WASTE HEAT RECOVERY HEAT EXCHANGER USING SUPERCRITICAL CARBON DIOXIDE

A Major Thesis Submitted in Partial Fulfilment of the requirements for the award of the degree of

MASTER OF TECHNOLOGY

IN

THERMAL ENGINEERING

Submitted by:

RASHIN KHERA

(2K15/THE/12)

SESSION: 2015-2017

Under the Guidance of

Dr. K. MANJUNATH

Department of Mechanical Engineering,
DELHI TECHNOLOGICAL UNIVERSITY,
Shahbad Daulatpur, Main Bawana Road, Delhi-110042

DECLARATION

I hereby declare that the work which being presented in the major thesis entitled **"Waste Heat Recovery Heat Exchanger using Supercritical Carbon Dioxide"** in the partial fulfilment for the award of the degree of Master of Technology in **"Thermal Engineering"** submitted to Delhi Technological University (Formerly Delhi College of Engineering), is an authentic record of my own work carried out under the supervision of **Dr. K. MANJUNATH,** Department of Mechanical Engineering, Delhi Technological University (Formerly Delhi College of Engineering). I have not submitted the matter of this dissertation for the award of any other Degree or Diploma or any other purpose what so ever.

Place: Delhi **RASHIN KHERA**

Date**:** **Roll No. 2K15/THE/12**

CERTIFICATE

This is to certify that **RASHIN KHERA**, (Roll no. **2K15/THE/12**), student of M.Tech., THERMAL ENGINEERING, Delhi Technological University, has submitted the dissertation titled **"Waste Heat Recovery Heat Exchanger using Supercritical Carbon Dioxide"** under my guidance towards the partial fulfilment of the requirements for the award of the degree of Master of Technology under my guidance and supervision.

Dr. K. MANJUNATH
Department of Mechanical Engineering
Delhi Technological University
Delhi-110042

ACKNOWLEDGMENT

It is said that gratitude is a virtue. This part is dedicated to special thanks that I would like to deliver to the people who helped me in making the fulfilment of this thesis project possible.

I take great pride in expressing my unfeigned appreciation and gratitude to my learned mentor **Dr. K. MANJUNATH** and Department of Mechanical Engineering, Delhi Technological University (Formerly Delhi College of Engineering), for their invaluable inspiration, guidance and continuous encouragement throughout this project work. His critics and suggestions on my work have always guided me towards perfection. This work is simply the reflection of his thoughts, ideas, concepts and above all his efforts. Working under his guidance has been a privilege and an excellent learning experience that I will cherish for a long time.

I express my deepest gratitude to Prof. R.S. MISHRA**,** Head of Department Mechanical Engineering Delhi Technological University (Formerly Delhi College of Engineering).

RASHIN KHERA

(2K15/THE/12)

ABSTRACT

Common power cycles discard a large portion of useful energy into the environment via exhaust gases. Through the use of supercritical power cycle, this wasted energy may be utilized for power generation and hot water production. Heat transfer between cycles occurs through a heat exchanger. To maximize heat exchanger effectiveness, a supercritical working fluid is used in the supercritical power cycle to better match the heating curve of the sensible heat source. Carbon dioxide is selected as the working fluid because it possesses a relatively low critical temperature which makes it attractive for low temperature waste heat applications. In contrast to many other working fluids, carbon dioxide is inert, abundant, non-flammable, and presents negligible environmental impact. The purpose of this study is to do thermal design of helical coil heat exchanger, second law analysis and Thermo-Economic analysis of the same to gain insight for future research in the field of waste heat recovery.

A program code is established using EES software to perform the calculations required for the waste heat recovery analysis considering real variation ranges of the main operating parameter such as length, diameter of the helical coil and temperature, mass flow rate of supercritical carbon dioxide at inlet condition. The effect of these parameters on system performance are investigated.

By the second law analysis we are able to combine both the effects of heat transfer and pressure drop in a single close form equation by which we are able to know the exact irreversibility occurring in the system which can be used to improve the performance. Thermoeconomic cost will increase as usual with increase of tube bundle length and diameter of the tube but it decreases with increase with increase in the mass flow rate and inlet temperature of SCO2.

The second law efficiency which is also known as rational efficiency is having a maximum value for varying the diameter of the tube which result in optimum value of the diameter. As mass flow rate increases rational efficiency decreases but there is increase in the rational efficiency as inlet temperature of SCO2 increases.

In entropy generation number behaviour also we are able to obtain minimum value for varying the diameter which can obtain optimum diameter corresponding to lower irreversibility. As usual behaviour as the mass flow rate of SCO2 increases entropy generation number increases while entropy generation number decreases by increases in inlet temperature of SCO2.

CONTENTS

LIST OF FIGURES

LIST OF TABLES

NOMENCLATURE

r	Shell radius (m)
L_m	Tube bundle length (m)
a	tube radius(m)
d	Tube diameter (m)
p	Pitch (m)
th	Tube thickness (m)
T	Temperature (K)
$\dot{m}$	Mass flow rate (Kg/s)
P	Pressure (kPa)
Q	Heat duty (W)
C_p	Specific heat (kJ/kg-K)
μ	Viscosity (Pa-s)
ρ	Density (kg/m^3)
k	Conductivity (W/m-K)
N	Number of tube
R_e	Reynold number
Pr	Prandtl number
R_{eff}	Radius of curvature (m)
h	Film coefficient (h)
U	Overall heat transfer coefficient (W/m-K)
A_s	Surface area (m^2)
N_b	Number of rotation of tube bundle
f	Friction factor
δP	Pressure drop (Pa)
Hg	Hagen number

n_{tr}	Number of effective tube bundle
ε	Effectiveness of heat exchanger
Ψ	Rational efficiency
$\dot{E}$	Exergy (W)
$\dot{I}$	Irreversibility (W)
N_s	Overall entropy generation number
Ns_H	Entropy generation number due to heat transfer
Ns_P	Entropy generation number due to pressure drop
i_e	Effective rate of return
TL	Technical life (year)
Φ	Operation maintenance factor
C_s	Cost associated with irreversibility
n_h	Number of hours per year
C_{cost}	Total cost of heat exchanger ($/Year)
$S_{gen,H}$	Entropy generation due to heat transfer (W/K)
$S_{gen,P}$	Entropy generation due to pressure drop
C_e	Cost of equipment (W/K)
R_f	Capital recovery factor
NTU	Number of transfer unit
EF	Exergoeconomic factor
EES	Engineering equation solver
HTTR	High temperature test reactor
AHTR	Advance High temperature reactor
IHX	Internal Heat Exchanger

Subscripts

ex	Exhaust gas in shell side
sco_2	Super critical carbon dioxide in tube side
i	Inner
o	Outer
tub	Turbulent

lam	Laminar
avg	average

Chapter 1

1. INTRODUCTION

1.1 MOTIVATION

Today in gradually changing world energy security, economic development and environment protection are not well balanced. Due to increase in consumption of fossil fuel there is more environmental issues like global warming, atmospheric pollution and ozone depletion. Another reason is due to fast development there may be shortage of energy. By all these reasons waste heat recovery pays more attention. Organic Rankine cycle, Binary Fluid cycle, Supercritical power cycle etc. are used for low grade waste heat recovery. Ecogen power system developed SCO_2 power cycle for waste heat recovery.

According to waste heat recovery report by U.S. DOE, industrial processes in the United States consumes approximately 35 quadrillion BTU(Quads) of energy per year BCS Inc. [1]. This amount is about 30% of total energy consumed in the United States This reports also estimates that around 25-50% of that energy is lost to waste heat BCS Inc. [1]. The report categorizes the waste heat based on the temperature of the waste products. The three waste heat groups are: low, medium, high temperature. Table 1.1 defines the temperature range for each source based on limited samples of industrial application. Table 1.1 also shows the amount of waste heat and work potential of each waste heat group. The waste heat and work potential is based on the reference temperature of 25º C.

Table 1.1: Yearly national unrecovered waste heat [1]

	Temperature Range		Waste Heat (1055 trillion joules per year)		Work Potential (1055 trillion joules per year)
	ºF	ºC	77F [25ºC]	300F [150ºC] *Reference*	77F [25ºC] *Reference*
Low	< 450	< 230	903	37	287
Med	450–1200	230-650	466	130	216
High	>1200	>650	108	89	86
Total	-	-	1478	256	589

This data indicates that low temperature heat source results in 60 % of total waste heat source. It is estimated that 290 trillion BTU per year or 33% of this low temperature heat can be recaptured into useful works. Low temperature heat source recovery presents the largest opportunity to recover among other discarded heat.

1.2 PINCH PROBLEM

A common method to convert waste heat to useful work is through a Combined cycle or Bottoming cycle. The Bottoming cycle may be a gas power or vapour power system in which heat is transfer between cycles via heat exchanger (HX). A popular type of heat exchanger is the heat recovery steam generator (HRSG) which combines an economiser, an evaporator and a super heater. Kehlhofer [2]. Marrero et al. use the steam product of a HRSG to power a bottoming cycle [3]. Utilizing a HRSG, combined power cycles capable of achieving 60 % thermal efficiency have been constructed Siemens Energy [4]. In a HRSG a hot Exhaust gases heats another working fluid from a liquid to a two phase mixture, a saturated vapour or superheated vapour.

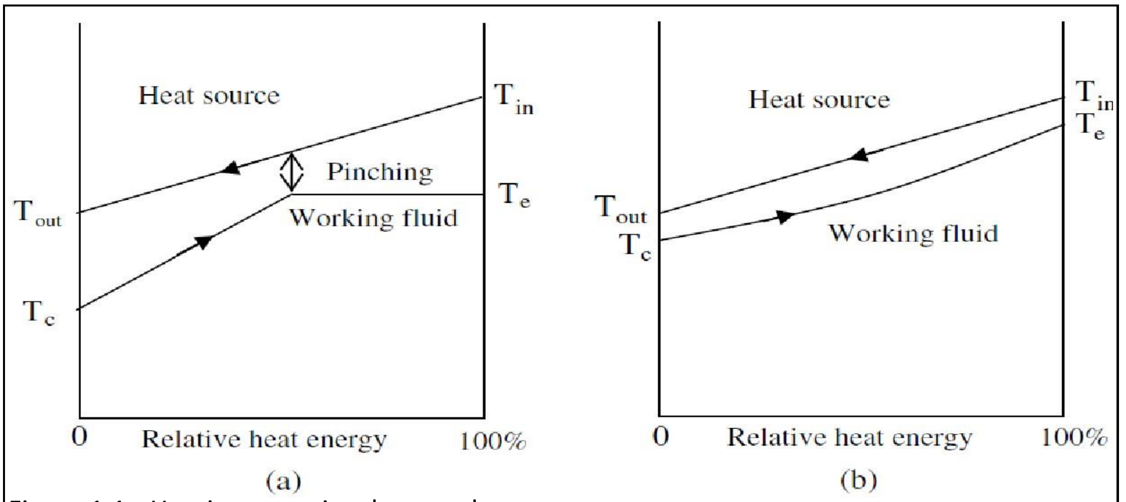


Figure 1.1 : Heating curve in a heat exchanger

Figure 1.1: Schematic representation chart of the heat transfer between waste heat and working fluid in the high temperature main heat exchanger. (a) ORC cycle; (b) CO2 Supercritical power cycle. [5]

Figure 1.1(a) shows an example of the cooling curve in a HRSG. Chen et al. [5].

Heat is supplied to the working fluid as it goes through a phase transition. Remember that temperature is constant of working fluid during phase change. The working fluid undergoes sensible cooling during which temperature continuously decreases. Pinch point is the minimum temperature difference between heat source and working fluid as shown in figure 1.1(a). The existence of Pinch point causes two desirable effects:

1. Heat transfer between two fluids is proportional to temperature difference between them. The temperature difference at the pinch point reduces the effectiveness of heat exchanger. As a result, the minimum heat transfer occurs at the pinch point. This reduces the total amount of heat that can be supplied to the working fluid.

2. In order to prevent the reversal direction of heat transfer, the average temperature difference between the fluids must be larger than the necessary with a single phase fluid (refer to figure 1.1(a)). These relative temperature difference on both side of the Pinch point would result in more Entropy production within the heat exchanger.

A proposed solution to the pinch problem is to use a single phase working fluid that more closely matches the heat source fluid temperature profile, Chen et al. This would result in sensible cooling or a "temperature glide" in the heat exchanger. Supercritical fluids remain in a single phase but compared to gases, have smaller specific volumes and better transport properties Kim et al. [6]. A system using a supercritical working fluid therefore has a relatively low volume to power ratio. This low volume to power ratio requires smaller system components to achieve the same power output. It is proposed by many authors to use supercritical fluids for application to waste heat recovery Chen et al.; Persichilli et al.; Cayer et al.; Velez et al.; Chen et al.; Austin and Sumathy [7-11].

Figure 1.1 (b) shows a schematic of behaviour of supercritical working fluid like CO_2 in a heat exchanger with a sensible heat source. This study will investigate the performance of Rankine cycle using supercritical CO_2 cycle for waste heat recovery.

1.3 Working fluid selection

There are many properties that should be considered when selected working fluid for utilising the energy into low grade heat source and waste heat.

For instances, the critical temperature and critical pressure indicates whether the cycle run as a Transcritical or Supercritical cycle.

Carbon Dioxide has favourable characteristics for the following reasons:

- Moderate critical pressure of 7.4 MPa.
- Relative low critical temperature(31^{o}C) is well suited for low temperature heat source.
- Abundance, non-flammability, non-toxic etc. Cayer et al. [10]
- It is environment-friendly with an ozone depletion potential (ODP) of zero and a global heating potential (GWP) of 1 over 100 years. McQuay Air Conditioning [12]
- Its thermophysical properties are well known even in the supercritical area.
- It is compatible with the standard materials and lubricants and is not harmful to the environment.
- It has potentially favourable thermodynamics and transports properties.
- limited research and information available for CO_2 power cycle with low temperature heat source. Velez et al. [11],

Table 1.2 list critical properties and environmental properties of common refrigerants that can be used as working fluid. [11]

Name	Refrigerant Number	Formula	Critical Temperature C(F)	Critical Pressure MPa (psi)	Ozone Depletion Potential	Global Heating Potential
Ammonia	R-717	NH_3	133 (270)	11.2 (1636)	0	0
Carbon Dioxide	R-744	CO_2	31 (88)	7.4 (1072)	0	1
Water	R-718	H_2O	374 (705)	22.1 (3205)	0	<1
Propane	R-290	$CH_3CH_2CH_3$	97 (206)	4.3 (619)	0	~0
Butane	R-600a	$CH_3CH_2CH_2CH_3$	152 (305)	3.8 (551)	0	~0
	R-22	$CHCIF_2$	96 (205)	5 (722)	0.055	1500

Table 1.2: Critical and environmental properties of common refrigerants.

Compared to other working fluid listed in Table 1.2, Carbon Dioxide has low critical temperature (31^{o}C) and relatively high critical pressure (7.4 MPa). Due to low temperature, a low grade heat source can give a supercritical cycle whose “gliding” temperature profile can give a better match to heat source temperature glide than other working fluid.

Chapter 2

2.1 Literature Review:

Chen *et al*; [13] In the current work, the performance of a CO_2 power cycle in utilizing the low-grade heat sources is simulated and the results are analysed with a focus on second law thermodynamics (i.e. exergy and entropy). Different system parameters that influencing the system exergy and entropy change are discussed. Engineering Equation Solver (EES) is used for simulation. The simulation results show that the matching of the temperature profiles in the system heat exchangers has crucial influences on their exergy destructions and entropy generations. It is also an essential factor that influences the system. They found that remaining other system working conditions and the heat source inlet conditions constant, while changing the CO_2 mass flow rate, it can be noticed that the exergy destruction and the entropy generation in the expander and the pump are almost constant with an increasing CO_2 mass flow rate. Furthermore, the exergy destruction and the entropy generation are increasing in the system gas heater, but decreasing in the gas cooler & condenser. The contributions of entropy generations from the expander and the pump are and are almost independent on the CO_2 mass flow rate. On the contrary, the contributions of the entropy generations by the heat exchangers are high and is decreasing from the gas cooler & condenser, while increasing from the gas heater with an increasing CO_2 mass flow rate. Moreover, the system thermodynamic efficiencies (thermal efficiency and exergy efficiency) are slightly decreasing with the increasing CO_2 mass flow rate.

