Performance Evaluation of a Hybrid Air Conditioning System Based on Transcritical CO2 Cycle

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ABSTRACT

A hybrid air conditioning system that uses CO_2 (R-744) as refrigerant is proposed for summer air conditioning applications. In the proposed system the cooling capacity of the transcritical CO_2 cycle is utilized for sensible cooling of process air while the heating capacity is used for regeneration of the desiccant wheel, which handles the latent heat load on the building. Three configurations (C1, C2 and C3) based on the location of desiccant wheel are analyzed using suitable thermodynamic and psychrometric equations. Results obtained for warm and humid climate show that among the three configurations, C1 yields the best performance in terms of the compressor power consumption while C2 yields the best performance in terms of utilization of gas cooler heating capacity. The performance of the hybrid system with different configurations is compared with that of a conventional air conditioning system that uses R-1234yf as refrigerant. It is observed that the compressor power required for the hybrid system is much less than that of R-1234yf system under high latent load conditions. Also, the compressor power variation is marginal under all room sensible heat factors for the hybrid system, whereas it increases steeply in a non-linear manner for the R-1234yf system as the room sensible heat factor decreases. In addition, the evaporator temperature is above 0ºC for the hybrid system under all load conditions, whereas it goes below 0ºC for the R-1234yf system at high latent loads. Results show that the environment friendly CO_2 based air conditioning systems can be made competitive in terms of energy efficiency by using the proposed hybrid cycles.

Keywords: *transcritical CO_2, hybrid cycle, sensible heat factor, desiccant wheel*

1. INTRODUCTION

In view of the growing global restrictions on the use of high global warming potential (GWP) refrigerants for various refrigeration and air conditioning applications, there is intense activity in the area of low GWP, natural refrigerants. Though several environment friendly refrigerants have emerged in the past two decades as replacements for the high GWP synthetic refrigerants, finding a suitable refrigerant for air conditioning applications (stationary as well as transport) is turning out to be a challenge. Propane (R-290) and carbon dioxide (R-744) are among the potential natural refrigerants for this application. However, as propane is flammable, there are some reservations about it finding wide acceptance globally. Carbon dioxide is a perfectly safe refrigerant. In addition, it has very low GWP and excellent thermophysical properties, it is available widely and is cheap. However, due to its low critical point (≈ 31ºC), the CO_2 system operates in transcritical mode for high ambient conditions. Transcritical operation offers the possibility of obtaining useful heating at relatively high temperatures in addition to refrigeration. Thus, CO_2 based air conditioning systems are very attractive when both cooling and heating effects can be utilized. However, a major drawback of the basic transcritical CO_2 based refrigeration systems is higher power consumption in comparison to the standard subcritical systems. Various modifications have been done to make CO_2 based air conditioning systems efficient. Barta et al. (2021) have reviewed the research conducted on CO_2 cycles for transportation and stationary refrigeration and air conditioning applications. They discussed many common cycle modifications and the broad array of applications. The common modifications discussed are multistage and parallel compression, compressor cooling, recovery of expansion work, subcooling, microchannel evaporators and economization (Flash tank). Based on the review, they concluded that there should be further improvements for the transcritical CO_2 cycle to be widely used by the automotive industry. Song et al. (2021) have investigated the cooling performance of a transcritical CO_2 air conditioning system for an electric bus using energy and exergy analyses. They have experimentally investigated the effects of the working conditions, such as the compressor speed, electronic expansion valve opening, inlet temperature and flow rate of indoor air on the energy performance. Their results show that with the increase of indoor air flow rate, exergy efficiency of the system improves and vice versa for increase in indoor and outdoor temperatures. The greatest percentage of exergy destruction occurs in the electronic expansion valve.

Yin et al. (2021) studied the effect of normalized charge on the temperature, pressure and performance of the transcritical CO_2 automotive air conditioning system. They proposed a novel standard to calculate the appropriate normalized charge and concluded that the optimal normalized charge range of the transcritical CO_2 air conditioning system should be set between 0.111 and 0.321. Hazarika et al. (2018) have experimentally investigated the effect of refrigerant charge and some operating parameters on the performance of transcritical CO_2 summer air conditioning system.

