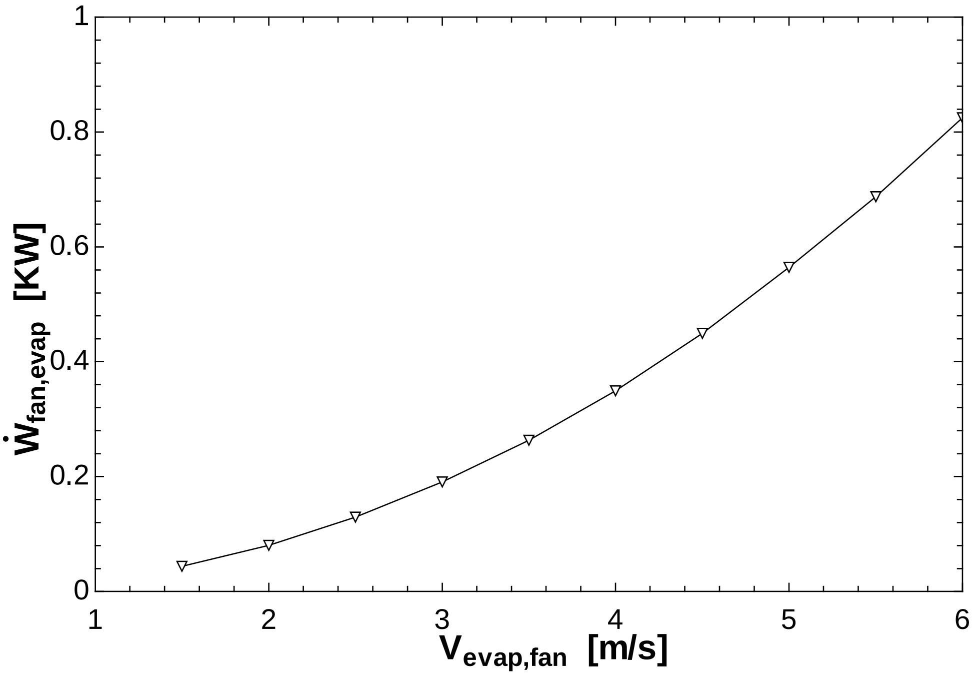


Figure 5.10 Effect of evaporator fan velocity variation on power input to evaporator fan.

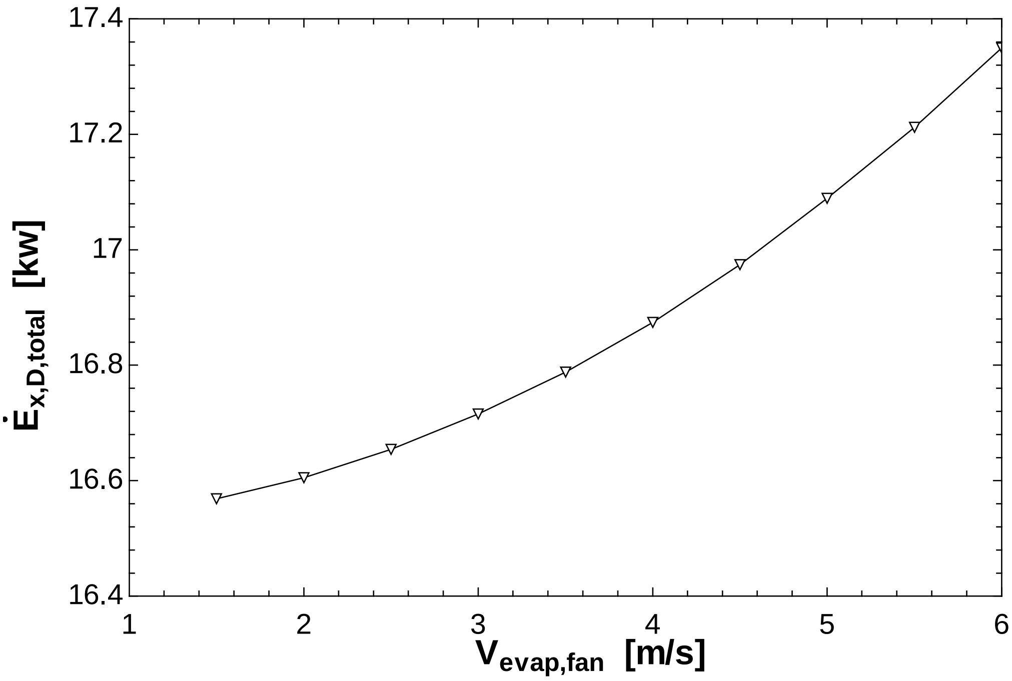


Figure 5.11 Effect of evaporator fan velocity variation on total exergy destruction

