Study of Effect of Heat Transfer through Fins in a Fin-and-tube Carbon Dioxide Gas Cooler on its Performance through Numerical Modeling

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ABSTRACT

In fin-and-tube heat exchangers operating at high temperatures such as Carbon Dioxide gas coolers, the heat transfer between neighboring tubes due to fin conduction affects the heat exchanger performance. In this research, a segment-by-segment heat exchanger model has been developed that can account for tube-to-tube conduction in two dimensions. The new model has been validated against experimental data for a gas cooler at 36 test conditions with total heat load varying between 6kW and 12kW. The refrigerant temperatures were measured at 27 locations on the different U-bends in the coil. The predicted heat load agreed within 3% and the refrigerant temperatures, agreed within 3.3K of measured values. The model is capable of simulating a heat exchanger that has both continuous and discontinuous i.e. cut fins. This paper investigates the effect of fin cuts on the performance of a Carbon Dioxide gas cooler through an exhaustive search method. In this approach, tubes are progressively separated through incremental fin cuts, and the change in overall heat load is studied. Further, the affect of fin cuts on refrigerant temperature profile through the heat exchanger is also studied.

1. INTRODUCTION

Refrigerant to air fin-and-tube heat exchangers are widely used in the refrigeration and air conditioning industry. They are used to transfer heat between air and working fluid (e.g., refrigerant, water, glycols etc.) In order to predict their performance accurately, and reduce design and development time, computer models are fast replacing physical prototypes. There are several such models and tools in the literature used to model both steady state and transient behavior that have been validated against experimental results. Several authors have carried out a review of existing heat exchanger models (Singh et al. (2008), Jiang et al. (2006), Liu et al. (2004), Oliet et al. (2002), Liang et al. (2001), Domanski (1999)).

Singh et al. found that most of the models did not account for tube-to-tube conduction through fins. This phenomenon was shown to be of significant importance in the experiments of Jin et al. (2004) and Zilio et al. (2007). Further. Zilio et al. showed that heat exchangers with discontinuous fins (where fins between tubes are cut to insulate tubes from each other) showed a higher heat load than heat exchangers with continuous fins, for same heat exchangers, and refrigerant and air inlet parameters. The applications considered in their experiments employed Carbon Dioxide as a refrigerant, in a gas cooler application. Payne et al. (2003) showed the improvement of evaporator performance due to discontinuous fins. Besides synthetic refrigerants, Carbon Dioxide is one of the few working fluids that are considered natural refrigerants. It gained prominence in the 1980s because of several favorable characteristics like non-flammability, negligible global warming potential and limited toxicity. To increase the accuracy of modeling of Carbon Dioxide gas coolers, Singh et al. developed and validated two heat exchanger models. The models, besides accounting for tube-to-tube heat conduction through fins, is capable of simulating heat exchangers where tubes can be isolated through discontinuous fins i.e., fin cuts on a tube-by-tube basis.

2. HEAT EXCHANGER MODEL

The model developed by Singh at al. is based on the heat exchanger model of Jiang et al. The model is introduced here. For in-depth understanding of the model, the reader is referred to the original publication.

To allow generalized circuitry, the model employs a junction-tube connectivity matrix which allows for tracking of refrigerant flow from inlet to outlet of the heat exchanger. The model distributes mass flow rate through circuits of different length, based on pressure drop. Tubes are divided into several tube-fin macro volumes which allows non-uniform distribution of air flow on coil face, as well as accurate calculation of heat load and pressure drop through the tubes. To solve for refrigerant side heat transfer, energy balance equations are employed which incorporate neighboring tube temperatures into the system of equations. Singh et al. developed two models for achieving this goal, the "resistance model" and the "conduction model". The reader is referred to the original paper for detailed information on the two models.

3. GAS COOLER

Jin et al. tested a carbon dioxide gas cooler at 36 different test conditions shown in Table 1 and Table 2. The heat exchange between air and carbon dioxide occurs in the supercritical region of carbon dioxide.