Wang *et al*; [14] The heat transfer characteristics of supercritical CO_2 cooled in the helically coiled tube are investigated experimentally and numerically. The inner diameter, coil pitch and coil radius of the helically coiled tube are 4 mm, 34 mm and 36 mm, respectively. The effects of heat flux, pressure and mass flux on the heat transfer are analysed based on 512 sets of experimental data, a new modified correlation is developed to calculate the heat transfer coefficients of supercritical CO_2 in the cooled helically coiled tube. To study the buoyancy effect, the heat transfer coefficients of upward flow are compared to that for horizontal flow, and it is found that the effect of buoyancy is negligible in the liquid-like region, but significant in other region under the experimental conditions. On this basis, the three buoyancy parameters: Bup, Grth/Grq and Ri are applied to predict the effect of buoyancy on the heat transfer. The three parameters overestimate the impact of buoyancy. Numerical analysis of the cooling heat transfer coefficient in helically coiled tube is conducted by using shear-stress transport (SST) model. The flow fields are analyzed and the effect of heat flux is mainly related to the distribution of specific heat (cp) in the radial direction of tube transverse section.

Jayachandriah *et al*; [15] They did the Fabrication of spiral coil heat exchanger by taking the outer diameter of the shell as 159.16 mm and inner diameter of the shell is 116 mm, outer diameter of the tube is 9.21 mm and inner diameter of tube is 8.91 mm, the coil pitch is taken as 24.11 mm, number of coil turns is taken is 14 and shell material is steel, tube material as copper and assembled, then the experiment is done by varying the mass flow rate of cold water and mass flow rate of hot water kept constant for first time and by varying the mass flow rate of hot water and mass flow rate of cold water kept constant then the values of effectiveness and LMTD was evaluated. They found that by changing the pitch lay out rate of heat transfer can be improved. By changing the temperature of tubes and medium rate of heat transfer can be improved. By changing the materials of tubes heat transfer rate can be improved.

Prabhanjan *et al*; [16] They determined the relative advantage of using a helically coiled heat exchanger versus a straight tube heat exchanger for heating liquids. The particular difference in this study compared to other similar studies was the boundary conditions for the helical coil. Most studies focus on constant wall temperature or constant heat flux, whereas in this study it was a fluid-to-fluid heat exchanger. Results showed that the heat transfer coefficient was effected by the geometry of the heat exchanger and the temperature of the water bath surrounding the heat exchanger. All tests were performed in the transitional and turbulent regimes. Use of a helical coil heat exchanger was seen to increase the heat transfer coefficient compared to a similarly dimensioned straight tube heat exchanger. Both heat exchanger had higher heat transfer coefficients when the bath temperature was increased, most probably due to increased buoyancy effects. Flow rate did not effect the heat transfer coefficient, most likely from the fact that the flow was turbulent and increasing the flow rate does not greatly change the wall effects. Temperature rise of the fluid was found to be effected by coil geometry and by the flow rate.

Coronel *et al*; [17] This study involved the determination of pressure drop and fiction factor (f) in helical heat exchangers under turbulent flow conditions. The experiments were conducted in helical heat exchangers, with coils of two different curvatures ratios (d/D = 0.114 and 0.078) at various flow rates (9.46 x 10^{-5} - 6.31 x 10^{-4} m^3/s) and end-point temperatures (20, 93.3, 121, 149C). The computed friction factor (f) in the helical heat exchanger was compared to published correlations, and it was found that the experimental data was in good agreement with them. In addition, correlations to determine pressure drop based on the Reynolds number, curvature ratio, and temperature were developed.

Dostal *et al*; [18] performed a systematic, detailed major component and system design evaluation and multiple parameter optimization under practical constraints of the family of supercritical CO2 Brayton power cycles for application to advanced nuclear reactors. The recompression cycle is shown to excel with respect to simplicity, compactness, cost and thermal efficiency. They found that the supercritical CO2 cycle is well suited to any type of nuclear reactor with core outlet temperature above ~ 500 oC in either direct or indirect versions. Intercooling, re-heating, re-compressing and pre-compressing were considered. The recompression cycle was found to yield the highest efficiency, while still retaining simplicity. Inter-cooling is not attractive for this type of cycle as it offers a very modest efficiency improvement. Re-heating has a better potential, but it is applicable only to indirect cycles. Economic analysis of the benefit of re-heating for the indirect cycle showed that using more than one stage of re-heat is economically unattractive. They found that thermal efficiency of the advanced design is close to 50% and the reactor system with the direct supercritical CO2 cycle is ~ 24% less expensive than the steam indirect cycle and 7% less expensive than a helium direct Brayton cycle.

Chen *et al;* [19] conducted the study that a CO2 transcritical power cycle shows a higher potential than an ORC when taking the behaviour of the heat source and the heat transfer between heat source and working fluid in the main heat exchanger into account. It is due to better temperature glide matching between heat source and working fluid. The CO2 cycle does not show any pinch limitation in the heat exchanger. They found that due to temperature gradients for the heat source and heat sink the thermodynamic mean temperature had been used as a reference temperature when comparing both the cycles. They developed the thermodynamic models in EES and calculated the relative efficiencies for both cycles. They obtained that when utilizing the low-grade waste heat with the same thermodynamic mean heat rejection temperature, a transcritical carbon dioxide power system gives a slightly higher power output than the organic Rankine cycle. They also found that the power system with carbon dioxide as a working fluid is also more compact and more environmental friendly than the one with organic working fluid as a working media.

Naphon [20] studied the thermal performance and pressure drop of the helical-coil heat exchanger with and without helical crimped fins. The heat exchanger consists of a shell and helically coiled tube unit with two different coil diameters. Each coil is fabricated by bending a 9.50 mm diameter straight copper tube into a helical-coil tube of thirteen turns. Cold and hot water are used as working fluids in shell side and tube side, respectively. The experiments are done at the cold and hot water mass flow rates ranging between 0.10 and 0.22 kg/s, and between 0.02 and 0.12 kg/s, respectively. The inlet temperatures of cold and hot water are between 15 and 25 °C, and between 35 and 45 °C, respectively. The cold water entering the heat exchanger at the outer channel flows across the helical tube and flows out at the inner channel. The hot water enters the heat exchanger at the inner helical-coil tube and flows along the helical tube. He found that Outlet cold water temperature increases with increasing hot water mass flow rate. An average heat transfer rate increases as hot and cold water mass flow rates increase. The friction factor decreases with increasing hot water mass flow rate. Inlet hot and cold water mass flow rates and inlet hot water temperature have significant effect on the heat exchanger effectiveness.

Sarkar; [21] conducted exergetic analyses and optimization of S-CO2 recompression cycle have been performed to study the effect of operating parameters on the optimum pressure ratio, energetic and exergetic efficiencies and component irreversibilities. Effect of isentropic efficiency, recuperator effectiveness and component pressure drop on the second law efficiency is presented as well. Results show that the effect of minimum operating temperature on the optimum pressure ratio and cycle efficiencies is more predominant than the maximum operating temperature, whereas the effect of maximum cycle pressure is significant only for lower values and the optimum pressure ratio leads to near critical minimum cycle pressure. Result shows that the irreversibilities of heat exchangers are significantly more compared to that of turbo-machineries and the effect of operating parameters on irreversibility is also more significant for recuperators compared to turbo-machines. Effect of isentropic efficiency of turbine is more predominant (about 2.5 times) than that of compressors and effect of high temperature recuperator (HTR) effectiveness is more predominant (about double) than that of low temperature recuperator (LTR) on the second law efficiency. Effect of pressure drop in reactor is more significant compared to others components on the second law efficiency reduction. Due to significant effect on the specific heat capacity variation near the critical point, the effect of minimum operating temperature on the optimum compressor pressure ratio and cycle efficiencies is more predominant than the maximum operating temperature. Result shows that the effect of operating conditions is more significant on the recuperator irreversibility compared to turbo-machine irreversibility.

Kharat *et al*; [22] developed a Correlation for heat transfer coefficient for flow between concentric helical coils. Existing Correlation is found to result in large discrepancies with the increase in gap between the concentric coils when compared with the experimental results. In the present study experimental data and CFD simulations using Fluent 6.3.26 are used to develop improved heat transfer coefficient correlation for the flue gas side of heat exchanger. Mathematical model is developed to analyse the data obtained from CFD and experimental results to account for the effects of different functional dependent variables such as gap between the concentric coil, tube diameter and coil diameter which affects the heat transfer. Optimization is done using Numerical Technique and it is found that the new correlation for heat transfer coefficient developed in this investigation provides an accurate fit to the experimental results within an error band of 3–4%.

Mandal *et al*; [23] conducted experiments for the first time in tube in tube helical heat exchanger at the pilot designt scale to investigate the fluid flow and heat transfer under turbulent flow conditions. The experiments were carried out with hot compressed air in the inner tube and cooling water in the outer tube in the counter current mode of operation. heat transfer calculations were based on physically realistic condition of fluid to fluid heat transfer. The inner tube Nusselt number of compressed air in the present study was found to be slightly higher than the data reported in the literature for ambient conditions while the friction factor values are within the range reported in the literature for ambient conditions. The friction factor as well as Nusselt number calculated for cooling water flowing in the outer tube was higher than the existing literature data. On the basis of the experimental results, new correlations for friction factor and Nusselt number in the inner as well as outer tubes were developed

Kocatepe *et al*; [24] Characteristics of heat transfer and pressure drop of a helically coiled corrugated flex tube in a heat exchanger were experimentally investigated in this study. A corrugated flex tube, 13 m long and 0.0254 m in diameter, was helically coiled in the heat exchanger. Water was selected as working fluid. Hot water flows through the tube, and cold water in the boiler is either stationary or in motion by a pump. It was found that the ratio of the friction coefficient of the present setup to that of the straight corrugated tube appears to increase as the flow rate decreases. The overall heat transfer coefficient increases only slightly as the flow rate inside the tube increases. They found that the corrugated tube considerably enhances heat transfer rate per unit length of the tube, compared to a smooth tube.

Tuo; [25] A reheat CO_2 transcritical power cycle with two stage expansion is presented to improve baseline cycle performance in this paper. First law analysis is carried out to investigate parametric effects on reheat cycle performance. The main results show that reheat cycle performance is sensitive to the variation of medium pressures and the optimum medium pressures exist for maximizing work output and thermal efficiency, respectively. Reheat cycle is compared to baseline cycle under the same conditions. More significant performance improvements by reheat are obtained at lower turbine inlet temperatures and higher maximum cycle pressure. Work output improvement is much higher than thermal efficiency improvement, because extra waste heat is required to reheat CO_2, which reduces the thermal efficiency. It is found that reheat cycle has great potential to improve thermal efficiency and especially work output of a CO_2 transcritical power cycle using a low-grade heat source.

Kacludis *et al***;** [26] They presented an overview on three exemplary applications: combined cycle gas turbines using a sCO_2-based bottoming cycle, bottoming cycle for a reciprocating engine generator sets, and waste heat to power (WH2P) from energy-intensive manufacturing processes. Supercritical CO_2 heat engines are scalable across a broad system size range − from 250kWe to 45MWe and above, with net electrical output to support the widest possible variety of industrial and utility-scale applications. The sCO_2 Cycle is thermal source agnostic − suitable with a wide range of heat sources from 400°F to 1000+°F with efficiencies up to 30 percent depending on the heat source. New energy production can be offset with recovered energy without increasing associated greenhouse emissions while improving overall energy production efficiency. The sCO_2 heat engine can add up to 35% more power to simple cycle gas turbines, 10% to 15% more power to reciprocating engines, and can significantly improve the energy efficiency and bottom line performance at steel mills, cement kilns, glass furnaces and other fuel-fired industrial processes by converting previously wasted exhaust & flue gas energy into usable electricity.

Shirgire *et al*; [27] determined the relative advantage of using a helically coiled heat exchanger against a straight tube heat exchanger. It was found that the heat transfer in helical circular tubes is higher as compared to Straight tube due to their shape. Helical coils offer advantage over straight tubes due to their compactness and increased heat transfer coefficient. The increased heat transfer coefficients are a consequence of the curvature of the coil, which induces centrifugal forces to act on the moving fluid, resulting in the development of secondary flow. The fluid to fluid heat exchange was taken into consideration, most of the investigations on heat transfer coefficients was for constant wall temperature or constant heat flux. The result shows that the heat transfer coefficient is affected by the geometry of the heat exchanger. Helical coil heat exchanger is superior in all aspect studied here. The effectiveness of heat exchanger greatly affected by hot water mass flow rate and cold water flow rate. When cold water mass flow rate is constant and hot water mass flow rate increased the effectiveness decreases. Increase in cold water mass flow rate for constant hot water mass flow rate resulted in increase in effectiveness. For both helical coil and straight tube heat exchangers with parallel and counter flow configuration this result obtained. Helical coil counter flow is most effective in all these conditions and straight tube parallel flow heat exchanger is least effective.

Manjunath *et al*; [28] entropy generation analysis of an air-cooled natural draft wire-and-tube condenser commonly used in the domestic refrigerator is carried out to investigate geometrical and operating parameters. The Bejan number formulation is

used for the analysis because of its ability to relate the heat transfer to pressure drop entropy generation. The analytical study is carried out by varying geometrical parameters of the condenser such as tube outer diameter, wire diameter, number of wire pairs, number of tube rows, tube pitch, wire pitch and refrigerant (R134a) properties such as mass velocity, saturation temperature and dryness fraction

Anderson *et al;* [29] proposed a novel energy recovery device based on a SCO2 regenerative Rankine cycle for small-scale (1kW to 5kW) heat recovery from low temperature (200°C - 500°C) sources using small mass flow rates (20 – 60 grams/sec). They presented a prototype of cycle including details of key system components. They obtained result as recovery from low temperature sources approximately 200°C and above is feasible with an operating efficiency of 11%. Another result is that the overall thermodynamic efficiency of the system is highly dependent on the internal heat exchanger effectiveness.

Nassar *et al*; [30] developed a flexible design system which is starting from heat balance calculation, continues with sizing of turbomachinery flow path, through 1D/2D/3D aero and structural multidisciplinary optimization. They analysed four different embodiments of S-CO2 thermodynamic cycles using assumed component efficiencies and based on the actual design of the turbomachinery components. The cycle was again calculated and accurate performance of the cycle was predicted. They found that the turbine efficiency had significant influence on the overall cycle performance compared to the compressor efficiency. They found that simulation in AxCYCLE of different S-CO2 cycle embodiments, such as the recompression cycle for Concentrated Solar Power, nuclear reactor, simple recuperated Brayton cycle and cycle with sequential cooling and compression for waste heat showed that this tool allows quick and precise calculations of the cycle performance in a wide variety of schemes. They also observed that AxSTREAM allows selection of appropriate turbine type, main compressor type as well the recompression compressor type in a quick and efficient manner providing multi-parameter analysis during preliminary design.

Andhare *et al*; [31] In the present work the convective heat transfer coefficients of a helical coil heat exchanger are investigated experimentally. Three helical coils of different curvature ratio and pitch are arranged horizontally in a shell and are tested for counter flow arrangement. Hot water is made to flow through the helical coil and the cold water through the shell. The tube side and shell side flow rates were altered and appropriate instruments were used to measure the flow rates and temperatures of both the fluids. Tube side and shell side convective heat transfer coefficients were calculated using Wilson plots. Based on the curvature ratio and pitch ratio separate empirical correlations are proposed for tube side and shell side for 75 test runs. They found that the shell side heat transfer coefficients are larger than the tube side heat transfer coefficients considering the pitch ratio and curvature ratio.

Manjunath *et al*; [32] conducted comparison of a constructal heat exchanger (CHE) and normal heat exchanger (NHE) is analyzed by using second law analysis. Analysis is carried out by considering the three irreversibilities due to heat transfer, pressure drop, and production of the materials and the construction of the heat exchanger. The thermo-economic aspect of the heat exchanger is considered to further analyze the economic differences between the CHE and NHE. They found that there is an increase in the performance and a cost reduction in the CHE when compared to the NHE. They concluded that by considering the same heat transfer rates and temperature difference between balanced streams, the entropy generation number due to heat transfer decreases. The entropy generation numbers due to pressure drop and material irreversibility have a lower value for the CHE. This is due to an increase in effectiveness and the NTU in the case of the NHE as compared to the CHE. There is a decreasing behavior of the thermo-economic cost ratio for an increase in the number of pairing levels which indicates the advantages of the CHE over the NHE.

