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Varun Singh
University of Maryland

Vikrant Aute
University of Maryland

Reinhard Radermacher
University of Maryland

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Varun Singh[1]*, Vikrant Aute[2], Reinhard Radermacher[3]
Center for Environmental Energy Engineering
Department of Mechanical Engineering, University of Maryland
College Park, MD 20742 USA
Tel: 301-405-8726, Fax: 301-405-2025
Email: [1]vsingh3@umd.edu, [2]vikrant@umd.edu, [3]raderm@umd.edu
*Corresponding author

ABSTRACT

A heat exchanger is usually characterized by two types of thermodynamic losses. First of these two losses is associated with the heat transfer across a finite temperature difference. The second loss is due to pressure drop due to friction in a heat exchanger. The loss associated with heat transfer across a finite temperature difference can be mitigated by increasing the heat transfer area and reducing the local temperature difference through enhancements. However, increasing the heat transfer area can lead to greater overall frictional loss and higher pressure drop. This shows that the two losses are mutually conflicting, which points to the existence of an "optimum" heat exchanger design where these two losses are minimized. The two losses can be quantified by a single number which is the total entropy generated in the heat exchanger. Total entropy generation should be minimized to arrive at an optimum heat exchanger design. In this paper, the principle of entropy generation minimization is applied as an objective to an air-cooled tube-fin heat exchanger design optimization problem using a heat exchanger modeling tool. The solution space is analyzed to understand the usefulness of entropy generation as an optimization criterion.

1. INTRODUCTION

Air cooled heat exchangers are widely used in air conditioning as well as refrigeration applications. Any heat transfer process is characterized by two kinds of losses, heat transfer through a finite temperature difference and pressure drop in heat transfer fluids. Entropy generation combines the two processes in one property that can be evaluated. There is abundant work in the area entropy generation minimization and most of it is analytical in nature. Entropy generation minimization was first proposed by McClintock (1951) who developed equations for optimum design of fluid passages for a heat exchanger. Bejan (1977) examined the coupling losses due to heat transfer across a finite temperature difference and frictional pressure drop. He proposed the use of number of entropy generation units, $N_{S,}$ as a basic parameter in describing heat exchanger performance. Poulikakos and Bejan (1982) established the theoretical framework for the minimization of entropy generation for extended surfaces (fins). They developed an entropy generation rate formula for a general fin, and then applied the analytical methods and graphical results developed as a result, for selecting optimum dimensions of fins. Witte and Shamsundar (1983) proposed a thermodynamic efficiency concept for heat exchange devices. The efficiency was written in terms of mean absolute temperatures of the two fluids and the appropriate environment temperature. They applied the concept to typical heat exchange cases to demonstrate its usefulness. Sekulic (1986) defined enthalpy exchange irreversibility norm (EEIN) as a measure of internal heat exchanger irreversibility. Aceves-Saborio et al. (1987) extended the entropy generation minimization method to account for exergy of the material of construction. Sekulic (1990) proposed a second law quality of a heat exchange process in heat exchanger analysis. However he ignored the contribution of fluid friction. All of these studies involve an analytical approach to heat exchanger irreversibility analysis. With the maturity of heat exchanger modeling tools, the concept of entropy generation minimization can be applied more readily to heat exchanger design, selection and optimization. There are several heat exchanger models available in literature (Singh et al. 2008, Jiang et al. 2006, Liu et al. 2004, Oliet et al. 2002, Liang et al. 2001, Domanski et al 1999).

The objective of this paper is to use entropy generated in a heat exchanger in a multi-objective optimization of heat exchanger, with heat load and cost as primary objectives and entropy generation as a constraint. Further, heat load is replaced by entropy generation as the objective, and the solution space for the two optimizations is compared to establish the usefulness of entropy generation as an optimization objective. The optimization is carried out through exhaustive search which ensures that the variable space remains constant for all tests. This plays an important role when the solution space for different optimization problems is compared.

2. HEAT EXCHANGER MODEL

The heat exchanger model developed by Jiang et al. was employed. This model divides the heat exchanger into several tube-fin macro volumes, and employs the effectiveness-NTU (ε-NTU) method to solve for outlet states of the two fluids. To allow generalized circuitry, the model employs a junction-tube connectivity matrix which is used to track refrigerant flow from inlet to outlet of the heat exchanger. The computational sequence is generated at run-time based on heat exchanger circuitry. The model distributes mass flow rate through circuits of different length, based on pressure drop. In the current work, refrigerant mass flow was evenly distributed among circuits. Furthermore, the heat exchanger is discretized into tube-fin macro volumes to account for non-uniform distribution of air flow on coil face, as well as accurate calculation of heat load and pressure drop through the tubes.