Table 1 : Carbon dioxide gas cooler specifications

Parameters		
Number of Segments	10	---
Tube Configuration	Staggered	
Number of Tubes Per Bank	18	---
Number of Tube Banks	3	---
Tube Length	0.61	m
Tube OD	0.0084	m
Tube Thickness	0.406	mm
Tube Vertical Spacing	1	in
Tube Horizontal Spacing	0.625	in
FPI	17	fpi
Fin Thickness	0.0043	in
Fin Type	Slit	---
Coil Face Air Velocity	variable	ms^{-1}

Table 2: Carbon dioxide gas cooler test conditions

No	Inlet Air Temp [°F (°C)]	Ref MFR [$lbmin^{-1}$ (gs^{-1})]	Inlet Pressure [psia (MPa)]	Air Frontal Velocity [fpm (ms^{-1})]
1			1,300 (9.0)	200, 400, 600 (1.0, 2.0, 3.0)
2	85 (29.4)	5 (38)	1,450 (10.0)	200, 400, 600 (1.0, 2.0, 3.0)
3			1,600 (11.0)	200, 400, 600 (1.0, 2.0, 3.0)
4			1,300 (9.0)	200, 400, 600 (1.0, 2.0, 3.0)
5		10 (76)	1,450 (10.0)	200, 400, 600 (1.0, 2.0, 3.0)
6			1,600 (11.0)	200, 400, 600 (1.0, 2.0, 3.0)
7			1,300 (9.0)	200, 400, 600 (1.0, 2.0, 3.0)
8	95 (35)	5 (38)	1,450 (10.0)	200, 400, 600 (1.0, 2.0, 3.0)
9			1,600 (11.0)	200, 400, 600 (1.0, 2.0, 3.0)
10			1,300 (9.0)	200, 400, 600 (1.0, 2.0, 3.0)
11		10 (76)	1,450 (10.0)	200, 400, 600 (1.0, 2.0, 3.0)
12			1,600 (11.0)	200, 400, 600 (1.0, 2.0, 3.0)

The specifications of the gas cooler are shown in Table 1. The heat exchanger, as shown in Figure 2, has three banks with continuous fins. Even though the heat exchanger has slit fins, they are not discontinuous (cut) fins and don't insulate tubes from each other. The comparison of experimental heat load and predicted heat load, which was carried out by Singh et al. is shown in Figure 1.

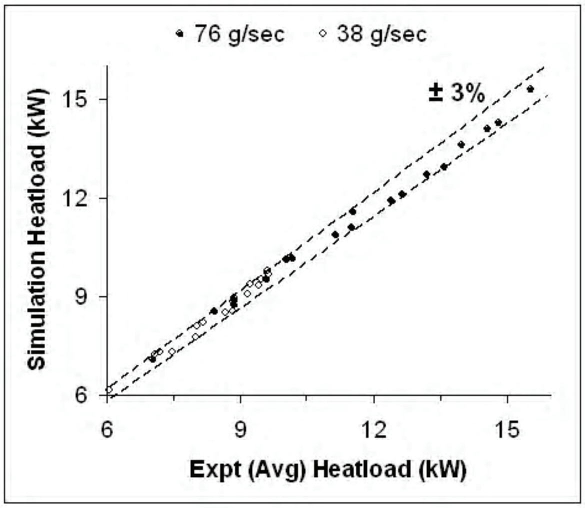


Figure 1: Comparison of experimental and predicted heat load for baseline gas cooler