Dia et al. (2020) analyzed a CO_2 heat pump air conditioning system with dedicated mechanical subcooling for yearly heating and cooling requirements of a residential building. D'Agaro et al. (2019) worked on utilization of heating and cooling capacity of CO_2 transcritical cycle to satisfy both refrigeration and air conditioning needs of a small size supermarket in Italy. They observed that in mild climatic conditions CO_2 transcritical plant can fully and effectively satisfy the needs of refrigeration and air conditioning at the supermarket.

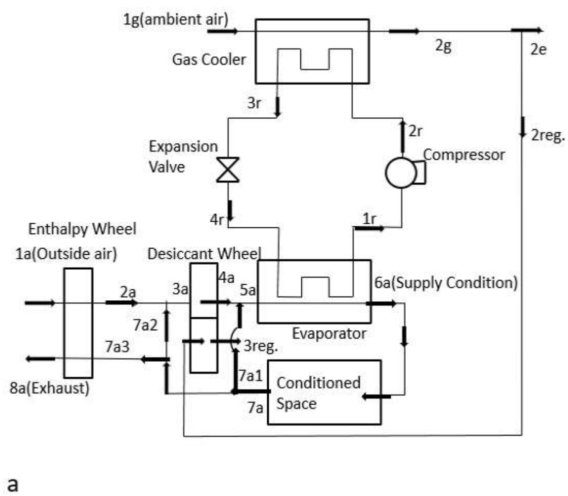
In case of air conditioning systems, it is well known that separation of sensible and latent loads is beneficial as the refrigeration system then can be operated at higher evaporator temperatures resulting in lower compressor power. Jin et al. (2019) studied a CO_2 air conditioning system in which the evaporator operates at two temperature levels to handle separately the sensible and latent loads of room. An ejector is used to produce the two different evaporator temperatures. Several studies are reported in literature on handling of latent load using desiccants. Li et al. (2021) have enumerated all possible airflow configurations of separate sensible and latent cooling (SSLC) systems. They have modeled simple vapour compression system to compare all the configurations, and found that there is only one unique basic airflow SSLC configuration with the series sensible and latent heat exchangers. Under standard conditions COP could be improved by 14.8%. Using desiccants for handling latent load requires regeneration energy. Liu (2020) proposed a novel solar hybrid rotary desiccant wheel air conditioning system, where solar energy is used to regenerate the desiccant.

In this work, a hybrid air conditioning system that uses a transcritical CO_2 based refrigeration cycle is proposed to meet the performance requirements that are on par or better than the systems based on conventional, synthetic refrigerants. The performance improvement is achieved by separating the sensible and latent loads and thereby operating the evaporator at a high temperature leading to low power consumption. In this system, the evaporator of the CO_2 refrigeration system handles only the sensible load of the room, while a desiccant wheel takes care of the latent load. The regeneration of the desiccant is achieved using the heat rejected by the gas cooler of the CO_2 cycle. An enthalpy wheel is used to reduce the ventilation load thereby further improving the overall performance. For the proposed hybrid cycle, three different configurations depending upon the location of the desiccant wheel are considered. In the first configuration (C1) desiccant wheel is placed before the evaporator. For 2nd configuration (C2) desiccant wheel is placed in the return air path after the conditioned space. Some amount of recirculated air is supplied to desiccant wheel and then mixed with process air stream. From the remaining air, some part is directly mixed with the process air stream before the cooling coil and the rest passes through the enthalpy wheel. In the third configuration (C3) desiccant wheel is located after evaporator, where only the required amount of cooled air from evaporator is supplied to desiccant wheel which satisfies the room latent load, and the remaining amount is mixed with the hot and dry air from desiccant wheel and supplied to the conditioned space. Performance of all these configurations is evaluated using suitable thermodynamic and psychrometric equations. The results are compared with a conventional R-1234yf based air conditioning system for different room sensible heat factors (RSHF).

