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ABSTRACT

Improving energy efficiency and reducing the cost of the appliance simultaneously is a continuing challenge for refrigerator manufacturers. While conventional system configuration is based on a single evaporator vapor compression cycle, there are several other system configurations that can offer benefits over it. For example, a dual evaporator (one evaporator for freshfood and one for freezer) offers several benefits such as increased efficiency, isolation of odors, and higher humidity levels in the freshfood compartment. Engineers typically use extensive experimentation to optimize the system. This approach takes significant time and resources. Although optimization studies exists for a conventional single evaporator cycle, studies for dual evaporator cycle optimization are limited. Most manufacturers do not explore complex architecture due to time consuming, labor intensive and expensive development procedures. This study presents experimental results obtained from a prototype dual evaporator refrigerator. Further, this study presents a physics-based model and a multi-objective optimization methodology that demonstrates how engineers can optimize a refrigeration system by considering multiple objectives simultaneously. The study presents example optimization results for simultaneously minimizing cost and maximizing performance within a specified design space. Optimization of the novel design uses a genetic algorithm-based optimizer in conjunction with a response surface based metamodel. Using optimization techniques, we can arrive at lowest cost design relatively quickly as shown in the analysis. More work needs to be done to validate optimized solutions as well as alternate methods to improve temperature control.

1. INTRODUCTION

Typically, there are two compartments in the domestic refrigerator: freshfood compartment and frozen food compartment. The dual evaporator system has one individual evaporator for each compartment. There are several advantages of it over a single evaporator system. A single evaporator unit usually has a lower airflow rate in the freshfood compartment, resulting in a large temperature variation. This reduces shelf life of the products being stored. With an individual evaporator in each compartment, the airflow rate in the freshfood compartment can be increased, which improves the temperature uniformity. With a dual evaporator system, the airflow for each compartment is isolated resulting in higher humidity in the freshfood compartment. Energy consumption is improved because of the reduction in defrost demand for the evaporator in the freezer compartment. Also, if the freshfood evaporator is operated at a higher temperature, irreversibility is reduced due to the temperature lift. Because of the isolation of airflows, the odors from each compartment are isolated. The major drawback of a dual evaporator system is an increase in material cost. Additional parts require additional tooling and manufacturing cost. Increasing brazed connections also increases opportunities of refrigerant leakage. Furthermore, a dual evaporator system adds complexity to the temperature controller. Various configurations have been explored by researchers to achieve a dual evaporator system. Lavanis et al. (1998) have presented an experimental investigation of an alternating evaporator cycle. Gan et al. (2000) have investigated the application of a dual system. Gerlach et al. (2003) presented an experimental investigation of a series connected evaporator with a single expansion device. Each configuration has its advantages and disadvantages as related to energy consumption and cost. In this study, a series connected evaporator configuration with two expansion devices is investigated.

2. SIMULATION MODEL DEVELOPMENT

Figure 1 shows the detailed schematic and thermodynamic representation of the system on a P-h diagram of the refrigeration cycle in current study. A capillary tube coupled with suction line heat exchanger was used as the primary expansion device. The intermediate expansion device is a simple adiabatic capillary tube.

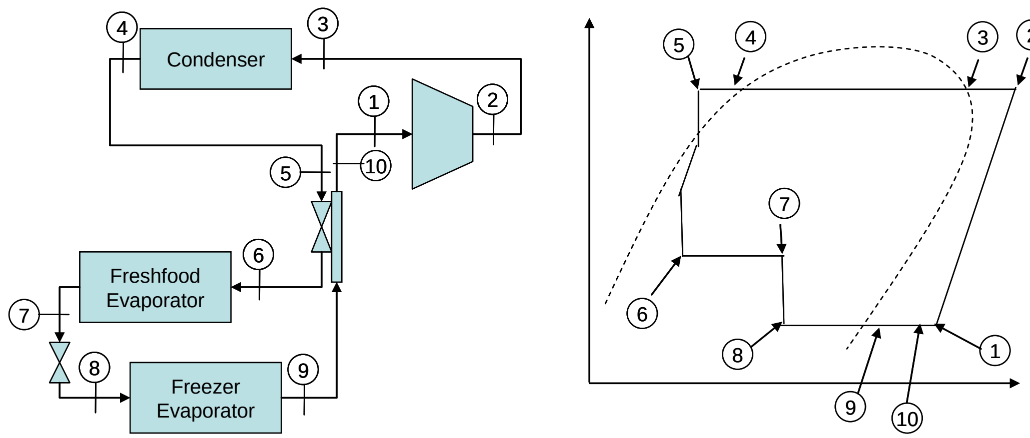

Figure 1: Schematic of a dual evaporator system and corresponding P-h diagram

The following is a list of components as shown in Figure 1:

1 – 2: Compressor
2 – 3: Discharge line (Refrigeration line connecting compressor and condenser)
3 – 4: Condenser
4 – 5: Liquid line (Refrigeration line connecting condenser and expansion device)
5 – 6: Capillary tube coupled with suction line heat exchanger (Primary expansion device)
6 – 7: Freshfood evaporator
7 – 8: Capillary tube (Intermediate expansion device)
8 – 9: Freezer evaporator
9 – 10: Suction line (Refrigeration line connecting evaporator and compressor)

The dual evaporator cycle simulation model was developed by modifying single evaporator system model presented by Gholap and Khan (2007). Modification includes adding a component model for the adiabatic capillary tube to represent the secondary expansion device. The new solver also includes an additional call to the evaporator model and necessary thermodynamic connections to represent the second evaporator.

3. EXPERIMENTAL INVESTIGATION

This section describes prototype development and validation of the simulation model against test data for the dual evaporator refrigerator. As previously described, we selected a dual evaporator unit for this study where both freshfood and freezer evaporators are connected in a series with an intermediate expansion device. The prototype refrigerator was created by modifying an existing bottom mount, single evaporator refrigerator. Figure 2 shows the freezer compartment with a view of the evaporator. It also shows the thermocouples installed to measure the refrigerant and compartment temperature.

Figure 2: Freezer compartment

This is a finned tube heat exchanger with a split fin density. The heat exchanger fins and tubes are made of aluminum. The fin density is highest at the air inlet and decreases along the airflow path. To simplify the modeling, we used the average fin density of 3.5 fins per inch. The fan motor assembly employed for this evaporator is single speed with a rated power consumption of 6 Watt at 115 V AC. Wind tunnel testing of this assembly showed that the fan motor assembly delivers 68 m^3/hr of airflow. Figure 3 shows the freshfood compartment with the evaporator mounted using temporary brackets. The evaporator used in this case is a continuous finned tube heat exchanger and the fin material is steel.

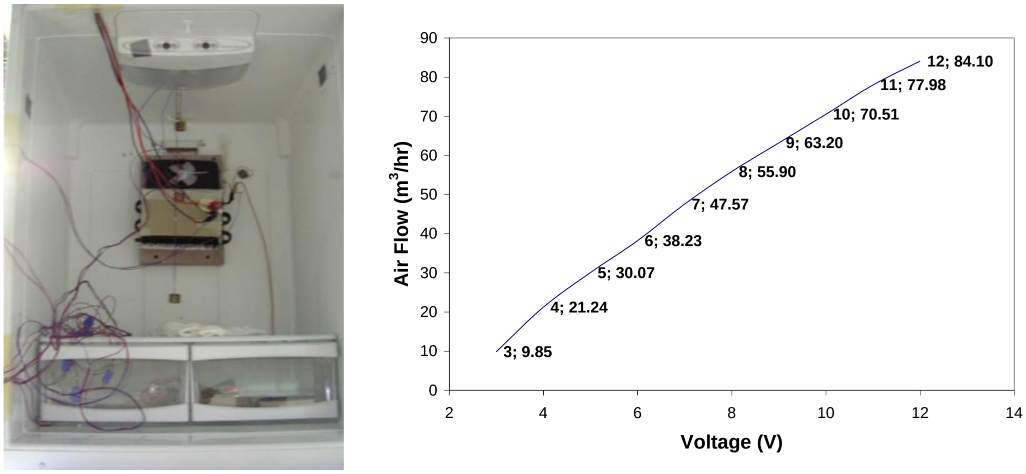

Figure 3: Freshfood compartment and fan assembly airflow characteristics

Figure 3 represents the fan-motor assembly used for this evaporator. The motor is variable speed with a rated voltage of 12 V DC. The speed of the motor can be changed by changing the voltage signal from 0 to 12 V. A wind tunnel study was conducted to determine the airflow characteristics of this assembly. Figure 3 shows the change in the airflow with the supplied voltage for this assembly. The airflow values obtained from this experiment were used as inputs for the simulation model.

