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ABSTRACT

This paper presents a design optimization study of a main engine driven roof top bus air conditioning system using a thermo economic approach. The design optimization is conducted by investigating the effect of geometrical and operational parameters of bus air conditioner system (Compressor, Condenser and Evaporator) which have a significant influence on the performance. Two objective functions including the total exergy destruction of the system (as a thermodynamic criterion) and the total product cost of the system (as an economic criterion), have been considered. Since these objectives are conflicting, no single design will satisfy both simultaneously, a multi-objective optimization procedure is used to find optimal design values for design variables.

Three scenarios including a single-objective thermodynamic optimized, a single-objective economic optimized and a multi-objective optimized are studied and compared with the base line. Detailed simulation models are typically complex and computationally demanding. An optimization algorithm requires several evaluations of such models, so response surface based metamodels for objective functions were used to save computational effort. A goal programming based optimization tool is used for multi-criteria optimization. In the case of multi-objective optimization, the Pareto frontier is generated to give guidance on the optimal trade-off on the multi objectives and aid the decision-making process for selection of the final solution.

Using Multi objective optimization, 4% product cost reduction and 36% exergy reduction translating to 21% running cost reduction could be achieved simultaneously. The results have shown that the multi-objective design more acceptably satisfies generalized engineering criteria than other two single-objective optimized designs.

1. INTRODUCTION

The use of a roof-top Bus air-conditioning (AC) system has been steadily growing in the emerging markets. An AC system is the second biggest energy consumer component in a bus. For this reason, globally researchers and manufacturers seek to improve bus air conditioning systems designs to reduce the fuel consumption rate without forfeiting passenger thermal comfort. In addition to fuel efficiency, especially in the emerging markets there is even more emphasis on the product cost. Thus, there is a need to develop an affordable and efficient bus AC system featured by low product cost, economical operation in terms of energy usage and stable passenger thermal comfort.

Thus, the design optimization of a bus AC system should take into account the compromise between an economical system size and efficient refrigeration. This leads to a thermoeconomic optimization exercise, which combines economic and exergy analysis. The concept of thermoeconomic was developed by El-Sayed and Evans (1970), where the thermoeconomic objective function was optimized. Wall (1991) introduced a method of thermoeconomics to refrigeration engineering to improve the construction of technical systems by an analysis of effects of potential solutions in terms of cost and to minimize the total life cycle cost. The compressor capital cost is determined according to the equation suggested by Wall (1991).

Khamis and Musa (2008) presented a methodology of a design optimization technique for assessing the best configuration of a finned-tube evaporator of a vapor compression cycle for a roof-top bus air-conditioning (AC) system at a specified cooling capacity using a thermoeconomic approach. The methodology has been conducted by studying the effect of some operational and geometrical design parameters for the evaporator on the entire cycle exergy destruction or irreversibility, AC system coefficient of performance (COP), and total annual cost. The optimization scenario was carried out by varying one of the selected parameters, while the other parameters are kept at the same values as the base configuration parameters.

For optimization of a vapor compression refrigeration system, a designer may consider one or more of the thermodynamic and economic as the objective function. If only the thermodynamic criterion is considered, the system will be an ideal system from thermodynamic point of view, but it might not be able to pass the economic criterion. On the other hand, by considering only the economic criterion, the system will be the cheapest one, but this system might not be a well-designed system from thermodynamic points of view. Both of these systems are not acceptable from a comprehensive engineering point of view. Thus the simultaneous consideration of all or some of these criteria will provide better option for engineers. This goal can be obtained by multi-objective optimization techniques. In this way, we will have a system that satisfies all of the optimization criteria as much as possible simultaneously.

H. Sayyaadi and M. Nejatolahi (2010) presented optimization with multiple criteria applied to cooling tower assisted vapor compression refrigeration machine. A thermodynamic model based on energy and exergy analyses and an economic model according to the Total Revenue Requirement method have been developed and have been considered simultaneously.

Detailed simulation models are typically complex and computationally demanding. Response surface based metamodels for objective functions were used to save computational effort. Response surface methodology (RSM) was adopted based on the book by Raymond H. Myers, Douglas C. Montgomery and Christine M. Anderson-Cook (2009).

