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EFFECTS OF VARIOUS PARAMETERS ON THE EFFICIENCY OF A CO_2 HEAT PUMP: A STATISTICAL APPROACH

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ABSTRACT

The influence of variables such as; refrigerant amount, chilling and cooling water temperature, throttle valve opening, cooling and chilling water flow rates, on the efficiency (coefficient of performance – COP) of a water to water carbon dioxide heat pump was investigated. Design of experiments was done using design-expert® 6 software for regression analysis. A response surface method known as central cubic design was used to provide optimum results with minimum experiments. Through multiple regression analysis, an empirical equation relating the COP to the variables was derived. Analysis of variance revealed that these regressions are statistically significant at 95% confidence level compounded with a very low standard deviation and a high adequate precision. The close relationship between the predicted COP values and the actual values further proves the worthiness of the empirical equation. It was observed that cooling water temperature had the highest influence followed by the chilling water temperature. Surprisingly, the amount of the refrigerant was third followed by the throttle valve opening. Understandably, chilling water flow rate had the least effect on the COP. Through response surface diagrams, the interactive influence of the variables were also observed. The COP values arrived at varied from 1.545 to 6.914 although if the variables were optimized fully within the scope of this study, a value of up to 11.8 could be achieved. Still, if the variables range is increased further, higher COP could be achieved. Finally, a discussion was done in a bid to explain these results.

1. INTRODUCTION

1.1. Refrigerants

Refrigerants have passed through several stages of evolution. At the beginning, the early refrigeration equipment used any working fluid as a refrigerant. As long as the refrigerant worked, there was little regard on its safety or environmental impact. Some were even highly reactive. Most of these fluids were natural gases and solvents e.g. NH_3, CO_2, water, SO_2, etc. As concerns of safety and performance became more real, the refrigeration industry started to look for new and improved refrigerants. Artificial refrigerants known as fluorochemical refrigerants (Chlorofluorocarbons – CFCs and Hydrochlorofluorocarbons – HCFCs) were then created (HCFCs were made by substituting one or more chlorine and/or fluorine compounds in CFCs so as to make them chemically less stable and reduce their atmospheric life (Bhatia)). Apart from being refrigerants, CFCs and HCFCs also were used as solvents, aerosol propellants, fire suppressants and blowing agents. Their use in refrigeration was encouraged because they were stable, efficient and safe. Gradually, they caused the decline of the use of natural fluids in refrigeration except NH_3 which was (and still is) the most popular refrigerant in large food processing plants (Bensafi and Thonon, 2007, Calm, 2008).

1.2. Environmental Effects

Even though CFCs and HCFCs were the most desirable refrigerants, their long-term effects were not investigated until when their accumulation in the stratosphere partially destroyed the ozone layer after around 40 years of use. Bromine and chlorine from anthropogenic chemicals react catalytically with ozone molecules thereby destroying them. The result is a thinner ozone layer and thus reduced shield against harmful ultraviolet-B light from the sun. In their study of effects of CFCs and HCFCs on the ozone layer, Mario J. Molina and F. S. Rowland described how these chemicals disintegrate in the stratosphere and produce chlorine atoms which ultimately destroy the ozone layer. Unfortunately, because of the stability of these chemicals, they can stay in the atmosphere for 40 – 150 years after being released (Molina and Rowland, 1974). Molina and Rowland's study created a global attention on the effect of CFCs and HCFCs. An international convention (Vienna convention) was organized to discuss these effects. This resulted in the signing of an international agreement known as the Montreal Protocol in 1987 which focused on reduction and final elimination of the ozone depleting substances (ODS) (United Nations, 1987). Apart from refrigerants, other uses of CFCs and HCFCs like propellants and blowing agents were also phased out (United Nations environment Programme, 2007).

As a substitute of CFCs and HCFCs, hydrofluorocarbons (HFCs) were invented. Most HFCs have negligible ozone depleting potential (ODP) and had characteristics similar to CFCs and HCFCs, therefore were taken as ideal replacements. However, like all artificial refrigerants, HFCs also have negative environmental impacts. Global warming, which means the rise of global average temperatures, was not in check until early 2000s when some of its effects started to be realized. Due to global warming, there has been melting of polar ice which increases the sea level and increase in air and ocean average temperatures which affect also the wind patterns. A lot has been argued and discussed concerning global warming. It is still an ongoing issue but most findings observe that concentration of greenhouse gases (GHGs) in the atmosphere is the cause of global warming. Greenhouse gases are gases which can absorb heat especially in the infrared range. When the sun heat the earth with its energy, most of this heat is reflected back to space by earth so as to balance the surface temperature. If GHGs are concentrated in the atmosphere, they absorb this reflected heat and thus retain the heat on earth, which results in rise of surface temperature. Therefore there has been a general call on the reduction of these GHGs (Calm and Didion, 1998, Bensafi and Thonon, 2007).

Global warming potential (GWP) of a gas is its capacity to absorb heat as compared to an equal mass of CO_2 over a certain period of time when it is still in the atmosphere. Unfortunately, most HFCs in common use in refrigeration industry have very high GWP. Due to these concerns, another inter-governmental forum was organized to discuss and possibly formulate solutions to abate global warming. The United Nations framework convention on climate change formulated what is now known as the Kyoto protocol in a bid to reduce the emissions of these GHGs. Under the Kyoto protocol, emissions of GHGs are restricted and this includes most HFCs (United Nations, 1998). Although the contribution of refrigeration emissions to global warming is minute as compared to emissions from industrial and transport sector, its abatement is paramount because of the GWP value of the refrigerants involved (Calm, 2008).

When safely contained in a proper refrigerator, refrigerants do not cause any harm to the environment, but if the system leaks or during maintenance or its end of life, these hazardous gases are released to the atmosphere and that is when they become harmful. Furthermore, during their manufacture, toxic products are created and released which are not only harmful to the environment, but are also a health concern. Due to the uncertainty of these artificial refrigerants, their use will be controlled and limited because their full environmental impact is not known. Therefore, there is a general need for a more permanent solution. As safety and environmental concerns are becoming more important, their impact as a requirement is becoming more crucial to even overshadow reliability, performance and cost. This calls for more natural and freely existing materials. Natural refrigerants are chemicals which occur as a result of natural process and which can be used to produce a refrigerating effect. They are not synthetic although can also be created synthetically. The most common are air, water, NH_3, hydrocarbons (HC) and Carbon dioxide (CO_2 or R744), also known as the gentle five. Natural refrigerants have negligibly small ODP and GWP (Bensafi and Thonon, 2007).

1.3. CO_2 as a Refrigerant

CO_2 can be regarded as the best refrigerant because it is non-toxic, non-flammable and does not contribute to ozone depletion and negligibly to global warming as compared to the HFCs. CO_2 used as a refrigerant can be recycled from other industrial processes thus reused instead of being released to the atmosphere. CO_2 meets all the basic requirements of a refrigerant in that it is readily available and not expensive. Also, it has excellent thermo-physical properties and transport properties leading to good heat transfer; it is not corrosive and is compatible with various common materials. In addition, it has efficient compression properties and compact system design due to high volumetric capacity and high operating pressures (Nekså, 2002, Sarkar et al., 2004, Nekså et al., 1998). The only concern with the use of CO_2 as a refrigerant is it becomes a super critical fluid at 31.1 °C and 73.7 bars. For low critical temperature refrigerants, a trans-critical heat pump cycle would perform better. This cycle is advantageous in some applications such as domestic water heating (DHW) because of the good temperature fit between the CO_2 and the water in a counter-flow gas cooler (condenser in trans-critical cycles) (Lorentzen, 1994). Unfortunately, CO_2 trans-critical cycles operate at high heat rejection pressures. High pressure presents design and cost challenges in terms of component robustness and compressor capability. Typical capital cost of CO_2 heat pumps is approximately double the cost of convectional heat pumps (Hashimoto, 2006). Therefore, its efficiency of operation must be maximized to justify the capital cost.

This study investigates the effects of various parameters to the improvement of the system efficiency in terms of the coefficient of performance (COP). Variables which affect the performance of the system such as the refrigerant amount, environmental temperature, throttle valve opening and water flow rates were varied accordingly but within the equipment limits. These experiments were done with a statistical approach so as to improve repeatability and reliability of the results. More information on important parameters which influences COP and their interactive effects are shed in this study. Both direct and indirect effects of these parameters are thoroughly investigated. Furthermore, these experiments were done in South Africa where there is a warm tropical climate. In selecting these parameters, local environmental conditions were considered so as to observe the performance of such systems in these climatic conditions. Currently, information on experiments done in these climatic regions is very rare especially in the public domain.

2. EXPERIMENTAL SETUP

2.1. CO_2 Heat Pump Test Bed

In this study, a CO_2 trans-critical water to water test bed was used to study the output of the system in a tropical environment. Highly pure CO_2 was used as the refrigerant while normal municipal water was used in the cooling water (gas cooler water) and chilling water (evaporator water) systems. The system contains a compressor, an evaporator, a throttling device and a gas cooler as the basic equipment among other secondary or supporting devices like vapor-liquid and oil separators; and cooling and chilling water systems. A schematic diagram of the system is shown in figure 1. The compressor of the system is Italian designed, special piston, semi-hermetic, Dorin's CO_2 compressor of the second generation with a maximum output of 10 MPa and 110 ºC thus the maximum limit of the system. The power consumption of the compressor when operating optimally is 3 kW. The throttle device used adopts a manual throttle valve design so that the amount of refrigerant flowing can be adjusted accordingly.