2. SYSTEM DESCRIPTION

Figures 1, 2 and 3 represent the schematic diagram and psychrometric chart of C1, C2 and C3 configurations. The process undergone by air for all the configurations is represented in their corresponding psychrometric charts. From refrigerant side, all the configurations are composed of a simple, transcritical vapour compression system using CO_2 as refrigerant. The hot air from gas cooler is used for regeneration of desiccant wheel as represented by (2reg.-3reg.) process in all the configurations.

From air side, all the proposed hybrid air conditioning system configurations comprise of the same components, that is: an enthalpy wheel, cooling coil (evaporator) and a desiccant wheel. The working principle of C3 configurations is explained in detail. As shown in Figure 3, hot and humid ambient air (state 1a) for ventilation, passes through the enthalpy wheel and transfers heat and moisture to the exhaust air coming from the conditioned space (process 1a-2a for outside air and process 7a2-8a for exhaust air). Process air stream (state 2a) from the enthalpy wheel is then mixed with recirculated room air (state 7a1) from the conditioned space. The resulting process air at state 3a is supplied to the cooling coil (evaporator) where it is cooled sensibly. The sensible only cooling is achieved by operating the CO_2 evaporator at an appropriate temperature such that the cooling coil surface with which air comes in contact is above the dew point temperature of air at state 3a. The sensibly cooled air from the cooling coil is then split into two streams. One stream of air is supplied to the desiccant wheel at state 4a2, where it is dehumidified so that it can handle the latent load on the conditioned space. During dehumidification through desiccant wheel the process air temperature also increases (state 5a). The hot and dehumidified air from the desiccant wheel is then mixed with the remaining fraction of air (state 4a1). The resulting air at state 6a is then supplied to the conditioned space so that it can handle the room sensible and latent loads. A part of the return air from the conditioned space is re-circulated through the cooling coil after mixing with the outdoor air, while the remaining part is rejected to the ambient after passing through the enthalpy wheel. The working principle of configurations C1 and C2 can easily be understood from Figs. 1 and 2, respectively.



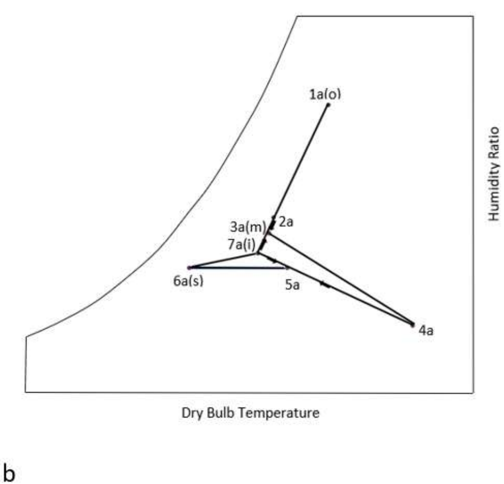


Figure 1: Schematic diagram and psychrometric chart for C1 configuration

3. MATHEMATICAL MODEL

The following simplifying assumptions are made for formulating the mathematical models of the systems described above:

1. All the processes are assumed to be steady state.
2. Refrigerant leaving the evaporator is assumed to be saturated.
3. Pressure losses in the heat exchangers and connecting pipelines are neglected.
4. All the system components are assumed to be adiabatic except the evaporator and gas cooler.
5. The power consumption by the desiccant and enthalpy wheel motors and fan motors is neglected.
6. The compression process is assumed to be reversible and adiabatic for both CO_2 and R-1234yf cycles.
7. For all the hybrid configurations, the coil temperature is assumed to be equal to the dew point temperature of the process air at the outlet of enthalpy wheel so that only sensible heat transfer takes place in evaporator.
8. A fixed minimum temperature difference of 5 K is considered for heat transfer between refrigerant and the cooling coil in evaporator and between refrigerant and temperature of air at outlet in gas cooler.

For the sake of brevity, mathematical model is given here for configuration C3 only.