This work has been presented here as an attempt for multiobjective optimization of a Bus Air-conditioning system. Objectives are the total exergy destruction and the total product cost of the system. The second law criterion (the total exergy destruction of the system) is considered as the thermodynamic objective function and the economic objective function is the total product cost of the system. Exergy relations are developed based on the book by Ibrahim Dincer and Marc A. Rosen (2009). Three optimization scenarios including the thermodynamic single-objective, the economic single-objective and multi-objective optimizations are performed. The multiobjective optimization scenario is conducted using a goal programing technique. The output of the multi-objective optimization is a Pareto frontier that yields a set of optimal points in addition to the optimal design. The exergetic and economic results obtained for systems obtained in the three optimization scenarios are compared and discussed.

2. BUS AC SYSTEM AND DESIGN VARIABLES

2.1 Bus AC system

The basic bus AC circuit consists of a compressor, a condenser coil, an expansion valve and an evaporator coil. Bus air conditioners typically employ an open-shaft compressor, which is belt driven from the bus main engine. Electrical power for the condenser fans and evaporator blowers are usually supplied from a 24 V DC alternator mounted on the bus engine. Figure 1. shows the layout of major components for a roof –mounted Bus AC system.
The Base line unit considered is a Thermo King standard rooftop bus air conditioner currently in production. The base line unit employs R-134a as the refrigerant and it has a nominal cooling capacity of 25 kW under Thermo King's standard rating conditions (Evaporator air inlet 27 °C DBT / 19 °C WBT and 35 °C ambient) at a fixed nominal compressor speed.

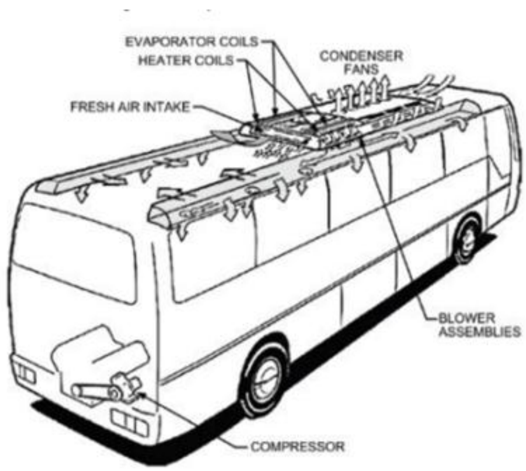

Figure 1 : Typical Roof Top Bus Air – Conditioning system

2.2 system components and design variables

Details of the main system Components considered for simulation are described below,

1. Compressor- A variable speed open type reciprocating compressor is considered, RPM is changed to represent different volumetric displacements. The performance characteristic (cooling capacity, power consumption etc.) of the compressor is derived from the compressor coefficients developed from testing. The volumetric efficiency, isentropic efficiency, etc. are predicted from the model for any given operating condition.
2. Condenser and Evaporator - A microchannel condenser is considered for condenser and a copper tube and aluminum fin heat exchanger is considered for evaporator. Performance of a these heat exchangers is affected by a multitude of factors related to its design and operation. These factors include the overall heat exchanger dimensions, the type of refrigerant-side and air-side heat transfer surfaces, fin spacing, and tube geometry, refrigerant pass design and air velocity distribution over the frontal heat exchanger surface. Typically during coil design, the outside dimensions are dictated by the available space, and most of the remaining parameters are imposed on the design engineer by established manufacturing practices, e.g., heat transfer surfaces or tube definition. In this analysis, for the condenser, coil length and tube width (and its associated tube and fin configuration) is treated as design variables. The coil height and the pass configuration are held constant. In a Rooftop Bus AC the height of the evaporator coil is defined by the unit maximum height, so number of tubes per row is defined by the tube face pitch for a particular standard tube outside diameter. So Finned length (perpendicular to the air velocity), tube outside diameter (and its associated tube pitches, with their corresponding fin pattern) and Fin pitch are the variables considered for the evaporator. In both the coils, a uniform and constant air velocity across the coil is assumed so the coil face area defines the airflow across the coil.
3. Expansion device- an isenthalpic expansion model is used.
4. All the above components are connected using refrigerant lines which do account for the refrigerant pressure drop and the heat ingress to the system.

