International Journal of Engineering Sciences Paradigms and Researches (IJESPR)
Vol. 48, Special Issue, (TAME-2019, April 4-5, 2019)
(An Indexed, Referred and Impact Factor Journal approved by UGC- Journal No. 42581)
ISSN (Online): 2319-6564
www.ijesonline.com

SECOND LAW ANALYSIS OF WASTE HEAT RECOVERY HEAT EXCHANGER USING SUPERCRITICAL CARBON DIOXIDE

Rashin Khera[1*], K. Manjunath [2]

[1]Research Scholar, Department of Mechanical Engineering, Delhi Technological University, Delhi, India

[2]Associate Professor,Ch. Brahm Prakash Government Engineering College Jaffarpur, Delhi 110073, India
Email id: rashinkhera.92@gmail.com

Abstract
The purpose of this study is analysis of second law of the helical coil heat exchanger for research in the field of waste heat recovery.By the second law analysis we can combine both the effects of heat transfer and pressure drop in a single equationto know the exact irreversibilities occurring in the system. The performance parameters used for the analysis are entropy generation number and second law efficiency. The parametric study has been carried out to know the behaviour of the systems by varying the tube diameter, length, mass flow rate of SCO2 (supercritical carbon dioxide) and inlet temperature of SCO2. The second law efficiency or rational efficiency is having a maximum value atoptimum diameter which is 0.015 m; it decreases with mass flow rate but increase with inlet temperature of SCO2. Entropy generation number is also having minimum value at optimum diameterwhich is 0.015 m. It increases with mass flow rate of SCO2 but decreases with inlet temperature of SCO2.

Keywords*: waste heat recovery, supercritical carbon dioxide, second law analysis, parametric analysis, Entropy generation minimization.*

1.Introduction

Basic power cycles dispose of an extensive segment of useful energy into the environment by means of exhaust gases. Using supercritical power cycle, this wasted energy might be used for powergeneration and production of hot water. Supercritical power cycle utilizes a supercritical working fluid for maximum effectiveness of heat exchanger. Carbon dioxide is chosen as the working fluid since it has a generally low critical temperature which makes it alluring for medium temperature waste heat applications

Chen et al. [1] simulated the performance of a CO2 power cycle in utilizing the low-grade heat sources and the results was analysed with a focus on second law thermodynamics.**Dostal***et al*. [2] performed parametricoptimization of supercritical CO2 Brayton power cycles for application to advanced nuclear reactors. **Sarkar** [3] conducted exergetic analyses and parametric optimization of S-CO2 recompression cycle. **Rao***et al*; [4] provided heat transfers and pressure drop characteristics of heat exchangers using SCO2 as working fluid based on many parameters such as size, mass flux, inlet temperature and pressure, etc.**Wang***et al*.[5] carried outthe heat transfer characteristics of supercritical CO2 cooled in the helically coiled tube.**Prabhanjan** et al. [6] determined advantages of using a helically coiled heat exchanger instead of straight tube heat exchanger for heating liquids.**Coronel***et al*. [7] determined pressure drop and fiction factor (f) correlations in helical coil heat exchangers under turbulent flow conditions based on the Reynolds number, curvature ratio, and temperature was developed.**Andhare***et al*. [8] investigated convective heat transfer coefficients of a helical coil heat exchanger experimentally **Manjunath***et al*. [9] conducted second law analysis of unbalanced heat exchangers by varying L/D ratio for parallel and counter flow configurations.

A program code is established with help of EES software for calculation required for the waste heat recovery analysis and variation of 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.

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 (W/m^2-K)
U	Overall heat transfer coefficient (W/m^2-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 because of heat transfer
Ns_P	Entropy generation number by virtue of pressure drop
$S_{gen,H}$	Entropy generation because of heat transfer (W/K)
$S_{gen,P}$	Entropy generation by the virtue of pressure drop (W/K)
NTU	Number of transfer unit
EES	Engineering equation solver

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

2.Analysis

In this section of heat exchanger's thermal design and Second law are analysed. 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.