3. APPROACH

This study investigates the applicability of entropy generation minimization to heat exchanger design in terms of improving a heat exchanger's performance. Number of entropy generation units describes the irreversibility rate of a heat exchanger using a non-dimensional number. $N_S \rightarrow 0$ implies an almost perfect heat exchanger where both temperature difference and pressure drop losses approach zero. On the other hand, a high N_S implies high losses owing to temperature difference or pressure drop or both. The following section outlines the formulation implemented for the current work, and first developed by Bejan (1977)

3.1. Entropy Generation

Figure 1 shows a segment of a heat exchanger of length dx. Fluid is flowing from left to right and the fluid control volume is shown by the dashed line. Entropy generated by such a unit of heat exchanger is given by the following equation.

$$d\dot{S}_{gen} = \dot{m}ds - \frac{q'dx}{T + \Delta T} \tag{1}$$

Number of entropy generation units is defined as

$$N_S = \frac{Td\dot{S}_{gen}}{q'dx} \tag{2}$$

Given the segmented nature of Jiang et al.'s model, it readily facilitates the use of entropy generation units to analyze the irreversibility losses in a heat exchanger. It should be noted that pressure drop is incorporated in the *ds* term.

3.2. Effect of Wall Fluid ΔT on the Number of Entropy Generation Units

As shown in Figure 1, Bejan (1977) illustrated the existence of an optimum $\Delta T/T$ where the proper trade off between fluid friction losses ($N_{S,\ \Delta P}$) and heat transfer losses ($N_{S,\ \Delta T}$) occurs. When $\Delta T/T < (\Delta T/T)_{opt}$, the heat transfer losses are small compared with the fluid friction losses. When $\Delta T/T > (\Delta T/T)_{opt}$ the number of entropy generation units is dominated by pressure drop losses. However, Bejan assumed that the air side behavior would mimic the refrigerant side behavior in terms of irreversibility. Though this might be true for entropy generated due to heat transfer, it is not necessarily true for entropy generated due to pressure drop. In the current study, authors combine the entropy generated on both sides of the heat exchanger, which is non-dimensionalized based on local flux and local wall temperature, as shown in equation 2 to obtain number of entropy generation units.

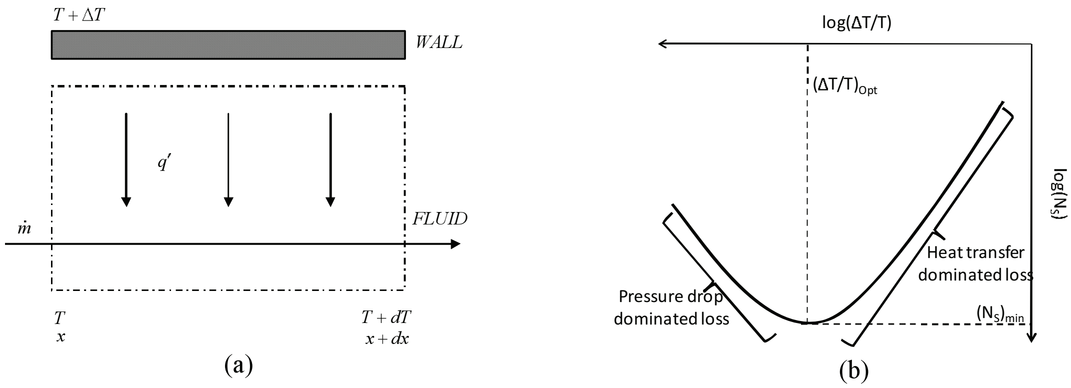


Figure 1: Schematic of entropy generation and optimum point of operation.

To reduce the irreversibility losses in a heat exchanger, the entropy generated throughout the heat exchanger must be minimized. To evaluate this, the authors developed a metric as described in equation 3, to obtain the total entropy generated in a heat exchanger. This was facilitated by the segmented nature of Jiang et al.'s heat exchanger model.

$$Metric = \sum_{nMax} N_S \tag{3}$$

Where, nMax is the total number of segments in the heat exchanger.