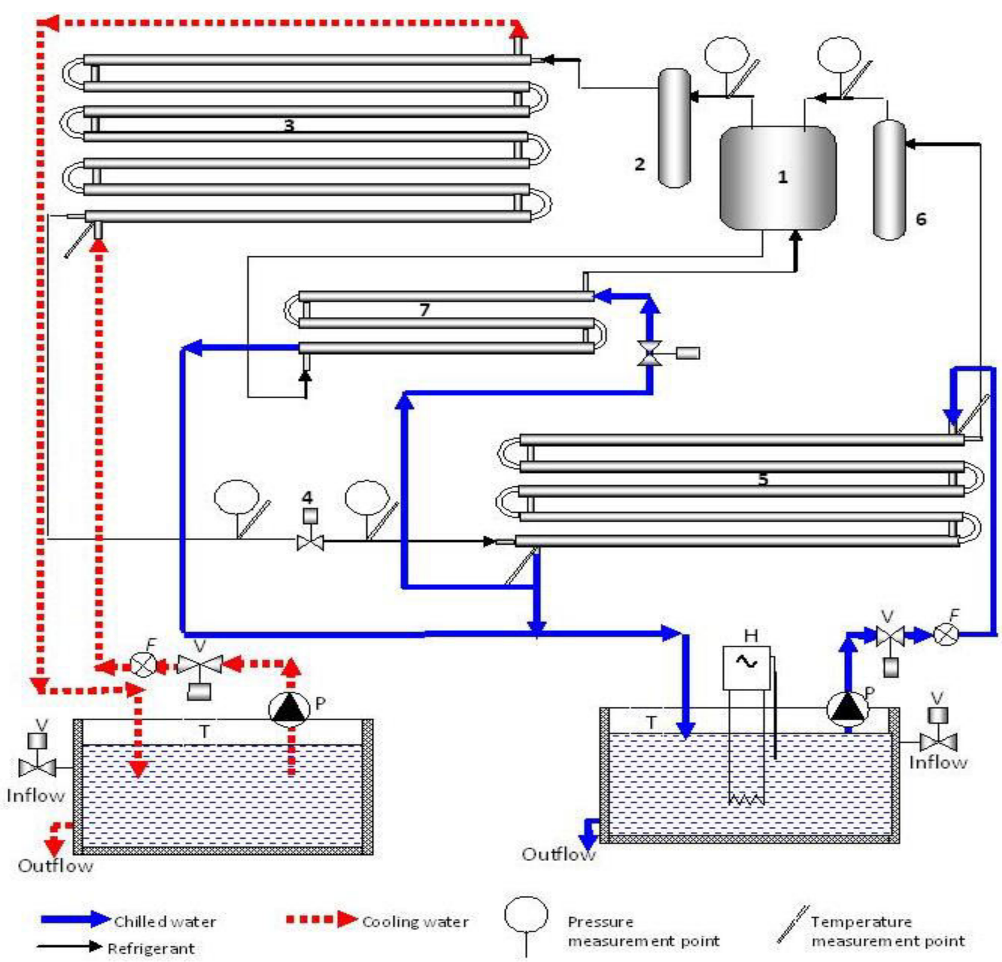

Figure 1 Schematic diagram of the CO_2 heat pump test bed; (1) compressor, (2) oil separator, (3) gas cooler, (4) thermal expansion valve, (5) evaporator, (6) gas-liquid separator, (7) lubricating oil heat exchanger, (*F*) flow meter, (V) water yield control valve, (P) water pump, (H) heater, (T) water tank.

Both the evaporator and gas cooler in this system have the tube in tube design because of the heat transfer and viscosity characteristics of the heat transfer fluids. CO_2 flows in the inner tube while water flows in the annulus of the outer tube. In the evaporator, there are three inner tubes while in the gas cooler there are four inner tubes. The inner diameter of the outer tube in both the gas cooler and the evaporator is 26 mm while the diameter of the inner tube varied according to the appliance. Due to the refrigerant characteristics, the evaporator inner tube has a diameter of 8 mm while the gas cooler inner tube had a diameter of 6 mm. All tubes had 1 mm thickness which is enough to counter the pressure experienced in them. The gas cooler has 12 passes whereas the evaporator has 10 passes, each pass in both having a length of 1580 mm.

The system also contains temperature, pressure and flow-rate sensors before and after each major component. Signals from these sensors are captured and stored by a data acquisition system. The data acquisition system hardware uses PC Auto industrial control software (Force supervisory control configuration software) to collect and store operating and system measurement parameters at each state point and reflect it in the data acquisition interface. To minimize losses to the environment, all pipes, heat exchangers and tanks were lagged. During experimentation, the test rig was left to run for some time until all the readings stabilized then the output recorded in terms of temperature and pressure of refrigerant at the state points, power consumed by the compressor (actual electric power) and the flow rate and temperature of chilling and cooling water systems (which provided the actual heat transferred to or from the water). The efficiency of the system (C.O.P) was then derived from the ratio of the actual heat transferred to the cooling water to the actual compressor power consumption.

2.2. Design of Experiments

In a bid to investigate the effect of input variables to the efficiency of the heat pump, optimization experiments were conducted. The variables considered were; the refrigerant amount, the water temperature and flow rate (both chilling and cooling) and the throttle valve opening. Table 1 indicates the minimum and maximum values of these variables as used in the experiments. The system full capacity refrigerant amount (design charge amount) is 4.2 kgs. The range experimented here varied between undercharged to overcharged conditions so as to investigate the effect of charge amount. On the other hand, considering the environmental conditions experienced in a tropical region like South Africa where this study was conducted, the water temperature was varied between 10 ºC (winter) to 30 ºC (summer). Due to the maximum output pressure and temperature limitations of the available compressor, the throttle valve minimum opening could not be less than 20%. It will also be noticed that the maximum value of the cooling water flow rate is higher than that of the chilling water flow rate. The reason behind this is that, the chilling water system is also used to cool the compressor oil. Therefore the difference between the cooling water flow rate and the chilling water flow rate is the water used for oil cooling (refer to figure 1).

Table 1 Range of variables

Name	Units	Low	High
A: Refrigerant amount (Ref Amt)	**kg**	1.26	5.25
B: Chilling water temperature (Ev Temp)	**deg C**	10	30
C: Cooling water temperature (GC Temp)	**deg C**	10	30
D: Throttle valve opening (Th V)	**%**	20	100
E: Cooling water flow rate (GCW Fl)	**kg/s**	0.192	0.383
F: Chilling water flow rate (EvW Fl)	**kg/s**	0.158	0.317

Design of experiments using design-expert® version 6 software (Stat-Ease Inc.) was used in these experiments for regression analysis. Response surface methodology (RSM) is a statistical method in design-expert® 6 that uses quantitative data from appropriate experiments to determine regression model equations and operating conditions. A standard RSM design called the central composite design (CCD) is suitable for investigating linear, quadratic, cubic and cross product effect of variables. It also helps to optimize the effective parameters and provide a lot of information with a minimum number of experiments as well as to analyze the interaction between the parameters. In addition, the empirical model that relates the response to the variables is used to obtain information about the process (Stat-Ease inc., 2002).

CCD comprises a two level full factorial design (2^6 = 64), twelve axial points and fourteen center points therefore a total of 90 experiments were conducted. The center points were used to determine the experimental error and the reproducibility of the data. The alpha (α) value, which is the distance of axial point from the center, was fixed at 2.82843 to make the design rotatable. The experiment sequence was randomized in order to minimize the effects of the uncontrolled factors. Each response of the COP was used to develop a mathematical model that correlates the COP to the input variables through first order, second order, third order and interaction terms, according to the following third order polynomial equation:

$$Y = b_0 + \sum_{j=1}^{n} b_j x_j + \sum_{i,j=1}^{n} b_{ij} x_i x_j + \sum_{j=1}^{n} b_{jj} x_j^2 + \sum_{k,i,j=1}^{n} b_{kij} x_k x_i x_j + \sum_{j=1}^{n} b_{jjj} x_j^3 \qquad \text{(i)}$$

Where Y is the predicted COP, b_0 is the first term, b_j is the linear effect, b_{ij} is the first order interaction effect, b_{jj} is the quadratic effect, b_{kij} is the second order interaction effect, b_{jjj} is the cubic effect, x_i, x_j and x_k are coded variables and n, the number of variables (in this study, n = 6) (Stat-Ease inc., 2002). The significance of the third-order model as shown in equation 1, was evaluated by analysis of variance (ANOVA). The insignificant coefficients were eliminated after the f (fisher)-test and the final model was obtained. Additional experiments were carried out to verify the predicted model and the associated optimal conditions.