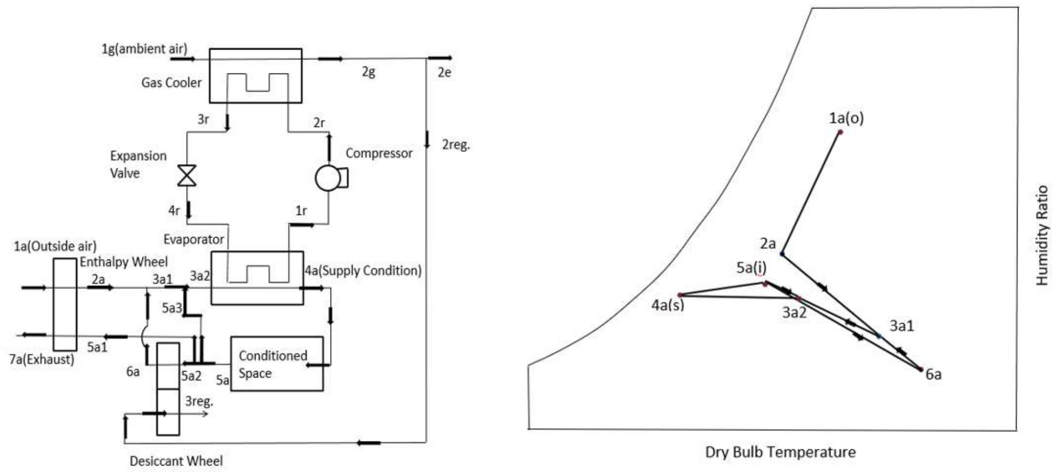


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Figure 2: Schematic diagram and psychrometric chart for C2 configuration

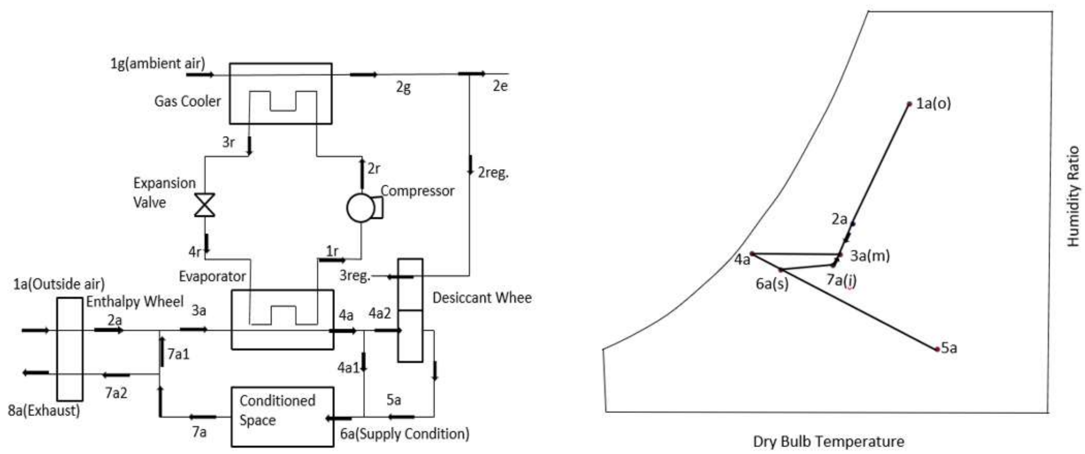

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Figure 3: Schematic diagram and Psychrometric chart of C3 configuration

For Configuration, C3:

Evaporator:

Air Side: The air side cooling capacity of the Evaporator is given by equation (1).

$$Q_e = m_s c_p (T_{3a} - T_{4a}) \tag{1}$$

Refrigerant Side: Refrigerant mass flow rate is given by:

$$m_r = Q_e / (h_{1r} - h_{4r}) \tag{2}$$

Gas Cooler:

Refrigerant Side: The heating capacity of the gas cooler is calculated by below Equation.

$$Q_{gc} = m_r (h_{2r} - h_{3r}) \tag{3}$$

Air Side: Amount of air supplied to the gas cooler is calculated by the Equation (4)

$$Q_{gc} = m_g (h_{2g} - h_{1g}) \tag{4}$$

Compressor:
Power supplied to the compressor is calculated by Equation (5).