2.1 Thermal Design of Heat Exchanger

In this type of heat exchanger, the number of tubes (N_{sco2}) in the bundle can be determined as follows[10]

$$N_{sco2} = (R_{out} - R_{in}) * \frac{L_m}{p^2} \quad (1)$$

A. Flow Parameter:

Velocity of Supercritical Carbon dioxide is given as [10]

$$\mathrm{Vel}_{sco2} = \frac{m_{sco2}}{\rho_{sco2} * \frac{\pi}{4} * d_i^{\,2} * N_{sco2}} \quad (2)$$

Reynold number in tube side is given as [10]

$$Re_{sco2} = \frac{\rho_{sco2} * Vel_{sco2} * d_i}{\mu_{sco2}} \quad (3)$$

Reynold number for exhaust gas in shell side [10]

$$Re_{ex} = \frac{\rho_{ex} * Vel_{ex} * d_o}{\mu_{ex}} \quad (4)$$

Velocity for exhaust gas in shell side [10]

$$Vel_{sco2} = \frac{\dot{m}_{ex}}{\rho_{ex} * \pi * (R_{out}^2 - R_{in}^2) * \left[1 - \left(\frac{d_o}{p}\right)\right]} \quad (5)$$

Prandtl number in the shell side [10]

$$Pr_{ex} = \frac{\mu_{ex} * c_{p_{ex}}}{k_{ex}} \quad (6)$$

Prandtl number in the tube side [10]

$$Pr_{sco2} = \frac{\mu_{sco2} * c_{p_{sco2}}}{k_{sco2}} \quad (7)$$

B. 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:

Tube Radius [10]

$$a = \frac{d_i}{2} \quad (8)$$

Radius of Curvature: [10]

$$R_{eff} = \frac{R_{in} + R_{out}}{2} \quad (9)$$

Based on that, Nusselt number for helical coiled tube can be estimated as follows Shah et al. [13]:

Nusselt number for straight pipe [10]

$$Nusselt_0 = 0.022 * Re_{sco2}^{0.8} * Pr_{sco2}^{0.5} \quad (10)$$

Nusselt number for helical coil, Shah et al. [11]

$$Nusselt_{sco2} = Nusselt_0 * \left[1.0 + 3.6 * \left[1 - \left(\frac{a}{R_{eff}}\right)\right] * \left(\frac{a}{R_{eff}}\right)^{0.8}\right] \quad (11)$$

heat transfer coefficient in the tube side can be given as follows: [10]

$$h_{sco2} = Nusselt_{sco2} * \frac{k_{sco2}}{d_i} \quad (12)$$

Shell side:

Nusselt number for shell, zukauskas [12]

$$Nusselt_{ex} = 0.27 * Re_{ex}^{0.63} * Pr_{ex}^{0.36} \quad (13)$$

heat transfer coefficient in the tube side can be given as:

$$h_{ex} = Nusselt_{ex} * \frac{k_{ex}}{d_o} \quad (14)$$

Overall heat transfer:

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

Overall heat transfer coefficient: [10]

$$U = \frac{1}{\frac{1}{h_{sco2}} + \frac{1}{h_{ex}}} \quad (15)$$

Average temperature difference: [10]

$$\delta T_{avg} = T_{ex} - T_{sco2} \quad (16)$$

Heat transfer Surface Area: [10]

$$A_s = \frac{Q}{U * \delta T_{avg}} \quad (17)$$

Tube length: [10]

$$L_{t_{middle}} = \frac{A_s}{\pi * \left(\frac{d_i + d_o}{2}\right) * N_{sco2}} \quad (18)$$

Number of rotation of tube bundle: [10]

$$N_b = \frac{L_{t_{middle}}}{\pi * (R_{out} + R_{in})} \quad (19)$$

C. 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 tube side can be given by Kakac and Liu [13]:

Friction factor in helical coiled Tubes: [13]

$$f = \left[0.0084 \left[Re_t * \left[\frac{\left(\frac{R_{in} + R_{out}}{2}\right)}{a}\right]^{-2}\right]^{-0.2} * \left[\frac{\left(\frac{R_{in} + R_{out}}{2}\right)}{a}\right]^{-0.5}\right] \quad (20)$$

Pressure drop in helical tube side: [13]

$$\delta P_{sco2} = 4 * f * \frac{L_{t_{middle}}}{d_i} * \rho_{sco2} * \frac{Vel_{sco2}^2}{2} \quad (21)$$

Shell Side:

Pressure drop in the tube bundles in cross flow can be given by Martin, Shah &Sekulic [14]:

Pitch to Diameter Ratio:

$$p_{ratio} = \frac{p}{d_o} \quad (22)$$

For inline tube bundles, Hagen number is given by, Martin [14]:

$$Hg_{lam} = \frac{140 * Re_{ex} * (p_{ratio}^{0.5} - 0.6)^2 + 0.75}{\left[p_{ratio}^{1.6} * \left(\frac{4 * p_{ratio}^2}{\pi} - 1\right)\right]} \quad (23)$$

$$Hg_{tub} = \left[\left[0.11 + \frac{0.6 * \left(1 - \frac{0.94}{p_{ratio}}\right)^{0.6}}{(p_{ratio} - 0.85)^{1.3}}\right] * 10^{0.47 * (-0.5)} + 0.015 * (p_{ratio} - 1)^2\right] * Re_{ex}^{1.9} \quad (24)$$

Total Hagen number is given as

$$Hg = Hg_{lam} + Hg_{tub} \quad (25)$$

Number of effective tube bundles:

$$n_{tr} = N_b * \left(\frac{L_m}{p}\right) \quad (26)$$

Pressure drop in shell side:

$$\delta P_{ex} = \frac{{\mu_{ex}}^2 * n_{tr} * Hg}{\rho_{ex} * d_b^2} \quad (27)$$

2.2 Second law efficiency analysis of heat exchanger

Number of transfer unit –NTU is defined as

$$NTU = \frac{U * A_s}{C_{min}} \quad (28)$$

Min heat capacity is of exhaust gas (C_h)

$$C_{min} = cp_{ex} * \dot{m}_{ex} \quad (29)$$

Max heat capacity is of sco2(C_c)

$$C_{max} = cp_{sco2} * \dot{m}_{sco2} \quad (30)$$

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

$$C = \frac{C_{min}}{C_{max}} \quad (31)$$

Assuming flow in helical coil heat exchanger as counter flow. The effectiveness is given as [11]

$$\varepsilon = \frac{1 - \exp(-NTU * (1 - C))}{1 - C * \exp(-NTU * (1 - C))} \quad (32)$$

The second law efficiency which is the ratio of desired exergy output to exergy used.Kotas [15]
Rational efficiency or Exergetic efficiency:

$$\psi = \frac{\dot{E}_{desiredoutput}}{\dot{E}_{used}} \quad (33)$$

The $\dot{E}_{desiredoutput}$ is the sum of all exergy output from the system while $\dot{E}_{used}$ is the exergy input into system and irreversibilities are the exergy destruction as

$$\dot{E}_{used} = \dot{E}_{desiredoutput} + \dot{I} \quad (34)$$

$\dot{E}_{desiredoutput}$ Can be expressed in terms of effectiveness and inlet temperature and

$$\dot{E}_{desiredoutput} = C_{max} * \left[\varepsilon * C * (T_{ex} - T_{sco2}) - T_o * \ln\left\{1 + \left(\frac{\varepsilon * C * (T_{ex} - T_{sco2})}{T_{sco2}}\right)\right\}\right] \quad (35)$$

Irreversibility is the product of the reference temperature and entropy generation rate.

Irreversibility given by $\dot{I}$

$$\dot{I} = T_o * N_s * C_{min} = T_o * \dot{S}_{gen} \quad (36)$$

Entropy generation number is defined as the ratio of entropy generation to minimum heat capacityrate, i.e., C_{min}[10]
Entropy generation number

$$N_s = N_{s_H} + N_{s_P} \quad (37)$$

Entropy generation number are given as
Entropy generation number because of heat transfer

$$N_{s_H} = \ln\left[1 + \varepsilon * \left\{\left(\frac{T_{ex}}{T_{sco2}}\right) - 1\right\}\right] + \left[\left(\frac{1}{C}\right) * \ln\left\{1 - C * \varepsilon * \left(1 - \frac{T_{sco2}}{T_{ex}}\right)\right\}\right] \quad (38)$$

Entropy generation number because of pressure drop

$$N_{s_P} = \left(\frac{R_{sco2}}{cp_{sco2} * C}\right) * \left(\frac{\delta P_{SCO2}}{P_{SCO2}}\right) + \left(\frac{R_{ex}}{cp_{ex}}\right) * \left(\frac{\delta P_{ex}}{P_{ex}}\right) \quad (39)$$

3. Results and Discussion

The basic input parameters which are taken from typical marine gas turbine[16], are used for the heat exchanger design and are summarized below in the table 1 to 3. 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.