3. RESULTS AND DISCUSSION

3.1. Response Matrix

Optimization process of various variables affecting the COP was analyzed by design of experiments in design-expert® 6. Table 2 shows the experimental design matrix and response of the experiments in terms of the COP. 14 runs (4, 18, 25, 36, 44, 47, 56, 71, 73, 74, 82, 86, 88 and 89) at the center point of the design were used to determine the experimental error. The results were reliably consistent. The COP ranged from 1.545 to 6.914 although with all the variables put at their optimum values within the range used in this study, a maximum COP of 11.8 was achieved (and proved by extra practical experiments). It is believed that even better COPs could be achieved when variable values are beyond the limits of this study. Still, the average COP value was 4.04 which means that at all conditions, the heat pump will be working better than when traditional water heating methods (e.g. using fuels or geysers) are applied.

Table 2 Design of experiments matrix and its response

Run	VARIABLES						RESPONSE
	A: Ref Amt kg	B: Ev Temp ºC	C: GC Temp ºC	D: Th V %	E: GCW Fl kg/s	F: EvW Fl kg/s	COP
1	2.550	16.464	16.464	74.142	0.254	0.210	3.675
2	3.960	16.464	16.464	45.858	0.321	0.210	4.758
3	2.550	23.536	23.536	74.142	0.254	0.265	1.979
4	3.255	20.000	20.000	60.000	0.288	0.238	4.520
5	3.960	16.464	23.536	74.142	0.321	0.210	2.577
6	3.960	23.536	23.536	45.858	0.254	0.210	4.625
7	2.550	16.464	23.536	45.858	0.321	0.265	3.004
8	2.550	16.464	16.464	74.142	0.321	0.265	4.296
9	2.550	16.464	23.536	45.858	0.254	0.210	2.539
10	2.550	23.536	16.464	45.858	0.254	0.265	5.438
11	3.960	23.536	16.464	74.142	0.321	0.265	5.831
12	3.960	23.536	16.464	74.142	0.254	0.210	4.900
13	2.550	23.536	16.464	45.858	0.321	0.210	5.612
14	2.550	23.536	23.536	74.142	0.321	0.210	2.021
15	3.960	16.464	16.464	45.858	0.254	0.265	4.414
16	3.960	16.464	23.536	74.142	0.254	0.265	2.479
17	3.960	23.536	23.536	45.858	0.321	0.265	5.163
18	3.255	20.000	20.000	60.000	0.288	0.238	4.433
19	2.550	23.536	23.536	45.858	0.254	0.210	2.513
20	2.550	23.536	16.464	74.142	0.254	0.210	4.889
21	2.550	23.536	16.464	74.142	0.321	0.265	5.286
22	3.960	23.536	16.464	45.858	0.254	0.265	6.076
23	3.960	16.464	23.536	45.858	0.321	0.265	4.195
24	3.960	23.536	23.536	74.142	0.321	0.210	4.184
25	3.255	20.000	20.000	60.000	0.288	0.238	4.360
26	2.550	16.464	16.464	45.858	0.321	0.210	4.363
27	2.550	16.464	23.536	74.142	0.321	0.210	1.552

Table 2 Design of experiments matrix and its response (cont.)

Run	VARIABLES						RESPONSE
	A: Ref Amt kg	**B: Ev Temp ºC**	**C: GC Temp ºC**	**D: Th V %**	**E: GCW Fl kg/s**	**F: EvW Fl kg/s**	**COP**
28	3.960	23.536	16.464	45.858	0.321	0.210	6.405
29	3.960	16.464	16.464	74.142	0.321	0.265	4.048
30	2.550	16.464	16.464	45.858	0.254	0.265	4.609
31	3.960	23.536	23.536	74.142	0.254	0.265	4.206
32	2.550	16.464	23.536	74.142	0.254	0.265	1.664
33	3.960	16.464	23.536	45.858	0.254	0.210	3.271
34	2.550	23.536	23.536	45.858	0.321	0.265	2.825
35	3.960	16.464	16.464	74.142	0.254	0.210	3.733
36	3.255	20.000	20.000	60.000	0.288	0.238	4.341
37	3.960	23.536	16.464	74.142	0.321	0.210	5.498
38	2.550	23.536	23.536	74.142	0.321	0.265	2.030
39	3.960	23.536	23.536	45.858	0.321	0.210	4.914
40	2.550	16.464	23.536	45.858	0.254	0.265	1.857
41	3.960	16.464	23.536	74.142	0.254	0.210	2.446
42	3.960	16.464	16.464	45.858	0.254	0.210	4.064
43	2.550	16.464	23.536	45.858	0.321	0.210	2.067
44	3.255	20.000	20.000	60.000	0.288	0.238	4.419
45	2.550	16.464	16.464	74.142	0.321	0.210	3.916
46	3.960	16.464	16.464	45.858	0.321	0.265	5.052
47	3.255	20.000	20.000	60.000	0.288	0.238	4.380
48	3.960	23.536	23.536	45.858	0.254	0.265	4.912
49	3.960	16.464	23.536	74.142	0.321	0.265	2.654
50	2.550	23.536	16.464	45.858	0.254	0.210	5.324
51	2.550	16.464	16.464	74.142	0.254	0.265	3.851
52	2.550	23.536	16.464	45.858	0.321	0.265	5.859
53	3.960	23.536	16.464	74.142	0.254	0.265	5.124
54	2.550	23.536	23.536	74.142	0.254	0.210	2.082
55	2.550	23.536	23.536	45.858	0.254	0.265	2.500
56	3.255	20.000	20.000	60.000	0.288	0.238	4.433
57	3.960	23.536	16.464	45.858	0.254	0.210	5.692
58	2.550	23.536	16.464	74.142	0.254	0.265	4.871
59	3.960	23.536	23.536	74.142	0.254	0.210	3.620
60	2.550	16.464	16.464	45.858	0.254	0.210	4.452
61	3.960	23.536	16.464	45.858	0.321	0.265	6.633
62	2.550	16.464	23.536	74.142	0.321	0.265	1.545
63	2.550	23.536	16.464	74.142	0.321	0.210	5.115
64	2.550	16.464	16.464	45.858	0.321	0.265	4.752

Table 2 Design of experiments matrix and its response (cont.)

Run	VARIABLES						RESPONSE
	A: Ref Amt kg	B: Ev Temp ºC	C: GC Temp ºC	D: Th V %	E: GCW Fl kg/s	F: EvW Fl kg/s	COP
65	2.550	23.536	23.536	45.858	0.321	0.210	2.675
66	3.960	16.464	16.464	74.142	0.254	0.265	3.780
67	3.960	23.536	23.536	74.142	0.321	0.265	4.274
68	2.550	16.464	23.536	74.142	0.254	0.210	1.649
69	3.960	16.464	23.536	45.858	0.254	0.265	3.164
70	3.960	16.464	23.536	45.858	0.321	0.210	3.332
71	3.255	20.000	20.000	60.000	0.288	0.238	4.433
72	3.960	16.464	16.464	74.142	0.321	0.210	3.930
73	3.255	20.000	20.000	60.000	0.288	0.238	4.414
74	3.255	20.000	20.000	60.000	0.288	0.238	4.419
75	3.255	20.000	20.000	60.000	0.192	0.238	3.510
76	3.255	20.000	20.000	60.000	0.288	0.317	4.593
77	1.260	20.000	20.000	60.000	0.288	0.238	1.664
78	3.255	20.000	20.000	60.000	0.383	0.238	4.475
79	3.255	20.000	30.000	60.000	0.288	0.238	2.301
80	3.255	30.000	20.000	60.000	0.288	0.238	6.500
81	3.255	10.000	20.000	60.000	0.288	0.238	2.978
82	3.255	20.000	20.000	60.000	0.288	0.238	4.467
83	3.255	20.000	20.000	60.000	0.288	0.158	3.918
84	3.255	20.000	20.000	100.000	0.288	0.238	3.749
85	3.255	20.000	20.000	20.000	0.288	0.238	6.194
86	3.255	20.000	20.000	60.000	0.288	0.238	4.487
87	5.250	20.000	20.000	60.000	0.288	0.238	4.147
88	3.255	20.000	20.000	60.000	0.288	0.238	4.487
89	3.255	20.000	20.000	60.000	0.288	0.238	4.453
90	3.255	20.000	10.000	60.000	0.288	0.238	6.914

3.2. Analysis of Variance (ANOVA)

From statistical point of view, there are three tests required to evaluate the model, these are, significance of factor test, R-squired test and lack of fit test. The significance test was indicated by the Fisher variance ratio (the F-test value) and its associated probability (Prob>F). The model equation was evaluated by F-test ANOVA which revealed that these regressions are statistically significant at 95% confidence level (table 3). As a general rule, the greater the F-value is from unity, the more certain it is that the empirical model describes the variation in the data about its mean and the estimated significant terms of the variables are real. The values of prob>F which are 0.05 or less indicate significance. Quadratic model was suggested to be the best because its prob>F is less than 0.05 (<0.0001) and due to its least significant lack of fit test. Quadratic model shows the significance of adding quadratic terms to the mean and blocks when linear and two factor interaction terms are already in the model.