3.3. Entropy Generation Units and Heat Exchanger Performance

While it is evident that entropy generation units quantify the irreversibility losses in a heat exchanger, it is important to understand the heat load and pressure drop performance of a heat exchanger when the entropy generation is minimized. To study this, an R134a condenser and a hot water coil were chosen. A parametric study was performed by varying the tube length while the air flow rate was kept constant. As shown in Figure 2 and Figure 3, increase in heat load flattens out after a certain point due to pinching, but refrigerant side pressure drop continues to increase. When the metric described in equation 3 is compared with various tube lengths, it is evident that the minimum entropy is generated when the heat load gain diminishes but the refrigerant side pressure drop continues to increase. This gives an optimum tube length in terms of minimum entropy generation.

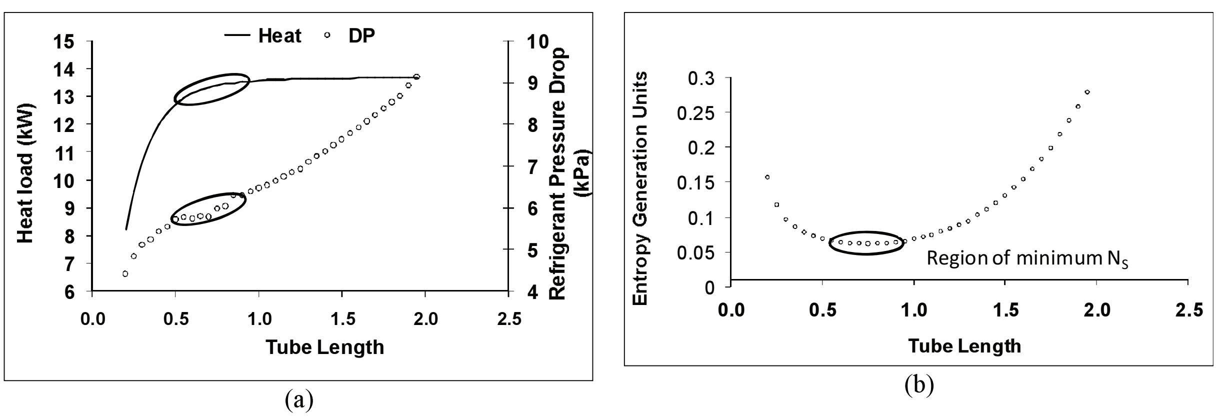
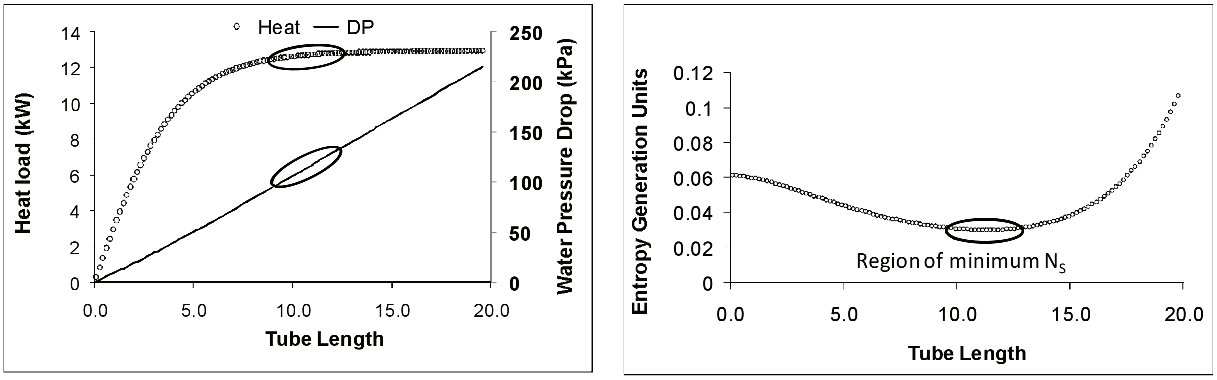