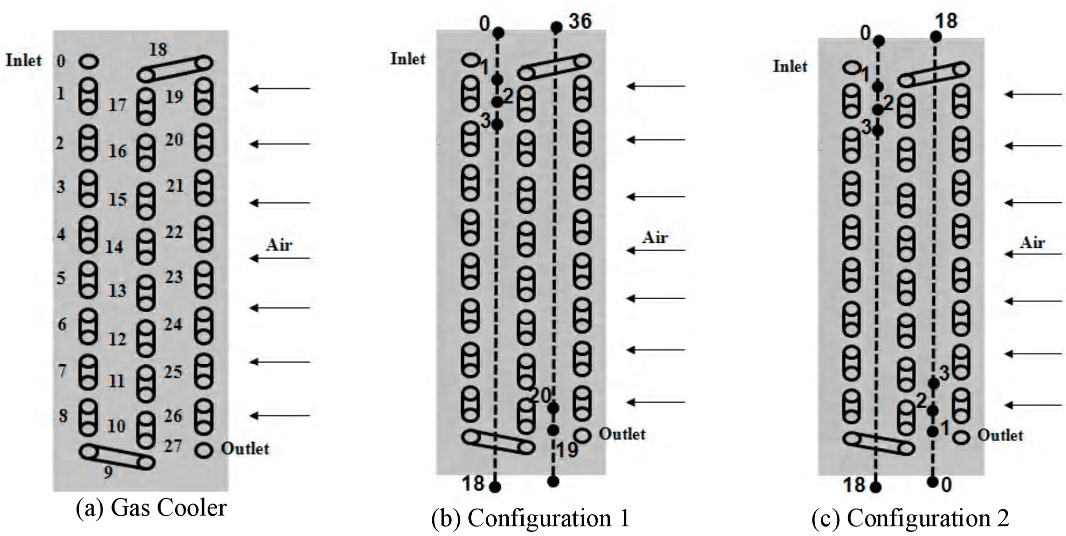


Figure 2: Schematic of the gas cooler with two cut configurations simulated

4. SIMULATION STUDY

To study the effect of cuts on overall performance of the gas cooler, two configurations of cuts were chosen. The configuration, or the sequence in which fins are cut to separate tubes, directly influences the degree of performance alteration of the gas cooler. The configurations chosen for this study are shown in Figure 2.

The first configuration shows that a total of 36 different cut sizes were simulated for all 36 test cases. Cut 0-1 was the first cut, 0-2 being the second and so on till 18, till one bank was completely insulated. Similarly, cut in the second bank were initiated from bottom to top, all the way until the three tube banks were insulated from each other.

The second configuration shows that a total of 18 different cut sizes were simulated for all 36 test cases. Cut 0-1 was the first cut, which was initiated on both the banks with intent to insulate the tubes with the highest temperature gradient. This was carried on incrementally till cut 0-18, which is identical to 0-36 state of configuration 1.

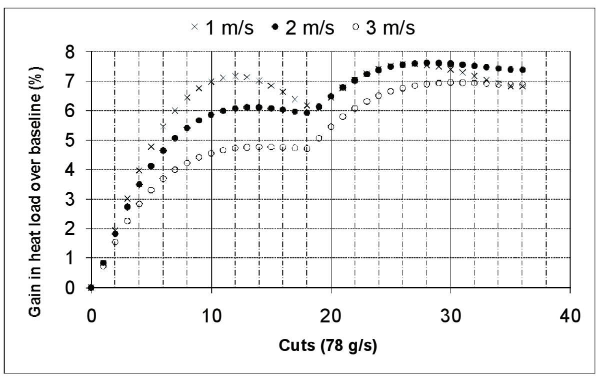
The underlying reason for selecting these two configurations was separation of the highest temperature tubes through discontinuous fins. The highest temperature gradient between tubes leads to the highest heat transfer due to conduction, and this leads to significant performance degradation.