Manjunath *et al*; [33] Heat exchangers are thermal systems which are used extensively, have a major role in energy Conservation aspect and preventing global heating. This paper is based on reviews of scientific work and provides a state-of-the art review of second law of thermodynamic analysis of heat exchangers. Initially, the basics of heat exchangers are briefly provided along with second law analysis which also includes two-phase flow analysis and thermoeconomic analysis. Following this, detail literature survey based on performance parameters such as entropy generation, exergy analysis, production and manufacturing irreversibilities (cumulative exergy destruction associated with the production of material and manufacturing of component or assembly of components) and two phase fluid loss of heat exchangers is presented including constructal law applied to analyse heat exchangers. Constructal theory along with second law analysis can be used for the systematic design of heat exchangers. This review high lights the importance of first and second law investigations of heat exchangers leading to energy conservation. They found that the entropy generation analysis is preferred over exergy analysis, as entropy generation is a process quantity and does not need reference temperature. While, exergy analysis is an availability function consisting of enthalpy term, which is combination of state variable and process variable. Analysis of heat exchangers based on entropy generation as performance parameter has been carried out in terms of non-dimensional numbers in different ways. The entropy generation rate equation is divided by different terms like heat capacity rate, the ratio of heat transfer rate with reference or fluid inlet temperature, the ratio of heat transfer rate with fluid temperature difference, reference entropy generation rate and maximum entropy generation rate. Also, the analysis has been carried out in terms of different ways of defining entropy generation numbers like irreversibility distribution ratio, Bejan number and second law efficiency terms, enthalpy exchange irreversibility norm, quality of energy transformation and relative entropy generation.

Ahn *et al;* [34] conducted a review of supercritical CO2 power cycle technology and current status of research and development. In this paper the current development progress of the S-CO2 cycle is introduced and a quick comparison of various S-CO2 layouts is presented in terms of cycle performance. They found that the S-CO2 cycle can achieve relatively high efficiency within the mild turbine inlet temperature range (450-600 degree centigrade) compared with other power conversion systems. The main benefit of the S-CO2 cycle is the small size of the overall system and its application includes not only the next generation nuclear reactors but also conventional water-cooled reactors, coal power designts, and several renewable energy sources. To evaluate the S-CO2 cycle performance, various countries constructed and demonstrated S-CO2 integral system test loops and similar research works are ongoing in Korea as well. However, to evaluate the commercial S-CO2 power systems, development of a large scale (> 10 MW) prototype S-CO2 system is necessary. The research activities are focused on a large scale S-CO2 power system and various foreign research institutions and Korean researchers are attempting to realize the future power system that can significantly transform the energy industry around the world.

Mandore *et al*; [35] They dealt with the performance improvement of the helical tube in TCHE with parallel and counter flow configuration of various correlations with specific data. The readings of mass flow rate and temperature difference of hot oil fluid and water are recorded. Logarithmic Mean Temperature Difference LMTD, capacity ratio, universal heat transfer coefficient and effectiveness are calculated and compared for Parallel flow and Counter flow configuration The increased in the intensity of secondary's flow developed in fluid flow it increases Nu, which gives effective value in proposed helical coil in TCHE in counter flow configuration. They found that Designed Helical coil in coil heat exchanger in counter flow configuration is 1.27 times effective than the Helical coil in coil heat exchanger in parallel flow configuration.

Seyyedvalilu *et al*; [36]; They investigated numerically on the influence of different parameters such as coil radius, coil pitch and diameter of tube on the hydrodynamic and heat transfer characteristics of helical double tube heat exchangers using the CFD software which is based on the principles of heat transfer, fluid mechanics and thermodynamics. The results indicated that heat transfer augmentation occurs by increasing of the inner Dean Number, inner tube diameter, curvature ratio and by the reduction of the pitch of heat exchanger coil. By increasing the radius of coils, the secondary flow effects due to centrifugal forces diminishes and flow of fluid through the coils tends to flow in a straight path and as a result, the friction coefficient decreases consequently. Increasing the pitch of heat exchanger, leads to decrease of overall heat transfer coefficient of that. But this effect is negligible. So it can be concluded that, tube pitch is a parameter that doesn't have a great influence on analysis trend. Increasing of number of coils means longer heat transfer path and observed decrease in Nusselt Number due to number of coils and this decrease in Nusselt Number due to number of coils indicates that flow is thermally developing and in large numbers the flow will be fully developed.

Manjunath *et al*; [37] conducted analytical analysis of unbalanced heat exchangers to study the second law thermodynamic performance parameter through second law efficiency by varying length-to-diameter ratio for counter flow and parallel flow configurations. The study is carried out by giving special influence to geometric characteristics like tube length-to-diameter dimensions; working conditions like changing heat capacity ratio, changing the value of maximum heat capacity rate on the hot stream and cold stream separately and fluid flow type, i.e., laminar and turbulent flows for a fully developed condition. Further, second law efficiency analysis is carried out for condenser and evaporator heat exchangers by varying the effectiveness and number of heat transfer units for different values of inlet temperature to reference the temperature ratio by considering heat transfer irreversibility. They found that The total rational efficiency, which also includes the irreversibility due to pressure drop,has a lower value as compared to rational efficiency defined by the heat transfer term only and having an increasing trend, attaining maximum value and then showing decreasing behaviour.

Zhao *et al*; [38] studied Supercritical CO2 Brayton cycles (SCO2BC) including the SCO2 single-recuperated Brayton cycle (RBC) and recompression recuperated Brayton cycle (RRBC), and flexible thermodynamic and economic modeling methodologies was presented. The main purpose of this paper is to study the thermodynamic and economic performance of SCO2BC, including RBC and RRBC, and its optimization. The effects of the key parameters on the cycle thermodynamic performance are investigated, and the comparative analysis on RBC and RRBC is conducted. Based on the thermodynamic and economic models, the Pareto-based multiobjective optimization is conducted for RRBC, with the maximum exergy efficiency and the lowest cycle cost as its objectives. After that, ANN is selected to establish the relationship of the key cycle parameters and accelerate the parameters query process.

Xu *et al*; [39] An experimental investigation on the heat transfer characteristics of supercritical CO2 during gas cooling process in a helically coiled tube is conducted. The experimental data are obtained over a mass flux range of 79.6–238.7 kg m−2 s−1, an inlet pressure range of 7.5–9.0 MPa and a mean bulk temperature of 23.0–53.0 °C. The effects of mass flux, bulk temperature and pressure on the heat transfer coefficient for helically coiled tubes are investigated. A comparative analysis of the gravitational buoyancy and the heat transfer coefficient is carried out between helically coiled tubes and straight tubes. A new heat transfer correlation of the supercritical CO2 in the horizontal helically coiled tube is proposed based on the experimental data. The maximum error between the predicted results of the new correlation and the experimental data is 20%.

Rao *et al*; [40] They provide an overview of the published studies that are relevant to the flow behaviour and heat transfer characteristics of scCO2. The review of available works displays that the heat transfers and pressure drop characteristics of heat exchangers using scCO2 as working fluid rely on many parameters such as tube shape and size, mass flux, inlet temperature and pressure, type of process (heating or cooling) etc. Moreover, application of scCO2 in cooling and heating processes and the available empirical correlations for heat transfer are also discussed. They concluded that in the cooling process, the heat transfer coefficient decreased as the system inlet pressure increased above the critical pressure. The heat transfer coefficient also decreased when the mass flux is increased. On the other hand, in most researches it was observed that the smaller the inner tube diameter the higher the heat transfer coefficient will be. However, few research works showed that the Nusselt number decreased when the inner tube diameter is decreased. The pressure drop in a system decreased when the system inlet pressure is increased. However, it increased with increasing mass flux.

Vishvakarma *et al*; [41] They presented a brief review of heat transfer through helical coil heat exchangers. They proposed that heat transfer characteristics of double pipe helical heat exchangers are available in the literature, there exists no published experimental or theoretical analysis of a helically coiled heat exchanger considering fluid-to-fluid heat transfer, which is the subject of this work. After validating the methodology of CFD analysis of a heat exchanger, the effect of considering the actual fluid properties instead of a constant value is established. Heat transfer characteristics inside a helical coil for various boundary conditions are compared. It is found that the specification of a constant temperature or constant heat flux boundary condition for an actual heat exchanger does not yield proper modelling. Hence, the heat exchanger is analysed considering conjugate heat transfer and temperature dependent properties of heat transport media. They found that the effectiveness of helical coil heat exchanger is found to be higher when compared to that of the straight tube heat exchanger for all the inlet temperatures.

Sharma *et al*; [42] conducted analysis of a waste-heat-recovery-system (WHRS) using supercritical carbon dioxide (SC-CO2) based regenerative recompression Brayton cycle (RRCBC) for shipboard applications following energy and exergy analyses. The influence of key operating parameters such as, the gas compositions, turbine and compressor inlet temperatures, pressure drop irreversibility, pressure ratio etc., on the overall performance of the system including the exergy destruction rate has been investigated. The results show that the proposed integration improves the overall efficiency of the system by 10% while the net power is found to be increasing up to 25% of the rated power. It is also found that the topping gas turbine exhaust gas compositions and temperatures have a significant influence on the WHRS performance. It is found that the exergy destruction in the heat exchangers is significantly higher which is about four times higher as compared to that in the turbomachinery. From the exergy balance, it is found that the precooler, HRHE and the regenerators are the most important components from the exergetic performance point of view. Further, it is found that there is an optimum pressure ratio at which the cycle attains the maximum efficiency for a typical set of operating parameter. Again, the variation in any of the operating parameters, will result in the significant changes not only in the optimum pressure ratio but also the corresponding thermal and exergetic efficiency of the cycle.

2.2 Research Gap:

From the literature review following are the gaps in the literature which is having scope for this research work.

1. Very less research has been carried out for waste heat recovery by using supercritical carbon dioxide.
2. There is wide scope for second law analysis of heat exchanger which uses entropy generation number as performance parameter for heat exchanger analysis which is used in waste heat recovery.
3. There is less research is carried out for helical heat exchanger which can be used for exhaust waste heat recovery in the application of Diesel generators, Marine gas turbine, Nuclear power generation, Gas turbine power generation etc.
4. There is no literature foundwhich uses second law thermoeconomic analysis applied for waste heat recovery heat exchanger which uses carbon dioxide as working fluid

Chapter 3

System Description

3.1 Helical Coil Heat Exchangers:

3.1.1Advantages:

Because of the conservative structure and high heat exchange coefficient, helical loop heat exchangers find broad use in mechanical applications, for example, power generation, atomic industry, process designts, heat recuperation frameworks, refrigeration, the food industry, and so forth. Abdulla; Bai et al.; Futagami and Aoyama ; Jensen and Bergles; Patankar et al.; Xin et al. [43-48].

1. The helical stream way gives higher shear rates and turbulence at a given pressure drop, which can bring about film coefficients up to 40% higher than those accomplished with numerous equivalent shell and tube units.
2. Helical geometry grants handling of high temperatures and outrageous temperature differentials without high initial stress or expensive extension joints.
3. High-pressure capacity and the capacity to completely clean the service liquid stream region add to the exchanger's preferences.
4. Cleaning the case side stream area is easily managed. The case can be unbolted and the whole case get together evacuated for cleaning, examination or substitution.
5. The coil design's smallness additionally gives benefit, on the grounds that the exchanger requires insignificant floor space.
6. The high shear stress and incited turbulence of helically coiled exchangers decrease the inclination for fouling.
7. Economical unit choice is conceivable because of approved film coefficients, full utilization of accessible LMTD and insignificant required thicknesses.

3.1.2 Various Arrangements:

Although different arrangements are accessible, the essential and most normal design comprises of a progression of stacked helically coiled tubes. The tube closes are associated with manifolds, which go about as liquid passage and leave areas. The tube package is built of various tubes stacked on each other, and the whole package is put inside a case, or shell (Fig. 3.1).

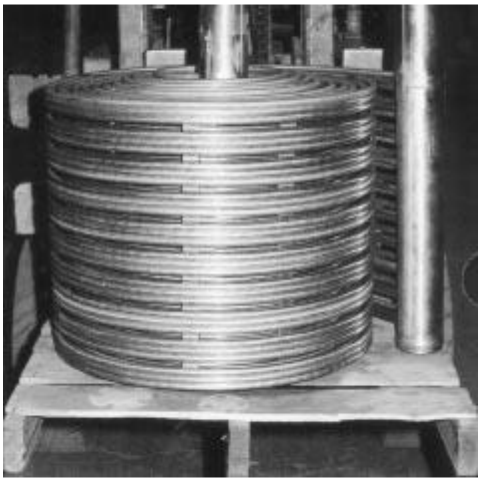


Fig. 3.1: Bundle for a water vaporizer in a supercritical water-extraction process. [49-52].

A simple thermal design of helical coil heat exchanger has been conducted for AHTR. Figure 3.2 and 3.3 show the schematic of the reference helical coil heat exchanger and the tube bundles used in this design. This heat exchanger is intermediate heat exchanger for HTGR IHX. Kato et al. [53]

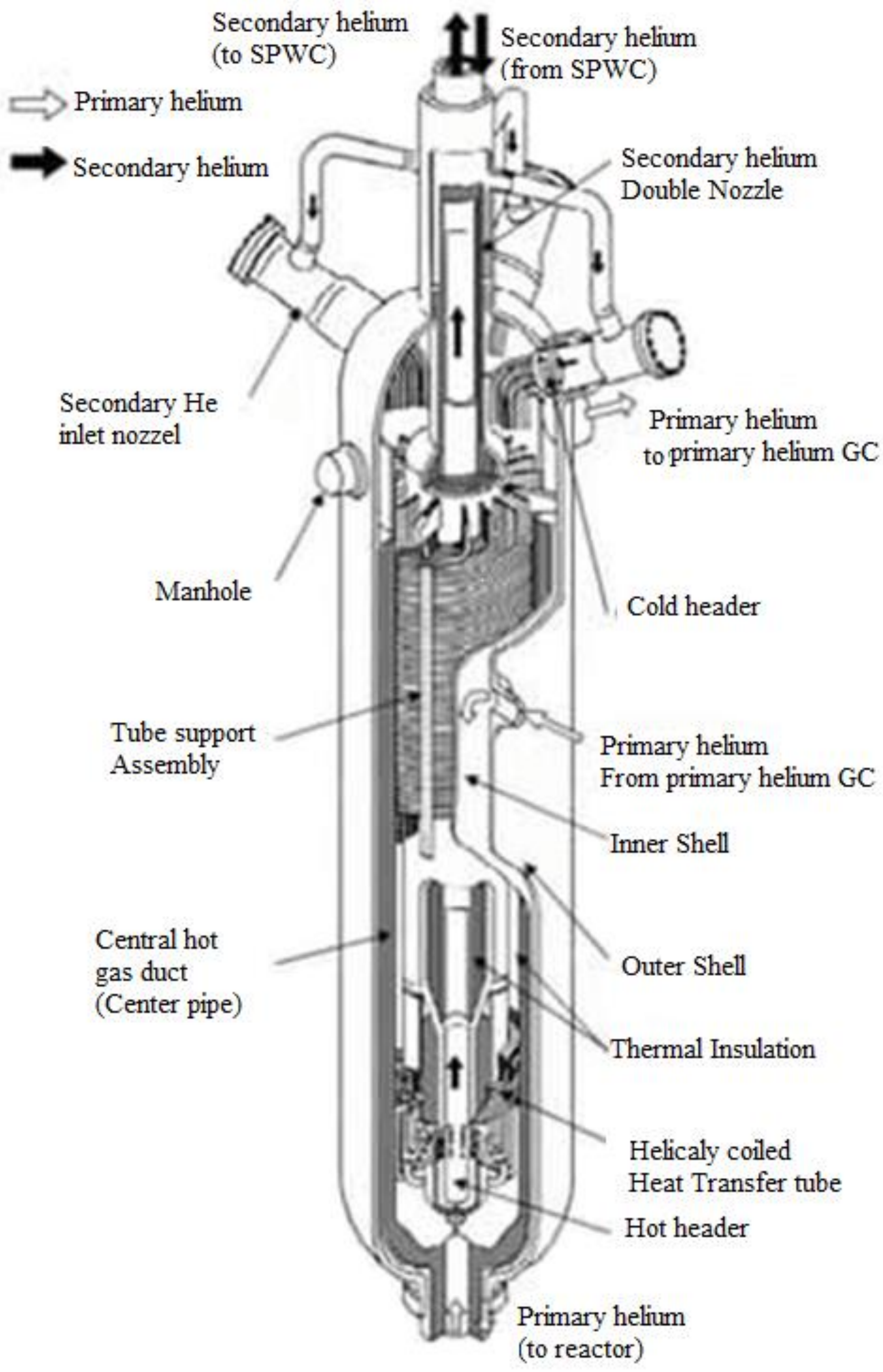


Fig 3.2 - Schimatic of HTTR IHX [54]