In order to have an optimum bus AC design for a given cooling capacity in terms of product cost, running cost, or both together as objective functions, simulations had to be run to generate cases with different design variables and constraints which have direct influences on them. The design variables and the levels considered for each variable is shown in the Table 1. Constant Air velocity of 3 m/s is considered for condenser and to avoid condensate water carry over at higher velocities towards downstream of the evaporator constant air velocity 2.5 m/s is considered for Evaporator. Evaporator air inlet 27 °C DBT / 19 °C WBT and 35 °C ambient are considered. A fixed condenser sub cooling of 5 °C and Evaporator superheat of 12 °C is assumed. Two refrigerants (R134a and R407C) are considered, since the component models are different for different refrigerants, the simulations for DOE are run separately for the two refrigerants.

Table 1: Design Variables and levels

Variable			
Compressor RPM	2000	2250	2500
Condenser Face length(m)	1.2	1.4	1.6
Condenser tube width (mm)	25.4	18	
Evaporator Face length(m)	1.2	1.5	1.8
Evaporator Tube OD (mm)	7	8	
Evaporator Fin pitch (FPI)	12.7	14	

For the construction of a quadratic response surface model with 6 variables at least 28 function evaluations are required to estimate the tuning parameters. Considering the full factorial design a total of 216 runs for each refrigerant (R134a and R407C) are performed. For all these cases all the major output parameters like cooling capacity, power consumption, refrigerant mass flow and COP along with values of pressure and temperatures at all state points are captured, which are necessary for exergy destruction calculation for the total system.

2.3 Refrigeration system simulation tool

An in-house refrigeration system balancing tool is used to simulate the steady state performance of the AC system. In this tool, each component model is represented using a set of nonlinear equations of input and output variables. When each of these components is interconnected to build a closed loop refrigeration system, these set of nonlinear multi-equations are solved simultaneously for output variables using Newton-Raphson method with the help of an initial guess value routine.

3. COST FUNCTIONS

3.1 Exergy Analysis

Only a part of the input power to the compressor and different fans is transferred to the air that is cooled in cabin. The rest of it is used in overcoming irreversibilities in different components. An exergy analysis gives the magnitude of irreversibility in different components and helps in assessing the overall COP of the system. The exergy loss in any component refers to the difference between exergy input and exergy output. Thermodynamic processes in refrigeration systems release large amount of heat to the environment. Heat transfer between the system and the surrounding environment takes place at a finite temperature difference, which is a major source of irreversibility or exergy destruction (ED). All the refrigeration system components contribute to the air-conditioning (AC) system ED. Exergy balancing for a steady state process is given by the Equation (1).

$$Ex_{in} - Ex_{Out} + Ex_Q + Ex_w = Ex_{des} \tag{1}$$

Where Ex_{des} is the total Exergy destruction. Ex_Q Exergy flow due to heat transfer in the control volume. Ex_w Exergy associated with the work interaction with the control volume. Ex_{in} and Ex_{Out} are the exergies of the inlet and outlet streams of refrigerant or air interacting with the control volume.

Exergy rate of the Air entering and leaving both the condenser and evaporator by treating dry air and water vapor mixture as ideal gases and the total flow Exergy of humid air per kg Dry air is expressed as below.

$$\begin{aligned} ex = {} & (C_{p,a} + \omega C_{p,v})T_0\left(\frac{T}{T_0} - 1 - ln\frac{T}{T_0}\right) \\ & + (1 + 1.6078\omega)R_aT_0\, ln\frac{P}{P_0} \\ & + R_aT_0\left((1 + 1.6078\omega)\, ln\frac{(1 + 1.6078\omega_0)}{(1 + 1.6078\omega)} + 1.6078\omega\, ln\frac{\omega}{\omega_0}\right) \end{aligned} \tag{2}$$

The dead state of air is at ambient temperature, T_0, ambient pressure, P_0, ambient humidity ratio, ω_0. The significance of first term in the equation is the effect of temperature on Exergy, Second term relates to the effect of pressure on Exergy and Third term relates to the effect of Humidity ratio on Exergy.

Exergy of Refrigerant at each State point is calculated as

$$ex = h - h_0 - T_0(s - s_0) \tag{3}$$

Exergy Rate at each state point is given by

$$Ex = m\ (ex) \tag{4}$$

The dead state of the refrigerant is taken at ambient temperature, T_0, and at a pressure, P_0, equal to the saturation temperature of the refrigerant at T_0.