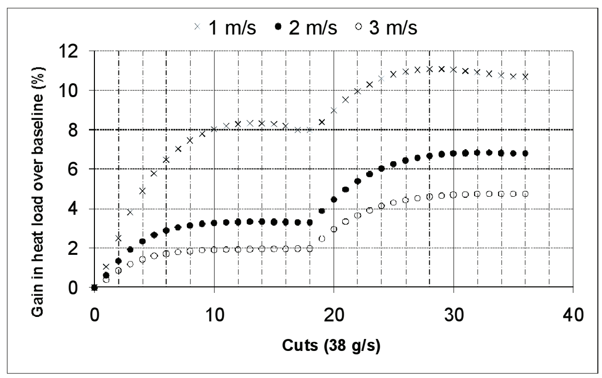
5. RESULTS

The results for studies on the two configurations are presented in this section. Firstly, the two configurations are analyzed independently, and effect of changing air speed is presented. This is followed by the comparison of the two configurations in terms of performance enhancement as a function of length of the cut.

5.1. Configuration 1

To understand the response of heat load to different air velocities, average percent gain in heat load, over baseline, was plotted for all three different velocities on $1ms^{-1}$, $2ms^{-1}$ and $3ms^{-1}$. The results show that for the lowest air velocity test cases gain the most over base heat load for test conditions where mass flow rate for refrigerant is $38gs^{-1}$. However, for 76 gs^{-1}, the heat load gain for the lowest air velocity test condition is higher than other cases for the first 12-18 cuts. However, after cuts ranging from 19-36, the overall heat load gain for air velocities 2 ms^{-1} and $3ms^{-1}$ is higher than the gain for $1ms^{-1}$.

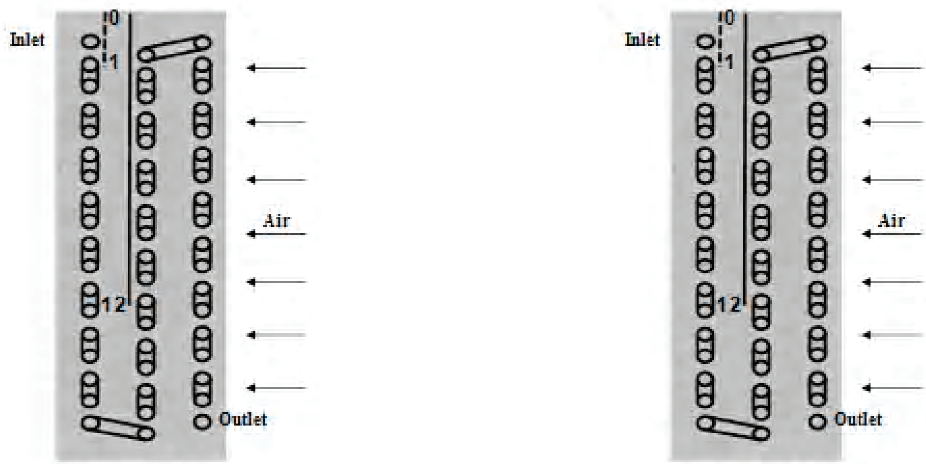




(a) (b)

Figure 3: Gain in heat load, over baseline, for Configuration 1

It is interesting to note that for the test case with 76 gs^{-1} mass flow rate with 1ms-1 air velocity, the overall heat load gain due to cuts ranging from 1-18 peaks at 12 cuts and then diminishes. To better understand this phenomena, the heat exchanger temperature profiles of case with 12 cuts is compared that with 18 cuts, and the case with 1 cut is compared to the case with 12 cuts, as shown in Figure 4.