Refrigeration connections were made as per the schematics of the system to achieve the desired refrigeration system. As shown in Figure 4, the majority of connections were made from the rear exterior of the unit to minimize interference with the compartment airflow. The primary expansion device which was a capillary tube suction line heat exchanger was installed on the rear in such a way that the addition of extra tubing would be minimized. The secondary expansion device which was a simple adiabatic capillary tube was installed inside the freshfood compartment. Both expansion devices were installed in such a way that they could be easily accessed and modified during the experimental optimization process. Figure 4 also shows the prototype unit installed at the test station.

Figure 4: Test unit setup

A direct current power supply as seen on the top of the refrigerator in Figure 4 was used to change the voltage provided to the freshfood fan. Two thermocouples measuring the ambient temperatures were placed on the side of the refrigerator at a distance of one foot apart. Gaps created in the door gaskets due to the passage of the thermocouple were carefully sealed to avoid any infiltration. The unit was placed in such a way that the rear wall has minimal impact on the condenser airflow. Temperature control was achieved using the existing controller without significant modification. The controller recorded the temperature of the freezer compartment to determine whether to turn the unit on or off. The control signal for varying the freshfood damper position was disabled and air passages connecting the freezer and freshfood damper were carefully sealed. There was no direct control of the freshfood compartment in the dual evaporator system. This strategy works well during the design process since most of the dual compartment refrigerators are designed in such a way that the heat load is equally divided. If needed, the control strategy can be improved in the future to provide more precise control of the freshfood compartment.

Because the refrigeration design described above is new, the refrigerant charge and the expansion device needed to be optimized. Initial simulation model runs were conducted to identify the approximate size of expansion devices. Since it is easier to reduce the length of the capillary tube rather than add to the length, longer tube lengths were selected in the beginning of the optimization study. Also, the optimization study was begun with the smallest refrigerant charge since it is easier to add the refrigerant without disturbing the setup or refrigerant connection.

For the first test, a 10.4 mm internal diameter (ID) and 2.743 m long capillary tube was used as the primary expansion device. Since the restriction requirement for the secondary expansion device was much lower, a capillary tube measuring 19.5 mm ID and 1.219 m long was used as an initial guess. Several tests were performed with variations in capillary tube dimensions, refrigerant charge, freezer control settings and freshfood fan airflow. The ambient temperature was kept constant at 32.2 ^{0}C until optimum performance was observed.

The objective of the experiment was to find a design which satisfied the following performance objectives:

- Lower compressor energy consumption
- Freshfood compartment integrated average temperature of 7.22 +/- 1.66 ^{0}C
- Freezer compartment integrated average temperature of -15 +/- 1.66 ^{0}C
- Lower or negligible superheat in each evaporator

Data collected from the test showing optimum performance is presented in Figures 5 and 6.

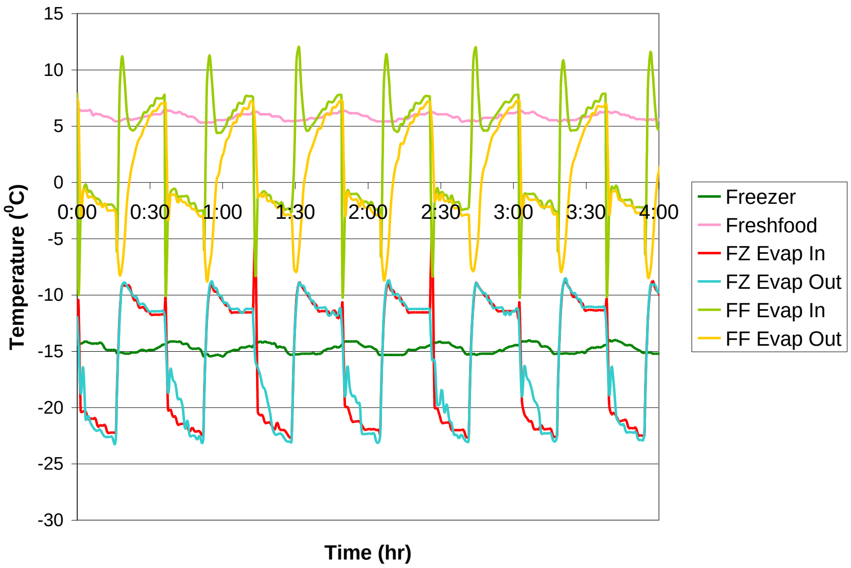

Figure 5: Compartment and evaporator temperatures

Figure 5 shows that both evaporators have very little or zero superheat during the compressor on cycle. As the unit is shut off, the freshfood evaporator inlet is influenced by the hot gas from the condenser, causing a quick surge in the temperature. At the same time, the freshfood evaporator outlet is influenced by the freezer evaporator.