Exergy consumption in each component and Exergy transferred between the components to the load and the environment is evaluated. Exergy destructions are obtained from Exergy balances for all the system components are expressed as follows:

Exergy destruction of the compressor is given below:

$$W_{Comp} = m_r\ (h_{2\_isen} - h_1)/\eta_{isen} \tag{5}$$

$$Ex_{des\ Comp} = m_r\ (ex_1 - ex_2) + W_{Comp} \tag{6}$$

W_{Comp} is the compressor work in kW, evaluated from equation (5). Or alternatively can be obtained from the test data; η_{isen} is the isentropic efficiency of the compressor; ex_1 and ex_2 are the exergy state points at Inlet and outlet of the compressor respectively; m_r is the mass flow rate of the refrigerant.

Exergy Destruction for condenser is given by the equation below:

$$Ex_{des\ Cond} = m_r\ (ex_2 - ex_3) + m_{air\ Cond}\ (ex_{air\ in} - ex_{air\ out}) + W_{fan} \tag{7}$$

$$W_{fan} = (\Delta P\ \ m_{air\ Cond}) / (\rho_{air}\ \eta_{stat}) \tag{8}$$

ex_2 and ex_3 are the exergy state points of Inlet and outlet of the condenser respectively on refrigerant side; $ex_{air\ in}$ and $ex_{air\ out}$ are the exergy state points of Inlet and outlet of condenser respectively on air side; m_r and $m_{air\ Cond}$ are the mass flow rates of refrigerant and air. ΔP is the air side pressure drop across the heat exchanger; η_{stat} is the static efficiency of the fan; ρ_{air} is the density of air.

Exergy destruction for expansion devise is by the equation below:

$$Ex_{des\ TXV} = m_r\ (ex_3 - ex_4) \tag{9}$$

ex_3 and ex_4 are the exergy state points at Inlet and outlet of the Expansion Valve respectively; m_r mass flow of refrigerant.

Exergy Destruction for evaporator is given by the equation below:

$$Ex_{des\ evap} = m_r\ (ex_4 - ex_1) + m_{air\ evap}\ (ex_{air\ in} - ex_{air\ out}) + W_{fan} \tag{10}$$

$$W_{fan} = (\Delta P\ \ m_{air\ evap}) / \rho_{air} \tag{11}$$

ex_4 and ex_1 are the exergy state points of Inlet and outlet of the evaporator respectively on refrigerant side; $ex_{air\ in}$ and $ex_{air\ out}$ are the exergy state points of Inlet and outlet of evaporator respectively on air side; m_r and $m_{air\ evap}$ are the mass flow rates of refrigerant and air. ΔP is the air side pressure drop across the heat exchanger; η_{stat} is the static efficiency of the blower; ρ_{air} is the density of air.

Total Exergy destruction of the AC system is equal to sum of Exergy destruction in each component and is given by the equation (12).

$$Ex_{\text{des Total}} = Ex_{des\ Comp} + Ex_{des\ Cond} + Ex_{des\ TXV} + Ex_{des\ evap} \tag{12}$$

3.2 Cost Calculations

In this analysis two cost contributions are considered, the total product cost and annual operating cost. The product cost consists costs majorly of compressor, evaporator, condenser, evaporator blower, condenser fan, alternator and other system component costs (thermostatic expansion valve, liquid receiver, filter-dryer, piping network) which are not significantly affected by the variation of the design parameters and hence not considered in the optimization.

- Evaporator Cost: the evaporator cost consists of material cost and conversion cost (fabrication cost) .The evaporator considered is a Copper tube and aluminum fin type coil, and fabrication cost is typically based on a multiple of material cost. Evaporator cost was calculated using cost of copper (USD 6.4 kg) and aluminum (USD 1.9 kg) from the London Metals Exchange and with appropriate conversion costs. In order to calculate the Evaporator cost Copper weight and Aluminum fin weight are calculated for all coil variants.
- Condenser Cost: The condenser considered is a micro channel brazed aluminum coil, the cost of condenser consists of Material cost and conversion cost (fabrication cost). For material cost aluminum (USD 1.9 kg) from the London Metals Exchange and with appropriate conversion costs. Fabrication cost is typically a multiple of material cost.
- Compressor cost: The compressor capital cost is determined according to the equation suggested by Wall (1991) as follows:

$$C_1 = a_1 k_1 \frac{V_2}{0.9-\eta_1} \frac{p_3}{p_2} \ln \frac{p_3}{p_2} \tag{13}$$

η_1 is the isentropic compressor efficiency is obtained by curve fitting the experimental data using

$$\eta_1 = 0.9343 - 0.04478 \frac{p_3}{p_2} \tag{14}$$

The annuity factor a_1 for compressor capital investments is defined as

$$a_1 = \frac{r}{1-(1+r)^{-n_1}} \tag{15}$$

Assuming the interest rate to be 5% and 10 years as the depreciation time horizon, this leads to the value of the annuity factor a_1 = 0.1295 years^{-1} . k_1 , Compressor cost per volumetric rate (Vs) is used based on internal cost information.