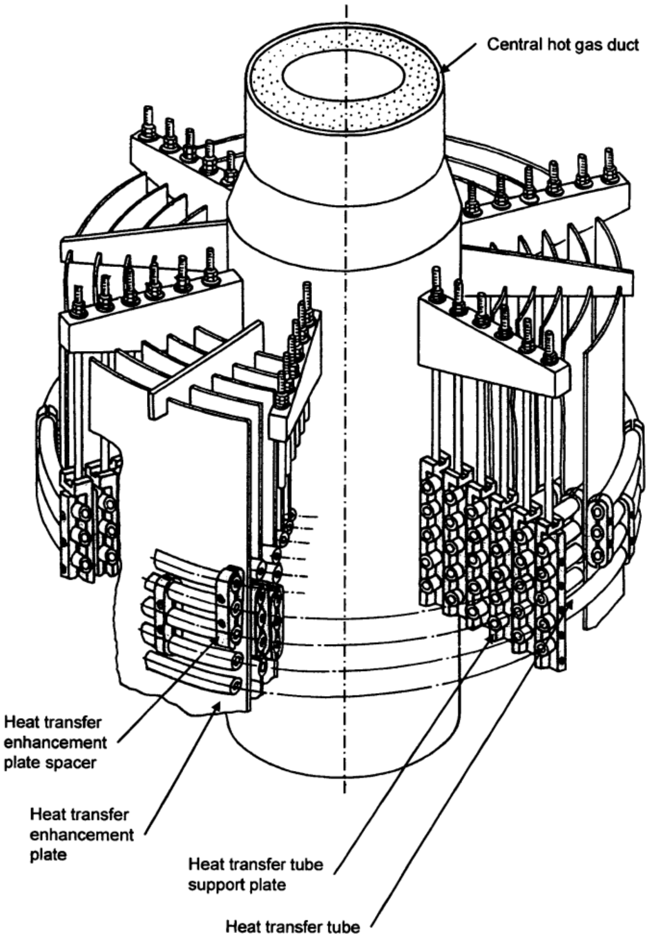


Fig. 3.3 Structural diagram of heat transfer tube bundle of HTTR [54]

3.1.3 Limitations

There are not very many constraints for the utilization of helically coil heat exchangers. For the most part, a pressure farthest point of 10,000 psig covers the larger part of utilizations. Temperature limits are controlled by development materials, similar to the erosion rates. Surface zones of 1 to 650 sq. ft. are accessible, and utilizing units in arrangement or parallel may broaden this range generously.

3.2.4 Applications

The utilization of helically coil exchangers keeps on expanding. Applications incorporate fluid heating/cooling, steam radiators, vaporizers, cryogenic cooling and vent consolidating. Below are the points of interest for standard services in which helical exchangers are considered.

• Sample Cooling

• Analyser Pre-cooling

• Seal Coolers

• Condensers

• Cryogenic Vaporizers

• Compressor Inter- and After-Coolers

• General Applications

3.2 Echogen's Technology

Echogen is doing commercialisation of waste heat recovery power system. The Echogen engine consists of five parts condenser, pump, turbine, exhaust and recuperator heat exchanger. System checking and control are done by valve and sensors. (Figures 3.4 and 3.5).

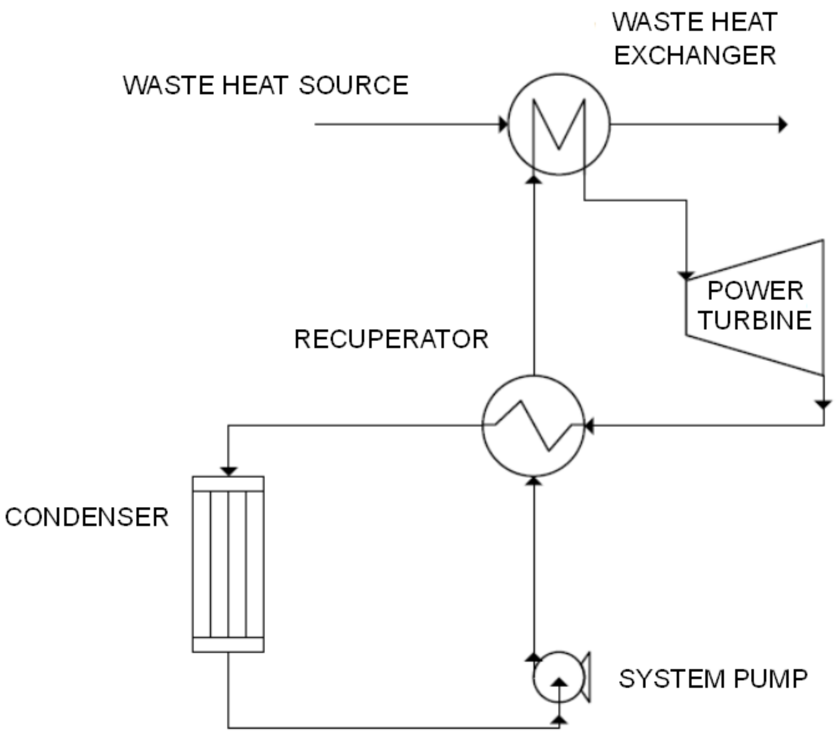


Figure 3.4: Simplified process flow diagram for the Echogen heat engine [55]

Subordinate parts (valves and sensors) give system checking and control. Waste heat via waste heat exchanger comes in exhaust stack of gas turbine or other outside heat sources with 200C to more prominent than 540C working temperature go (400F to more prominent than 1000F). Echogen's innovation can give incorporated power, heating, and additional cooling through an adaptable framework design, that can be arranged for control, cogeneration or trigeneration.

SCO_2 engine can be used for 250kWe to 50MWe industrial application with efficiencies up to 30 percent. This Echogen engine can be used for various application like gensets, industrial waste heat, internal combustion engine etc.[55-57]

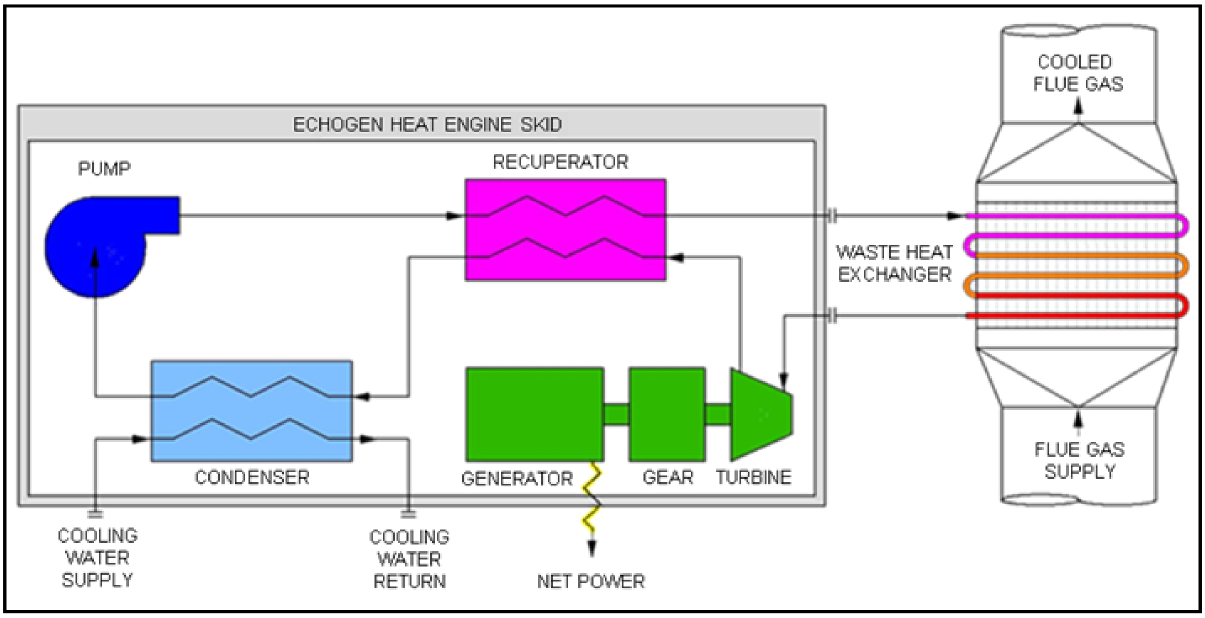


Figure 3.5: The Echogen Cycle. [56]

Echogen Cycle Description:

 Liquid CO2 pumped to supercritical pressure

 Internal system heat recycled at recuperator

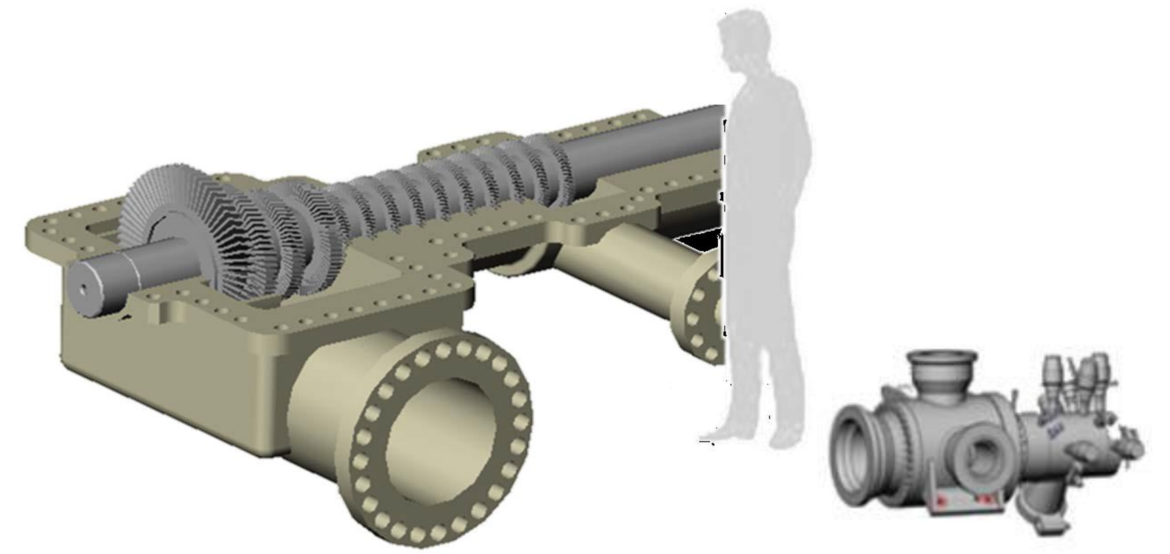
 Turbine exhaust or industrial waste heat added at waste heat exchanger

 High energy sCO2 expanded at power turbine drives generator

 Generator produces electrical power to customer specifications

Advantages of supercritical CO2 over steam

Supercritical CO2 is a perfect working liquid for closed-loop power generation applications. It is a minimal cost liquid that is nonpoisonous, noncombustible, non-corrosive and promptly accessible. The high fluid density of sCO2 empowers to a great degree conservative turbomachinery designs. Figure 3.4 compares Echogen 10 MWe CO2 turbine with 10 MWe steam turbine. CO2 turbine are of single stage generally but steam turbine is of many stages and have more complex structure. Echogen turbine uses 9.5 inches or 0.24 m impeller for turbine and create electrical power which are enough for around 8000 homes.[57-58]



10 MWe steam Turbine ***Echogen's 10 MWe SCO_2 power turbine***

Figure 3.6: *Echogen's 10 MWe sCO_2 power turbine compared to a 10 MWe steam turbine.* [58]

A comparison between shell tube and highly compact microchannel for dimensions for equal heat duty is given in table.

Table 3.1: Comparison of Shell & Tube and Highly Compact HX Technologies [57]

Recuperator Type	Dimensions (m) [in]	Pressure (Kg) [lbs]
Shell & Tube	4-shells, 0.25 dia. x 6.09 lg. [10 dia. X 240 lg.]	7,711 [17,000]
Highly Compact Microchannel	0.58 x 0.58 x 0.58 [23 x 23 x 13]	969.78 [2,138]

Chapter 4

Analysis

(1) THERMAL DESIGN OF HEAT EXCHANGER

A. *Design Input Parameters:*

The basic input parameters which are taken from GELM 2500 Marine Gas Turbine broacher [59], are used for the heat exchanger design and are summarized below:

Heat Exchanger Geometric Parameters:

The following summarizes the basic geometrical input parameter:

Shell outer radius: R_{out}= 3.5 m

Shell input radius: R_{in} = 1.5 m

Tube bundle length: L_m = 1.0 m

Tube inner diameter: d_i = 0.03 m

Tube pitch: p = 0.05 m

Tube thickness: th = 0.001754 m

Tube outer diameter: $d_o = d_i + 2 * th$

Heat exchanger operating conditions:

In this design, the exhaust gas is flowing in shell side and the operating condition are as follows:

1.Exhaust gas at **100 % load**

Temperature of exhaust gas: T_{ex} = 843 K

Mass flow rate of exhaust gas: $\dot{m}_{ex}$= 63 Kg/s

Pressure of exhaust gas: P_{ex} = 200 kPa

2.Exhaust gas at **71 % load**

Temperature of exhaust gas: T_{ex} = 798 K

Mass flow rate of exhaust gas: $\dot{m}_{ex}$ = 55 Kg/s

Pressure of exhaust gas: P_{ex} = 170 kPa

3.Exhaust gas at **49 % load**

Temperature of exhaust gas: T_{ex} = 761 K

Mass flow rate of exhaust gas: $\dot{m}_{ex}$ = 48 Kg/s

Pressure of exhaust gas: P_{ex} = 140 kPa

4. Supercritical CO_2:

Temperature of supercritical carbon dioxide: T_{SCO2} = 308 K

Mass flow rate of supercritical carbon dioxide: = 35 Kg/s

Pressure of supercritical carbon dioxide: P_{SCO2} = 20000 kPa

5. Ambient condition**:**

Ambient Temperature: T_o=300 K

6. Heat transfer duty:

Q = 20 * 10^6 W

B. Properties:

For designing heat exchanger, fluid properties should be defined first. The following summarizes the detail. In this heat exchanger cold fluid is supercritical carbon dioxide which is flowing in helical tube and hot fluid is exhaust gas which is flowing in shell side.

The following summarizes the basic properties of both the fluid.

1. Exhaust Gas:

(a) At 100% load

Density ρ_{ex} = 0.8266 kg/m³

Thermal conductivity k_{ex} = 0.0594 W/m-K

Heat Capacity ${c_p}_{ex}$ = 1.108 kJ/kg-K

Viscosity μ_{ex} = 0.00003763 kg/m-s

(b) At 71% load

Density ρ_{ex} = 0.7422 kg/m³

Thermal conductivity k_{ex} = 0.05705 W/m-K

Heat Capacity ${c_p}_{ex}$ = 1.098 kJ/kg-K

Viscosity μ_{ex} = 0.00003635 kg/m-s

(c) At 49% load

Density ρ_{ex} = 0.6409 kg/m³

Thermal conductivity k_{ex} = 0.05506 W/m-K

Heat Capacity $c_{p_{ex}}$ = 1.090 kJ/kg-K

Viscosity μ_{ex} = 0.00003528 kg/m-s

2. Supercritical Carbon Dioxide:

Density ρ_{sco2} = 866.4 kg/m³

Thermal conductivity k_{sco2}= 0.09851 W/m-K

Heat Capacity $c_{p_{sco2}}$ = 2.2 kJ/kg-K

Viscosity μ_{sco2} = 0.00008368 kg/m-s

C. Other Parameter:

In this type of heat exchanger, the number of tubes (N_{sco2}) in the bundle can be determined as follows:

Number of tube as [51]

$$N_{sco2} = (R_{out} - R_{in}) * \frac{L_m}{p^2} \quad 4.1$$

Other design parameter will be determined after heat exchanger designs.