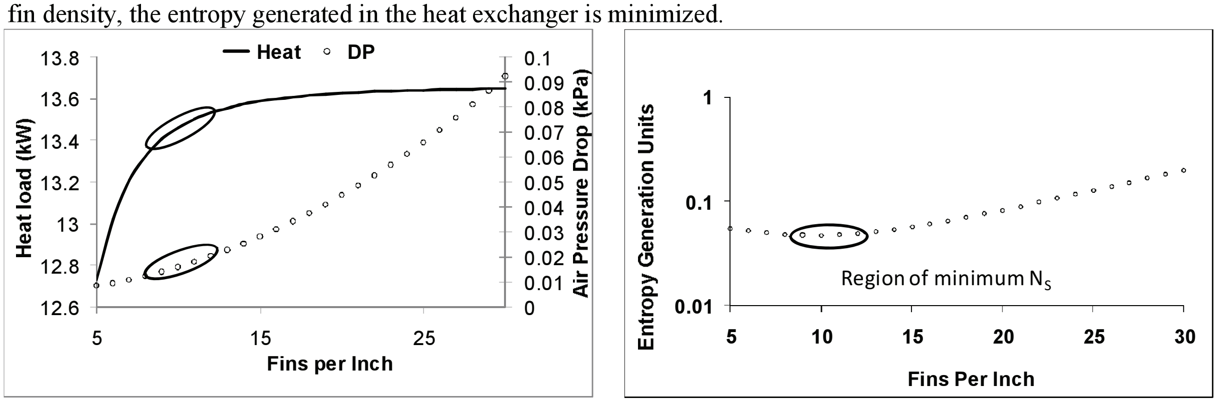
Figure 2: Effect of varying tube length for R134a condenser on entropy generation, heat load and refrigerant pressure drop



(a) (b)

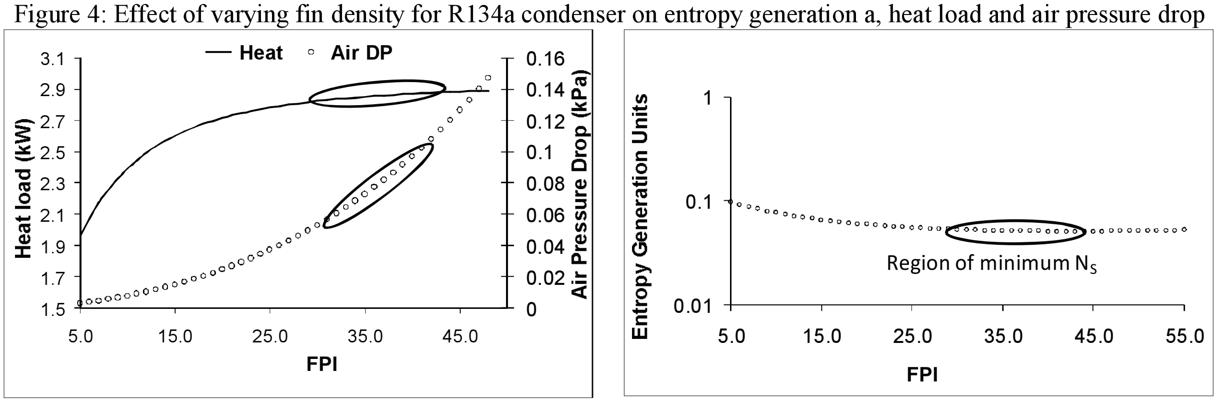
Figure 3: Effect of varying tube length for hot water coil on entropy generation, heat load and refrigerant pressure drop

Similarly, if a parametric study is conducted with varying fin density, the results as shown in Figure 4 and Figure 5, show that gain in heat load diminishes after a certain point but air side pressure drop continues to increase. At that fin density, the entropy generated in the heat exchanger is minimized.



(a) (b)

Figure 4: Effect of varying fin density for R134a condenser on entropy generation a, heat load and air pressure drop



(a) (b)

Figure 5: Effect of varying fin density for hot water coil on entropy generation a, heat load and air pressure drop

From these results, it can be inferred that entropy generation is minimized when fluid pressure drop continues to increase whereas heat load gain disappears. It is evident that minimized entropy generation helps quantify the pinch off location in a heat exchanger. It is also clear that in some cases like a study on fins per inch for a hot water coil, minimization of entropy generation might lead to economically expensive designs.