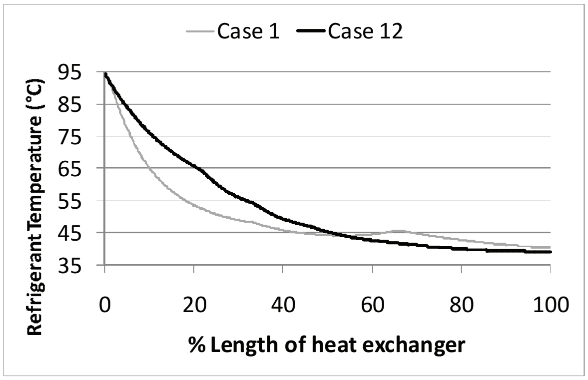
(a) (b)

Figure 4: Two cases compared to understand diminishing gain after 12 cuts for cuts 1-18

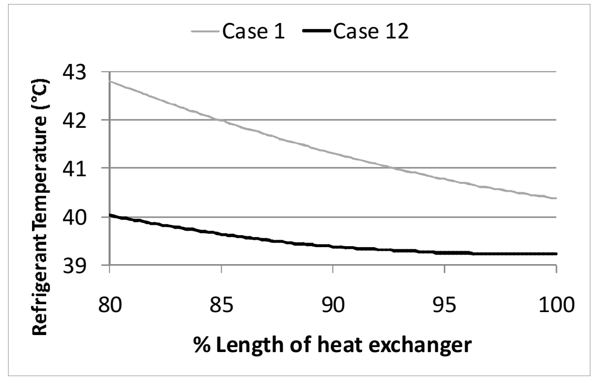
Upon comparing the refrigerant temperature profiles for case with 1 cut and 12 cuts (Figure 5), it is seen that the refrigerant temperature drops rapidly for the case with 1 cut in the first 40% of the heat exchanger area due to fin conduction, but thereafter, the refrigerant temperature increases due to fin conduction. For case with 12 cuts, the fin conduction doesn't influence the later part of the heat exchanger, leading to a steady drop in refrigerant temperature. It should be noted that the drop in the last 20% of the heat exchanger length is more significant for case with 1 cut than the case with 12 cuts.

However, for case 12 and case 18 (Figure 6), the comparison shows that the refrigerant temperature of the case with 12 cuts increases in the last 5 % of the heat exchanger. This is due to the neighboring bank temperature being higher for case 12 than for case 18, which leads to heat conducted to the last 5% of the heat exchanger. Air at $1ms^{-1}$ doesn't have enough heat capacity to remove that excess heat leading to lower heat load for case 18.

This phenomenon is explained in Figure 7, where heat conducted due to fins, heat gained by refrigerant and heat gain by air, are shown for the last segment of the heat exchanger. A positive value of heat indicates that the refrigerant (QRef), air (QAir) and segment due to conduction (Qfin) are gaining heat, whereas negative value implies the opposite. It should be noted that the air side heat load of the last segment is nearly the same for all three cases but the heat conducted by fins is markedly different, as is the refrigerant heat load. Qfin is the highest for case 18 which has the highest temperature gradient with the neighboring tube, followed by case 12 and case 1. Since the air side heat loads are nearly same, due to same air inlet state in all three cases, and nearly same refrigerant temperature, the refrigerant heat load (QRef) compensates for the difference between Qfin and QAir, maintaining the energy balance in the segment. This validates the refrigerant temperature behavior in the last 20% of the heat exchanger length for cases 1, 12 and 18, and explains the degrading performance of the heat exchanger from cut 12 to cut 18.

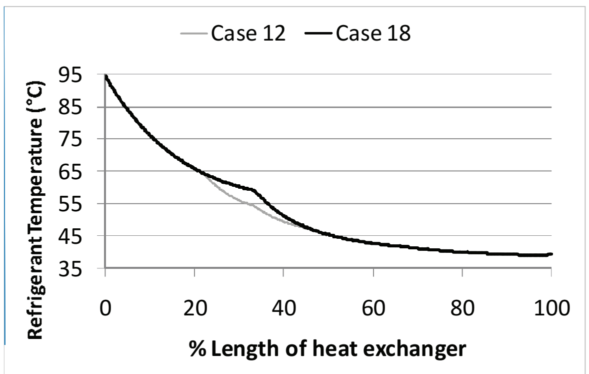


(a)