D. Flow Parameter:

Based on the heat exchanger design conditions and input parameters, flow parameters of the heat exchanger can be estimated can be estimated as follows:

Velocity of Supercritical Carbon dioxide is given as [51]

$$\mathrm{Vel}_{sco2} = \frac{\dot{m}_{sco2}}{\rho_{sco2} * \frac{\pi}{4} * {d_i}^2 * N_{sco2}} \quad 4.2$$

Reynold number in tube side is given as [51]

$$Re_{sco2} = \frac{\rho_{sco2} * \ Vel_{sco2} * d_i}{\mu_{sco2}} \quad 4.3$$

Reynold number of exhaust gas in shell side [51]

$$Re_{ex} = \frac{\rho_{ex} * \ Vel_{ex} * d_o}{\mu_{ex}} \quad 4.4$$

Velocity of exhaust gas in shell side [51]

$$Vel_{sco2} = \frac{\dot{m_{ex}}}{\rho_{ex} * \pi * \left(R_{out}{}^2 - R_{in}{}^2\right) * \left[1 - \left(\frac{d_o}{p}\right)\right]} \quad 4.5$$

Prandtl number in the shell side [51]

$$Pr_{ex} = \frac{\mu_{ex} * c_{p_{ex}}}{k_{ex}} \quad 4.6$$

Prandtl number in the tube side [51]

$$Pr_{sco2} = \frac{\mu_{sco2} * c_{p_{sco2}}}{k_{sco2}} \quad 4.7$$

E. Heat Transfer Correlations:

In this part, heat transfer for the helical coiled heat exchanger will be estimated.

Tube Side:

In the tube side, heat transfer correlations are based on the heat transfer in the helical coiled tubes. To estimate it, some geometrical features should be defined first as follows:

Radius of Tube: [51]

$$a = \frac{d_i}{2} \quad 4.8$$

Radius of Curvature: [51]

$$R_{eff} = \frac{R_{in} + R_{out}}{2} \quad 4.9$$

Based on that, Nusselt number for helical coiled tube can be estimated as follows Shah et al. [60]:

Nusselt number for straight pipe [51]

$$Nusselt_0 = 0.022 * \ Re_{sco2}{}^{0.8} * \ Pr_{sco2}{}^{0.5} \quad 4.10$$

Nusselt number for helical coil Shah et al. [60]

$$Nusselt_{sco2} = Nusselt_0 * \left[1.0 + 3.6 * \left[1 - \left(\frac{a}{R_{eff}}\right)\right] * \left(\frac{a}{R_{eff}}\right)^{0.8}\right] \quad 4.11$$

From the above equation, heat transfer correlation in the tube side can be estimated as follows: [51]

$$h_{sco2} = Nusselt_{sco2} * \frac{k_{sco2}}{d_i} \quad 4.12$$

<u>Shell side:</u>

Nusselt number for shell can be estimated as follows, zukauskas [51]

$$Nusselt_{ex} = 0.27 * Re_{ex}{}^{0.63} * Pr_{ex}{}^{0.36} \quad 4.13$$

From the above equation, heat transfer correlation in the tube side can be estimated as follows:

$$h_{ex} = Nusselt_{ex} * \frac{k_{ex}}{d_o} \quad 4.14$$

<u>Overall heat transfer:</u>

In this part, the overall heat transfer coefficient for the helical coiled heat exchanger is estimated. Since the tube thickness is small, effect of heat transfer resistance at the wall has been neglected.

Overall heat transfer coefficient: [51]

$$U = \frac{1}{\frac{1}{h_{sco2}} + \frac{1}{h_{ex}}} \quad 4.15$$

Average temperature difference: [51]

$$\delta T_{avg} = T_{ex} - T_{sco2} \quad 4.16$$

Heat transfer Surface Area: [51]

$$A_s = \frac{Q}{U * \delta T_{avg}} \quad 4.17$$

Tube length: [51]

$$L_{t_{middle}} = \frac{A_s}{\pi * \left(\frac{d_i + d_o}{2}\right) * N_{sco2}} \quad 4.18$$

Number of rotation of tube bundle: [51]

$$N_b = \frac{L_{t_{middle}}}{\pi * \ (R_{out} + R_{in})}$$

F. Pressure Drop

In this part, pressure drop in shell and the tubes are estimated for the helical coiled heat exchanger:

Tube Side:

Pressure drop in the tube side can be estimated by the following equations Kakac and Liu [62]:

Friction factor in helical coiled Tubes: [62]

$$f = \left[0.0084\left[Re_t * \left[\frac{\frac{(R_{in}+R_{out})}{2}}{a}\right]^{-2}\right]^{-0.2} * \left[\frac{\frac{(R_{in}+R_{out})}{2}}{a}\right]^{-0.5}\right] \quad 4.19$$

Pressure drop in helical tube side: [62]

$$\delta P_{SCO2} = 4 * f * \frac{L_{t_{middle}}}{d_i} * \rho_{sco2} * \frac{\mathrm{Vel_{sco2}}^2}{2} \quad 4.20$$

Shell Side:

Pressure drop in the tube bundles in cross flow can be estimated by the following equations Martin, Shah & Sekulic [63-64]:

Pressure drop in shell side

Pitch to Diameter Ratio:

$$p_{ratio} = \frac{p}{d_o} \quad 4.21$$

Hagen number for inline tube bundles Martin [63]:

$$Hg_{lam} = \frac{140 * Re_{ex} * \left(p_{ratio}{}^{0.5} - 0.6\right)^2 + 0.75}{\left[p_{ratio}{}^{1.6} * \left(\frac{4 * p_{ratio}{}^2}{\pi} - 1\right)\right]} \quad 4.22$$

$$Hg_{tub} = \left[\left[0.11 + \frac{0.6 * \left(1 - \frac{0.94}{p_{ratio}}\right)^{0.6}}{(p_{ratio} - 0.85)^{1.3}}\right] * 10^{0.47*(-0.5)} + 0.015 * (p_{ratio} - 1)^2\right] * Re_{ex}{}^{1.9} \quad 4.23$$

Total Hagen number is given as

$$Hg = Hg_{lam} + Hg_{tub} \quad 4.24$$

Number of effective tube bundles:

$$n_{tr} = N_b * \left(\frac{L_m}{p}\right) \quad 4.25$$

Pressure drop in shell side:

$$\delta P_{ex} = \frac{\mu_{ex}{}^{2} * n_{tr} * Hg}{\rho_{ex} * d_o^2} \qquad 4.26$$

(2) SECOND LAW EFFICIENCY ANALYSIS OF HEAT EXCHANGER

Number of transfer unit –NTU is defined as

$$NTU = \frac{U * A_s}{C_{min}} \qquad 4.27$$

Min heat capacity is of exhaust gas (C_h)

$$C_{min} = cp_{ex} * \dot{m}_{ex} \qquad 4.28$$

Max heat capacity is of sco2 (C_c)

$$C_{max} = cp_{sco2} * \dot{m}_{sco2} \qquad 4.29$$

Ratio of C_{min} to C_{max} is C which is given as

$$C = \frac{C_{min}}{C_{max}} \qquad 4.30$$

Assuming flow in helical coil heat exchanger as counter flow

The effectiveness and number of heat transfer units (NTU) relationship for counter-flow heat exchangers is given [60]

$$\varepsilon = \frac{1 - \exp(-NTU * (1 - C))}{1 - C * \exp(-NTU * (1 - C))} \qquad 4.31$$

The second law efficiency called rational efficiency which is the ratio of desired exergy output to exergy used. Kotas [64]

Rational efficiency or Exergetic efficiency:

$$\psi = \frac{\dot{E}_{desiredoutput}}{\dot{E}_{used}} \qquad 4.32$$

The $\dot{E}_{desiredoutput}$ is the sum of all exergy transfers from the system, which constitutes the desired output by the system while $\dot{E}_{used}$ is the required exergy input for the process to be performed, which can be expressed in terms of irreversibilities as

$$\dot{E}_{used} = \dot{E}_{desiredoutput} + \dot{I} \qquad 4.33$$

We can express $\dot{E}_{desiredoutput}$ in terms of inlet temperature and effectiveness

$$\dot{E}_{desiredoutput} = C_{max} * \left[\varepsilon * C * (T_{ex} - T_{sco2}) - T_o * \ln\left\{1 + \left(\frac{\varepsilon * C * (T_{ex} - T_{sco2})}{T_{sco2}}\right)\right\}\right] \qquad 4.34$$

The irreversibility term in the second law efficiency expression is obtained as the product of the entropy generation rate and reference temperature. Expressing the entropy generation rate in terms of the minimum capacity rate and entropy generation number as

Irreversibility given by $\dot{I}$

$$\dot{I} = T_o * N_s * C_{min} = T_o * \dot{S}_{gen} \qquad 4.35$$

Defining the entropy generation number by dividing entropy generation by minimum heat capacity rate, i.e., *Cmin* [63]

Entropy generation number

$$N_s = N_{s_H} + N_{s_P} \qquad 4.36$$

Entropy generation number are given as

Entropy generation number due to heat transfer

$$N_{s_H} = \ln\left[1 + \varepsilon * \left\{\left(\frac{T_{ex}}{T_{sco2}}\right) - 1\right\}\right] + \left[\left(\frac{1}{C}\right) * ln\left\{1 - C * \varepsilon * (1 - \frac{T_{sco2}}{T_{ex}})\right\}\right] \qquad 4.37$$

Entropy generation number due to pressure drop

$$N_{s_P} = \left(\frac{R_{sco2}}{cp_{sco2} * C}\right) * \left(\frac{\delta P_{SCO2}}{P_{SCO2}}\right) + \left(\frac{R_{ex}}{cp_{ex}}\right) * \left(\frac{\delta P_{ex}}{P_{ex}}\right) \qquad 4.38$$

(3) THERMAL ECONOMIC ANALYSIS

Input data

Effective rate of return in % per year

i_e= 10 %

Technical life or life of cycle in years

TL = 10 year

Operation maintenance factor

ϕ= 1.06

Cost associated with irreversibility or electricity cost

C_s= 0.1 dollar/kwh

Number of operation hours in years

n_h= 3650 hrs / year

The total cost of the heat exchanger is given as the sum of the capital cost and the irreversibility penalty costs as provided in Aceves-Saborio and Reisted [66] and Gogus [67] as

Total cost of Heat Exchanger

$$C_{cost} = C_e * R_f * \phi + C_s * n_h * T_o * (\dot{S}_{gen_H} + \dot{S}_{gen_H}) \tag{4.39}$$

Entropy generation due to heat transfer

$$\dot{S}_{gen_H} = N_{s_H} * C_{min} \tag{4.40}$$

Entropy generation due to pressure drop

$$\dot{S}_{gen_P} = N_{s_P} * C_{min} \tag{4.41}$$

Cost of equipment in dollar

$$C_e = A_s * 3 * 2 \tag{4.42}$$

where Ce is cost of the equipment which is proportional to the surface area of the heat exchanger which includes material and production cost. R_f is the capital recovery factor given as

Capital Recovery factor:

$$R_f = \frac{i_e*(1+i_e)^{TL}}{(1+i_e)^{TL}-1} \tag{4.43}$$

i_e is the effective rate of return, TL is the technical life in years and ϕ is the operation maintenance factor. C_s is cost due to the irreversibilities which is equal to electricity cost and n_h is the number of operation hours in a year.

defined an important thermoeconomic factor for component evaluation which plays an important role in the optimisation of thermal systems. The variable is the exergoeconomic factor (EF) which is given by Bejan, Tsatsaronis, and Moran [68]

Exergoeconomic Factor:

$$EF = \frac{C_e*R_f*\phi}{C_{cost}} \tag{4.44}$$

In the thermoeconomic equation, the first part consists of non-exergy related costs (capital investment and operating and maintenance expenses) and the second part consists of exergy destruction cost. To know the relative significance of each of the part for evaluating the performance of a component EF is used. EF is the ratio of the non-exergy related cost to the total cost. A low value of EF proposes that cost savings in the entire system could be achieved by improving the component efficiency or reducing exergy destruction even if the capital investment will increase. A high value of EF proposes a decrease in the investment cost at the expense of its exergetic efficiency or destruction.

Chapter 5

Result and Discussion

This chapter includes the results obtained from the analysis and its discussion.

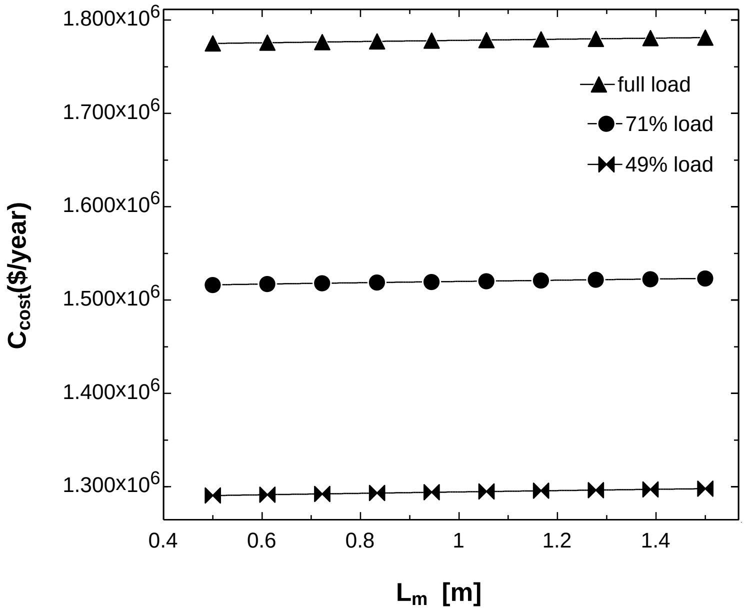


Fig 5.1 Total cost of Heat Exchanger at various load VS tube bundle length

Fig 5.1 demonstrate the total cost of heat exchanger at various load verses tube bundle length. Total cost of heat exchanger is increasing when tube bundle length increases at various load. The primary reason for this is, if tube bundle length increases then cost of material also increases. The secondary reason behind this is that total cost of heat exchanger depends on sum of entropy generation due to heat transfer and pressure drop and entropy generation due to pressure drop is increasing with increase in tube bundle length. Entropy generation due to heat transfer has no dominant effect on cost with increase in length. So total cost of heat exchanger is increasing when tube bundle length increases at various load.

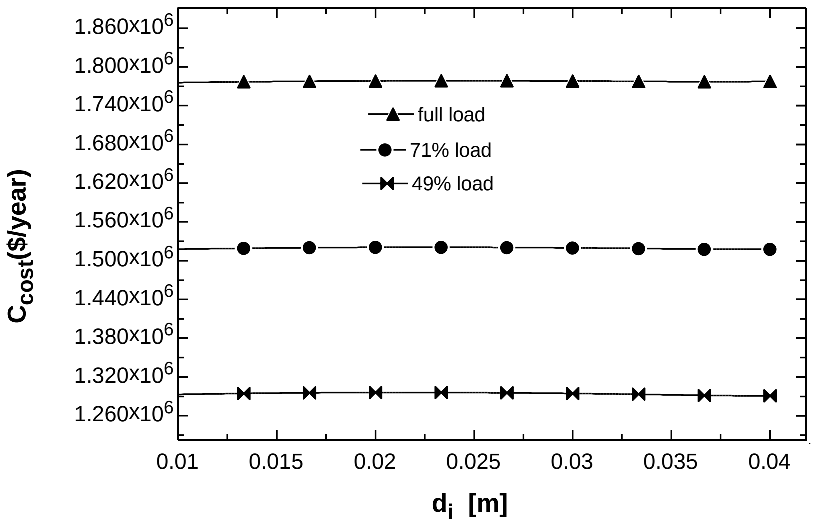


Fig 5.2 Total cost of Heat Exchanger at various load VS tube inner diameter

Fig 5.2 portrays total cost of heat exchanger verses tube bundle length at various load. Total cost of heat exchanger is increasing with increase in tube bundle diameter, obtains a maximum value at 0.025m tube inner diameter, decreases and get a minimum value at 0.0375m tube inner diameter and again increases with increase in inner tube diameter. So optimum value of cost is found at 0.0375 m diameter. Initially as diameter increases then investment cost or non-exergy cost increases which dominates the exergy destruction cost then overall cost of heat exchanger increases. After getting maximum value, as diameter increases pressure drop decreases, due to this entropy generation due to pressure drop also decreases so exergy destruction cost decreases and this cost dominates investment cost so overall cost of heat exchanger decreases.