$$W_c = m_r(h_{2r} - h_{1r}) \tag{5}$$

Expansion valve: Assuming isenthalpic expansion,

$$h_{3r} = h_{4r} \tag{6}$$

Enthalpy Wheel: The exit temperature and humidity ratio of process air are:

$$T_{2a} = T_o - \eta_{EW}(T_o - T_i) \tag{7}$$

$$w_{2a} = w_o - \eta_{EW}(w_o - w_i) \tag{8}$$

Desiccant Wheel:
The air temperature and humidity ratio at the outlet of desiccant wheel are calculated from the correlations proposed by Nia et al. (2006). They are given by Equations (9) and (10).

$$T_{5a} = g_1(N)g_2(T_{4a})g_3(d_t)g_4(T_{reg})g_5(w_{4a})g_6(D_h)g_7(U) \tag{9}$$

$$g_1(N) = -0.0002N^2 + 0.0112N + 0.4201$$

$$g_2(T_{4a}) = -0.0001\,T_{4a}{}^2 + 0.0275\,T_{4a} + 0.7993$$

$$g_3(d_t) = -18.79d_t{}^2 + 7.92d_t + 1.75$$

$$g_4(T_{reg}) = -0.0004T_{reg}{}^2 + 0.1255T_{reg} + 0.6757$$

$$g_5(w_{4a}) = 594.48w_{4a}{}^2 + 26.76w_{4a} + 3.79$$

$$g_6(D_h) = -0.039D_h{}^3 + 0.026D_h{}^2 + 0.603D_h + 0.0912$$

$$g_7(U) = -0.060U + 0.7973$$

The outlet humidity ratio of the desiccant wheel is calculated using desiccant wheel effectiveness:

$$\eta_{DW} = {(w_{4a} - w_{5a})}\Big/{w_{4a}} \tag{10}$$

Where η_{DW} is the effectiveness of the desiccant wheel given by (Nia et al., 2006):

$$\eta_{DW} = f_1(N)f_2(T_{4a})f_3(d_t)f_4(T_{reg})f_5(w_{4a})f_6(D_h)f_7(U) \tag{11}$$

$$f_1(N) = -0.0001N^2 + 0.0042N + 0.4474$$

$$f_2(T_{4a}) = -0.0001\,T_{4a}{}^2 - 0.003\,T_{4a} + 0.8353$$

$$f_3(d_t) = -21.67d_t{}^2 + 6.93d_t + 1.34$$

$$f_4(T_{reg}) = -0.0001T_{reg}{}^2 + 0.0355T_{reg} - 0.4924$$

$$f_5(w_{4a}) = 592.77w_{4a}{}^2 + 41.23w_{4a} + 1.283$$

$$f_6(D_h) = -0.0572D_h{}^3 + 0.0933D_h{}^2 + 0.6139D_h - 0.0922$$

$$f_7(U) = -0.0611U + 0.8376$$

The thermal energy required for regeneration of the desiccant wheel is provided by the gas cooler for all the configurations. The regeneration temperature is calculated using Equation (12) and the energy required for regeneration is calculated by Equation (13).

$$T_{2reg} = T_{2r} - 5 \tag{12}$$

$$Q_{reg} = m_{reg}\,(h_{1reg} - h_{2reg}) \tag{13}$$

The performance of the system is specified in terms of COP and overall COP. COP of the system is given by Equation (14) and the overall COP of the system is given by Equation (15).

$$COP = \frac{Q_e}{W_c} \tag{14}$$

$$COP_{overall} = \frac{Q_e + Q_{reg.}}{W_c} \tag{15}$$

4. RESULTS AND DISCUSSION

Results of the CO_2 based hybrid air conditioning system with different configurations are obtained at the optimum gas cooler pressure at which the COP is maximum. For comparison, a conventional air conditioning system that uses R-1234yf as refrigerant is considered as reference. The reference system does not have enthalpy and desiccant wheels. Table 1 shows the numerical values of the input parameters used for generating the results.