By using multiple regression analysis, the response (C.O.P) obtained in table 2 was correlated with the six variables studied using the polynomial equation 2 after excluding the insignificant terms identified using Fisher's test method as seen in table 3.

$$COP = 4.33 + 0.45A + 0.57B - 0.93C - 0.38D + 0.16E + 0.099F - 0.22A^2 - 0.086E^2 - 0.053F^2 + 0.2AB + 0.34AC - 0.053AD + 0.061AE - 0.099BC \quad \text{(ii)}$$

Where A, B, C, D, E and F are as defined in table 1. The coefficient of the full regression model equation and their statistical significance were determined and evaluated using ANOVA. Positive sign before terms indicate synergistic effect, while negative sign indicates antagonistic effect. Generally in all the significant term, most of the linear effects were synergistic except gas cooler water temperature and throttle valve opening (these antagonistic values are expected because, theoretically, the lower the cooling water temperature, the higher the heat transfer in the gas cooler and therefore the higher the COP. In addition, the lower the throttle valve opening, the higher the gas cooler pressure and thus the higher the heat transfer and COP but until a certain optimum pressure value). All the quadratic effects were antagonistic while the interactive effects were partly synergistic and partly antagonistic.

Quadratic antagonistic effects mean that at extreme values of the variables, they produce negative results (become detrimental to the COP). The synergistic interaction effects means that as one effect is increased in size, the other interacted effect will produce better results if it is also increased and vice versa. In this case, the refrigerant amount is seen as a reinforcement to the other variables, whereby, by increasing it, the other variables effects will be improved (except throttle valve opening) resulting in very good system efficiency. On the other hand, antagonistic interactive effect means an increment on one variable will lead to decay in the effect of the other interactive variable and thus a low COP. This is the case with refrigerant amount interaction with throttle valve opening and also the chilling water temperature interaction with cooling water temperature. These results have also been observed in other studies (Stene, 2007).

The coefficients of the variables in equation 2 represent the magnitude of the effect the variable has on COP which is dictated by its F value and prob > F. The effect of the variable on COP becomes high if its coefficient is high. The opposite happens if the coefficient is low. In this study, the linear effect of cooling water temperature had the highest coefficient (0.93) followed by the linear effect of chilling water temperature (0.57), and then the linear effect of refrigerant amount is third (0.45) and the linear effect of throttle valve opening is fourth (0.38). Surprisingly enough, the fifth highest effect is the interactive effect of refrigerant amount and cooling water temperature (0.34) then followed by the quadratic effect of refrigerant amount (0.22). The least effect among the significant variables was interactive effect of refrigerant amount and throttle valve opening (0.053).

From these coefficients, it is clear that the water inlet temperatures are very important in determining the COP of a CO_2 heat pump. Though it can also be seen that their quadratic effects are insignificant given that they had very low F values and high Prob>F values as seen in table 3. However, their interactive effects are still highly significant and important. It is surprising to note that the throttle valve opening, which directly affects the gas cooler pressure, does not have a lot of effect as compared to the first three variables, probably because it also affects the refrigerant flow rates. On the other hand, water flow rates effect is minute both linearly and in quadratic form, thus it is not a parameter of much concern. A lot more analysis and observations can be done from this polynomial equation alone especially concerning the magnitude of the coefficient and their specific signs. As it can be observed, with a proper selection of the variables, the COP will definitely be optimized. Still, caution should be taken because interactive effects have amplification tendencies because of contribution from two or more variables while their coefficient is not as effective as when only one variable is considered like in quadratic cases. In this study for example, it will not be surprising to realize that the

Table 3 ANOVA for response surface quadratic model

Source	**Sum of Squares**	**DF**	**Mean Square**	**F Value**	**Prob > F**
Model	144.96	27	5.37	126.26	< 0.0001
A	16.35	1	16.35	384.51	< 0.0001
B	25.71	1	25.71	604.55	< 0.0001
C	69.88	1	69.88	1643.41	< 0.0001
D	11.45	1	11.45	269.25	< 0.0001
E	2.02	1	2.02	47.41	< 0.0001
F	0.78	1	0.78	18.29	< 0.0001
A^2	5.82	1	5.82	136.83	< 0.0001
B^2	0.01	1	0.01	0.15	0.7016
C^2	0.01	1	0.01	0.23	0.6315
D^2	0.16	1	0.16	3.66	0.0608
E^2	0.88	1	0.88	20.59	< 0.0001
F^2	0.33	1	0.33	7.86	0.0069
AB	2.61	1	2.61	61.45	< 0.0001
AC	7.48	1	7.48	175.98	< 0.0001
AD	0.18	1	0.18	4.29	0.0429
AE	0.24	1	0.24	5.63	0.021
AF	0.07	1	0.07	1.67	0.2016
BC	0.63	1	0.63	14.74	0.0003
BD	0.01	1	0.01	0.26	0.6121
BE	0.02	1	0.02	0.51	0.4768
BF	0.00	1	0.00	0.00	0.9512
CD	0.05	1	0.05	1.24	0.2709
CE	0.14	1	0.14	3.22	0.0781
CF	0.02	1	0.02	0.54	0.4666
DE	0.09	1	0.09	2.02	0.1602
DF	0.05	1	0.05	1.08	0.3025
EF	0.15	1	0.15	3.48	0.067

quadratic effect of chilling water temperature had more influence to COP than the interactive effect of refrigerant amount with cooling water flow rate or the quadratic effect of cooling water flow rate having more influence than the interactive effect of cooling and chilling water temperatures.

R value is very high for the model (0.9916) therefore the variability of the response could accurately be explained by the mathematical model of equation 2. On the other hand, the value of R^2 for the model is 0.9833 which implies that 98.33% of the total variation in the COP responses is attributed to the experimental variables studied as stipulated by the model. This is further stressed by the low value of the standard deviation (0.21), the high value of Adequate Precision (45.74) – which means the model can be used to navigate the design space, and the closeness between the Adjusted R-squired (0.9755) and the Predicted R-squired (0.9523). The lack of fit test compares the residual error to the experimental error (pure error) from replicated design points. It is this test which was used to select the quadratic model over the linear and the two factor interaction (2FI) model, whereby all of them had model Prob>F values of less than 0.05 (< 0.0001) but the lack of fit test for the linear and 2FI model had very large F value as compared to the lack of fit test for the quadratic model. Although this implies that the lack of fit of the quadratic model is significant relative to the pure error, still, its significance is way lower as compared to the linear and 2FI models. However, caution should be taken because large residuals might results from fitting the model when the lack of fit is significant.

3.3. Graphical Response and Diagnostics

A plot of predicted COP results and actual (experimental) results (figure 2) further validates the mathematical model due to its linearity with the line of unit slope (perfect fit with points corresponding to zero error). This plot proves that the model describes the connection between the variables and COP adequately within the range of the variables studied.

Figure 3, 4 and 5 are 3D plots of the interaction between the variables and their effect to COP. The X and Y-axis values of these figures are the real values of variables. These response surfaces facilitate a straight-forward examination of the effects the variables exert on the COP of the heat pump.

Figure 3 shows the response surface of the COP with varying refrigerant amount (A) and throttle valve opening (D), the other four variables were held constant at their mid-levels. The response surface has a sinusoidal behavior towards refrigerant amount and a linear mini-max behavior towards throttle valve opening. Similarly, the response surface has a linear mini-max behavior both towards the chilling (evaporator) water temperature (B) and cooling (gas cooler) water temperature (C) as shown in figure 4. On the other hand, the response surface has a sinusoidal behavior both towards the cooling water flow rates (E) and chilling water flow rates (F), with its peak point at higher flow rates.

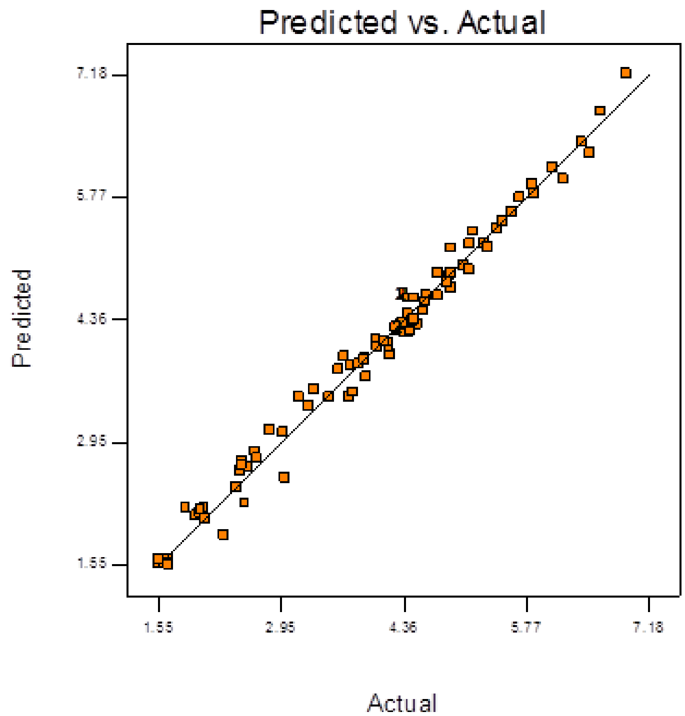

Figure 2 Model Predicted responses versus Actual (Experimental) responses.