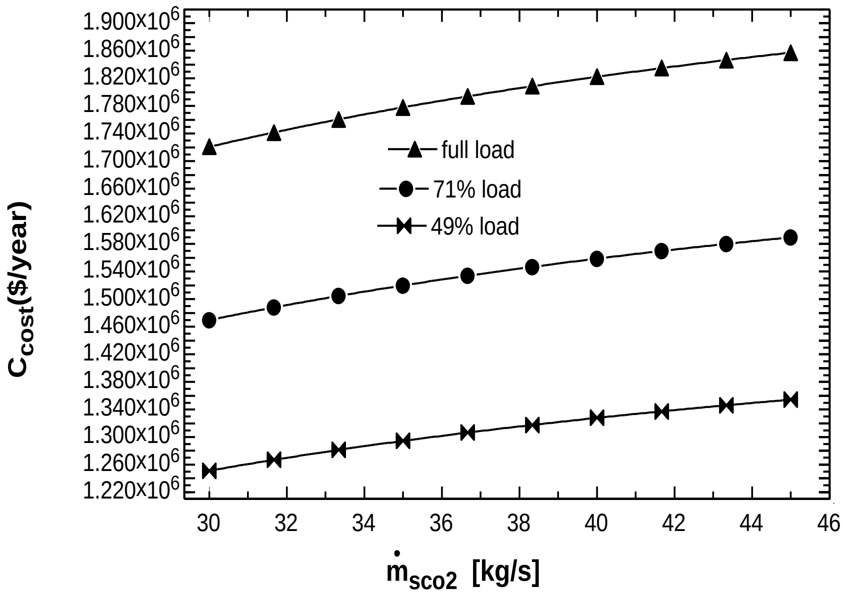


Fig 5.3 Total cost of Heat Exchanger at various load VS SCO2 mass flow rate

Fig 5.3 demonstrate total cost of heat exchanger verses supercritical CO2 mass flow rate at various load. Total cost of heat exchanger is increasing with increase in supercritical CO2 mass flow rate at various load. This is due to as mass flow rate increases, pressure drop also increases, due to this entropy generation due to pressure drop also increases so exergy destruction cost increases and here investment cost is constant so overall cost of heat exchanger increases.

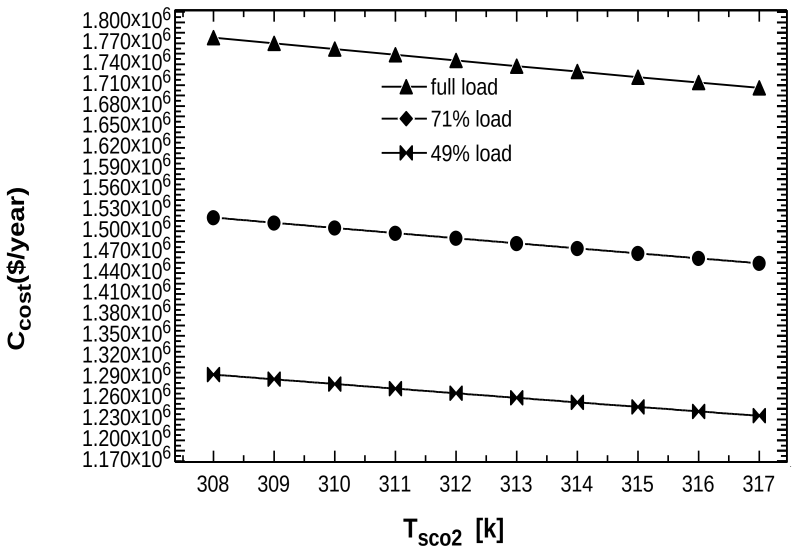


Fig 5.4 Total cost of Heat Exchanger at various load VS SCO2 inlet temperature

Fig 5.4 portrays total cost of heat exchanger verses supercritical CO2 inlet temperature at various load. Total cost of heat exchanger is decreasing with increase in supercritical CO2 inlet temperature. As temperature of cold fluid is increasing, temperature difference between two fluids is decreasing so entropy generation due to heat transfer is decreasing. This will decrease exergy destruction cost and here investment cost is constant so total cost of heat exchanger will decrease.

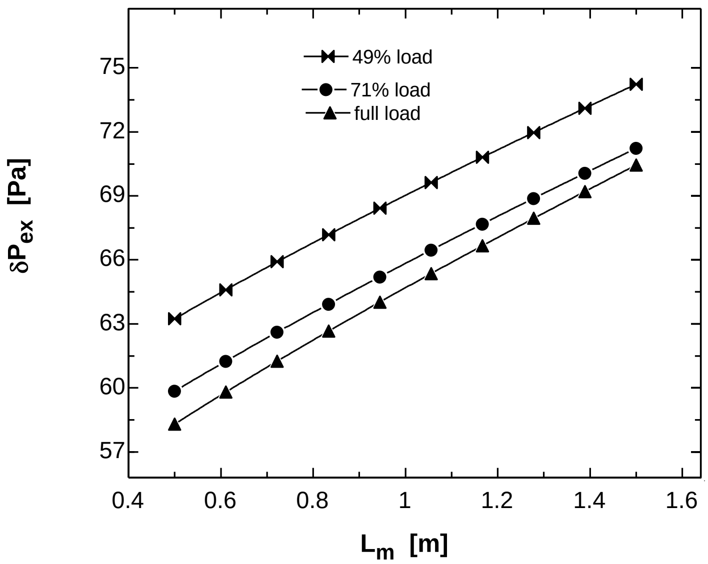


Fig 5.5 Pressure drop of Exhaust gas in Shell side VS Tube bundle length

Fig 5.5 demonstrate pressure drop of exhaust gas in shell side verses tube bundle length at various load. Pressure drop of exhaust gas in shell side is increasing when tube bundle length increases. This is due to Pressure drop is directly proportional to length.

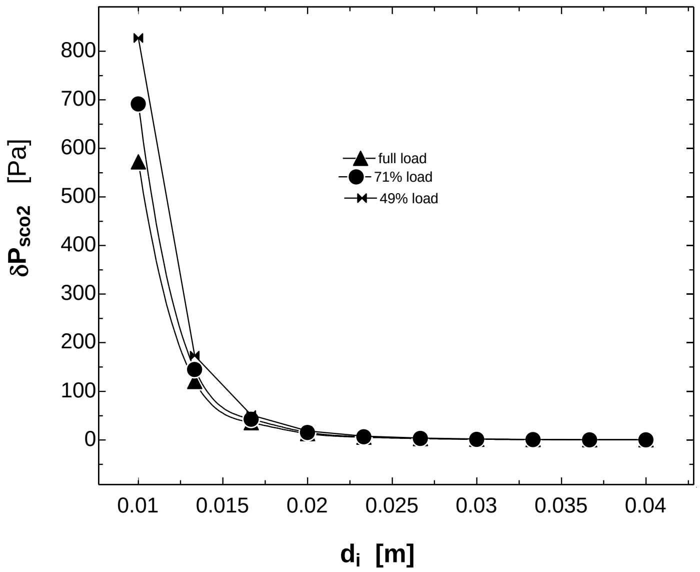


Fig 5.6 Pressure drop of SCO2 in Coil side VS tube inner diameter

Fig 5.6 portrays pressure drop of SCO2 in coil side verses tube inner diameter at various load. There is steep decrement in pressure drop of SCO2 in coil side up to 0.0175 m tube inner diameter after that it decreases gradually up to 0.025 m diameter, after that it decreases slightly up to 0.03 m diameter after that it becomes linear with increase in diameter. Because pressure drop is inversely proportional to square of diameter. So as the diameter increases there is sudden drop in pressure and there is no effect of load after 0.03m value of diameter on pressure drop.

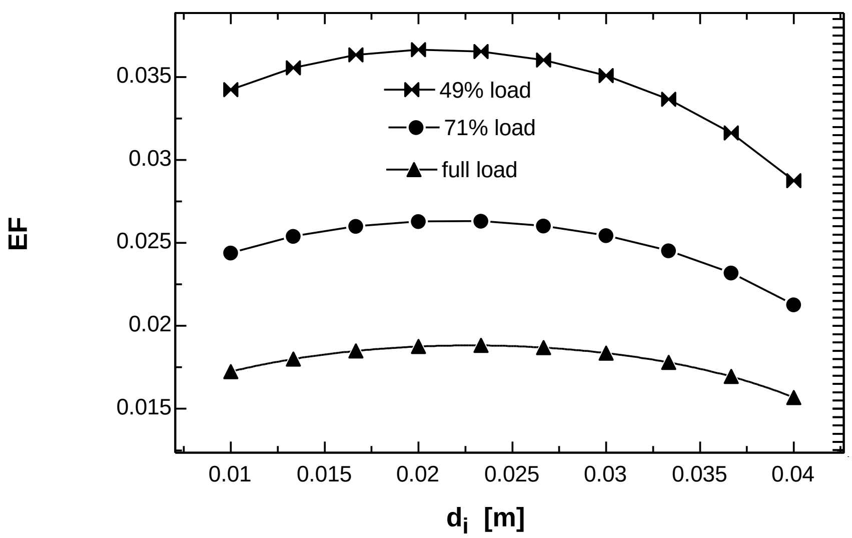


Fig 5.7 Exergoeconomic Factor of Heat Exchanger at various load VS tube inner diameter

Fig 5.7 illustrate exergoeconomic factor of heat exchanger at various load verses tube inner diameter. Exergoeconomic factor increases up to 0.02 m diameter and becomes maximum between 0.02 m and 0.025 m for different load and after that it is decreasing. A low value of EF proposes that cost savings in the entire system could be achieved by improving the component efficiency or reducing exergy destruction even if the capital investment will increase. A high value of EF proposes a decrease in the investment cost at the expense of its exergetic efficiency or destruction.

As diameter increases then pressure drop decreases and entropy generation due to pressure drop decreases so exergy destruction cost decreases but as diameter increases then investment cost increases which dominates other cost so EF increases. At higher values of diameter exergy destruction cost dominates so EF decreases.

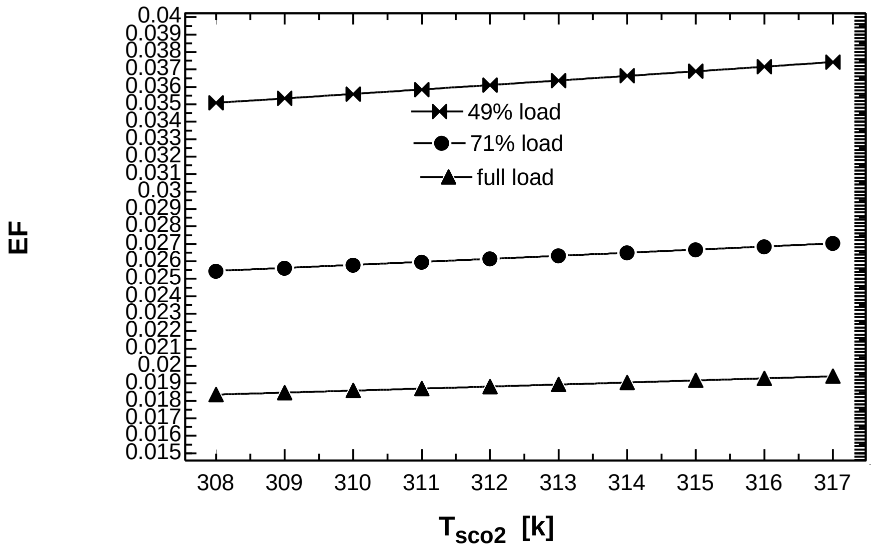


Fig 5.8 Exergoeconomic Factor of Heat Exchanger at various load VS SCO2 inlet temperature

Fig 5.8 portrays exergoeconomic factor of heat exchanger at various load verses supercritical CO2 inlet temperature. The exergoeconomic factor of heat exchanger increases slightly with increase in supercritical CO2 inlet temperature in tube side. As temperature of cold fluid is increasing, temperature difference between two fluids is decreasing so entropy generation due to heat transfer will decrease. This will decrease exergy destruction cost and here investment cost is constant so EF will increase.

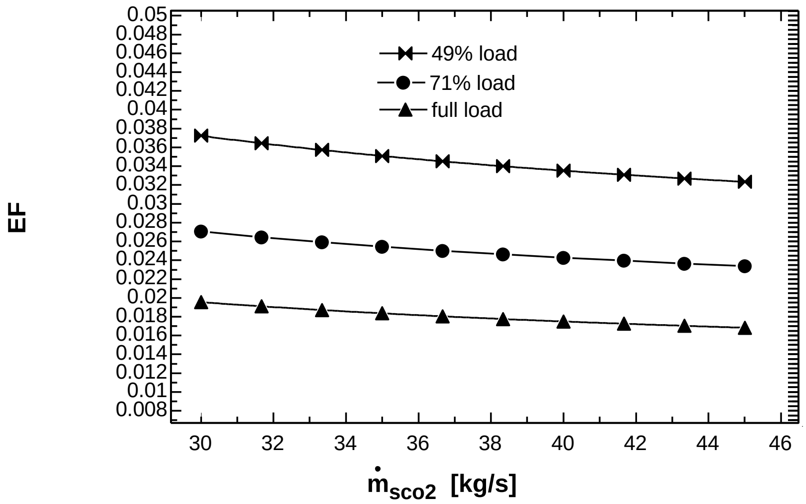


Fig 5.9 Exergoeconomic Factor of Heat Exchanger at various load VS SCO2 mass flow rate

Fig 5.9 demonstrate exergoeconomic factor of heat exchanger at various load verses supercritical CO2 mass flow rate. Exergoeconomic factor of heat exchanger decreases with increase in supercritical CO2 mass flow rate. This is due to as mass flow rate increases, pressure drop also increases, due to this entropy generation due to pressure drop also increases so exergy destruction cost increases and here investment cost is constant so overall cost of heat exchanger increases therefor EF decreases.

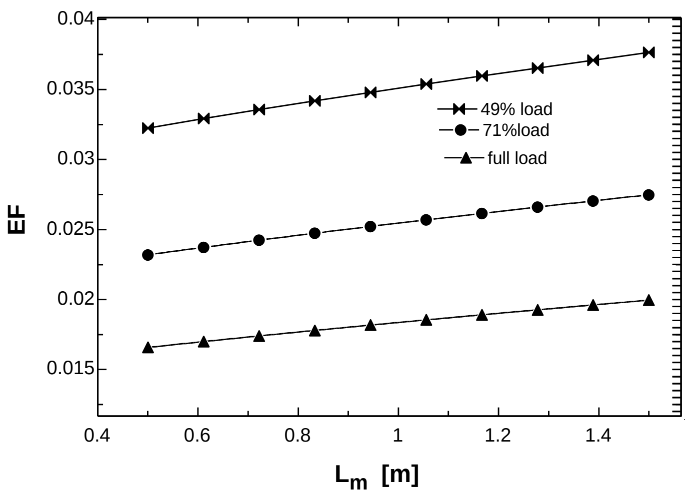


Fig 5.10 Exergoeconomic Factor of Heat Exchanger at various load VS Tube bundle length

Fig 5.10 illustrate exergoeconomic factor of heat exchanger at various load verses tube bundle length. Exergoeconomic factor of heat exchanger is slightly increasing with increase in tube bundle length. As length increases investment cost increases thus EF increases.