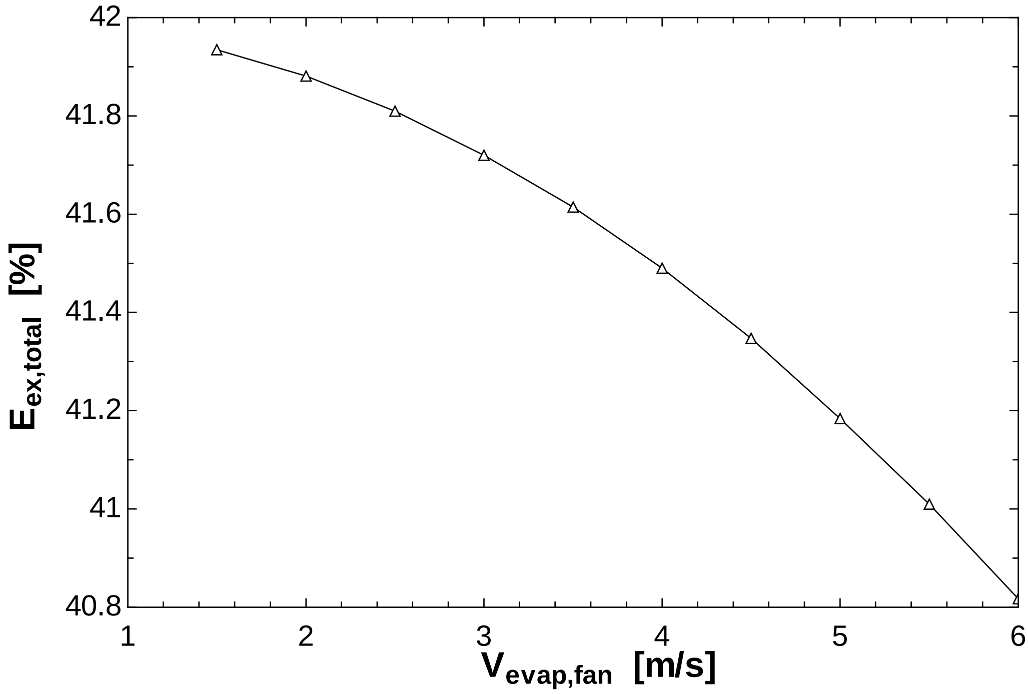


Figure 5.12 Effect of evaporator fan velocity variation on total exergetic efficiency

5.2.3 Effect of variation in LTS isentropic efficiency

Figures 5.13 through 5.16 show the effects of increase or decrease in η_{LTS} on COPs, costs, the total exergy destruction and the second law efficiency of the system respectively. Figure 5.13 shows the effect of η_{LTS} on COPs as that; increase of it from 0.62 to 0.85 causes roughly a ~35% increase on COP_{LTS} and a ~ 20% increase on COP_{system}, while the COP_{HTS} remains constant. Figure 5.14 shows the effects of η_{LTS} increase on operation cost, total purchase cost and total annual cost, as that; increase of it from 0.62 to 0.85 causes an ~8% linear decrease on C_{total}, a~5% linear decrease on C_{purch}, and a ~15% decrease on C_{OP}. Figure 5.15 shows the effect of increase in η_{LTS} on total exergy destruction. It is noted from the figure that; increase of η_{LTS} in the range causes to a ~25% decrease in total exergy destruction. Figure 5.16 shows the relationship between η_{LTS} and second law efficiency, its noted from the figure that; increase of η_{LTS} from 0.62 to 0.85 causes a ~20% almost linearly increase in second law efficiency.