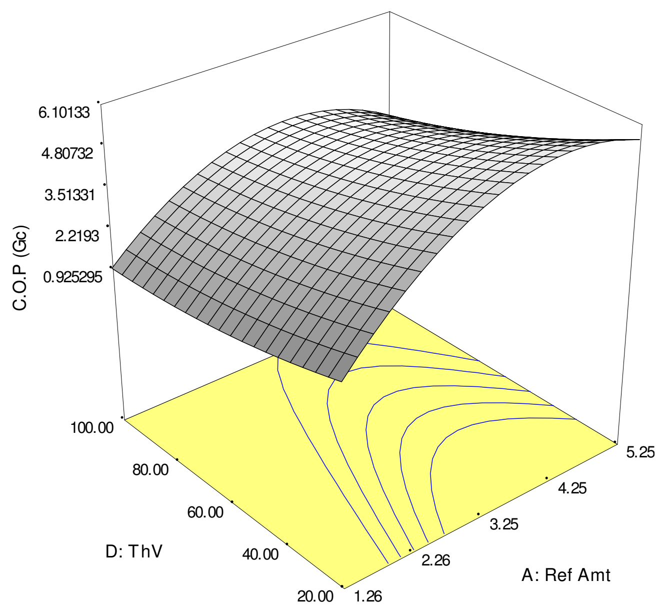

Figure 3 Interactive effect of refrigerant amount and throttle valve opening on the COP.

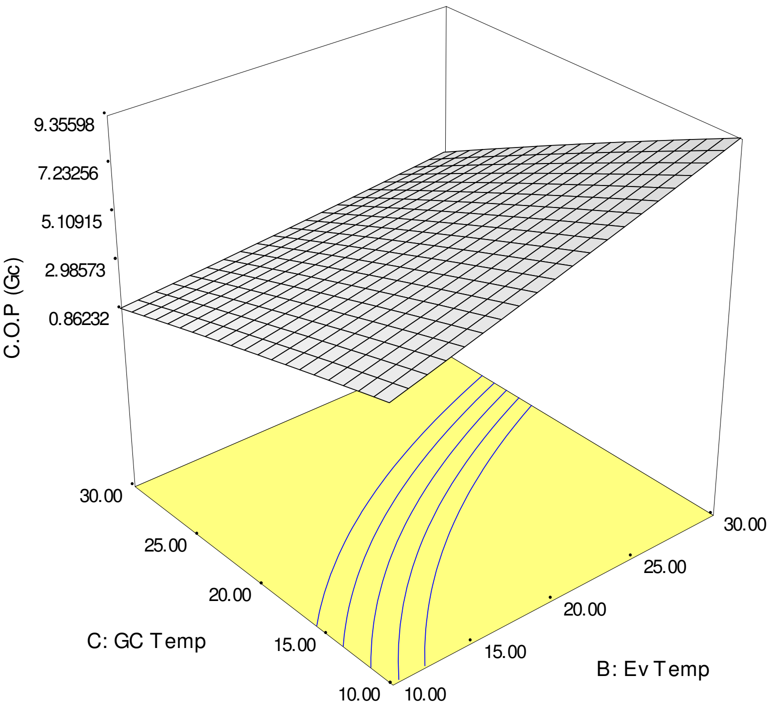

Figure 4 Interactive effect of evaporator and gas cooler water temperatures on the COP.

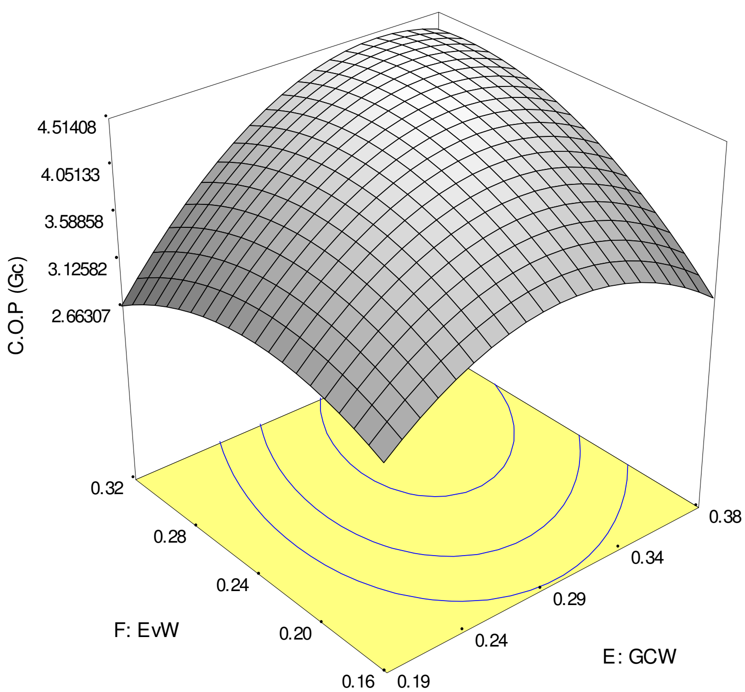

Figure 5 Interactive effect of evaporator and gas cooler water flow rates on the COP.

Mini-max response means that the optimum value of the variable was not achieved in the given range of the variable. If an optimum point is achieved, then a sinusoidal behavior will result which indicates that the optimum value of the respective variable is within the range considered (as with the case of refrigerant amount and water flow rates). A sinusoidal behavior results when the effect of the variable on the response surface varies non-linearly. These variables, accordingly, have a quadratic effect on the response surface and their optimum point is in the range employed. It can be seen that according to equation 2, refrigerant amount and the cooling and chilling water flow rates had a significant quadratic value and thus produce a sinusoidal response surface.

3.4. Discussion

3.4.1 Refrigerant amount

From the polynomial equation 2, refrigerant amount has an effect both linearly, in quadratic form and in interaction. This means that refrigerant amount has a great influence in COP determination. Practically, effect of refrigerant amount is experienced especially when there is system leakage (less refrigerant amount) or after recharge. In this study, its synergistic linear effect coupled with its antagonistic quadratic effect means that as the refrigerant amount is being added, the COP improves until an optimum point is reached where any more addition of the refrigerant amount results in a reduction of the COP. For any one system, there is always an optimum amount of the refrigerant. This optimum amount is not necessarily the full design amount of the system, but an amount in which the heat exchangers and the compressors work very efficiently (Sarkar et al., 2009, Atik and Aktaş, 2011). In the equipment under study here, the optimum point, within the range of throttle valve opening considered, was approximately 4.01 kgs of refrigerant (figure 3).

With the interactive effects, it can be seen that all the interaction between refrigerant amount and other variables had synergistic effect except its interaction with throttle valve opening. This can be expected because as you increase the refrigerant amount, the highest COP can only be achieved from high optimum gas cooler pressure (Stene, 2007). It is a known fact that optimum gas cooler pressure is essential for high COP because the gas cooler pressure has marked influence on the specific enthalpy due to the s-shape of the isotherm in supercritical region. For supercritical operation, high side pressure is determined by the relationship between refrigerant charge (mass), inside volume and temperature (Sarkar, 2005). This high gas cooler pressure can be realized by reducing the throttle valve opening. It must be noted that these results are realized within the range of test in this study. This means that different results might be realized if for example higher amount of refrigerant are used or when the throttle valve is closed beyond 20%. This factor can be stressed by the fact that the magnitude of this interactive effect was the lowest thus it asserts very weak influence to COP actualization.

It is interesting to note that the interactive effect of refrigerant amount and cooling water temperature was synergistic instead of being antagonistic. The theory behind it can be that as the refrigerant amount is increased, the required inlet cooling water temperature can also be increased so as to ensure the inlet temperature to the throttle valve and thus to the evaporator and finally the compressor is high enough to guarantee an efficient compression process. Also, as the refrigerant amount increases, the compression ratio decreases and thus higher volumetric efficiencies of the compressor are achieved. On the other hand, as the volumetric ratio decreases, the refrigerant mass flow rate increases which means a higher heat transfer even with higher cooling water temperature (Cho et al., 2007). Therefore, even if the heat transferred in the gas cooler remains constant, the compressor power intake on the other hand is less and thus a high COP results. Other studies realized similar results (Rieberer, 1998). From figure 3, it can be observed that the refrigerant amount has a sinusoidal relationship to COP (more pronounced at lower values of throttle valve opening). This supports the fact that after the optimum refrigerant amount value, the COP will diminish in value.

Refrigerant charge is an important parameter that affects the performance of a heat pump system. There is an optimum charge at which the system yields the best COP. Performance deterioration is more severe at undercharged condition than at overcharged condition mostly because of the effects of less mass flow rates and reduced compressor efficiency (Kim et al., 2005). The CO_2 system performance is reported to be more sensitive to system refrigerant charge compared to that in conventional systems because of its low liquid to vapor density ratio. This makes expansion losses the dominant factor of the overall system performance (Cho et al., 2005). Other studies report that too much refrigerant charge reduces the gas cooler heat transfer area and thus results with a high output pressure. On the other hand, an insufficient charge lead to less evaporation and reduce cooling capacity drastically (Wang, 2000). If the CO_2 charge is too small, the receiver will be emptied and the evaporator will be underfed at high operating pressures. This will in turn result in excessive superheating and reduced evaporation temperature, and as a consequence poor system performance (Stene, 2004). Therefore, the refrigerant charge must be controlled more precisely in a CO_2 system than in other conventional systems to achieve high performance under various operating conditions.