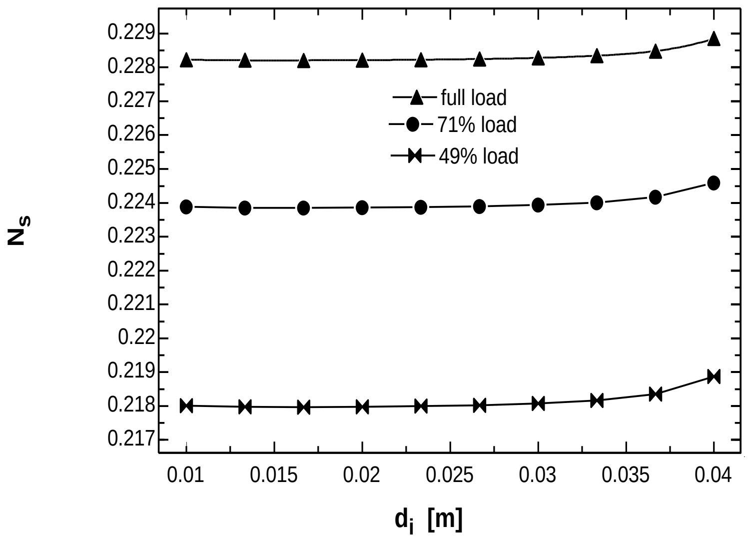


Fig 5.11 Overall Entropy Generation Number of Heat Exchanger VS tube inner diameter

Fig 5.11 demonstrate overall entropy generation number of heat exchanger at various load verses tube inner diameter. Initially overall entropy generation number is decreasing with increase in tube inner diameter and obtains a minimum value at 0.015 m diameter and after that it is increasing with increase in tube inner diameter. Initially when diameter increases then pressure drop decreases and entropy generation due to pressure drop decreases because it dominates here. After getting minimum value of overall entropy generation number it starts increasing because entropy generation due to heat transfer will increase and dominate with increment in diameter. Here optimum diameter can be achieved by second law analysis which can't be obtained by first law thermodynamics analysis.

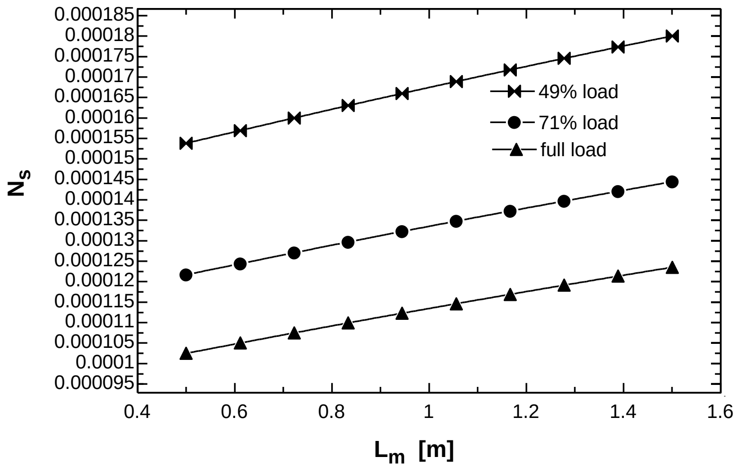


Fig 5.12 Entropy Generation Number due to Pressure Drop VS Tube bundle length

Fig 5.12 illustrate entropy generation number due to pressure drop verses tube bundle length. Entropy generation number due to pressure drop is increasing with tube bundle length. As length increases pressure drop will increase so entropy generation number due to pressure drop will increase. So overall entropy generation will increase.

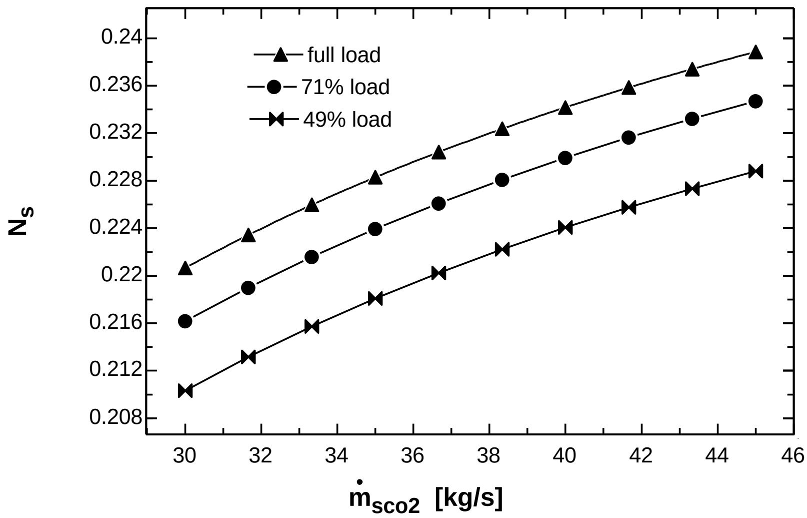


Fig 5.13 Overall Entropy Generation Number of Heat Exchanger VS SCO2 mass flow rate

Fig 5.13 illustrates Overall entropy generation number of heat exchanger at various load verses SCO2 mass flow rate. Overall entropy generation number increases with increase in SCO2 mass flow rate. This is due to as mass flow rate increases, pressure drop also increases, due to this entropy generation due to pressure drop will increase which dominates entropy generation due to heat transfer therefore overall entropy generation number increases with increase in SCO2 mass flow rate.

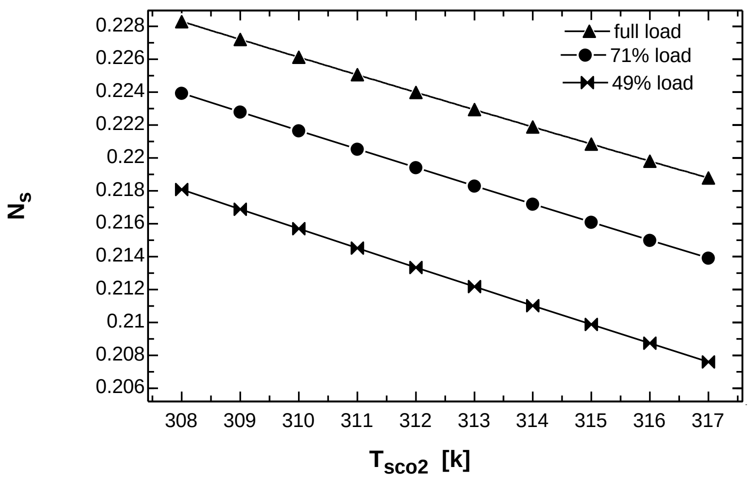


Fig 5.14 Overall Entropy Generation Number of Heat Exchanger VS SCO2 inlet temperature

Fig 5.14 demonstrate Overall entropy generation number of heat exchanger verses SCO2 inlet temperature. Overall entropy generation number is decreasing with increase in SCO2 inlet temperature. As temperature of cold fluid is increasing, temperature difference between two fluids is decreasing so entropy generation due to heat transfer will decrease and it will dominate entropy generation due to pressure drop therefore overall entropy generation number will decreasing with increase in SCO2 inlet temperature.

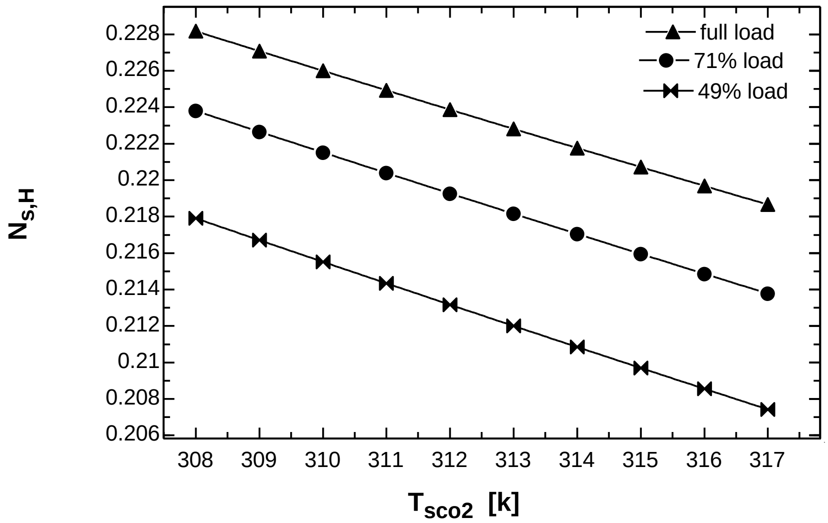


Fig 5.15 Entropy Generation Number due to Heat Transfer VS SCO2 inlet temperature

Fig 5.15 illustrates Entropy generation number due to heat transfer verses SCO2 inlet temperature. Entropy generation number due to heat transfer is decreasing with increase in SCO2 inlet temperature. As temperature of cold fluid is increasing, temperature difference between two fluids is decreasing so entropy generation due to heat transfer will decrease therefore overall entropy generation number will decrease with increase in SCO2 inlet temperature.

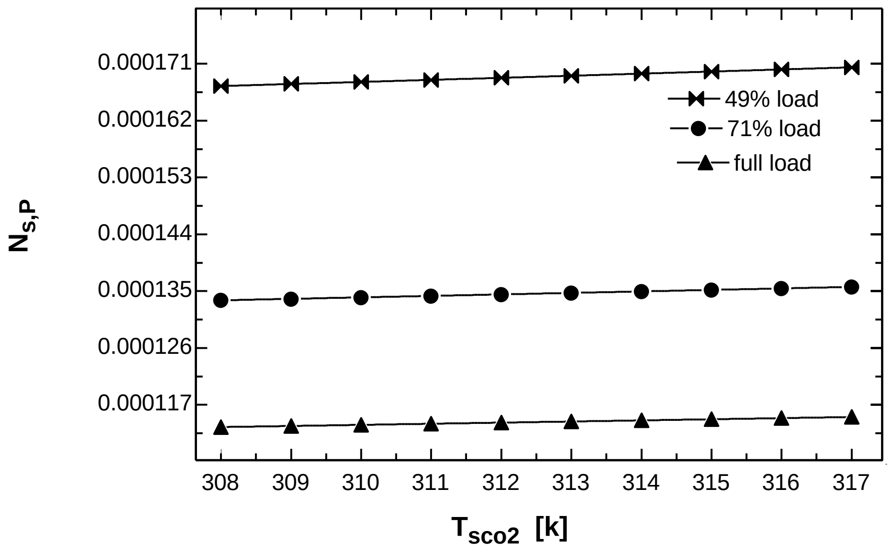


Fig 5.16 Entropy Generation Number due to Pressure Drop VS SCO2 inlet temperature

Fig 5.16 demonstrates Entropy Generation Number due to Pressure Drop at various load verses SCO2 inlet temperature. Entropy Generation Number due to Pressure Drop is slightly increasing with increase in SCO2 inlet temperature. This is due to when temperature increases then density of fluid will increase, this will increase pressure drop thus entropy generation due to pressure drop will increase so entropy generation number will also increase.

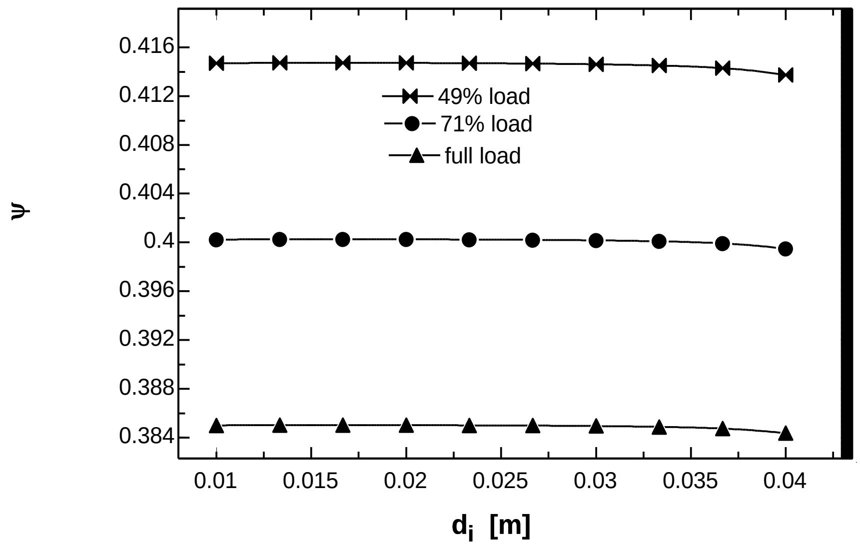


Fig 5.17 Rational Efficiency of Heat Exchanger at various load VS tube inner diameter

Fig 5.17 portrays Rational Efficiency of Heat Exchanger at various load verses tube inner diameter. Rational Efficiency of Heat Exchanger is increasing with increase in tube inner diameter and becomes maximum at 0.015 m diameter. After this, it slightly decreases up to 0.0325 m diameter and then decreases steeply. Initially when diameter increases then pressure drop decreases so entropy generation due to pressure drop will decrease and it dominates here. Rational Efficiency increases, becomes maximum at 0.015 m diameter. After getting maximum value of Rational Efficiency it starts decreasing because entropy generation due to heat transfer will increase and dominate here with increase in diameter. Here optimum diameter can be achieved by second law analysis which can't be obtained by first law thermodynamics analysis.

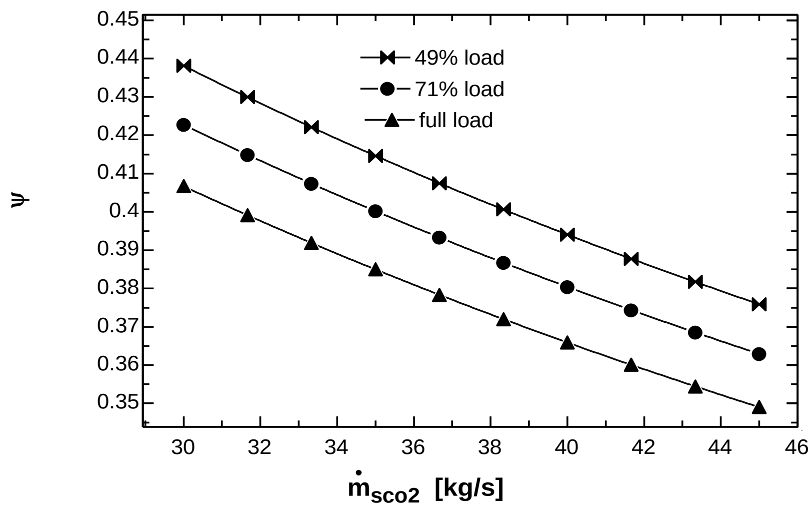


Fig 5.18 Rational Efficiency of Heat Exchanger at various load VS SCO2 mass flow rate

Fig 5.18 Illustrates Rational Efficiency of Heat Exchanger at various load verses SCO2 mass flow rate. Rational Efficiency is decreasing with increase in SCO2 mass flow rate. This is due to as mass flow rate increases, pressure drop also increases, due to this entropy generation due to pressure drop also increases therefore rational efficiency will decrease.

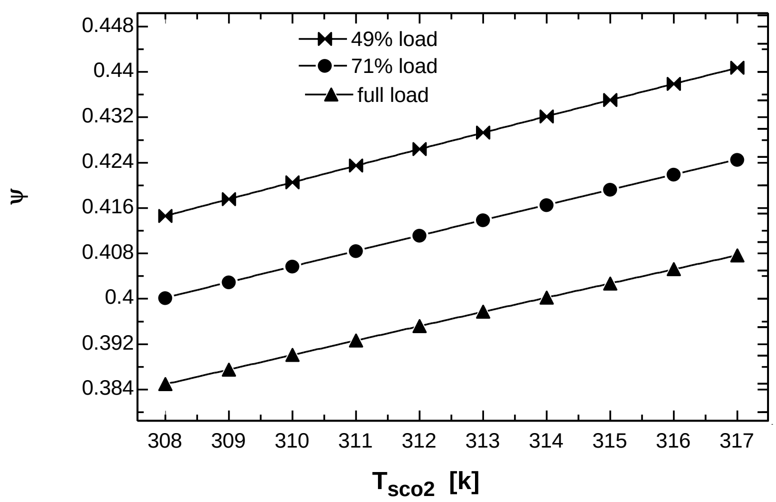


Fig 5.19 Rational Efficiency of Heat Exchanger at various load VS SCO2 inlet temperature

Fig 5.19 demonstrates Rational Efficiency of Heat Exchanger at various load verses SCO2 inlet temperature. Rational Efficiency of Heat Exchanger is increasing with increase in SCO2 inlet temperature. As temperature of cold fluid is increasing, temperature difference between two fluids is decreasing so entropy generation due to heat transfer will decrease therefore rational efficiency will increase.