4. OPTIMIZATION STUDY

To better understand the usefulness of entropy generation minimization as a parameter, two optimization problems were set up as shown in Table 1. Problem A minimized cost and maximized heat load of a given heat exchanger for four variables, viz., tube length, tube vertical spacing, tube horizontal spacing, and fin density. Problem B minimized entropy generation and minimized cost of a given heat exchanger for the same variables as problem A. Because the optimization was carried out by exhaustive search, the design points for the two problems were same. This allowed the comparison of the solution spaces of the problem A and problem B.

Table 1: Problem formulation

PROBLEM A	**PROBLEM B**
Objectives: minimize material cost maximize heat load	Objectives: minimize material cost minimize entropy generation units
Variables: Tube length [0.3m, 0.6m] Tube vertical spacing [19.05mm,38.1mm] Tube horizontal spacing [12.7mm, 25.4mm] Fins per inch [10, 25]	Variables: Tube length [0.3m, 0.6m] Tube vertical spacing [19.05mm,38.1mm] Tube horizontal spacing [12.7mm, 25.4mm] Fins per inch [10, 25]

4.1. Hot Water Coil

The specifications of the hot water coil tested are shown in Table 2.

Table 2: Specification of hot water coil

Parameters		
Number of Segments	10	---
Tube Configuration	Staggered	
Number of Tubes Per Bank	8	---
Number of Tube Banks	2	---
Tube Length	0.8	m
Tube OD	0.0084	m
Tube Thickness	0.3	mm
Tube Vertical Spacing	1	in
Tube Horizontal Spacing	0.625	in
FPI	22	fpi
Fin Thickness	0.0043	in
Fin Type	Louver	---
Coil Face Air Flow Rate	185	cfm

For problem A, the number of design points that were Pareto optimal contained 60 points out of a total of 1800 design points evaluated. For problem B, the number of design points on the Pareto optimal front was 63 out of 1800 design points evaluated. Out of 63 points found as a solution for problem B, 55 points were the same as the ones in the solution space for problem A. Figure 6 shows the comparison of the two solution spaces. It is evident that even the points that are not present in both the solutions also lie close to the Pareto optimal front. It is important to note that coils with minimum entropy generation units tend to be more costly from a material requirement point of view.

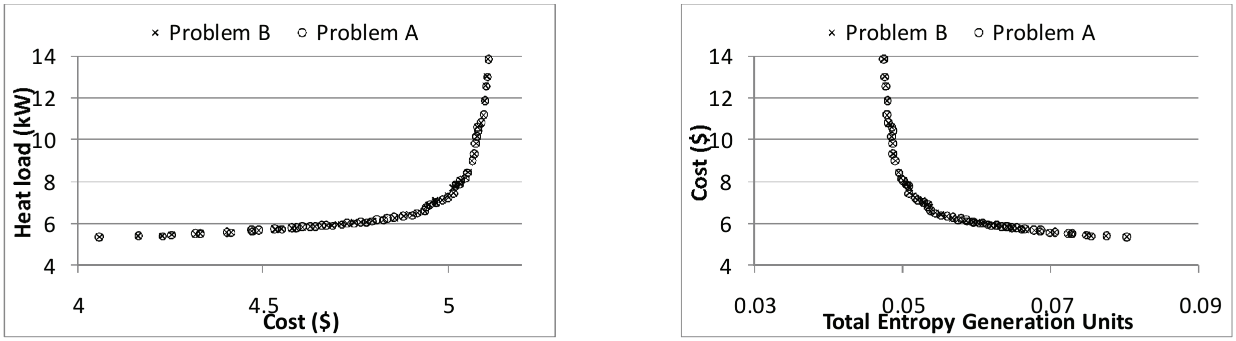


Figure 6: Comparison of solution spaces for Problem A and Problem B

In order to investigate points that are not common to the two solution spaces, it is important to study the refrigerant pressure drop. Comparing cost and heat load per unit refrigerant side pressure drop in Figure 7, it is evident that points unique to the solution space of problem B have a better heat load per unit pressure drop performance when compared to solution of problem A. However, 4 out of 5 points unique to the solution space of problem A have equal or lesser cost than the unique solutions of problem B. This implies that using entropy generation units as an objective in problem B has successfully accounted for losses associated with pressure drop as well. This proves the usefulness of entropy generation minimization as an optimization criteria that includes heat transfer as well as pressure drop.

