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Design Integration of Dedicated Outdoor Air System with Variable Refrigerant Flow System

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ABSTRACT

Increase in energy consumption by Heating Ventilation Air Conditioning, HVAC sector and improvement in Indoor Air Quality, IAQ, according to ASHRAE standards has led to need of replacement of popular Variable Air Volume, VAV system with sustainable and cost effective systems. Variable Refrigerant Flow, VRF systems have emerged as energy efficient HVAC system in recent years, due to application of inverter driven variable speed compressors in its outdoor units, which offer good part load efficiency. In order to meet IAQ, Dedicated Outdoor Air Systems, DOAS have emerged as efficient systems where more Outdoor Air, OA, can be introduced in the conditioned space while limiting the load and energy required, especially for warm and humid climates. This paper investigates different DOAS configurations that can be integrated with VRF system to achieve thermal comfort and minimize life cycle cost. Various non-compressor based DOAS configurations in literature have been discussed. Cooling and Liquid Desiccant Dehumidification, LDD, of OA by Indirect Evaporative Cooling, IEC, is identified as a suitable option for integration with conventional air-conditioning systems. Psychrometric analysis of the configuration selected for integration with VRF system, for cities from each climate zone of India, is presented. Theoretical analysis for heat and mass transfer has been carried out based on patented nine enhanced passage aluminium extrusion for deployment in Mumbai. Overall heat transfer coefficient on outdoor air side is calculated as 13.89 W/m^2K. Mass transfer coefficient on outdoor and indoor air side is obtained as 0.03 m/s and 0.04 m/s respectively. Assumed outdoor air conditions are 31.0 DBT and 80% rh. Water evaporated on indoor air side and water condensed in outdoor air side is obtained as 0.34 g/s and 0.37 g/s respectively. LMTD for the heat transfer process is obtained as 4.2°C for a counter flow heat and mass exchange. Energy and cost saving calculations suggest savings of 2,71 INR/day from heat recovery and 38.6% increase in COP of parallel VRF system.

1. INTRODUCTION

In countries like India with a tropical climate, the Heating Ventilation and Air Conditioning, HVAC, systems in the commercial and building sector are responsible for major part of energy consumption. About 30 to 70% of electrical power consumed in various facilities is for HVAC. After the introduction of ASHRAE Standard 62.1 ventilation requirements became strict in order to maintain proper Indoor Air Quality, IAQ. It was realized that Variable Air Volume, VAV system, which is the popular air-conditioning technique, was not appropriate in the changed context. Treating Outdoor Air, OA, typically consumes 20 to 40% of energy required for air-conditioning.

Handling sensible and latent loads separately was one of the energy efficient options while maintain good IAQ. Dedicated Outdoor Air Systems, DOAS, are increasingly being considered in air-conditioning systems to improve IAQ and reduce energy consumption. Outdoor Air, OA, is treated separately, and is not a part of the air conditioning system. Unlike in VAV, before being introduced in the air conditioned space while limiting the increase in energy cost. This improves IAQ, as more OA can be introduced in the conditioned space while limiting the load and energy required. OA is cooled and dehumidified such that it is capable of handling all or part of the space latent load and some portion of sensible load.

A separate air conditioning equipment handles the remaining space sensible/latent load. There can be various choices for this air conditioning system like Variable Refrigerant Flow, VRF systems, Radiant Panels, RP, Chilled Beams, Fan Coil Units and Packaged Unitary Equipment. VRF systems have emerged as energy efficient HVAC system in recent years, due to application of inverter driven variable speed compressors in it is outdoor units, which offer good part load efficiency. VRF systems are typically installed in industrial spaces, rooms and laboratories. Indoor units of VRF system either do not have or have a limited provision for supply of OA. To meet indoor air quality requirement, VRF integrated with DOAS ventilation system is recommended.

Appropriate combination of DOAS and VRF system can be superior in terms of thermal comfort and life cycle cost. For any choice of sensible cooling equipment, knowledge of proper integration strategies of DOAS with these systems is essential for energy efficient operation.

The main objective of this paper is to study and compare different non-compressor based DOAS configurations that can be integrated with VRF system, identify suitable DOAS configurations and analyze results.