Chapter 6

Conclusions and Future Scope

1. Overall cost of Heat Exchanger:

a. Cost is increasing by increase in tube bundle length.

b. As diameter increases initially cost increases and becomes maximum after that it decreases.

c. By increase in mass flow rate of SCO2 overall cost of Heat Exchanger decreases.

d. As temperature of SCO2 increases overall cost of Heat Exchanger decreases.

2. Pressure Drop:

a. As tube bundle length increases pressure drop of exhaust gas in shell side increases.

b. As tube diameter increases, pressure drop of SCO2 in tube decreases steeply and becomes constant.

3. Exergoeconomic Factor:

a. As diameter increases initially EF increases and after attaining maximum value it starts decreases.

b. As temperature of SCO2 increases EF increases.

c. As mass flow rate of SCO2 increases EF decreases.

d. As tube bundle length increases EF increases.

4. Rational Efficiency:

a. As diameter increases rational efficiency initially starts increasing and then after getting maximum it decreases. By this the optimum diameter of the tube is obtained.

b. As mass flow rate of SCO2 increases rational efficiency decreases.

c. As temperature of SCO2 increases then rational efficiency increases.

5. Overall Entropy Generation Number:

a. It initially starts decreasing with increase in diameter and then after getting minimum value it starts increasing. By this the optimum diameter of the tube is obtained which is having lower irreversibility. This cannot be obtained by the first law analysis alone.

b. As mass flow rate of SCO2 increases then Overall Entropy Generation Number increases.

c. As temperature of SCO2 increases Overall Entropy Generation Number decreases.

d. As tube bundle length increases Overall Entropy Generation Number increases.

The above results are useful for the thermal design of heat exchangers based on exergy and thermoeconomic analysis. Here optimum diameter can be achieved by second law analysis which can’t be obtained by first law thermodynamics analysis. Second law of thermodynamic combines entropy generation due to heat transfer and due to pressure drop which is not possible from first law of thermodynamics.

Future scope

1. The same analysis can be extended for other types of compact heat exchangers according to the different application.
2. This analysis can also be further used in research field of diesel gensets, gas turbine power production, nuclear reactor power application and other heat recovery projects.
3. Same analysis can be extended for cooling application by using same working fluid. By using transcritical carbon dioxide cycle we are able to achieve co-generation by which power and cooling application can be obtained for waste heat recovery.

REFERENCES

1. Johnson, I., Choate, W.T. and Davidson, A., 2008. *Waste heat recovery. Technology and opportunities in US industry*. BCS, Inc., Laurel, MD (United States).
2. Kehlhofer, R, 1997. *Combined-Cycle Gas and Steam Turbine Power Plants*; Penn Well Publishing Company: Tulsa, Oklahoma.
3. Marrero, I.O., Lefsaker, A.M., Razani, A. and Kim, K.J., 2002. Second law analysis and optimization of a combined triple power cycle. *Energy Conversion and Management*, 43(4), pp.557-573.
4. Siemens Energy. Combined Cycle site. http://www.energy.siemens.com/us/en/ industries-utilities/power/processes/combined-cycle.htm (accessed 3/31/13).
5. Chen, Y., Lundqvist, P., Johansson, A. and Platell, P., 2006. A comparative study of the carbon dioxide transcritical power cycle compared with an organic Rankine cycle with R123 as working fluid in waste heat recovery. *Applied Thermal Engineering*, 26(17), pp.2142-2147.
6. Kim, M-H.; Petterson, J.; Bullard, C.W. Fundamental process and system design issues in CO2 vapor compression systems. *Progress in Energy and Combustion Science, 30,* pp. 119-174.
7. Chen, Y., Lundqvist, P. and Platell, P., 2005. Theoretical research of carbon dioxide power cycle application in automobile industry to reduce vehicle's fuel consumption. *Applied Thermal Engineering*, 25(14), pp.2041-2053.
8. Persichilli, M., Kacludis, A., Zdankiewicz, E. and Held, T., 2012. Supercritical CO2 power cycle developments and commercialization: why sCO2 can displace steam ste. *Power-Gen India & Central Asia*.
9. Austin, B.T. and Sumathy, K., 2011. Transcritical carbon dioxide heat pump systems: A review. *Renewable and Sustainable Energy Reviews, 15(8)*, pp.4013-4029.
10. Cayer, E., Galanis, N., Desilets, M., Nesreddine, H. and Roy, P., 2009. Analysis of a carbon dioxide transcritical power cycle using a low temperature source. *Applied Energy, 86(7),* pp.1055-1063.
11. Vélez, F., Segovia, J., Chejne, F., Antolín, G., Quijano, A. and Martín, M.C., 2011. Low temperature heat source for power generation: exhaustive analysis of a carbon dioxide transcritical power cycle. *Energy, 36(9),* pp.5497-5507.
12. McQuay Air Conditioning. Refrigerants Application Guide,2000. Technical Report No. AG 31-007,

13. Chen, Y., Lundqvist, P. and Workie, A.B., 2011. Second Law Analysis of a Carbon Dioxide Transcritical Power System in Low-grade Heat Source Recovery.
14. Wang, K., Xu, X., Wu, Y., Liu, C. and Dang, C., 2015. Numerical investigation on heat transfer of supercritical CO 2 in heated helically coiled tubes. *The Journal of Supercritical Fluids, 99,* pp.112-120.
15. Jayachandraiah, B. and Jagaesh, R., 2014. Fabrication and Design of Spiral Tube Heat Exchanger. *American International Journal of Contemporary Scientific Research, 1(1),* pp.11-15.
16. Prabhanjan, D.G., Raghavan, G.S.V. and Rennie, T.J., 2002. Comparison of heat transfer rates between a straight tube heat exchanger and a helically coiled heat exchanger. *International Communications in Heat and Mass Transfer, 29(2),* pp.185-191.
17. Rao, N.T., Oumer, A.N. and Jamaludin, U.K., 2016. State-of-the-art on flow and heat transfer characteristics of supercritical CO 2 in various channels. *The Journal of Supercritical Fluids, 116,* pp.132-147.
18. Dostal, V., Driscoll, M.J. and Hejzlar, P., 2004. *A supercritical carbon dioxide cycle for next generation nuclear reactors* (Doctoral dissertation, Massachusetts Institute of Technology, Department of Nuclear Engineering).
19. Chen, Y., Lundqvist, P., Johansson, A. and Platell, P., 2006. A comparative study of the carbon dioxide transcritical power cycle compared with an organic Rankine cycle with R123 as working fluid in waste heat recovery. *Applied Thermal Engineering, 26(17),* pp.2142-2147.
20. Naphon, P., 2007. Thermal performance and pressure drop of the helical-coil heat exchangers with and without helically crimped fins. *International Communications in Heat and Mass Transfer, 34*(3), pp.321-330.
21. Sarkar, J., 2009. Second law analysis of supercritical CO 2 recompression Brayton cycle. *Energy, 34(9*), pp.1172-1178.
22. Kharat, R., Bhardwaj, N. and Jha, R.S., 2009. Development of heat transfer coefficient correlation for concentric helical coil heat exchanger. *International Journal of Thermal Sciences, 48(12),* pp.2300-2308.
23. Mandal, M.M. and Nigam, K.D., 2009. Experimental study on pressure drop and heat transfer of turbulent flow in tube in tube helical heat exchanger. *Industrial & Engineering Chemistry Research, 48(20),* pp.9318-9324.

24. Kocatepe Y., Ahn H., Aydin C.2006, “Experimental Investigation of a Heat Exchanger With A Helical Coil Made Of Corrugated Tubes”, Ulusal Isı Bilimi ve Tekniği Kongresi.
25. Tuo, H., 2011. Parametric analysis of a reheat carbon dioxide transcritical power cycle using a low temperature heat source. *In 2nd International Conference on Environmental Engineering and Applications, IPCBEE (Vol. 17).* pp. 33-37.
26. Kacludis, A., Lyons, S., Nadav, D. and Zdankiewicz, E., 2012. Waste Heat to Power (WH2P) Applications Using a Supercritical CO2-Based Power Cycle. *Power-Gen International*, pp.11-13.
27. Shirgire, N.D. and Kumar, P.V., 2013. Review on Comparative Study between Helical Coil and Straight Tube Heat Exchanger. *IOSR Journal of Mechanical and Civil Engineering (IOSR-JMCE) e-ISSN, 2278,* p.1684.
28. Kaushik, S.C. and Manjunath, K., 2014. Entropy generation and thermoeconomic analysis of the wire-and-tube condenser. *International Journal of Ambient Energy, 35(2)*, pp.80-93.
29. Anderson, K.R., Wells, T., Forgette, D., Okerson, R., DeVost, M., Cunningham, S. and Stuart, M., 2014. Waste Heat Energy Supercritical Carbon Dioxide Recovery Cycle Analysis and Design. *Journal ISSN: pending, 1(1)*.
30. Nassar Abdul, Moroz Leonid, Burlaka Maksym, 2014. Designing Supercritical CO2 Power Designt Using an Integrated Design System. *Gas Turbine India Conference*.
31. Andhare A., Kridesigni V M, Modak J P,2014. Thermal Analysis of a Helical Coil Heat Exchanger. *International Journal of Innovative Research in Advanced Engineering, Volume 1 Issue 12,* pp. 135-143.
32. Manjunath, K. and Kaushik, S.C., 2014. Entropy Generation and Thermo-Economic Analysis of Constructal Heat Exchanger. *Heat Transfer—Asian Research, 43(1),* pp.39-60.
33. Manjunath, K. and Kaushik, S.C., 2014. Second law thermodynamic study of heat exchangers: A review. *Renewable and Sustainable Energy Reviews, 40,* pp.348-374.
34. Ahn, Y., Bae, S.J., Kim, M., Cho, S.K., Baik, S., Lee, J.I. and Cha, J.E., 2015. Review of supercritical CO 2 power cycle technology and current status of research and development. *Nuclear Engineering and Technology, 47(6),* pp.647-661.
35. Mandore S., Kolhe K.P.,2015. A Review on Performance Enhancement of Tube Coil Heat Exchanger by using Helical Tube. *International Journal of Research in Mechanical Engineering & Technology Vol. 5, Issue 1,* pp. 81-83.

36. Seyyedvalilu M. H., Ranjbar S.F.,2015. The Effect of Geometrical Parameters On Heat Transfer and Hydro Dynamical Characteristics of Helical Exchanger. *International Journal of Recent advances in Mechanical Engineering (IJMECH) Vol.4, No.1*, pp. 35-46.
37. Manjunath K., Kaushik S. C.,2015. Second Law Efficiency Analysis of Heat Exchangers. *Heat Transfer—Asian Research, 44 (2),* pp. 89-108.
38. Zhao H., Deng Q., Huang W., Wang D.,2016. Thermodynamic and Economic Analysis and Multi-Objective Optimization of Supercritical CO2 Brayton Cycles. *Journal of Engineering for Gas Turbines and Power, Vol. 138,* pp. 1-9.
39. Xu X. X., Liu C., Dang C.,2016. Experimental investigation on heat transfer characteristics of supercritical CO2 cooled in horizontal helically coiled tube, *International journal of refrigeration 67,* pp. 190-201.
40. Rao N. T., Oumer A.N., Jamaludin U.K.,2016, State-of-the-art on flow and heat transfer characteristics of supercritical CO2 in various channels. *The Journal of Supercritical Fluids, 116,* pp.- 132–147.
41. Vishvakarma S., Kumbhare S., Thakur K. K.,2016. A Review On Heat Transfer Through Helical Coil Heat Exchangers, *International Journal of Engineering Sciences & Research Technology*, pp. 607-612.
42. Sharma O.P, Manjunath K., Kaushik S.C.,2017 .Thermodynamic analysis and optimization of a supercritical CO2 regenerative recompression Brayton cycle coupled with a marine gas turbine for shipboard waste heat recovery. *Thermal Science and Engineering Progress 3*, pp. 62–74.
43. Abdulla M. A., 1994, A four-region, moving-boundary model of a once through, helical coil team generator, *Nuclear Energy, 21(9),* pp. 541-562
44. Bai, B., L. Guo, Z. Feng, and X. Chen. 1999, Turbulent heat transfer in a horizontally coiled tube. *Heat Transfer-Asian Research, 28(5),* pp. 395-403.
45. Futagami, K., Aoyama, Y., 1988, Laminar heat transfer in helically coiled tubes, *International Journal Heat and Mass Transfer, Vol. 31*, pp. 387-396
46. Jensen M. K. and Bergles A. E., 1981, Trans. ASME 103, pp. 660-666
47. Patankar S., Pratap V. S. and Spalding D. B., 1974. *Journal of Fluid Mechanics*, 62, pp. 539-551.
48. Xin, R. C., A. Awwad, Z. F. Dong and M. A. Ebadian., 1996, An investigation and comparative study of the pressure drop in air-water two-phase flow in vertical helicoidal pipes. *International Journal Heat and Mass Transfer, Vol. 39(4),* pp. 735-743.

49. Minton P.E.,1970. *Designing Spiral Tube Heat Exchangers*, Chemical Engineering, p. 145.
50. Noble, M.A., Kamlani, J.S., and McKetta, J.J., 1992.Heat Transfer in Spiral Coils, Petroleum Engineer, p. 723.
51. "Heliflow Coolers and Heaters," Bulletin H-84-3, Graham Manufacturing Co. Inc.
52. "Heliflow Heat Exchangers," Bulletin HHE-30, Graham Manufacturing Co. Inc.
53. Kato K., D. F. Wilson, and C. W. Forsberg, 2007, Assessment of Silicon Carbide Composites for Advanced Salt-Cooled Reactors, ORNL/TM-2007/168, Revision 1, September 2007.
54. Sabharwall, P., 2011. *Feasibility study of secondary heat exchanger concepts for the advanced high temperature reactor* (No. INL/EXT-11-23076). Idaho National Laboratory (INL).
55. Echogen Power Systems LLC 2012, Supercritical CO2 Heat Engines for Power Generation –Combined Cycle Gas Turbine (CCGT) and Concentrating Solar Power (CSP) Useable (Waste) Heat to Power: Echogen Power Systems LLC.
56. Persichilli, M., Held, T., Hostler, S., and Zdankiewicz, E., 2011Transforming Waste Heat to Power through Development of a CO2-Based-Power Cycle, presented at 16th International Symposium for Compressor Users and Manufacturers, St. Petersburg, Russia.
57. Persichilli, M., Held, T., Hostler, S., Zdankiewicz, E., and Klapp, D., 2011 "*Transforming Waste Heat to Power through Development of a CO2-Based-Power Cycle*," presented at Electric Power Expo 2011, Rosemount, IL U.S.A.
58. Dostal V., Drsicoll. M., and Hejzlar, P, 2004. *A Supercritical Carbon Dioxide Cycle for Next Generation Nuclear Reactors*, MIT-ANP-TR-100, The MIT Center for Advanced Nuclear Energy Systems.
59. *GELM 2500* Marine Gas Turbine
60. Shah, R. K., and D. P. Sekulic, 2003, *Fundamentals of Heat Exchanger Design*, Hoboken, NJ, John Wiley & Sons.
61. Zukauskas, A., and R. Ulinskas, 1988, *Heat Transfer in Tube Banks in Crossflow*, Hemisphere Publishing, Washington, DC.
62. Kakac, S., and H. Liu, 1998, *Heat Exchangers: Selection, Rating, and Thermal Design*, CRC Press, Boca Raton, FL.
63. Martin, M., 1992, *Heat Exchangers*, Hemisphere Publishing, Washington, DC.

64. Kotas TJ. 1985. *The exergy method of thermal designt analysis*. Butterworths.
65. Bejan A.1997. The concept of irreversibility in heat exchanger design: counter flow heat exchangers for gas-to-gas applications. *Journal of Heat Transfer,* 99, pp. 374–380
66. Aceves-Saborio, R. J., and G. M. Reisted. 1989. An Extension to Irreversibility Minimization Analysis Applied to Heat Exchangers. *Journal of Heat Transfer 111 (1),* pp. 29–36.
67. Gogus, Y. A., 2005. Thermoeconomic Optimization. *International Journal of Energy Research, 29 (7),* pp. 559–580.
68. Bejan, A., G. Tsatsaronis, and M. Moran. 1995. *Thermal Design and Optimization*. New York: Wiley.