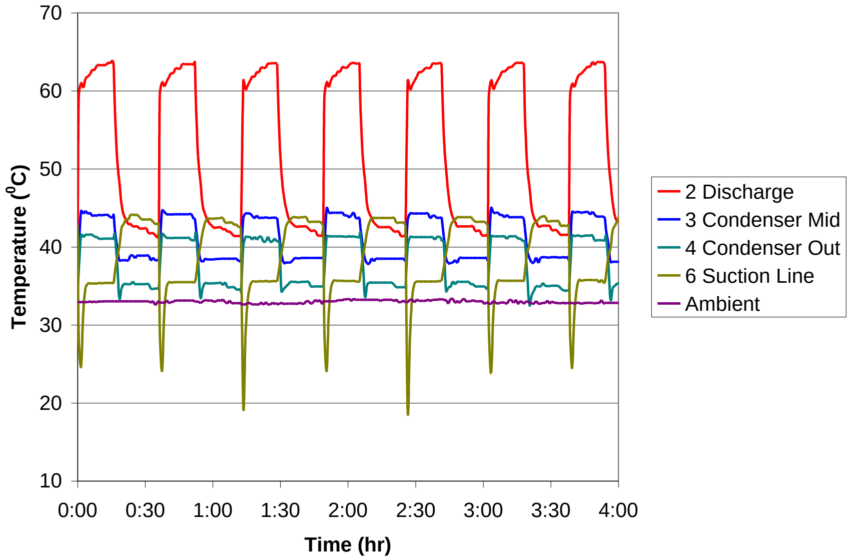

Figure 6: High pressure side temperatures

During the off cycle, refrigerant migrates to the coldest place in the system i.e., the evaporator and suction line. As the compressor is turned on, the refrigerant in the suction line is boiled off very quickly causing a sudden drop of temperature as seen in Figure 6. It is very important to make sure that suction line temperature is maintained sufficiently high to avoid any compressor damage caused by liquid refrigerant going back to the compressor.

4. MODEL VALIDATION

The simulation model was first compared with the data obtained from the test conducted at 32.2 °C ambient. Table 1 summarizes the findings of this comparison. Instantaneous data obtained from the unit just before the unit is turned off was used for the comparison of the thermodynamic state points. Overall the model predictions are in good agreement with the data with the exception of the suction line temperature. This discrepancy is due to the fact that suction line heat loss to the ambient is not modeled in this simulation. Another important point to note here is that the simulation model calculates energy consumption using a runtime fraction predicted for the freezer compartment. The table shows runtime estimates for both the freezer and freshfood sections. The difference between values of runtime is due to the errors associated in calculating the heat load for each compartment.

Table 1: Comparison between test data and simulation model

	Unit	32.2 ^{0}C Ambient			23.6 ^{0}C Ambient		
Input							
Freezer	^{0}C	-14.86			-15.00		
Freshfood	^{0}C	6.04			3.28		
Ambient (Measured)	^{0}C	32.22			23.65		
Output							
		Test	Simulation	Difference	Test	Simulation	Difference
Condenser Mid	^{0}C	43.77	42.90	0.87	33.51	34.73	-1.22
Liquid Line	^{0}C	40.55	42.89	-2.34	30.24	32.72	-2.48
Suction Line	^{0}C	35.34	20.49	14.85	26.41	14.19	12.22
Freezer Evaporator Inlet	^{0}C	-21.95	-22.49	0.64	-25.01	-23.87	-1.14
Freezer Evaporator Outlet	^{0}C	-22.67	-22.35	-0.32	-22.06	-23.82	1.76
Freshfood Evaporator Inlet	^{0}C	-2.34	-4.25	1.90	-4.94	-5.87	0.93
Freshfood Evaporator Outlet	^{0}C	-2.85	-4.28	1.43	-5.13	-5.10	-0.04
Runtime Fraction Freezer	%	44.65	42.28	2.37	31.77	30.74	1.03
Runtime Fraction Freshfood	%	44.65	40.26	4.39	31.77	34.55	-2.78
Energy	KW-hr	1.467	1.478	-0.01	1.098	1.038	0.06

Typically, a domestic refrigerator operates at temperatures lower than 32.2 °C ambient conditions. Table 1 also shows a comparison of the simulation results with data collected at 23.6 °C ambient. As can be seen, the model predictions are in close agreement with the data.