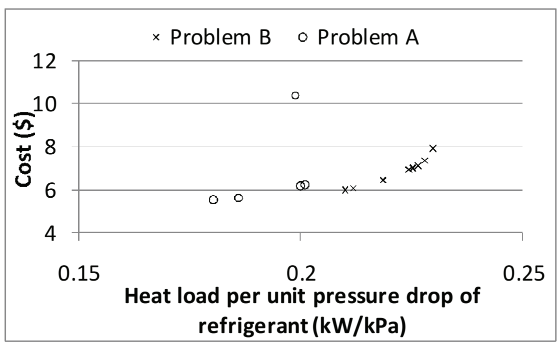


Figure 7: Comparison of unique points in solution spaces of problems A and B

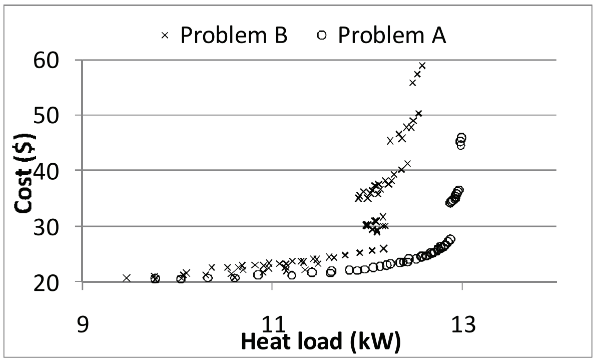
4.2. Condenser

The specifications of the R134a condenser are shown in Table 3.

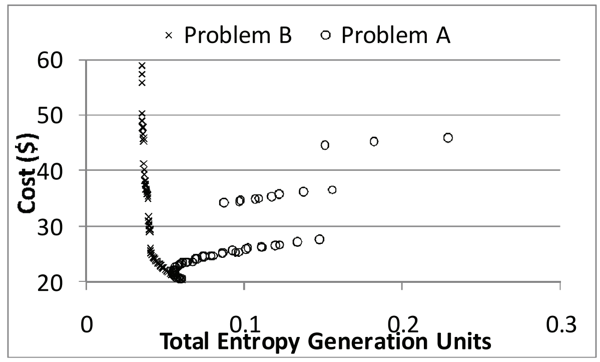
Table 3: Specification of R134a condenser

Parameters		
Number of Segments	10	---
Tube Configuration	Staggered	
Number of Tubes Per Bank	28	---
Number of Tube Banks	3	---
Tube Length	0.45	m
Tube OD	0.01	m
Tube Thickness	0.3	mm
Tube Vertical Spacing	0.75	in
Tube Horizontal Spacing	0.5	in
FPI	15	fpi
Fin Thickness	0.0043	in
Fin Type	Louver	---
Coil Face Air Flow Rate	938	cfm

The mass flow rate of the condenser was chosen such that for all design points evaluated, the entire length of the condenser is two phase. The motivation behind this was to study the usefulness of entropy generation minimization in a two-phase region where refrigerant temperature drop is insignificant but the pressure drop is substantial. The solution for problem A yields 51 points on the Pareto optimal solution, whereas the solution for problem B yielded 72 points. Amongst these solution sets, only 3 points were common. The rest of the solutions in each of the sets were unique. Figure 8 shows that solution to problem B yields points with lower total entropy generation but those points don't necessarily have the highest heat load. Further, it should be noted that these three points have very low material cost.



(a)



(b)

Figure 8: Comparison of solution spaces of problem A and problem B for condenser

To better understand the reason behind this, it is important to closely examine a design point in each solution which has the same heat load but completely different total entropy generation units. For this, the design points with heat load closest to 12.5kW were selected from the two solution spaces. Figure 9 shows the entropy generated in every segment of the heat exchanger with respect to local temperature gradient between bulk fluid and the wall. The superimposed curves represent the general shape of the entropy generation units profile as proposed by Bejan and shown here in Figure 1. It is evident that the heat exchanger from the solution space of problem B has much lesser number of total entropy generation units, when compared to the point from solution space for problem A. The main reason behind this behavior is the lack of any substantial change in approach temperature over the heat exchanger length, when compared to substantial pressure drop.