Table 1: Heat Exchanger Input Geometric Parameters

R_{out}(m)	R_{in} (m)	L_m (m)	d_i(m)	P (m)	th (m)
3.5	1.5	1.0	0.03	0.05	0.001754

Table 2: Heat Exchanger Input Operating Parameter for Exhaust Gas and SCO2

	Exhaust Gas 100% load	Exhaust Gas 71% load	Exhaust Gas 49% load	Supercritical CO_2
T (K)	843	798	761	308
$\dot{m}$ (Kg/s)	63	55	48	35
P (KPa)	200	170	140	20000

Table 3: Properties of Exhaust gas and SCO2 at various load

	Exhaust Gas 100% load	Exhaust Gas 71% load	Exhaust Gas 49% load	Supercritical CO_2
P (Kg/m^3)	0.8266	0.7422	0.6409	866.4
K (W/m-K)	0.0594	0.05705	0.05506	0.09851
C_p (kJ/kg-K)	1.108	1.098	1.090	2.2
μ (kg/m-s)	0.00003763	0.00003635	0.00003528	0.00008368

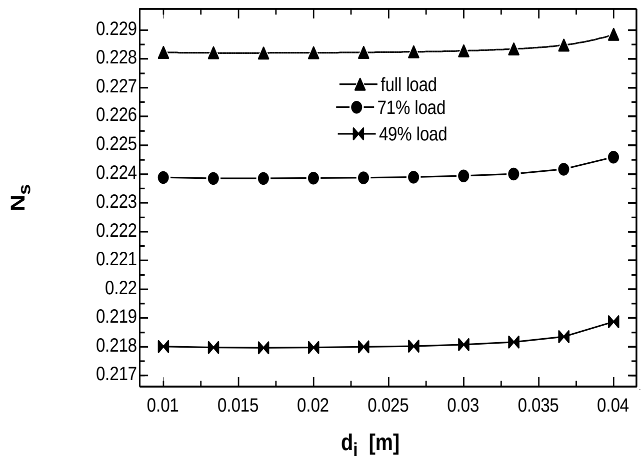

Fig 1Overall Entropy Generation Number of Heat Exchanger VS tube inner diameter

Fig 1demonstrate 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 by virtue of pressure drop decreases because it dominates here. After getting minimum value of overall entropy generation number it starts increasing because entropy generation because of 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.

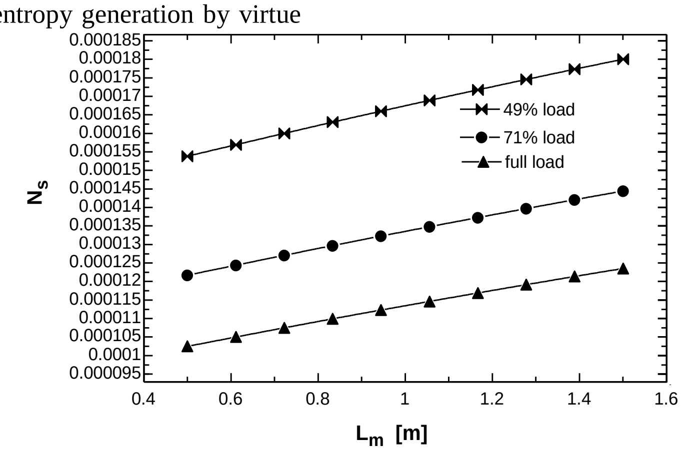

Fig 2Entropy Generation Number due to Pressure Drop VS Tube bundle length

Fig 2 illustrate entropy generation number because of pressure drop verses tube bundle length. Entropy generation number by virtue of pressure drop is increasing with tube bundle length. As length increases pressure drop will increase so entropy generation number by virtue of pressure drop will increase. So overall entropy generation will increase.

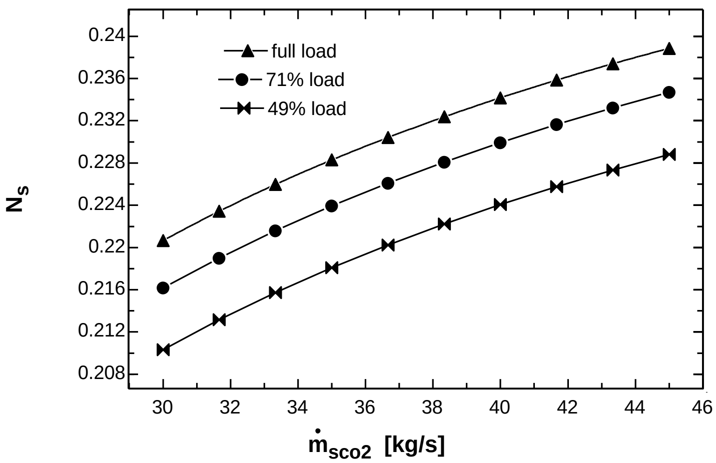

Fig 3 Overall Entropy Generation Number of Heat Exchanger VS SCO2 mass flow rate

Fig 3illustrates 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 because of pressure drop will increase which dominates entropy generation as a result of heat transfer therefore overall entropy generation number increases with increase in SCO2 mass flow rate.