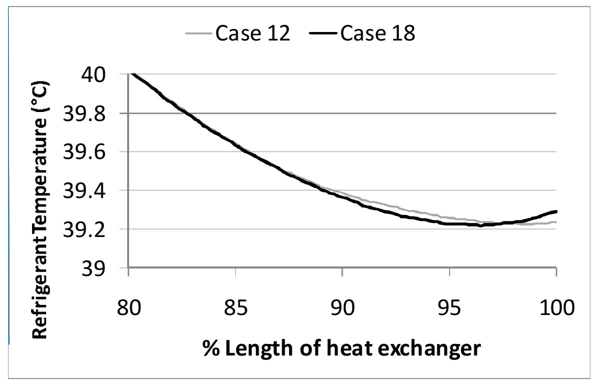


(b)

Figure 5: Comparison of refrigerant temperature profile for case 1 and case 12, for Configuration 1



(a)



(b)

Figure 6: Comparison of refrigerant temperature profile for case 12 and case 18, for Configuration 1

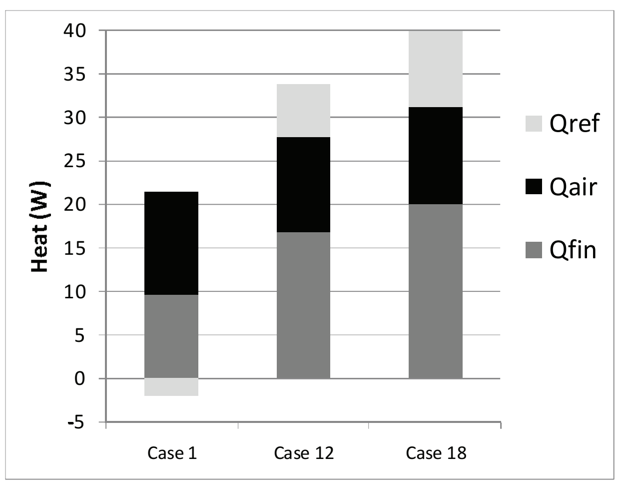
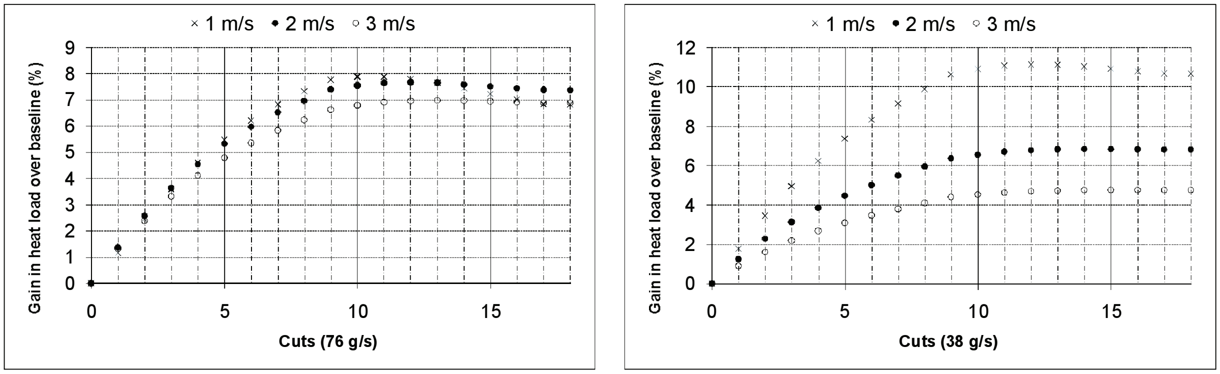


Figure 7: Heat gained by refrigerant, air and segment (due to conduction) in the last heat exchanger segment

5.2. Configuration 2

For configuration 2, there are a total of 18 incremental cuts with both the slabs being cut simultaneously as shown in Figure 2. When gain in heat load is compared (Figure 8), the maximum gain is for the lowest air velocity test conditions for 38gs^{-1} test cases. However for refrigerant mass flow rate 76gs^{-1}, all three air velocity test cases (1ms^{-1}, 2ms^{-1}, 3 ms^{-1}) show a comparable gain in heat load. Similar to the configuration 1, for refrigerant mass flow rate 76

gs^{-1} and 38 gs^{-1}, and air velocity $1ms^{-1}$, after a certain number of cuts, any further cuts lead to a diminishing gain in heat load.