5. OPTIMIZATION

For this paper, we have considered overall energy consumption and heat exchanger material cost as dependent variables to form a multi-objective optimization problem. Overall energy consumption is calculated by adding energy consumed by the compressor, and fan motors for a 24-hour period. Heat exchanger material costs are calculated using the individual weight of the heat exchanger multiplied by the rate per unit weight of the corresponding metal in the international market. The aluminum alloy (evaporator raw material) rate was obtained from London Metal Exchange and the rate for low carbon steel (condenser raw material) was obtained from MEPS International Ltd. Several algorithms are used to perform multi-objective optimization. These algorithms are classified as either gradient based or non-gradient based. One of the important classes of non-gradient based methods is the evolutionary algorithm also known as the genetic algorithm. Srinivas and Deb (1994) proposed one such algorithm called Non-dominated Sorting Genetic Algorithm or NSGA. This was later modified by Deb et al. (2002) which eliminated higher computational complexity, lack of elitism and the need for specifying the sharing parameter. This algorithm is called NSGA-II which is coupled with the objective functions developed in this study for optimization. This optimization method has been widely used within the HVAC field (Gholap and Khan 2007, Huang et al., 2016, 2017).

The details of each step for conducting numerical optimization are as follows:

1. Parameterization: The upper and lower bound are identified for each variable selected for optimization. These bounds are decided by several constraints such as manufacturability, space, material, tooling cost,

etc. Based on the ease of availability or standardization requirements, design variables are identified as either discrete or continuous.

2. Experimental Design: Here an experimental design is created using the design space defined earlier. This process helps reduce the number of simulation runs. The statistical software package Minitab® was used to create an experimental design.
3. Simulation: A simulation model is used in this step to estimate values of response for each experimental run. Responses are the desired dependent variables which are optimized.
4. Objective Function: Values obtained for responses from the simulation run are used to develop metamodels. Regression analysis is used to obtain the coefficients of the objective functions.
5. Optimization: Metamodels or objective functions are coupled with the optimization algorithm to obtain optimal solutions. Optimization is run over several generations until convergence is achieved.
6. Validation: A simulation model is run for each design from the Pareto Optimal solutions obtained from step five. This is done to eliminate error associated with the regression based metamodel. If the errors are too high, the experimental design must be revised to include more runs.

6. RESULTS & DISCUSSION

The prototype unit developed as part of the experimental investigation presented earlier was used as a baseline design in this analysis. The first step in the analysis was selecting the design variables. There are two evaporators and one condenser. For each heat exchanger, two variables were chosen for the optimization study. The details of each variable along with their baseline value and upper and lower bound are presented in Table 2.

Table 2: Heat exchanger design variables

Heat Exchanger	Design Variable	Symbol	Unit	Type	Baseline	Lower Limit	Upper Limit
Condenser	Wire Spacing	P_w	mm	Continuous	5.2	4	6
	Number of Wires	$N_{w,cond}$	#	Continuous	190	175	205
Freshfood Evaporator	Width	$Width_{evap,FF}$	m	Continuous	248	.2	.3
	Fin Density	$FPI_{evap,FF}$	#/25.4	Discrete	5	3	6
Freezer Evaporator	Width	$Width_{evap,FZ}$	m	Continuous	.362	.3	.4
	Fin Density	$FPI_{evap,FZ}$	#/25.4	Discrete	3.5	3	6

As mentioned in the prototype development process, the refrigerator's original evaporator and condenser were retained. The refrigerator was then fitted with an additional heat exchanger, acting as a freshfood evaporator. This heat exchanger was selected based on availability and overall size. The next step in the optimization process was to create Design of Experiments (DOE) simulation runs. The method based on the Box-Behnken design was utilized to generate simulation runs. For six design variables, the Box-Behnken method prescribed 54 experimental runs. These experimental points were used to develop objective functions used for the optimization. The simulation model was executed for these combinations of design variables. It was observed that the convergence of the dual evaporator model was very sensitive to the guess values when compared to the single evaporator model convergence. Figure 7 shows the DOE runs and the baseline design.