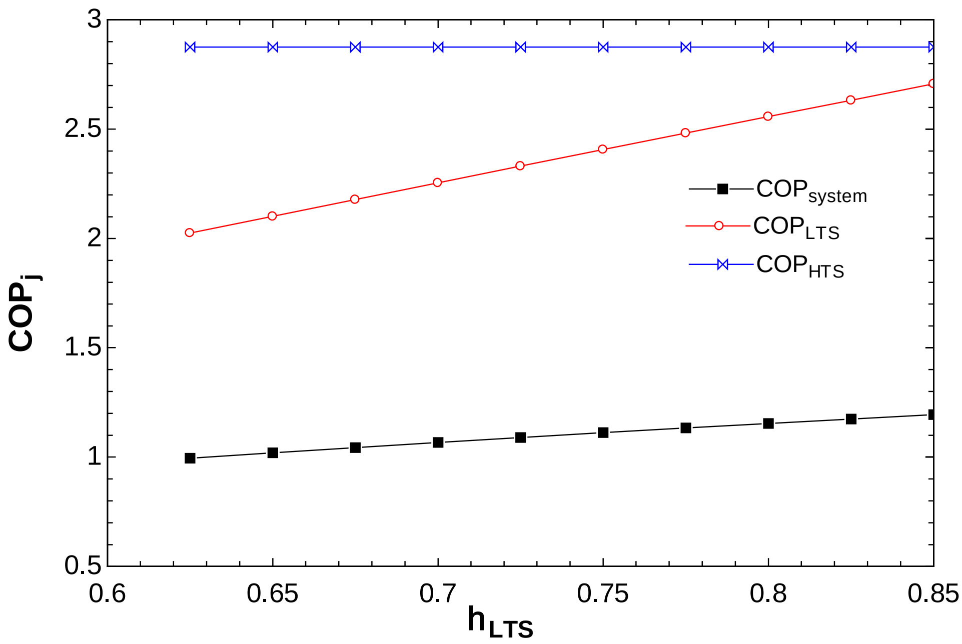


Figure 5.13 Effect of isentropic efficiency variation of LTS compressor on COPs

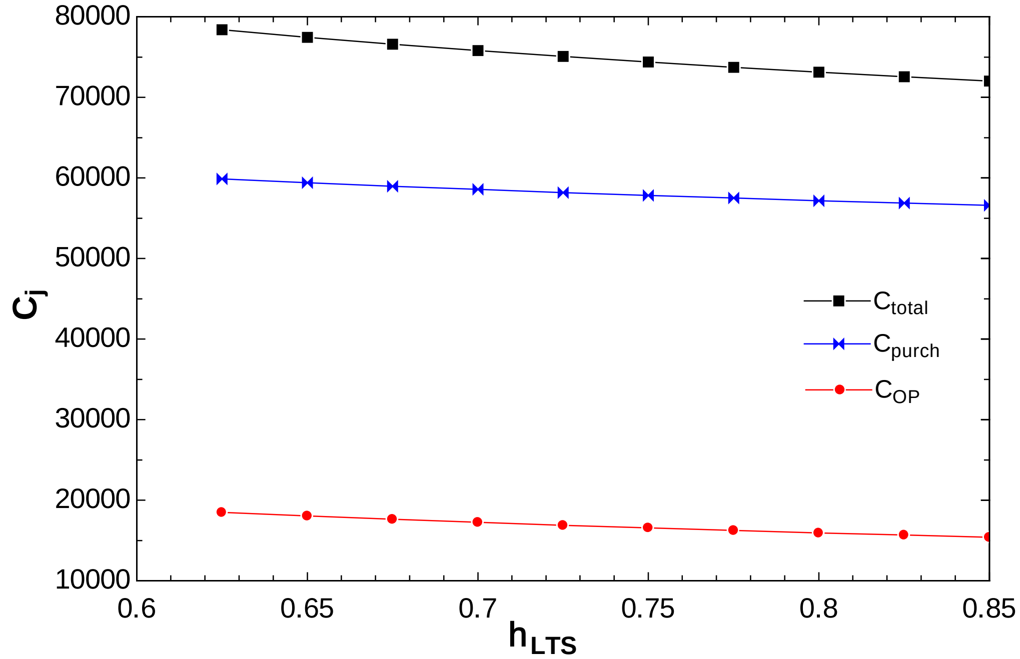


Figure 5.14 Effect of isentropic efficiency variation of LTS compressor on cost

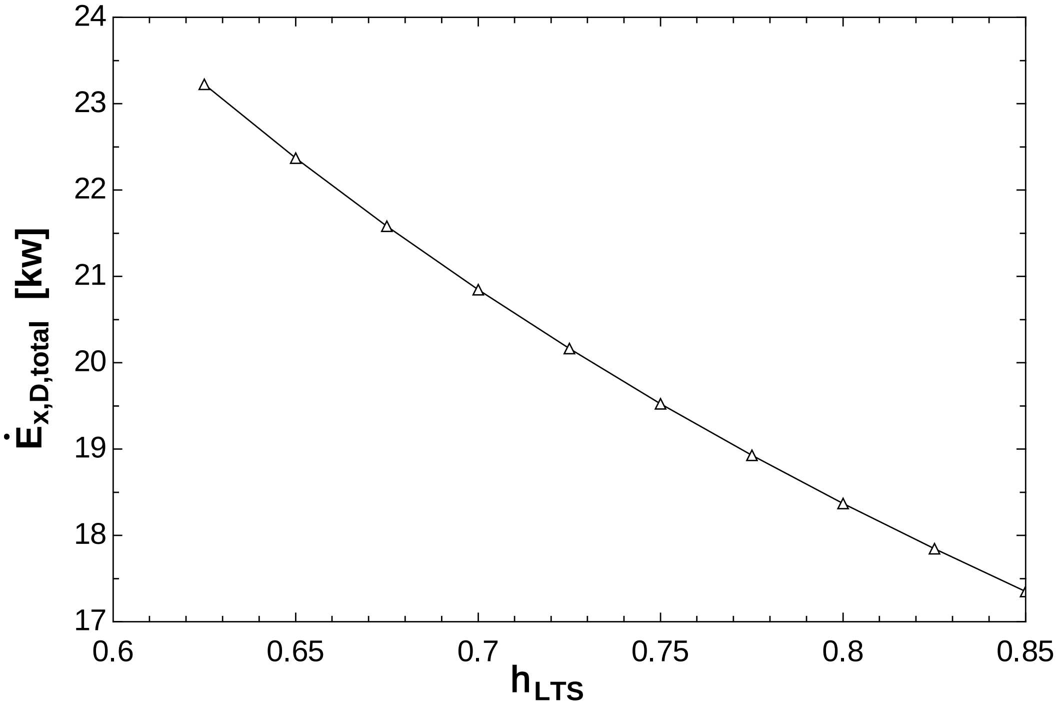


Figure 5.15 Effect of isentropic efficiency variation of LTS compressor on total exergy destruction.