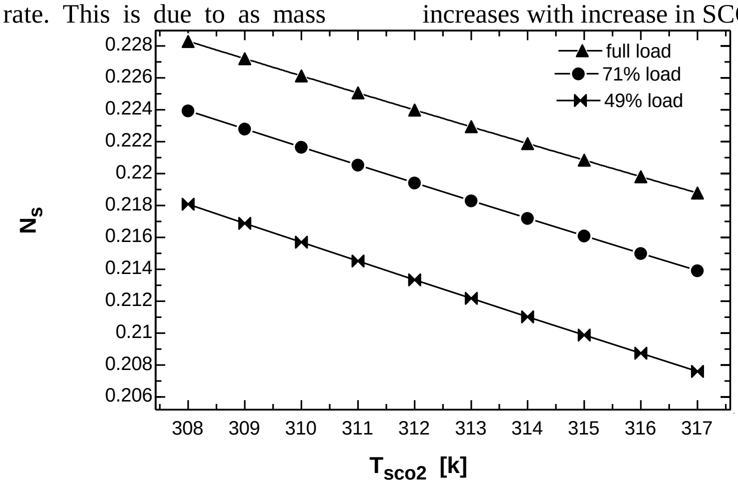

Fig 4Overall Entropy Generation Number of Heat Exchanger VS SCO2 inlet temperature

Fig 4demonstrate 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 because of heat transfer will decrease and it will dominate entropy generation by virtue of pressure drop therefore overall entropy generation number will decreasing with increase in SCO2 inlet temperature.

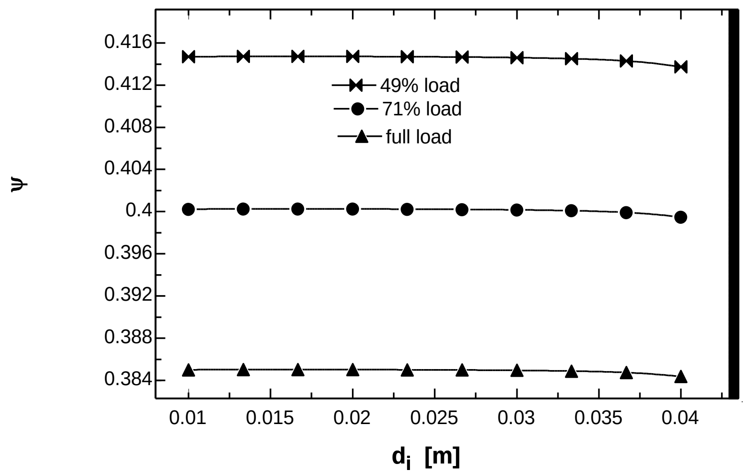

Fig 5 Rational Efficiency of Heat Exchanger at various load VS tube inner diameter

Fig 5portrays 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, becomes maximum at 0.015 m diameter and then decreases. Initially when diameter increases then pressure drop decreases so entropy generation by virtue of pressure drop will decrease and it dominates here sorational Efficiency increases. It starts decreasing because entropy generation as a result of to heat transfer will increase.

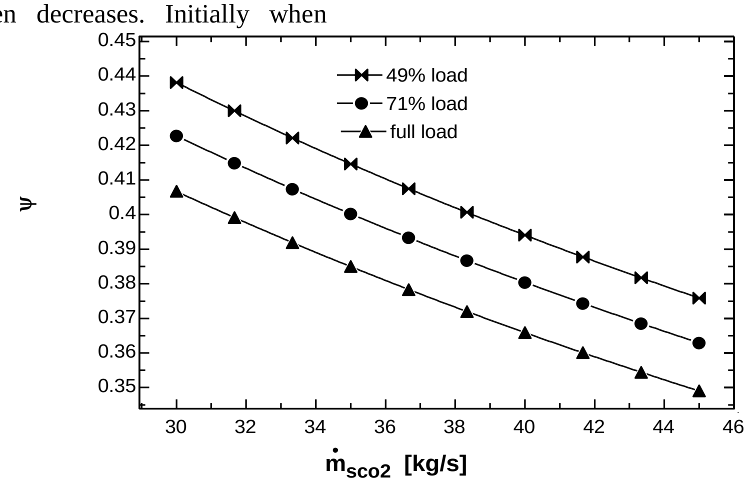

Fig 6 Rational Efficiency of Heat Exchanger at various load VS SCO2 mass flow rate

Fig 6Illustrates 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 as a result of to as mass flow rate increases, pressure drop also increases, due to this entropy generation because of pressure drop also increases therefore rational efficiency will decrease.