2. DOAS CONFIGURATIONS FOR DIFFERENT SITES

Traditionally compressor based DOAS were used in parallel to the cooling equipment, however due to high electricity consumption, complexity and low COP, they may not be regarded as sustainable or environmentally friendly. One of non-compressor based DOAS technologies that is being investigated is Direct Evaporative Cooling, DEC which has been used by people over centuries in one form or another for thermal comfort purposes. Wu *et al*. (2009) theoretically investigated heat and mass transfer in DEC areas. Heidarinejad *et al*. (2008, 2009) showed that DEC offers several advantages over mechanical vapor compressions systems. Some of these advantages include low cost, OA supply and less power consumption. However, the process of cooling the air by converting the sensible heat to the latent heat through water evaporation results in the primary air temperature decrease but in turn increases the humidity of the air and creates the discomfort to the occupants. This restricts DEC to only hot and dry climates where low latent internal load is required. This shortcoming can however be overcome by using Indirect Evaporative Cooling technology in which air can be cooled without humidifying. The primary air is cooled by the secondary air that is evaporatively cooled using the air to air heat exchangers, such as plate-type, tube-type and heat pipe type between the primary and secondary air (Jiang and Xie, 2010).

Rane and Chavan (2014a, 2014b) experimentally tested a novel modular configuration of DOAS with evaporative precooling of Indoor Air, IA, using rotating contacting device and plastic heat exchanger. Experimental results show that the effectiveness of plastic heat exchanger is 82.6% for counter-flow mode of heat exchange between evaporatively pre-cooled IA and OA. Overall heat transfer coefficient reported was 18.0 W/m^2K. Low pressure drop in the range of 40 Pa and 18 Pa was reported, on exhaust air and supply air sides respectively. Leading to low parasitic power, 27 W. Volume of plastic heat exchanger per 1000 cfm of air handling is 12.9 ft^3. It requires 64% of volume of plastic heat exchangers listed in literature survey. The pressure drop in this heat exchanger is also low, which is 39% of the pressure drops, reported in literature. The wet bulb effectiveness for IEC of OA is 101.5%.

Rane and Chavan (2014), have presented the process of adiabatic cooling followed by continued Diabatic Heating and Humidification, DHH of EA in patented channel profile to cool the OA. Thus, a significant enhancement can be achieved using the benefits of evaporation without humidifying the OA. The system proposed uses aluminium extrusion channels with small grooves which makes it compact, rigid and effective due to simultaneous heat and mass transfer. Novel crests and troughs in patented Al extrusions increase the heat transfer area by a factor of about $\pi/2$. The cross section area remains constant to keep the velocity constant.

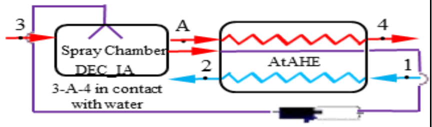
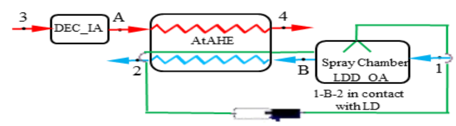
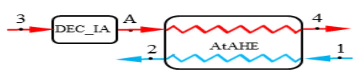
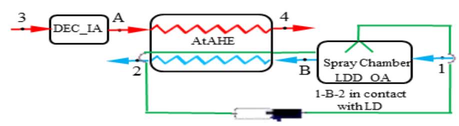
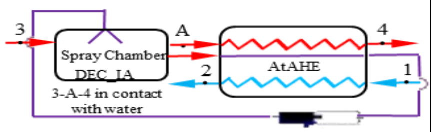
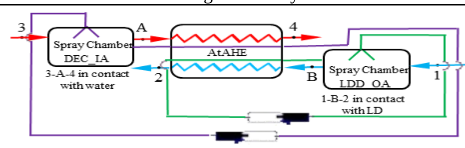
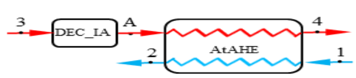
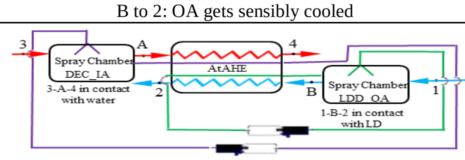
Compressor based dehumidification is energy intensive because process air has to be cooled below its DPT and sometimes reheated to ensure comfortable supply of indoor air. Dehumidification through desiccates (liquid or solid) can be a potential alternative to these dehumidification systems. As a rule desiccant systems are generally better choice if DPT is below 4^{o}C (Mumma, 2001). Desiccant systems are energy efficient because they allow direct expansion systems to work at higher evaporator temperature. Liquid desiccant system can be adiabatic or diabetic. Internally cooled diabetic liquid desiccant system is better than adiabatic system. Liquid desiccant can be regenerated by low grade heat sources. A potential non-compressor DOAS could be the combination of IEC with Liquid Desiccant Dehumidification, LDD.

In this paper, we will be studying four DOAS configurations as listed below.