Table 1: Numerical values of input parameters

Input Parameter	value	Unit
Outside dry bulb temperature	33	°C
Outside wet bulb temperature	27.6	°C
Room dry bulb temperature	26	°C
Room relative humidity	50%	-
Room Total Load	5	kW
Volume flow rate of outdoor air	0.05	m^3/s
Hydraulic diameter of channel, (D_h)	2.33	mm
Thickness of desiccant coating (d_t)	0.2	mm
Velocity (U)	3	m/s
Wheel Speed (N)	19	RPH

In the comparative analysis when the Room Sensible Heat Factor (RSHF) is varied, the value of enthalpy wheel effectiveness is taken as 0.75, and when the enthalpy wheel effectiveness is varied, RSHF value is assumed to be equal to 0.75. The RSHF is the ratio of the sensible load on the conditioned space to the total load on the conditioned space.

4.1 Performance comparison between hybrid and conventional air conditioning systems:

Figure 4 shows the effect of RSHF on compressor power consumption and COP. Since the refrigerant side temperatures remain constant, COP of all the hybrid air conditioning systems is equal and constant for all RSHF values. However, as evaporator temperature increases with RSHF, COP of the conventional system increases with RSHF and is greater than the COP of the hybrid systems for all RSHF values. Comparing the performance of the hybrid systems with the conventional system on the basis of COP could be misleading. For the same room total load (5 kW), due to higher ventilation load, the load on the evaporator of conventional R-1234yf system is higher than the hybrid systems for all RSHF values. Since the compressor power consumption is directly proportional to cooling load on evaporator, it can be higher or lower than that of the hybrid cycle depending upon the COP value. As the evaporator temperature of the conventional system, C decreases as RSHF decreases, the COP of the conventional system decreases and compressor power input increases as shown. The power consumption of the conventional R-

1234yf system increases by around 45% as the RSHF decreases from 0.8 to 0.63. Below a RSHF value of 0.63, the coil temperature of the conventional system drops below 0ºC, hence results are not shown below this RSHF value. Due to much higher COP, the power consumption of the conventional system is lower than the power consumption of the hybrid systems when the RSHF value is above 0.66. For lower RSHF values (higher room latent loads), hybrid system performs better with lower compressor power consumption. For lower RSHF values, the compressor power of the conventional system increases non-linearly and sharply, implying that the conventional systems are not good for high latent load applications. Among the three hybrid configurations considered in the present study, the configuration C1 is seen to offer the best performance in terms of compressor power followed by configuration C2 and C3 as shown in the figure. It is observed that for all the three configurations, the compressor power decreases only marginally with RSHF unlike the conventional system. Thus, the hybrid systems do not require complicated and expensive compressor controls even when the RSHF value changes widely. However, the conventional systems need suitable compressor control due to highly non-linear variation of the power input with RSHF as shown in the figure.

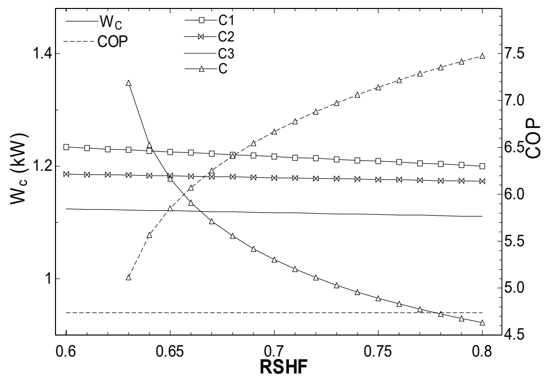

Figure 4: Variation of compressor power input with RSHF

Results show that among the three hybrid configurations, the required supply mass flow rate i.e., air flow rate to conditioned space is largest for C3 (Figure 5). However, supply air mass flow rate of conventional air conditioning system is lowest as compared to the hybrid system and it increases with RSHF as shown in Figure 5. The supply mass flow rate of air increases with RSHF for the conventional system as the temperature difference between air and refrigerant decreases with RSHF.

As shown in Figure 6, for all the three hybrid configurations, the cooling coil temperature is constant and is independent of RSHF as it is assumed to be equal to the dew point temperature of the process air at the exit of enthalpy wheel. However, as the cooling coil of the conventional system has to handle both sensible and latent loads, its temperature drops steeply as RSHF decreases as shown. Since the refrigerant temperature in the evaporator is assumed to be 5 K lower than the coil temperature, the evaporator temperature remains constant for hybrid systems whereas it decreases as RSHF decreases for the conventional system. As a result of this, the pressure ratio of the conventional system increases as RSHF decreases, whereas it is constant and much lower for the hybrid system. Lower pressure ratio is highly beneficial as it leads to compact and efficient compression process.