- Condenser Fans and Evaporator blowers cost: Typically the fans and blowers are standard parts and only number of fans and blowers vary with different air flow requirements. Cost of these components is considered based on a cost factor as a function of airflow \$ 0.0364 per m^3/hr for condenser and \$ 0.1 per m^3/hr for Evaporator.
- Alternator Cost: In Bus AC DC power is needed to power the fans and blowers and other control components. The Alternator cost is calculated as a function of the total Fan and Blower power requirement.

The annual operating cost is proportional to the system input power, which is the sum of the compressor power and the fans and blowers' power. The system input power is inversely proportional to the system efficiency COP for a fixed cooling capacity. The AC system input power is considered as a part of the total consumed power from the power produced by the bus main engine. Therefore, the operating or running cost is represented by the annual fuel cost needed to produce the required input power to drive the AC system. The annual operational cost, CO, is given as

$$CO = \frac{Q_e \times C_1 \times H_{year}}{\eta_{overall} \times CV \times \rho_{fuel}} \times \frac{1}{COP} \tag{16}$$

H_{year} : annual operational hours, C_1 : the price of diesel fuel in \$ / liter , $\eta_{overall}$: overall efficiency , *CV*: Fuel Calorific value and ρ_{fuel} : density of fuel.

4. OPTIMIZATION METHOD

4.1 Response surface

Response Surface Methodology (RSM) is useful for the modeling and analysis of programs in which a response of interest is influenced by several variables and the objective is to optimize this response. By careful design of experiments, the objective is to optimize a response (output variable) which is influenced by several independent variables (input variables). The application of RSM to design optimization is aimed at reducing the cost of expensive analysis methods and their associated numerical noise. There are two basic concepts in RSM, first the choice of the approximate model and, second, the plan of experiments where the response has to be evaluated.

The first step in RSM is to find a suitable approximation to the true relationship. If the response can be defined by a linear function of independent variables, then the approximating function is a first-order model. If there is a curvature in the response surface, then a higher degree polynomial should be used. The approximating function with *i* variables is called a second-order model as expressed in Equation (17). The response *y* depends on the independent variables and the experimental error term, denoted as ε.

$$y = \beta_0 + \sum_{i=1}^{k} \beta_i x_i + \sum_{i=1}^{k} \beta_{ii} x_i^2 + \sum_{i<j} \beta_{ij} x_i x_j + \varepsilon \tag{17}$$

In order to get the most efficient result in the approximation of polynomials the proper experimental design must be used to collect data. Once the data are collected, the Method of Least Square is used to estimate the parameters in the polynomials. The response surface analysis is performed by using the fitted surface.

The next important aspect of RSM is the design of experiments (DOE) which is the selection of the points where the response should be evaluated. The choice of the design of experiments can have a large influence on the accuracy of the approximation and the cost of constructing the response surface.

There are many designs available for fitting a second-order model. The factorial designs are widely used in experiments, when an experimenter needs to evaluate the joint effects of several controllable factors on the response. When the measurements on the response variable contain all possible combinations of the levels of the factors, the design is called a complete factorial experiment. Fractional Factorial Design, allows an experimenter to get information on the main effects and the low-order interactions.

These simulations are performed using the in-house refrigeration system balancing tool to evaluate the steady state performance and the thermodynamic quantities of all state points in the AC system. The output information is captured in the data file, which is loaded in Matlab to perform all the required exergy and cost calculations and developing the required response surfaces vectors F. An X matrix is formed from the Variables. The tuning parameters vector ***a*** for the particular response is obtained by solving, ***a*** *X* = ***f***. These Response surfaces are used in the optimization for constructing the objective, constraints and output functions.