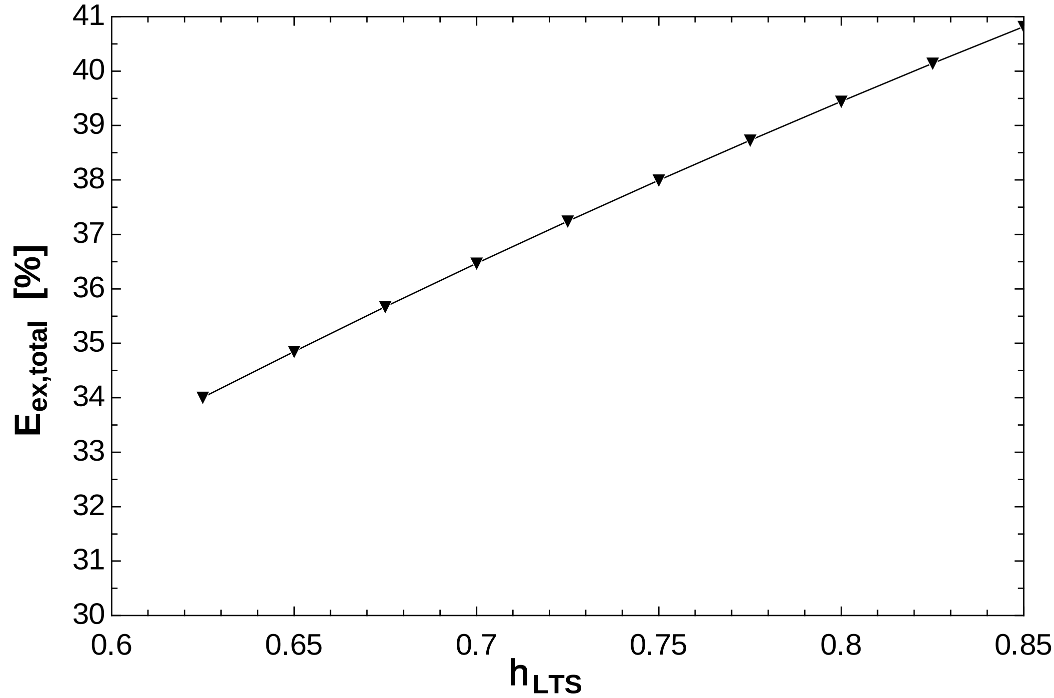


Figure 5.16 Effect of isentropic efficiency variation of LTS compressor on exergetic efficiency

5.2.4 Effect of isentropic efficiency variation on other important parameters

Figures 5.17 through 5.20 show the effects of increase or decrease in η_{HTS} on COPs, costs, the total exergy destruction and the second law efficiency of the system respectively. Figure 5.17 shows the effect of the increase in η_{HTS} on COPs. As shown from the figure, increase of η_{HTS} 0.625 to 0.85 causes a ~20% linear increase on COP_{system} and a ~34% linear increase on COP_{HTS} while COP_{LTS} remains constant. Figure 5.18 shows the effects of η_{HTS} increase on operation cost, total purchase cost and total annual cost, as that; increase of it from 0.62 to 0.85 causes an operation cost decrease and the purchase cost which causes to a ~6.4% decrease in the total annual cost, meaning that; the η_{HTS} it has less effect on the total annual cost compared to the effect of the η_{LTS}. Figure 5.19 shows the effect increase in η_{HTS} on total exergy destruction. It's observed from the figure that; increase of η_{HTS} in the range causes a~24% second order decrease in total exergy destruction. It can also be understand that the order of magnitude of the effects on η_{LTS} and η_{HTS} on total exergy destruction are same. Figure 5.20 shows the relationship between the η_{HTS} second law efficiency. Its observed from the figure that; increase of η_{HTS} from 0.625 to 0.85 causes a ~19% almost linear increase in second law efficiency. Its mean is that; there is very small difference about the effects of η_{LTS} and η_{HTS} on second law efficiency. Eventually it is clear that the η_{LTS} is a critical parameter in the design of any cascade refrigeration system and it should be taken into account like other parameters. If the present parameters taken into account accurately in the design of cascade refrigeration system it can be avoid the extra costs, large loss of energy and environment effect will be decrease.