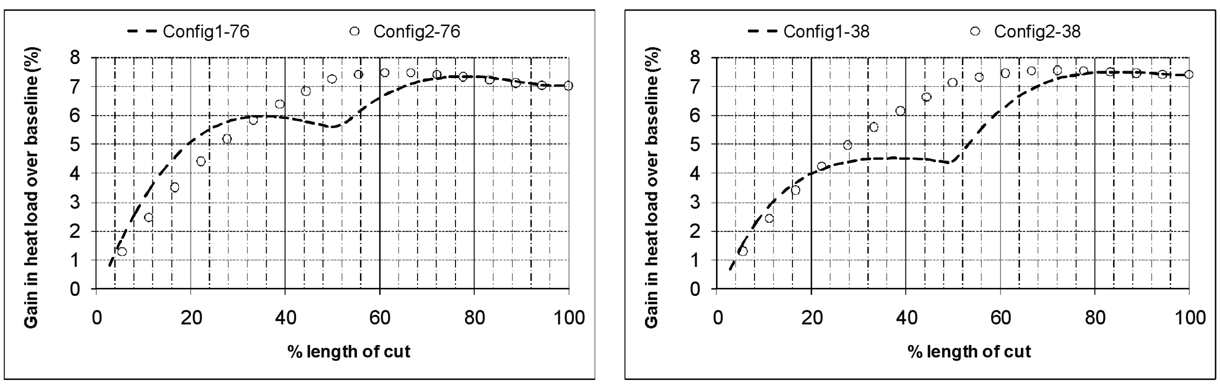


(a) (b)

Figure 8: Gain in heat load, over baseline, for Configuration 2

5.3. Comparison of configuration 1 and configuration 2

To highlight the significance of location of cut in addition to the length of cut, along with the dependence of refrigerant flow rate and conditions, the average gain in heat load for all three velocity test conditions ($1ms^{-1}$, $2ms^{-1}$, 3 ms^{-1}) is compared for non-dimensionalized cut length for $38gs^{-1}$ and 76 gs^{-1} refrigerant mass flow rate test conditions. From Figure 9, it is evident that if only up to 36% of the heat exchanger can be cut, configuration 1 gives greater gain than configuration 2 for $38gs^{-1}$ refrigerant mass flow rate. For cut length between 36% and 80%, configuration 2 provides greater heat load gain than configuration 1. For 76 gs^{-1} mass flow rate test conditions, the heat load gain in configuration 2 is higher than configuration 1 if the cut length is between 20% and 80%. For all other cut lengths, the gain in heat load from the two configurations is similar.



(a) (b)

Figure 9: Comparison of configuration 1 and configuration 2

6. CONCLUSIONS

A heat exchanger model capable of simulating a heat exchanger with certain tubes insulated from each other through discontinuous fins is introduced. Using the model, the improvement in heat exchanger performance due to discontinuous fins is shown. Two different configurations of cut patterns were modeled. It is shown that with increasing cut length, the gain in heat load increases in most cases. It was shown that for certain test conditions, the gain in heat load can be up to 12% over the baseline. The different trends of gain in heat load for the two configurations, given the same length of cut (or discontinuous fins) showed the importance of location of cut with respect to tube temperatures and air states. This highlights the importance of optimum location of fin cuts for varying refrigerant state and air inlet maldistribution.

NOMENCLATURE

$\dot{Q}$	Heat transfer rate, W
Ref	Refrigerant, CO_2

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