Figure 7 depicts that the ratio of regeneration mass flow rate to total gas cooler mass flow rate decreases with RSHF. This is as expected since increase in RSHF means less latent load and less mass of water adsorbed by the desiccant wheel. The results indicate that the proposed system is obviously beneficial when the latent loads are high as both cooling and heating capacities of the CO_2 refrigeration system are then properly utilized. Lower the regeneration air flow rate ratio means lower is the utilization of the heating capacity. Among the three hybrid configurations, the utilization of gas cooler heating capacity is highest for C2 configuration and lowest for C3 except at high RSHF.

4.2 Variation of hybrid system performance with enthalpy wheel effectiveness

As shown in Figure 8, as enthalpy wheel effectiveness increases, the required supply air mass flow rate to conditioned space decreases for all the configurations for an RSHF value of 0.75. Among the three configurations, the supply air flow rate is highest for C3 configuration, while it is almost same for the other two configurations. Lower supply air flow rate is advantageous as it leads to more compact ducting and/or lower fan power, hence higher enthalpy wheel effectiveness is obviously desirable.

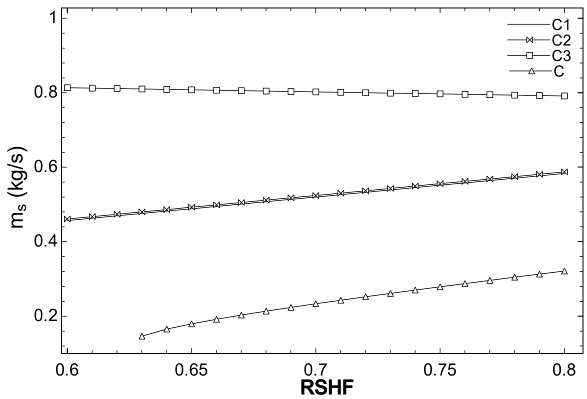

Figure 5: Variation of mass flow rate of supply air to conditioned space with RSHF

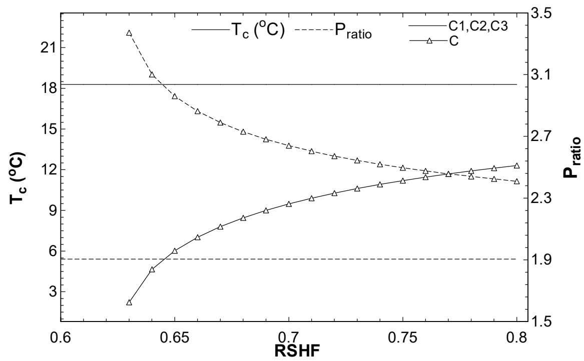

Figure 6: Variation of cooling coil temperature and compressor pressure ratio with RSHF

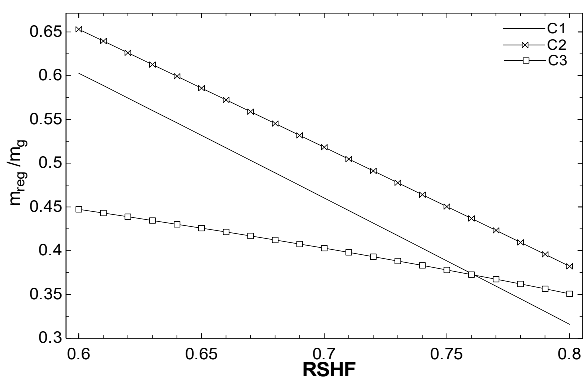

Figure 7: Variation of ratio of regeneration air mass flow rate to total gas cooler mass flow rate with RSHF

With the decrease in supply mass flow rate, the cooling coil and evaporator temperatures decrease and refrigerant temperature at gas cooler inlet temperature increases. Increased gas cooler inlet temperature in turn increases the regeneration temperature and desiccant wheel effectiveness as shown in Figure 8.