Charge optimization of the trans-critical CO_2 cycle is directly related with optimizing the gas cooling pressure to maximize the COP (or heating capacity). The second derivative of the pressure with respect to the enthalpy at constant temperature for CO_2 becomes zero in supercritical region near the critical point where the gas cooler outlet condition is approaching. This characteristic would result in the gas cooler capacity increase in superior degree compared to that of the compressor power increase with the gas cooling pressure increasing until it reaches the pressure corresponding to the reflection point. Higher refrigerant charge will result in higher system operating pressures and the higher ambient temperature will result in higher optimum gas cooling pressure (Fernandez et al., 2010, Hwang and Radermacher, 1998).

Entropy generation can be used to analyze the second law efficiency for the trans-critical CO_2 system under various charging conditions. Expansion loss is the dominant factor affecting system performance at undercharged condition while the gas cooler loss became the major parameter at overcharged conditions. The losses in the gas cooler increases as the amount of refrigerant charge increase due to the increase of the heat transfer rate. The enthalpy at the inlet of the evaporator significantly varies with a change of the gas cooler pressure; this change greatly varies the cooling capacity. Generally, with an increase of gas cooler pressure, the enthalpy at the exit of the gas cooler decreases. This results to lower quality at the inlet of the evaporator.

Although the refrigerant temperature at the exit of the gas cooler gradually increases with the addition of refrigerant charge at undercharged conditions, the quality at the inlet of the evaporator decreases due to a reduction in the enthalpy at the exit of the gas cooler, thus increasing the cooling capacity. However, for overcharged conditions, the performance of the CO_2 system decreases with refrigerant charge because the enthalpy difference across the compressor increases more than that across the gas cooler. Furthermore, compression ratio decreases with the addition of the refrigerant charge; this decrease causes an increase in the refrigerant mass flow rate. Therefore, the compressor power consumption slowly but continuously increases with normalized charge, while the cooling capacity rapidly increases at lower normalized charges and then the slope gradually decreases with an increase of the normalized charge (Cho et al., 2005).

3.4.2 Chilling water temperature

Chilling water temperature or the heat source temperature has a strong linear effect and some interaction effects (equation 2). Its linear effect is the second strongest after cooling water temperature. A linear effect signifies a direct effect to the response. Therefore, the heat source temperature exercises a strong direct effect to the system COP. Lack of its significant quadratic effect means that this parameter's effect does not change as its magnitude changes. The more you increase it (positive coefficient), the more it will improve the COP, a fact supported by figure 4. Still, the intensity of this behavior is dependent on the other interactive effects. Its synergistic interaction effect with refrigerant amount and antagonistic effect with cooling water temperature was as expected where with the refrigerant amount, an increase in any of them improves the COP, on the other hand, the cooling water temperature must reduce for the COP to improve.

Heat source temperature affects a lot of other parameters such as the gas cooler outlet temperature which in turn directly determines the heat transfer in the gas cooler (Duddumpudi, 2010). Heat pump COP values increase from a minimum at high gas cooler exit temperature and low evaporator temperature to a maximum at low gas cooler exit temperature and high evaporator's temperature. Therefore, for maximum system efficiencies, the evaporator's temperature must be as high as possible while the gas cooler exit temperature as low as possible (a factor which is made possible by low cooling water temperature). This statement is further emphasized by the coefficient of variable B and C in equation 2. In fact, in some studies, the effect of these temperatures was greater than some equipment enhancement like internal heat exchangers (Sarkar, 2005), while in others, these temperatures, if optimized, can improve the exergy efficiency of the overall system (Wang et al., 2010, Chen et al.).

Higher chilling water temperature ensured complete evaporation of the refrigerant and a possible superheat before compressor intake thus it has a high enthalpy. This reduces the compressor work input while it improves its performance due to the state of the refrigerant at entry and at the same time, increases the maximum cycle temperature thus more heat output (Skaugen, 2002). Lower system performance associated with lower chilling water temperatures can either be due to reduced refrigeration capacity due to lower heat source temperature, or due to increased specific compression powers coupled by lowered compression performance due to high pressure ratios encountered. These pressure ratios also cause the volumetric capacity of the compressor to be lowered due to a lower volumetric efficiency. Refrigeration capacity is further reduced by lower mass flow rates when chilling water temperatures are low. The lower mass flow rates are caused by the low density experienced at the evaporator. As the chilling water temperature increases, the gas cooler optimum pressure also increases (Adriansyah, 2001). Consequently, CO_2 heat pumps should preferably be designed for a moderate optimum gas cooler pressure. The optimum pressure depends on evaporation temperature, gas cooler exit temperature and compressor isentropic efficiency. The evaporation and gas cooler exit temperatures depend on the chilling and cooling water temperatures (Sarkar et al., 2007).

3.4.3 Cooling water temperature

Cooling water temperature or heat sink temperature has the strongest coefficient in equation 2 and thus has the greatest effect in COP determination. This temperature determines the amount of heat transfer to occur in the gas cooler, whereby the lower the temperature (thus an antagonistic coefficient) the higher the heat transfer and COP. Other researchers also had similar observations (Sarkar et al., 2010). As with the chilling water temperature, the cooling water temperature does not have a significant quadratic effect and thus exerts direct effects on the COP. Its antagonistic effect is so strong to even affect its interaction effect with chilling water temperature which means that in an overall scenario, if both the chilling and cooling water temperatures were to be the same (for example where both are supplied from the same source e.g. ground water), then for a maximum COP, the cooling water requirements should first be met (i.e. the cooling water temperature is the controlling factor). Figure 4 also supports these observations by its linear mini-max behavior. It is also clear from figure 4 that the intensity of one temperature effect depends on the other. This fact is also indicated by the presence of a significant temperature interaction coefficient in equation 2.

Cooling water temperature is an important parameter in that it directly determines the gas cooler exit temperature (Sarkar, 2005). A low gas cooler exit temperature is a pre-condition for high efficiency (Duddumpudi, 2010). It is believed that at a higher exit gas cooler temperature, the refrigerant quality is still very high thus causing higher losses both in throttle valve and the evaporator (Agrawal and Bhattacharyya, 2008). This explains the antagonistic nature of the cooling water temperature coefficient in equation 2. It is a well-known fact that the gas cooler exit temperature is a very important parameter in COP determination. Many studies support this allegation (Kim et al., 2004, Sarkar et al., 2004). In some studies, the optimum operational conditions are dependent on this exit temperature (Bensafi and Thonon, 2007, Stene, 2007, Sarkar, 2005) while in others a direct relationship between the exit temperature and COP exists (Kim et al., 2004, Skaugen, 2002). Apart from increasing the optimum gas cooler pressure, a high refrigerant temperature at the gas cooler exit means less heat transfer in the evaporator and higher compressor power consumed because a higher refrigerant temperature was delivered there (Sarkar et al., 2004). Care should be taken, however, for the exit temperature not to exceed the critical temperature because if it exceeds, the gas cooler operating pressure cease to be dependent of the exit temperature and thus not controlled by it (Sawalha, 2008).

When cooling water temperature is increased, the approach temperature (difference between the refrigerant temperature and the cooling water temperature at the gas cooler outlet) is reduced thus less heat transfer in the gas cooler. This reduces the system performance drastically because of thermodynamic losses. The rise in cooling water temperature has more devastating effect on the thermal efficiency of the system as compared to the exergetic efficiency (DiPippo, 2004). In addition, exergy losses and other irreversibility in the throttle valve also increase rapidly as the cooling water temperature increases due to the increase in the quality of refrigerant at the exit. This adds frictional and momentum effect resulting in higher pressure and exergy losses in the evaporator as well (Agrawal and Bhattacharyya, 2009). The maximum COP for a heat pump unit decreases quite rapidly when the cooling water temperature increases because of reduced heating capacity ratio. The average gradient ($\partial\Delta_{TA}/\partial p_{GC}$) becomes steep when the cooling water temperature is low, where ΔT_A is the temperature approach and p_{GC} the gas cooler (high-side) pressure. The compressor power input will also affect the COP, since the optimum high-side pressure increases slightly when the cooling water temperature is increased (Stene, 2004, Skaugen, 2002).

There is a general decrease of refrigerant heat transfer coefficient in the evaporator when the gas cooler pressure increases at higher cooling water temperature. This is due to the change of vapor quality at the entry of the evaporator which increases as the cooling water temperature increases. This increase in vapor quality is more pronounced at lower gas cooler pressures. Therefore, at higher cooling water temperature, the coefficient of heat transfer is affected negatively. The gas cooler refrigerant heat transfer coefficient is affected especially in the vicinity of the critical point. Here there are large changes in thermo-physical properties of the refrigerant and thus gives the heat transfer coefficient a sharp peak. A lower gas-cooling pressure is better for lower ambient case, while a higher gas-cooling pressure is better for higher ambient case. At lower cooling water temperatures, a lower gas-cooling pressure offers the highest COP, while at higher cooling water temperatures, a higher gas-cooling pressure offers the highest COP (Hwang and Radermacher, 1998). Still, the CO_2 cycle system performance is sensitive to gas cooler exit temperature because a minute change in the exit temperature can produce huge change in the exit enthalpy due to the associated change in the specific heat especially in conditions nearing the critical point (Sarkar, 2005).