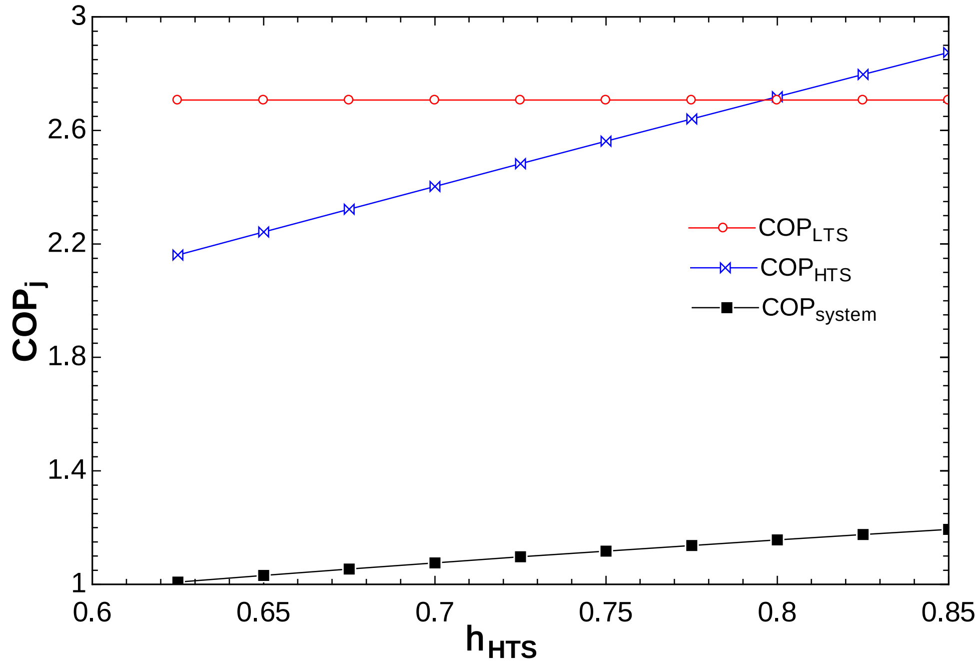


Figure 5.17 Effect of isentropic efficiency variation of HTS compressor on COPs

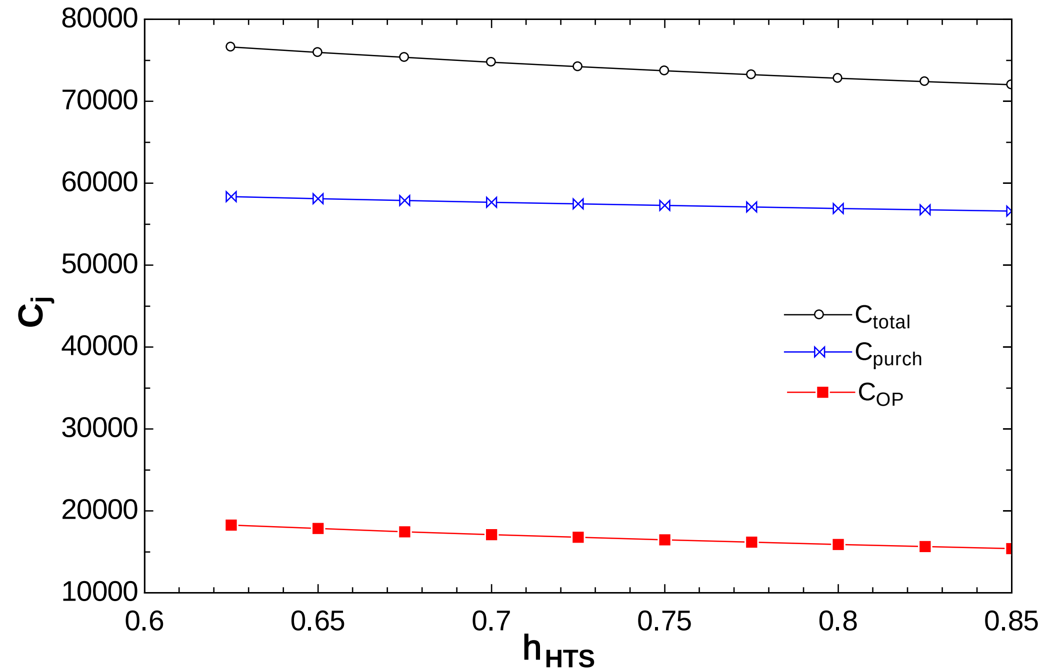


Figure 5.18 Effect of isentropic efficiency variation of HTS compressor on costs

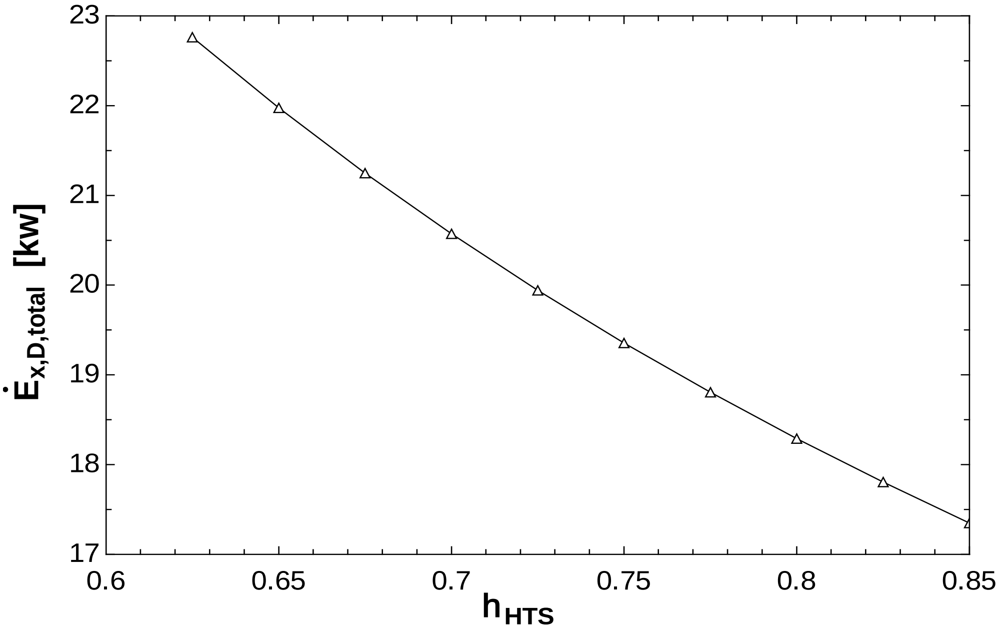


Figure 5.19 Effect of isentropic efficiency variation of HTS compressor on total exergy destruction

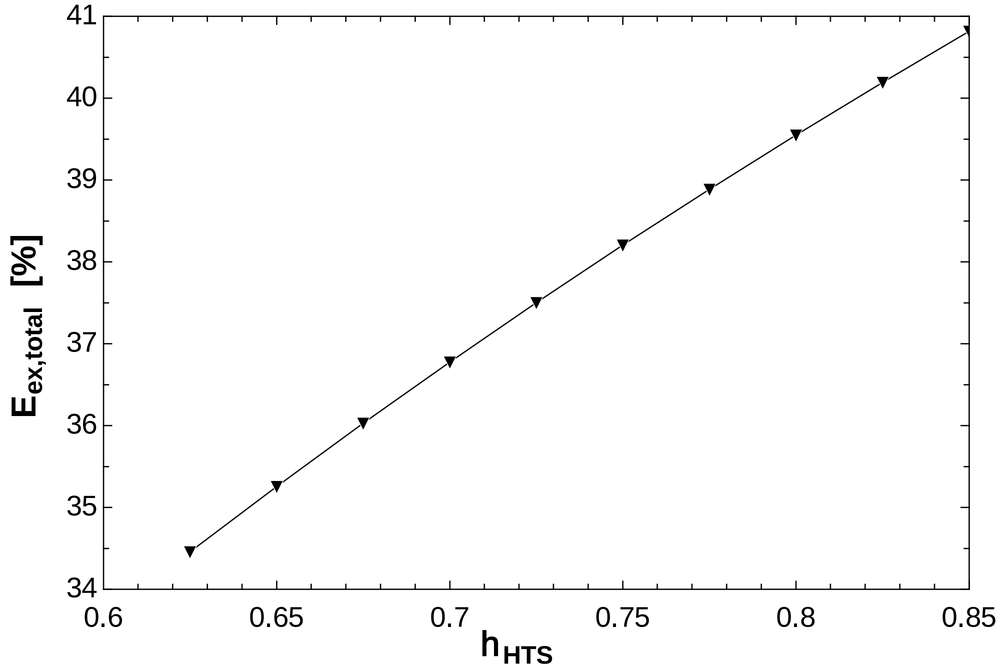


Figure 5.20 Effect of isentropic efficiency variation of HTS on exergetic efficiency

Table 5.5 Result of initial (optimize) case

$Q_{L,LTS}$	35 KW
T_0	33 ℃
T_c	44 ℃
$T_{e,HTS}$	-11.5 ℃
$T_{c,LTS}$	-8 ℃
T_e	-55 ℃
T_L	-45 ℃
$A_{cas,cond}$	122.8 m^2
A_{evap}	169.8 m^2
A_{cond}	294.4 m^2
$\dot{W}_{HTS,comp}$	14.81 KW
$\dot{W}_{LTS,comp}$	12.11 KW
$\dot{W}_{fan,cond}$	1.578 KW
$\dot{W}_{fan,evap}$	0.825 KW
$V_{fan,evap}$	6 m/s
$V_{fan,cond}$	6 m/s
$\dot{E}x_{in}$	29.32 KW
$\dot{E}x_{out}$	11.97 KW
$\dot{E}x_{D,total}$	17.35 KW
ϵ	40.82%
COP_{system}	1.194
COP_{HTS}	2.875
COP_{LTS}	2.707
η_{LTS}	0.85
η_{HTS}	0.85
C_{total}	72004 $