The overall COP of all the three configurations is seen to decrease with enthalpy wheel effectiveness as shown in Figure 9. Among the three configurations, because of the better utilization of gas cooler heating capacity, C2 has the highest overall COP as compared to the other two configurations. For low enthalpy effectiveness value, C3 configuration overall COP is more than C1 configuration as shown in Figure 9. Thus, from overall performance point of view, C2 configuration is the best. Figure 9 also shows the variation of cooling load on evaporator with enthalpy wheel effectiveness. As expected, it decreases with the increase in enthalpy wheel effectiveness.

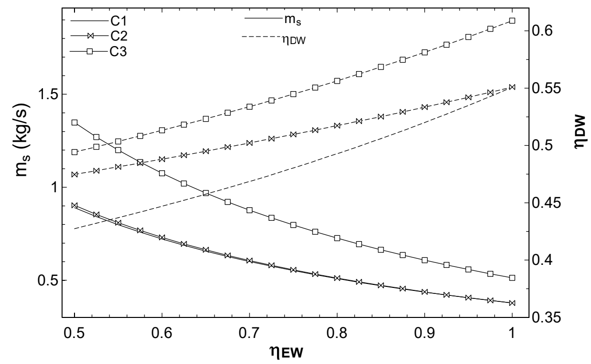

Figure 8: Variation of supply air mass flow rate and desiccant wheel effectiveness with enthalpy wheel effectiveness

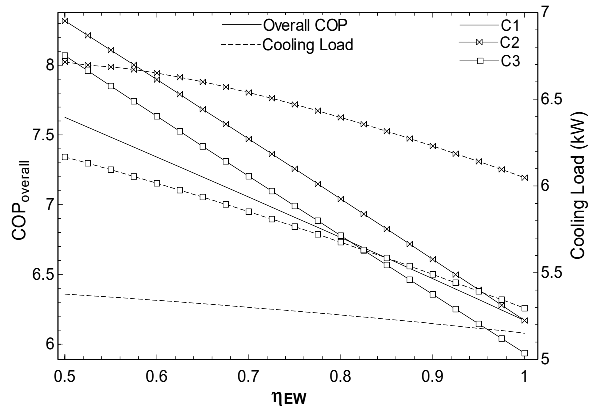

Figure 9: Variation of overall COP and cooling coil load with enthalpy wheel effectiveness

6. CONCLUSIONS

From the present analysis it is observed that the CO_2 based hybrid air conditioning systems perform better compared to the conventional R-1234yf based air conditioning systems for high latent load applications (low RSHF values). Also, the compressor power variation with RSHF is marginal for the proposed hybrid systems whereas it is significant for the conventional system. Due to sub-zero coil temperatures, the conventional system without desiccant wheel cannot operate for low RSHF values, whereas there is no such problem for the hybrid systems. Thus hybrid air conditioning systems are efficient option for warm and humid climates with high ventilation and room latent loads. Among the three hybrid configurations studied, on the basis of compressor power consumption, C1 configuration is the best while on the basis of overall COP, C2 configuration is the best. Since the power consumption of the hybrid systems is higher than conventional system at low latent loads (high RSHF), further improvements, for example, use of ejector in place of expansion valve etc., are needed for the CO_2 based air conditioning systems. It is expected that with these improvements, future air conditioning systems can be made economical using the environment friendly and safe CO_2 refrigerant.

NOMENCLATURE

COP	coefficient of performance	
d_t	thickness of desiccant coating	(mm)
D_h	diameter of desiccant wheel channel	(mm)
h	enthalpy	(kW/kg)
m	mass flow rate	(kg/s)
N	desiccant rotor speed	(RPH)
P	pressure	(kPa)
Q	cooling/heating capacity	(kW)

RSHF	room sensible heat factor	
T	temperature	(°C)
U	velocity	(m/s)
W	compressor power	(kW)
w	humidity ratio	(kg/kg)
Subscripts		
a	air conditioning	
c	compressor, cooling coil	
DW	desiccant Wheel	
EW	Enthalpy Wheel	
e	evaporator	
g/gc	gas cooler	
i	inlet	
m	mixing	
o	outlet	
r	refrigeration	
reg	regeneration	
s	supply condition	
η	Effectiveness	

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