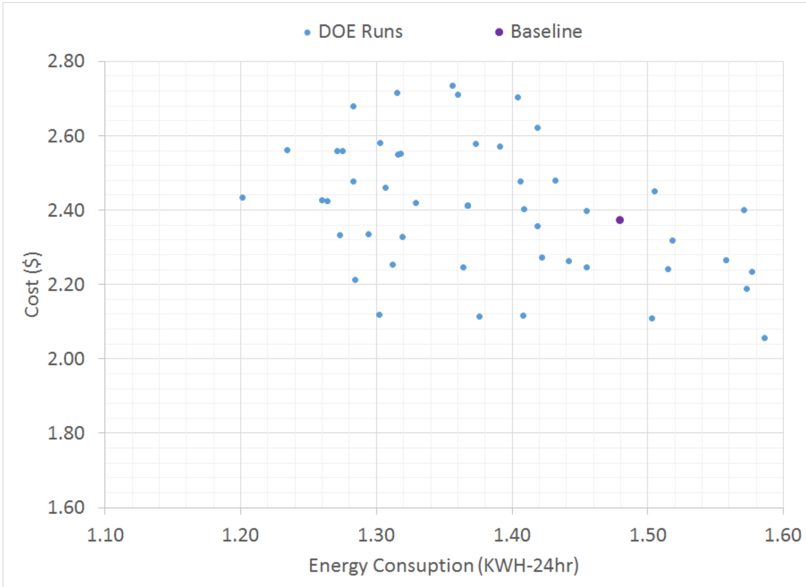

Figure 7: Simulation results for dual evaporator case

Objective functions were created using the data generated for energy and cost. Objective functions were added to the optimization code along with the necessary input such as the lower and upper bounds of the variables. Figure 8 shows the Pareto front obtained at the 50th generation of the dual evaporator case. The solutions are divided into four groups based on the values of energy consumption and heat exchanger cost. Group 1 represents solutions with the lower energy consumption and higher costs. Group 4 represents solutions with the lower costs and higher energy consumption.

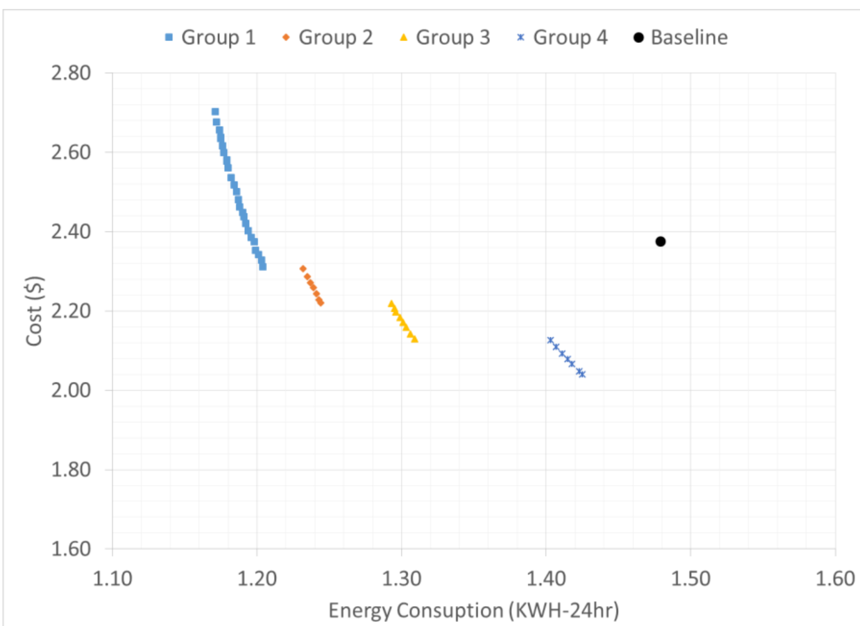

Figure 8: Pareto optimal solutions for dual evaporator case

The detailed results for Group 1 solutions are presented in Table 3. The optimum value of the condenser wire spacing (P_w) is equal to its lower limit. Considering both condenser variables, it seems that the condenser size can be reduced without compromising overall efficiency. In the case of the freshfood evaporator, both width ($Width_{evap,FF}$) and fin density ($FPI_{evap,FF}$) are optimum at their lower limit. This indicates that the current freshfood evaporator is oversized for the application.