3.4.4 Throttle valve opening

Throttle valve opening as a variable has a reasonable effect on the COP. From equation 2, throttle valve opening had an antagonistic linear effect (the less it is the higher the efficiency) and an interactive effect with refrigerant amount. Its lack of quadratic effect means that an optimum value was not reached in the range considered in this study. Figure 3 also supports these observations. A small extension of the figure still reveals a linear mini-max behavior more pronounced at higher amounts of refrigerant which explain the interactive effect. Throttle valve opening directly affects the refrigerant mass flow rates and system high side pressure. To reduce the throttling losses it is important to achieve lowest possible temperature in the gas cooler outlet. Every heat exchanger has a minimum temperature difference between hot and cold fluid, below which, the heat exchanger is not able to exchange heat. The pinch point is where this temperature limit occurs. The pinch-point has to be located in the gas cooler outlet so as to achieve the lowest possible outlet temperature. The location of the pinch-point is dependent on the gas cooler pressure and the CO_2 mass flow rate. An increase of pressure or mass flow rate moves the pinch-point towards the gas cooler outlet. When the pinch is at the gas cooler outlet more water can be heated than if the pinch point is inside the gas cooler (Christensen, 2009).

The high-side pressure is controlled by adjusting the opening of the throttle valve (Sarkar, 2005), thus temporarily changing the balance between the mass flow rate in the compressor and the valve. By reducing the valve opening, more CO_2 will accumulate in the gas cooler and piping, and the high-side pressure will rise until a new balance point for the mass flow rate in the compressor and the valve has been reached. The change in mass flow rate accordingly adjusts the heat transfer rate also (Sarkar, 2005, Kim et al., 2007). The extra CO_2 charge needed to increase the pressure is boiled off and transferred from the liquid reservoir in the receiver. When increasing the opening of the expansion valve, the high-side pressure will be reduced, and the surplus CO_2 is stored as liquid in the receiver. The low-pressure receiver should be designed to prevent possible liquid droplets from entering the suction line, as well as to provide sufficient volume to avoid excessive pressures if the system is inoperative at high ambient temperatures (Stene, 2004).

As it is widely known, at a single water inlet temperature, there is always an optimum pressure of operation, where above this pressure or below it the COP will always be less. In these systems, as the pressure increases the COP increases initially and then the added capacity no longer compensates for the additional work of compression and hence COP decreases. At the optimum pressure, the marginal increase in capacity equals marginal increase of work. High-side pressure regulation can be applied to maintain the maximum COP and/or to regulate the heating or cooling capacity (Sarkar, 2005). Evaporator outlet vapor fraction and the degree of superheat at compressor inlet can also be controlled by adjusting the throttle valve opening (Kim et al., 2007). Furthermore, there is a general decrease in pressure loss as the higher pressure increases. Pressure loss in pipes and heat exchanger is one of the highest causes of irreversibility in heat pumps. Its reduction means an improvement in the efficiency of the heat pump. Tolerable pressure drops in heat exchangers become higher as the pressure level increases, and this gives a possibility of improving heat transfer through higher flow velocities in high-pressure systems. This is of particular importance for single-phase heat transfer in the gas cooler of CO_2 systems. High pressure and proximity to the critical point causes increased specific heat, again leading to improved convective heat transfer. On the other hand, evaporator pressure drop leads to reduced temperature differences due to the corresponding drop in saturation temperature (Kim et al., 2004).

The effect of high gas cooler pressures in compressors can also be another possibility of the observed results. Compressors operate more efficiently at higher gas cooler pressures. Furthermore, the negative effect of pressure loss becomes less significant as the output pressure increases both in the compressor and gas cooler. Additionally, owing to the higher pressure level, the negative effect of valve pressure drops tends to be small in CO_2 compressors, thus giving higher efficiency (Kim et al., 2004). The optimum gas cooler pressure is a very important parameter, in that, the input power to the compressor is more or less proportional to the gas cooler pressure. At the same time, at low gas cooler pressure the cooling curve has an s- shape, because of the nonlinear c_p value. The shape becomes more linear with increasing pressure. Due to the s-shape of the isotherm in supercritical region, the gas cooler pressure has marked influence on the specific enthalpy. Since the throttling valve inlet condition and enthalpy determines the specific refrigeration effect, it is necessary to control the high side pressure (Sarkar, 2005).

At lower operating pressures (when the throttle valve is fully open or just slightly closed), the low heat output and COP was probably caused by low wall superheat. This superheat, which is needed to initiate nucleate boiling becomes low as the critical pressure is approached, thus low heat transfer characteristics are experienced (Kim et al., 2004). Still, due to the great variations in the specific heat capacity at pressures and temperatures above and near the critical point, the temperature - enthalpy slope is not constant and the isobars are not parallel. As a consequence, the change of the specific enthalpy difference in the gas cooler is not proportional to the change in the specific compressor work, and for each fixed outlet temperature from the gas cooler there will therefore be an optimum high-side pressure leading to a maximum COP (Stene, 2004).

3.4.5 Cooling water flow rate

Cooling water or heat sink flow rate had a linear, quadratic and interactive effect according to equation 2. The presence of a quadratic effect symbolizes an optimum amount in the study range (figure 5). This outcome was also observed in other studies (Fronk and Garimella, 2011). From figure 5, it can be observed that cooling water had a sinusoidal behavior more pronounced at higher values of chilling water flow rates. Its non-uniform optimum point depends on the other variables i.e. in the case of interactive effect between the cooling and chilling water flow rates, at lower chilling water flow rates, the optimum cooling water flow rate is low and at higher chilling water flow rates the optimum cooling water flow rate is high e.g. at chilling water flow rate of 0.16 kg/s, the optimum cooling water flow rate was approximately 0.3 kg/s but at chilling water flow rate of 0.317 kg/s, the optimum cooling water flow rate was approximately 0.35 kg/s (figure 5).

It is a well-known fact that heat transfer depends on the flow rate among other parameters like temperature difference and thermo-physical properties of the fluids. A lower flow rate means a low heat transfer because of either insufficient fluid for effective heat transfer (the fluid temperature increases rapidly thus it reaches a point where the temperature difference is insufficient for effective heat transfer) or because the fluid was simply not enough to cover the entire heat transfer surface area adequately. Generally, the water side heat transfer coefficient is reduced as the flow rate reduces. When the cooling water flow rate is maximized, more heat is transferred to the water and thus the gas cooler outlet temperature is reduced, therefore a low evaporator inlet temperature is achieved. Similarly, the opposite occurs when cooling water flow rate is minimized. Water flow rates seem to only affect the heat transfer but not the compressor power consumption. This was also observed in other studies (Sarkar, 2005, Jiang et al., 2009). In some studies, this improvement of heat transfer in the gas cooler is attributed to the elimination of the pinch effect by higher cooling water flow rates (Fronk, 2007).

Regulating mass flow rates of refrigerant or cooling medium (cooling water in this case) at a constant chilling water flow rate will affect the approach temperature. For a constant refrigerant mass flow rate, increasing mass flow rates of cooling medium will reduce refrigerant temperature at the outlet of the gas cooler thus improve the heating capacity (Reulens, 2009). If rejected heat is not utilized and just dissipated to the cooling medium, the refrigerant outlet temperature at the gas cooler can be made as low as possible with the cooling water flow rate adjustments, hence reducing the approach temperature (Adriansyah, 2001, Stene, 2004). For same size of gas cooler, the approach temperature reduces with increase in the operating pressure. The gas cooler capacity increases with increase in the operating pressure and also with the size of gas cooler. It also increases with increase in cooling medium flow rate as expected but the effect of increase in cooling medium flow rate is insignificant at higher pressure (Gupta and Dasgupta, 2014).

3.4.6 Chilling water flow rate

Chilling water flow rate had the least significant effect on COP both linearly and in quadratic form. It did not have any interactive effect probably because it does not catalyze other variables to perform better. Also, its influence on COP, even though minute, is important for consideration. Presence of a significant antagonistic quadratic effect symbolizes the negative impacts it has at extreme levels. This observation is also clear in figure 5 where a sinusoidal shape is created in the range considered. Similarly to cooling water flow rate, chilling water flow rate optimum values depend on the other variables. In the instance of its interaction with cooling water flow rates, the optimum value of chilling water flow rate increases as the cooling water flow rate increases. This is exactly what was observed with the cooling water flow rate optimum points as discussed before.

Optimum chilling water flow rates is paramount not only because of system efficiencies but also in safe operation of the equipment i.e. ensuring only superheated fluid enters the compressor especially if the system lacks a liquid-vapor separator. Apart from the direct effect that water flow rates have on the water side heat transfer, chilling water flow rates also assist in ensuring proper distribution of the heat source in the heat exchanger. With increase in water mass flow rate to evaporator, both the heating capacity and compressor work increase modestly due to minor increase in the suction temperature (increase in degree of superheat) and also discharge temperature. Although, cooling capacity has higher increase because of the increase in its water side heat transfer coefficient due to both the flow rate and turbulence (Sarkar, 2005, Jiang et al., 2009).