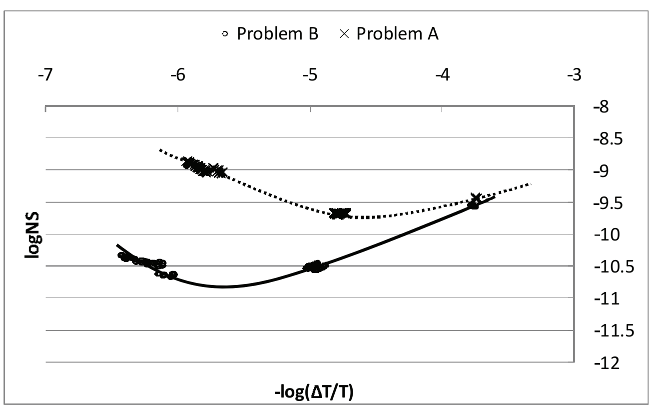


Figure 9: Entropy generation profile for two design points from problem A and problem B

5. CONCLUSIONS

The usefulness of entropy generation minimization in the design of a hot water coil and condenser was examined using a heat exchanger model. It was found that for a hot water coil, minimization of entropy generated and minimization of cost yields the same result as maximization of heat load and minimization of cost. However, for a completely two-phase condenser, the solution spaces for minimizing entropy generated and cost, and maximizing heat load and minimizing cost are almost unique except for a few cases. This result can be explained by examining the formulation of entropy generation units. The premise of entropy generation units is the co-existence of pressure drop and drop in finite temperature difference. However, this does not occur during phase change heat transfer. Therefore, the technique of entropy generation minimization using entropy generation units represents heat load accurately in a single phase flow for the given set of design variables and heat exchanger conditions. For a two-phase heat exchanger, entropy generation units could not be used, instead of heat load, for the given set of design variables and flow conditions.

REFERENCES

Aceves-Saborio, S., Ranasinghe, J., Reistad, G., M.,1987, "An extension to the Irreversibility Minimization Analysis Applied to Heat Exchangers", *J. Heat Transfer*, vol. 111, p. 29-36

Bejan, A., 1977, "The Concept of Irreversibility in Heat Exchanger Design: Counterflow Heat Exchangers for Gas-to-Gas Applications", *Trans. ASME*, vol. 99, p. 374-380

Jiang, H., Aute, V., Radermacher, R., 2006, CoilDesigner: A General-purpose Simulation and Design Tool for Air-to-Refrigerant Heat Exchangers, *Int. J. Refrig.*, vol. 29, p. 601-610.

Liang, S.Y., Wong, T.N., Nathan, G.K., 2001, Numerical and Experimental Studies of Refrigerant Circuitry of Evaporator Coils, *Int. J. Refrig.*, vol. 24, p. 823-833.

Liu, J., Wei, W., Ding, G., Zhang, C., Fukaya, M., Wang, K., Inagaki, T., 2004, A General Steady State Mathematical Model for Fin-and-Tube Heat Exchanger based on Graph Theory, *Int. J Refrig.*, vol. 27, p. 965-973

McClintock, F.A., 1951, "The Design of Heat Exchangers for Minimum Irreversibility", 1951 *ASME Annual Meeting*, Paper No. 51-A-108

Oliet, C., Perez-Segarra, C.D., Danov, S., Oliva, A., 2002, Numerical Simulation of Dehumidifying Fin-and-Tube Heat Exchangers: Modeling Strategies and Experimental Comparisons, p. 1-8.

Poulikakos, D., Bejan, A., 1982, "Fin Geometry for Minimum Entropy Generation in Forced Convection", *J. Heat Transfer*, vol. 104, p. 616-623

Sekulic, D., P., 1986, "Entropy Generation in a Heat Exchanger", *Heat Transfer Eng*, vol. 7, no. 1, p. 83-88

Sekulic, D., P., 1990, "The Second Law Quality of Heat Transformation in a Heat Exchanger", *J. Heat Transfer*, vol. 112, p. 295-300

Singh, V., Aute, V., Radermacher, R., 2008, Numerical Approach for Modeling of Air-to-Refrigerant Fin-and-Tube Heat Exchanger with Tube-to-Tube Heat Transfer, *Int. J. Refrig.*, doi. 10.1016/j.ijrefrig.2008.03.013.

Witte, L., C., Shamsundar, N., 1983, "A Thermodynamic Efficiency Concept for Heat Exchange Devices", *J. Eng. Power*, vol. 104, p. 199-203