4.2 Optimization Method

A minimization multi-objective decision problem with K objectives is defined as finding a vector x^* that minimizes a given set of K objective functions $f(x^*) = \{f_1(x^*),\ldots,f_K(x^*)\}$ given a decision variable vector $x = \{x_1,\ldots,x_n\}$ in the solution space X. The solution space X is generally restricted by a series of constraints, such as $g_j(x^*) = b_j$ for $j = 1,\ldots,m$, and bounds on the decision variables. In many real-life problems, objectives under consideration conflict each other and no solution vector X exists that minimizes all the K objective functions simultaneously. Hence optimizing X with respect to a single-objective often results in unacceptable results with respect to the other objectives. Thus a concept, known as the “Pareto optimum solution”, is used in multi-objective optimization problems. A feasible solution X is called “Pareto optimal” if there exists no other feasible solution Y that dominates solution X. The set of all feasible non-dominated solutions in X is referred to as the “Pareto optimal set”, and for a given Pareto optimal set, the corresponding objective function values in the objective space are called the “Pareto optimal frontier”. In the feasible area, all the points on the Pareto frontier are also optimal, but which of them should be selected as the final solution requires a decision-making process.

In this work, the Pareto frontier is found using Goal Programming approach. Goal Programming approach does not pose the question of maximizing multiple objectives, but rather it attempts to find specific goal values of these objectives. The problem transforms into one in which we have to find a solution whose value is as close as possible to the utopian set by introducing additional variables and constraints – these variables deal with the deviation from the goal for each objective and optimizing these slack/surplus variables to attain the goal values given in a goal vector. The relative importance of the goals is indicated using a weight vector.

The optimization is performed for a set of target system capacities from 15 kW to 35 kW in multiples of 5kW. The decision variables are extrapolated beyond the levels considered, to extend the scope of design. Optimization is performed for minimizing the product cost, exergy separately and simultaneously using Multi objective optimization.

4.3 Objective functions and constraints

Objective functions for single-objective and multi-objective optimizations in this study are the thermodynamic and economic objective functions denoted by Equations (18) and (19), respectively. In the single-objective thermodynamic optimization, the aim is minimizing the total irreversibility of the Bus AC refrigeration system. In the single-objective economic optimization, the aim is minimizing the total product cost of the Bus AC system.

$$J_{Thermodynamic} = Ex_{des\ Total} \tag{18}$$

$$J_{Product\ cost} = C_{Comp} + C_{Cond} + C_{Evap} + C_{evap\ Blower} + C_{cond\ Fan} + C_{alternator} \tag{19}$$

Following Constraints are applied based on limitations emanating from the practicality and experience,

g_1 : Discharge pressure < Max discharge pressure, (400 psi typically)
g_2 : COP $_{Min}$, (based on internal product development standard)
g_3 : Condensing temp > Ambient temp + ΔT
g_4 : Evaporating Temp < Inlet Air Temp – ΔT
g_5 : COP > COP $_{Carnot}$
g_6 : Evaporator PD_{ref} < Evaporator $PD_{Allowable}$
g_7 : Condenser PD_{ref} < Condenser $PD_{Allowable}$
g_8 : Evaporating temp < inlet air dew point temp , to allow for dehumidification.

All costs and exergy terms are non-negative, these are treated as additional constraints.
The target cooling capacity is defined as a nonlinear equality constraint.
Max discharge pressure, Minimum COP, ΔT, Evaporator $PD_{Allowable}$, Condenser $PD_{Allowable}$ are all defined by the designer as inputs.

5. RESULTS AND DISCUSSION

Table 2 shows the optimized values of the design variables, Objective function values, constraints and other outputs in the various optimization scenarios compared to the base line for a 25kW system capacity case.

Table 2: Results of Optimization Scenarios

	UNITS	Base line	R134a			R407C		
			Cost Optimized	Exergy optimised	Multi objective optimised	Cost Optimized	Exergy optimised	Multi objective optimised
Decision Variables								
Evap Tube dia	mm	8	6	6	6	8	8	9.52
Cond Tube width	mm	25.4	32	32	32	18	25.4	25.4
Compressor speed	rad/s	209	228	150	193	135	84	113
Evap finned length	m	1.5	1.14	1.71	1.1	1.34	1.98	1.55
Fin density	# per foot	152	169	200	169	189	200	182
Condenser coil length	m	1.6	1.11	2.56	2.0	1.63	2.45	2.29
Objective function values								
Capital Cost	$	3660	3150	4711	3525	3046	4462	3732
Exergy Destruction	kW	9.8	8.2	5.48	6.3	10.01	6.70	8.24
Other output Paramaters								
Running cost	$	17863	18353	11992	14113	18141	11618	14451
Discharge Pressure	psi	217	237	117	171	352	203	235
COP		2.19	2.17	3.23	2.76	2.13	3.37	2.86
Compressor Power	W	10785	13168	9026	10475	12749	8802	10677
Condensing Temp	C	54	63.3	37.1	40	57.7	37.4	42.4
Evaporating Temp	C	5.0	6.0	8.1	5.5	5.9	11.2	4.6