Table 3: Design variables values for Pareto Solutions from Group 1

Energy KWH-24hr	Cost ($)	P_w	$N_{w,cond}$	$Width_{evap,FF}$	$FPI_{evap,FF}$	$Width_{evap,FZ}$	$FPI_{evap,FZ}$
1.171	2.703	4	230	0.200	3	0.400	6
1.172	2.677	4	228	0.200	3	0.395	6
1.174	2.656	4	222	0.200	3	0.396	6
1.175	2.634	4	226	0.200	3	0.385	6
1.175	2.638	4	221	0.200	3	0.392	6
1.176	2.616	4	226	0.200	3	0.380	6
1.177	2.599	4	220	0.200	3	0.382	6
1.179	2.578	4	220	0.200	3	0.376	6
1.179	2.581	4	220	0.200	3	0.377	6
1.18	2.560	4	220	0.200	3	0.371	6
1.182	2.536	4	220	0.200	3	0.364	6
1.184	2.518	4	220	0.200	3	0.359	6
1.186	2.501	4	222	0.202	3	0.351	6
1.187	2.480	4	220	0.200	3	0.348	6
1.188	2.462	4	220	0.200	3	0.343	6
1.19	2.449	4	222	0.200	3	0.337	6
1.191	2.438	4	221	0.200	3	0.335	6
1.192	2.420	4	220	0.200	3	0.331	6
1.194	2.402	4	220	0.200	3	0.326	6
1.194	2.402	4	220	0.200	3	0.326	6
1.196	2.385	4	220	0.200	3	0.321	6
1.198	2.375	4	220	0.201	3	0.318	6
1.199	2.353	4	220	0.200	3	0.312	6
1.201	2.342	4	220	0.200	3	0.309	6
1.203	2.329	4	222	0.200	3	0.303	6
1.204	2.311	4	220	0.200	3	0.300	6

Overall optimization results are highly dependent on the choice of the design variables and limits. In this study, we chose variables that impact a heat exchanger's overall size which in turn impacts the material cost. A different set of variables may affect efficiency and energy consumption. The selection of these variables as well as a specific optimal solution will depend on the design strategy.

7. CONCLUSIONS

In this study a prototype series connected dual evaporator refrigeration system was created by modifying existing single evaporator refrigeration system. Several tests were carried out to optimize the thermodynamic performance by modifying the charge and expansion device. A detailed simulation model was developed for a refrigeration system including component level models. The heat exchangers were divided into several small elements along the flow streams. The system level model was validated with experimental data collected from performance tests of the refrigerator. These models were coupled with an optimization algorithm to conduct the design studies. Several optimization algorithms are available to perform single and multi-objective optimization. Each method has advantages and disadvantages based on the design problem under consideration. In this study, an enhanced genetic algorithm, NSGA-II, was employed for performing multi-objective optimization. Since this algorithm requires several calls to the objective function, it is impractical to include direct coupling with a complex simulation model. The simulation model was approximated by a response surface based metamodel. Minimization of energy consumption and heat exchanger material cost were the objectives under consideration. Optimization provided a set of Pareto optimal solutions. The results obtained had a group of solutions with both cost and energy consumption lower than the baseline design. It was found that the freshfood evaporator was oversized for the application. Optimization also provided guidance on the design of the condenser and freezer evaporator. The design process supported by the simulation and optimization tools presents an effective method for finding best trade-off design solutions for heat exchangers in the presence of two competing objectives for multiple design variables using a detailed refrigeration cycle simulation model. The designer can choose a solution based on the manufacturer's strategic direction regarding cost and energy consumption requirements. The prototype model development effort was limited to obtaining a thermodynamically working unit. The current control is based on the freezer compartment temperature. An independent temperature control can be added to the freshfood compartment. This would not only improve the overall energy performance of the refrigerator but also improve the temperature control over a wide range of ambient conditions. An experimental and numerical investigation of the application of a variable speed compressor provides another opportunity for further studies.

NOMENCLATURE

Subscript

cond	condenser
evap	evaporator
FF	freshfood
FZ	freezer
w	wire

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