5.3 Operation cost estimation of condenser fan and HTS compressor with variable temperature

As explained previously, the environment temperature variation for Gaziantep city considered to conduct energy and cost analysis of condenser fan and compressor of HTS an attempt to find the energy used in the condenser fan and HTS compressor, in order to approach the reality more, and reduce the electrical power used, also this analysis done after initial case or design analysis completed and optimized of parameters specified. Through considered analysis the air temperature is accepted as constant during each month. Then the heat rejection from the condenser to the outside air is accepted as constant for both condenser temperature cases in order to control the balance of pressure in HTS condenser through use of variable fan speed and two different speeds of HTS compressor. The process of calculation of energy consumption of the fan and the rest of the other variables by means of using the Eqs 3.78 up to 3.90 discussed in chapter three. The results of the analysis with variable environment temperature considered with two different condenser temperatures are available in Table 5.5 condenser temperatures available in table 5.5. As is shown from figure 5.21 the results refer that there is a cost saving by use this technic in the analysis, by 11 % of operation cost saving compare with the base case without variable temperature considered in the analysis, and by 8% of operation cost saving with variable temperature considered in the analysis at 44℃ used as a condenser temperature, while the analysis of previous case consider two temperatures of condenser, one of them used for summer time or at outside temperature more than 15℃ and the other one is used in winter or outdoor temperature less 15℃.

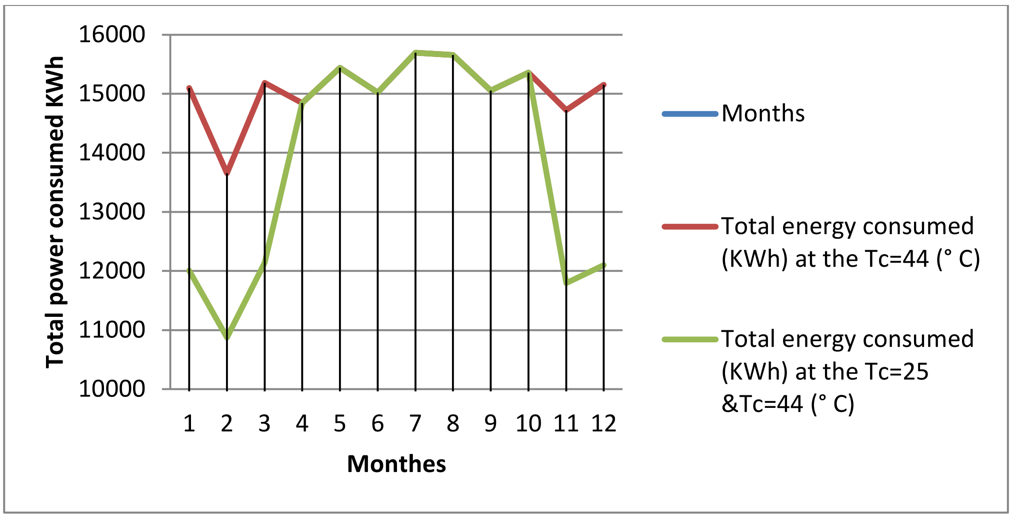
Table 5.6 Results of operation cost analysis with environment temperature variation

PARAMETERS	MONTHS											
	1	2	3	4	5	6	7	8	9	10	11	12
Number of days	31	28	31	30	31	30	31	31	30	31	30	31
Tc (° C)	25	25	25	0	0	0	0	0	0	0	0	20
Tc (° C)	44	44	44	44	44	44	44	44	44	44	44	44
Tair (° C)	1.1	2.7	8.2	15.75	20.5	23.35	27.9	27.35	23.6	16.55	10.15	5.9
Exergy input to compressors (KW) at Tc=25 (° C)	21.08	21.08	21.08	0	0	0	0	0	0	0	21.08	21.08
Exergy input to compressors (KW) at Tc=44 (° C)	26.91	26.91	26.91	26.91	26.91	26.91	26.91	26.91	26.91	26.91	26.91	26.91
Exergy input to condenser fan (KW) at Tc=25 (° C)	0.4349	0.4974	0.6779	0	0	0	0	0	0	0	0.7597	0.6017
Exergy input to condenser fan (KW) at Tc=44 (° C)	0.1445	0.1813	0.2988	0.5731	0.7547	0.9114	1.216	1.15	0.9678	0.6157	0.3567	0.2472
Operation cost of compressors (KWh) at Tc=25 (° C)	11762.64	10624.32	11762.64	0	0	0	0	0	0	0	11383.2	11762.64
Operation cost of compressors (KWh) at Tc=44 (° C)	15015.78	13562.64	15015.78	14531.4	15015.78	14531.4	15015.78	15015.78	14531.4	15015.78	14531.4	15015.78
Operation cost of condenser fan (KWh) at Tc=25 (° C)	242.67	250.69	378.27	0	0	0	0	0	0	0	410.24	335.75
Operation cost of condenser fan (KWh) at Tc=44 (° C)	80.63	91.37	166.73	309.47	421.12	492.15	678.53	641.7	522.61	343.56	192.62	137.94
Total energy consumed (KWh) at Tc=44 (° C)	15096.41	13654.0	15182.51	14840.87	15436.90	15023.56	15694.31	15657.48	15054.01	15359.34	14724.02	15153.72
Total energy consumed (KWh) at Tc=25 &Tc=44 (° C)	12005.31	10875.01	12140.91	14840.87	15436.90	15023.56	15694.31	15657.48	15054.01	15359.34	11793.44	12098.39
Operation cost of evaporator fan ($)					433.6							
C_{op} ($) at Tc=44 (° C) with initial design					15408							
C_{op} ($) at Tc=44 (° C) with temperature variation considered					14903.7							
C_{op} ($) at Tc=44 (° C) &Tc=25 (° C) with temperature variation considered					13712							
$C_{op,saving}$ (%) at Tc=44 (° C) &Tc=25(° C) with respect to temperature variation case					8.00%							
$C_{total, saving}$ (%) at Tc=44 (° C) &Tc=25 (° C) with respect to initial design					11.00%							