For the single-objective economic optimization case, 15% total product cost reduction could be achieved but with a 2% running cost penalty. For the single-objective thermodynamic optimization case, 38% exergy reduction could be achieved translating to a 34% running cost reduction but with 25% product cost penalty. Using Multi objective optimization, 4% product cost reduction and 36% exergy reduction (with 21% running cost reduction) could be achieved simultaneously. Figure 2. Shows the Pareto frontier for R134a, 25kW case with multi objective optimization. , It can be observed that separate cost optimization and Exergy optimization yields a Utopia Point (Product cost $ 3150, Exergy destruction 5.5 kW) which is not in the design space and is not achievable. The multi objective optimization yields the best solution in terms of cost and efficiency ($ 3525, 6.2 kW), which is represented by the minimum distance point from the utopia point on the Pareto optimal curve.

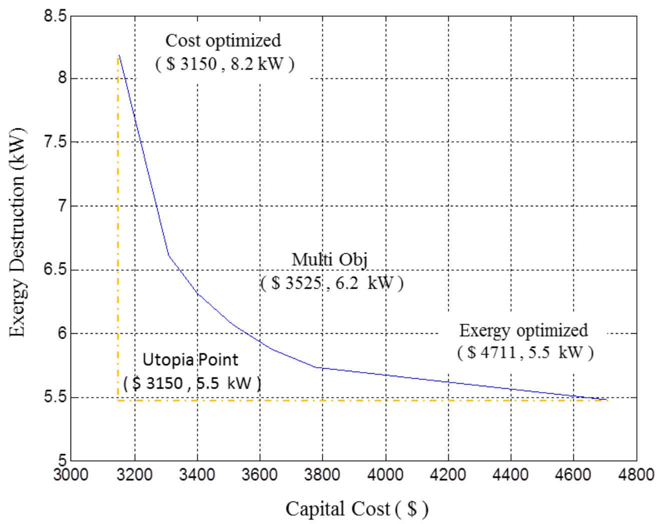

Figure 2: Pareto frontier (25 kW, R134a)

This Pareto frontier can further be used to aid the process of decision-making by considering the tradeoffs. Hence one of these points on the curve, instead of the selected point, could be selected as a final solution in the multi-objective optimization problem without significant increase in the distance to the equilibrium point.

6. CONCLUSIONS

The proposed model for the optimization of bus AC system including six design variables and their constraints is optimized, the optimization is performed for two refrigerants R134a and R407c. Response surface based metamodels for objective functions were used to save computational effort. Three optimization scenarios including thermodynamic single-objective, economic single-objective and multiobjective optimizations are performed and the results were compared. A goal programming based optimization tool is used for multi-criteria optimization.

The output of the multi-objective optimization is a Pareto frontier that yields a set of optimal points in addition to the optimal design. The results have shown that the multi-objective design more acceptably satisfies generalized engineering criteria than other two single-objective optimized designs.

As an improvement to the method, authors propose to continue the study incorporating genetic algorithm based optimization, so that discrete variable can be handled, and both refrigerants can be handled simultaneously in a single set of design of experiments.

NOMENCLATURE

m	Mass Flow rate	(kg/s)
E	Energy	(kW)
Ex	Exergy	(kW)
P	Pressure	(psi)

PD	Pressure Difference	(psi)
T	Temperature	(°C)
h	Enthalpy	(kJ/kg)
s	Entropy	(kJ/kg K)
Cp	Specific Heat	(kJ/kg K)
ω	Humidity Ratio	(g/kg)
C	Cost Function	($)
ΔT	Temperature delta	(°C)
g	Constraint Function	(--)
J	Objective Function	(--)
DBT	Dry bulb temperature	(°C)
WBT	Wet bulb temperature	(°C)

Subscript

In	Inlet to the Component
Out	Outlet of the Component
des	Destruction
0	Dead State or Reference State
1, 2, 3 and 4	Refrigeration cycle State Points
a	Dry air
v	Saturated air
ref	Refrigerant
Allowable	Allowable
Min	Minimum

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