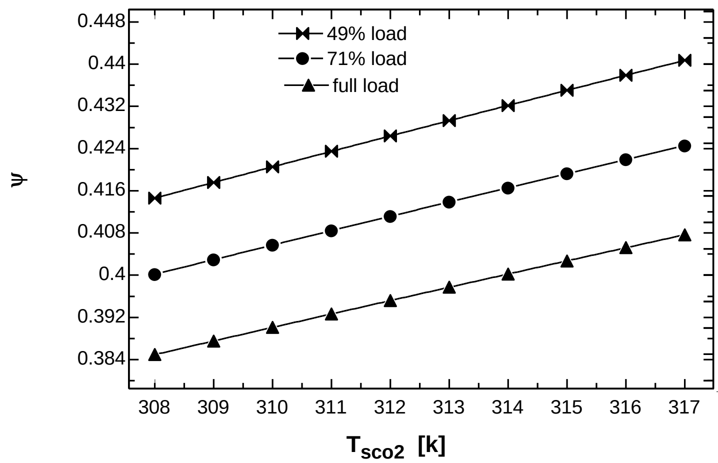

Fig 7 Rational Efficiency of Heat Exchanger at various load VS SCO2 inlet temperature

Fig 7demonstrates 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 as a result to heat transfer will decreasetherefore rational efficiency will increase.

4. Conclusions

From analysis of second law, the following are some of the conclusions to improve the overall performance of the system.The second law efficiency or rational efficiency is having a maximum value for varying the diameter of the tube, which results in optimum value of the diameter at 0.015 m. As mass flow rate increases,second law efficiency decreases, but it increases as inlet temperature of SCO2 increases. In entropy generation number behaviour also, we are able to obtain minimum value for varying the diameter and the value obtained is 0.015 mcorresponding to lower irreversibility. As the mass flow rate of SCO2 increases, entropy generation number increases but entropy generation number decreases for increase in inlet temperature of SCO2. These results are useful for heat exchanger's thermal design based on entropy generation minimization.Here optimum diameter can be achieved by second law analysis which can't be obtained easily by first law thermodynamics analysis. Second law of thermodynamic combines entropy generation by virtue of heat transfer and pressure drop which is not possible from first law of thermodynamics.

References

1. 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.

2. 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).

3. Sarkar, J., 2009. Second law analysis of supercritical CO 2 recompression Brayton cycle. *Energy*, *34(9*), pp.1172-1178.

4. Rao N. T., OumerA.N., Jamaludin U.K.,2016, State-of-the-art on flow and heat transfer characteristics ofsupercritical CO2 in various channels. *The Journal of Supercritical Fluids, 116*, pp.- 132–147.

5. 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.

6. 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.

7. CoronelP., SandeepK.P., "Pressure Drop and Friction Factor in HelicalHeat Exchangers Under No Isothermal and Turbulent Flow Conditions", Journal of Food Process Engineering 26, 2003, pp. 285-302.

8. 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.

9. Manjunath K., Kaushik S. C.,2015. Second Law Efficiency Analysis of Heat Exchangers. *Heat Transfer—Asian Research, 44 (2)*, pp. 89-108.

10. 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).

11. Shah, R. K., and D. P. Sekulic, 2003, *Fundamentals of Heat Exchanger Design*, Hoboken, NJ, John Wiley & Sons.

12. Zukauskas, A., and R. Ulinskas, 1988, *Heat Transfer in Tube Banks in Crossflow*, Hemisphere Publishing, Washington, DC.

13. Kakac, S., and H. Liu, 1998, *Heat Exchangers: Selection, Rating, and Thermal Design*, CRC Press,Boca Raton, FL.

14. Martin, M., 1992, *Heat Exchangers*, Hemisphere Publishing, Washington, DC.

15. Kotas TJ. 1985. *The exergy method of thermal designt analysis*. Butterworths.

16. http://www.geaviation.com/marine/engines/military/lm2500