5.4 Effect of environment temperature variation on total operation cost

Figure 5.21 shows the comparison between the system is working at 44°C as a condenser temperature throughout the year which shown in red color and at condenser temperature working at 44°C if the environment temperature is more than 15°C and at 25°C condenser temperature if the environment temperature is less than 15 °C which shown in green color. It is observed from the figure that; by system operate at two condenser temperatures causes the decrease in operation cost by ~8% in comparison with system operate at 44°C at environment temperature variation into account but the total operation cost of the system decrease by ~11% in comparison with optimization design.

Figure 5.21 Power consumed with variable temperature considered



5.5 Conclusions

Results of the thermodynamic and cost analyses of cascade refrigeration system are presented. The parameters studied include both condenser and evaporator fan velocities, and isentropic efficiencies of both HTS and LTS compressors. The effects of variable parameters on COPs, total annual cost, second law efficiency, total exergy destruction are presented.

It is concluded that these parameters are very important to take into account for design of cascade refrigeration system in order to avoid the extra energy and cost, and also decrease the environment effect which is desired in any design condition.

CHAPTER 6

CONCLUSIONS

The following conclusions can be drawn based on thermodynamic analysis of cascade refrigeration system presented in previous chapters.

1- The results of the optimization showed that with a constant cooling capacity (i.e., 35 kw), evaporator temperature, condenser temperature, temperature difference in cascade condenser, condenser temperature, ambient temperature and cold space temperature and compromising the above mentioned costs, a minimum was found for the objective function that reduced the annual cost of the system by 31% compared to the medium range selected. It is also demonstrated that while the other parameters were kept constant, the optimum values for decision variables are $V_{fan,evap}$=6 m/s, $V_{fan,cond}$=6 m/s, η_{HTS}=0.85 and η_{LTS}=0.85.

2- By increase of air velocity of HTS condenser the COP_{HTS} and COP_{system} slightly decrease, the operation cost of the system will increase gradually but the cost of purchasing and total annual cost, surface area of HTS air cooled condenser decrease more than twice times value between 1.5 m/s to 6 m/s, increase the total exergy destruction and decrease the second law efficiency.

3- Any increase in air velocity of LTS evaporator the COP_{LTS} and COP_{system} almost decrease, the operation cost of the system will increase gradually but the cost of purchasing and total annual cost, surface area of LTS air cooled evaporator decrease more than twice times value in the selected range 1.5 m/s to 6 m/s, increase the total exergy destruction and decrease the second law efficiency.

4- Increase in LTS isentropic efficiency lead to increase a ~35% in COP_{LTS} and 20% in COP_{system} , decrease the operation cost and purchasing cost, decrease a ~25% in total exergy destruction and the second law efficiency will be increase by ~20%.

5- It can be seen by increase the HTS isentropic efficiency lead to in significant increase in COP_{HTS} and slight increase in COP_{system}, decrease the operation, purchasing and total annual costs, decrease the total exergy destruction by ~24% and increase the second law efficiency.

6- If the cascade system worked in the cold months when T_c=25℃, and worked in the remaining months at T_c=44℃ the total annual operation cost for fans and compressors are 13712$.

7- If the system has worked throughout the year at Tc=44℃, the annual operation cost in a year is 15408$. Extend the use of two expansion valves tied in parallel to give the flow to evaporator for HTS at T_c =25 ° C and T_c=45 ° C with environment temperature variation taken into account the total annual operation cost decreases by 11% and the total annual cost decreased by 2.36. Finally in this study the sum of annual cost saving about 33.22%.

8- Furthermore, cascade refrigeration system with the other place instead the Gaziantep city may be optimized and compared between them, also with other refrigerants couples can be optimized and compared with the present system

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