4. CONCLUSION

The effects of six major variables on the COP of a water to water CO_2 heat pump were investigated in this study. Design of experiments was done using design-expert® 6 software for regression analysis. Central cubic design was used to optimize effective parameters while using the least number of experiments. Analysis of variance method was used to identify significant variables both linearly, in quadratic form and in interaction, and an empirical equation relating the COP to the variables was derived. All variables were significant but cooling water temperature had the highest effect on COP followed by chilling water temperature and then the refrigerant amount. The throttle valve opening follows after, and then the cooling water flow rates and finally the chilling water flow rates had the least significant effects. The effects of water temperatures on heat transfer were thought to be the main cause of their huge influence in COP. The COP value ranged from 1.545 to 6.914 but a maximum of 11.8 could be achieved if all the variables are put at their optimum values. Interactive effects of the variables are also observed in the study and their influence on the COP evidently exposed by use of response surface. It was theorized that these variables were affecting the system's COP by influencing related parameters like heat transfer, gas cooler pressure, gas cooler exit temperature and pressure losses.

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REFERENCES

ADRIANSYAH, W. 2001. Combined air conditioning and tap water heating plant using CO2 as refrigerant for Indonesian climate condition. Doctor of Engineering Doctorate, Norwegian University of Science and Technology.

AGRAWAL, N. & BHATTACHARYYA, S. 2008. Performance evaluation of a non-adiabatic capillary tube in a transcritical CO2 heat pump cycle. International Journal of Thermal Sciences, 47, 423-430.

AGRAWAL, N. & BHATTACHARYYA, S. 2009. Exergy assessment of an optimized capillary tube-based transcritical CO2 heat pump system. International Journal of Energy Research, 33, 1278-1289.

ATIK, K. & AKTAŞ, A. 2011. An experimental investigation of the effect of refrigerant charge level on an automotive air conditioning system. Journal of Thermal Science and Technology, 31, 11-17.

BENSAFI, A. & THONON, B. 2007. Transcritical R744 (CO2) heat pumps. Technician's Manual. Villeurbanne Cedex - France: Centre Technique Des Industries Aérauliques Et Thermiques.

BHATIA, A. RE: Selection Tips for Environmentally Safe Refrigerants. Type to CONTINUING EDUCATION AND DEVELOPMENT, I.

CALM, J. M. 2008. The next generation of refrigerants – Historical review, considerations, and outlook. International Journal of Refrigeration, 31, 1123-1133.

CALM, J. M. & DIDION, D. A. 1998. Trade-offs in refrigerant selections: past, present, and future. International Journal of Refrigeration, 21, 308-321.

CHEN, Y., LUNDQVIST, P. & WORKIE, A. B. Second Law Analysis of a Carbon Dioxide Transcritical Power System in Low-grade Heat Source Recovery.

CHO, H., RYU, C. & KIM, Y. 2007. Cooling performance of a variable speed CO2 cycle with an electronic expansion valve and internal heat exchanger. International Journal of Refrigeration, 30, 664-671.

CHO, H., RYU, C., KIM, Y. & KIM, H. Y. 2005. Effects of refrigerant charge amount on the performance of a transcritical CO2 heat pump. International Journal of Refrigeration, 28, 1266-1273.

CHRISTENSEN, Ø. 2009. Reversible R744 (CO2) heat pumps applied in public trains in Norway. Master of Science in Energy and Environment, Norwegian University of Science and Technology.

DIPIPPO, R. 2004. Second Law assessment of binary plants generating power from low-temperature geothermal fluids. Geothermics, 33, 565-586.

DUDDUMPUDI, V. S. 2010. Transcritical CO2 Air Source Heat Pump for Average UK Domestic Housing with High Temperature Hydronic Heat Distribution System. Master of Science, University of Strathclyde.

FERNANDEZ, N., HWANG, Y. & RADERMACHER, R. 2010. Comparison of CO2 heat pump water heater performance with baseline cycle and two high COP cycles. International Journal of Refrigeration, 33, 635-644.

FRONK, B. M. 2007. Modeling and Testing of Water-Coupled Microchannel Gas Coolers for Natural Refrigerant Heat Pumps. Master of Science, Georgia Institute of Technology.

FRONK, B. M. & GARIMELLA, S. 2011. Water-coupled carbon dioxide microchannel gas cooler for heat pump water heaters: Part II – Model development and validation. International Journal of Refrigeration, 34, 17-28.

GUPTA, D. K. & DASGUPTA, M. S. 2014. Simulation and performance optimization of finned tube gas cooler for trans-critical CO2 refrigeration system in Indian context. International Journal of Refrigeration, 38, 153-167.

HASHIMOTO, K. 2006. Technology and Market Development of CO2 Heat Pump Water Heaters (ECO CUTE) in Japan. Topical article. Japan: IEA Heat Pump Centre Newsletter.

HWANG, Y. & RADERMACHER, R. 1998. Theoretical Evaluation of Carbon Dioxide Refrigeration Cycle. HVAC&R Research, 4, 245-263.

JIANG, P.-X., ZHAO, C.-R., SHI, R.-F., CHEN, Y. & AMBROSINI, W. 2009. Experimental and numerical study of convection heat transfer of CO2 at super-critical pressures during cooling in small vertical tube. International Journal of Heat and Mass Transfer, 52, 4748-4756.

KIM, M.-H., PETTERSEN, J. & BULLARD, C. W. 2004. Fundamental process and system design issues in CO2 vapor compression systems. Progress in Energy and Combustion Science, 30, 119-174.

KIM, S. C., KIM, M. S., HWANG, I. C. & LIM, T. W. 2007. Heating performance enhancement of a CO2 heat pump system recovering stack exhaust thermal energy in fuel cell vehicles. International Journal of Refrigeration, 30, 1215-1226.

KIM, S. G., KIM, Y. J., LEE, G. & KIM, M. S. 2005. The performance of a transcritical CO2 cycle with an internal heat exchanger for hot water heating. International Journal of Refrigeration, 28, 1064-1072.

LORENTZEN, G. 1994. Revival of carbon dioxide as a refrigerant. International Journal of Refrigeration, 17, 292-301.

MOLINA, M. J. & ROWLAND, F. S. 1974. Stratospheric sink for chlorofluoromethanes: chlorine atomc-atalysed destruction of ozone. Nature, 249, 810-812.

NEKSÅ, P. 2002. CO2 heat pump systems. International Journal of Refrigeration, 25, 421-427.

NEKSÅ, P., REKSTAD, H., ZAKERI, G. R. & SCHIEFLOE, P. A. 1998. CO2-heat pump water heater: characteristics, system design and experimental results. International Journal of Refrigeration, 21, 172-179.

REULENS, W. 2009. Natural Refrigerant CO2. Diepenbeek: Katholieke Hogeschool Limburg.

RIEBERER, R. 1998. CO2 as working fluid for heat pumps. Doctor of Technology, Graz University of Technology.

SARKAR, J. 2005. Transcritical Carbon Dioxide Heat Pumps for Simultaneous Cooling and Heating. Doctor of Philosophy, Indian Institute of Technology.

SARKAR, J., BHATTACHARYYA, S. & GOPAL, M. R. 2004. Optimization of a transcritical CO2 heat pump cycle for simultaneous cooling and heating applications. International Journal of Refrigeration, 27, 830-838.

SARKAR, J., BHATTACHARYYA, S. & RAM GOPAL, M. 2007. Natural refrigerant-based subcritical and transcritical cycles for high temperature heating. International Journal of Refrigeration, 30, 3-10.

SARKAR, J., BHATTACHARYYA, S. & RAM GOPAL, M. 2009. Irreversibility minimization of heat exchangers for transcritical CO2 systems. International Journal of Thermal Sciences, 48, 146-153.

SARKAR, J., BHATTACHARYYA, S. & RAMGOPAL, M. 2010. Performance of a Transcritical CO2 Heat Pump for Simultaneous Water Cooling and Heating. International Journal of Applied Science, Engineering and Technology, 6, 57-63.

SAWALHA, S. 2008. Carbon Dioxide in Supermarket Refrigeration. Doctoral, Royal Institute of Technology.

SKAUGEN, G. 2002. Investigation of transcritical CO2 vapour compression systems by simulation and laboratory experiments. Doctor of Engineering Doctorate, Norwegian University of Science and Technology.

STAT-EASE INC. 2002. Design-Expert 6.0.6 ed. Minneapolis, Minessota: Stat-Ease Inc.

STENE, J. 2004. Residential CO2 Heat Pump System for Combined Space Heating and Hot Water Heating. Doctoral Degree, Norwegian University of Science and Technology.

STENE, J. 2007. Integrated CO2 Heat Pump Systems for Space Heating and Hot Water Heating in Low-Energy Houses and Passive Houses. International Energy Agency (IEA) Heat Pump Programme – Annex 32 – Workshop. Kyoto, Japan: International Energy Agency.

UNITED NATIONS 1987. Montreal Protocol on Substances that Deplete the Ozone Layer Montreal United Nations.

UNITED NATIONS 1998. Kyoto protocol to the united nations framework Convention on climate change. Kyoto.

UNITED NATIONS ENVIRONMENT PROGRAMME 2007. UNEP 2006 Report Of The Technology and Economic Assessment Panel. In: KUIJPERS, L. (ed.) Montreal Protocol On Substances that Deplete the Ozone Layer. Nairobi: UNEP

WANG, J., SUN, Z., DAI, Y. & MA, S. 2010. Parametric optimization design for supercritical CO2 power cycle using genetic algorithm and artificial neural network. Applied Energy, 87, 1317-1324.

WANG, S. K. 2000. Handbook of Air Conditioning and Refrigeration, New York, McGraw-HiII.