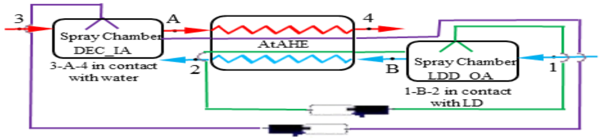
1. Indirect Evaporative Cooling, IEC of OA where IA undergoes Direct Evaporative Cooling, DEC and Air to Air Heat Exchange, AtAHE with OA
2. IEC of OA where IA undergoes DEC and diabatic humidification due to concurrent water flow and heat exchange with OA
3. Liquid Desiccant Dehumidification, LDD and IEC of OA where IA undergoes DEC and AtAHE with OA
4. LDD and IEC of OA where IA undergoes DEC and diabatic humidification due to concurrent water flow and heat exchange with OA

DOAS configurations are site specific, that is supply air conditions like dry bulb temperature, dew point temperature and supply airflow rate affect the performance of DOAS. Based on outdoor climate, India can be divided into six climatic zones based on climate that is predominant (more than six months) throughout the year. The following table enlists the climatic zones and best possible DOAS configuration for different seasons.

Table 1: DOAS Configurations for Different Indian Climate Zones for each Season

Sr #	**Climate Zone**	**Summer** Schematic and State Point Processes	**Monsoon** Schematic and State Point Processes
1.	Temperate Climate t = 25 to 30, rh < 75	 3 to A: IA undergoes evaporative cooling A to 4: IA gets diabatically heated and humidified 1 to 2: OA gets sensibly cooled	 3 to A: IA undergoes evaporative cooling A to 4: IA gets sensibly heated 1 to B: OA gets undergoes liquid desiccant dehumidification B to 2: OA gets sensibly cooled
2	Hot and Dry Climate t > 30, rh < 55	 3 to A: IA undergoes evaporative cooling A to 4: IA gets sensibly heated 1 to 2: OA gets sensibly cooled	 3 to A: IA undergoes evaporative cooling A to 4: IA gets sensibly heated 1 to B: OA gets undergoes liquid desiccant dehumidification B to 2: OA gets sensibly cooled
3	Warm and Humid Climate t > 25 rh > 55	 3 to A: IA undergoes evaporative cooling A to 4: IA gets diabatically heated and humidified 1 to 2: OA gets sensibly cooled	 3 to A: IA undergoes evaporative cooling A to 4: IA gets diabatically heated and humidified 1 to B: OA gets undergoes liquid desiccant dehumidification B to 2: OA gets sensibly cooled
4	Composite Climate when six months or more do not all fall within any of the other categories	 3 to A: IA undergoes evaporative cooling A to 4: IA gets sensibly heated 1 to 2: OA gets sensibly cooled	 3 to A: IA undergoes evaporative cooling A to 4: IA gets diabatically heated and humidified 1 to B: OA gets undergoes liquid desiccant dehumidification B to 2: OA gets sensibly cooled

Assumptions: $\varepsilon_{humidifier}$, ε_{he} **80%, temperatures 1% reliability:** $t_{1.db}$ **31.0°C,** $t_{1.wb}$ **27.4°C, volf 0.047 m³/s**



Processes

1-2	Cooling and dehumidification of outdoor fresh air
3-A	Evaporative precooling of indoor air
A-4	Heating and humidification of indoor exhaust air

State Points

1	Outdoor air
B	Dehumidified outdoor air
2	Dehumidified and sensibly cooled OA
3	Indoor Air
A	Evaporatively precooled indoor air
4	Evaporatively cooled air after diabatic heating and humidification

Heat Interactions

$\mathbf{Q_{12}}$ $= mf_{oa}\,(h_1 - h_2) = \mathbf{1.25\ kW} =$ 0.055 kg/s x (86.9 kJ/kg$_{da}$ – 64.2 kJ/kg$_{da}$)

$\mathbf{Q_{A4}}$ $= mf_{oa}\,(h_4 - h_A) = \mathbf{1.25\ kW} =$ 0.055 kg/s x (78.3 kJ/kg$_{da}$ – 55.6 kJ/kg$_{da}$)

Specific Parameters	Units	oa 1	2	3	ia A	4
$t_{.db}$	°C	**31.0**	**25.9**	**27.0**	21.0	27.3
$t_{.wb}$	°C	**27.4**	21.9	**19.5**	19.5	25.4
$t_{.dp}$	°C	26.3	20.2	15.7	18.7	24.8
ϕ, rh	%	**76.0**	71.1	**50.0**	86.9	86.9
W	g/kg$_{da}$	21.8	15.0	11.2	13.6	20.0
h	kJ/kg	**86.9**	**64.2**	**55.6**	55.6	78.3
mf	g/s	82			82	
	kg/min	4.9			4.9	
	kg/h	295			295	
volf	m³/s	0.07			0.07	
	cfm	150			150	

Water Consumed IEC	DHH	Total	Units
0.10	0.40	0.50	g/s
0.01	0.02	0.03	kg/min
0.48	1.26	1.74	kg/h

Water Consumption

$\mathbf{mf_{wc}}$ $= mf_{ia}\,(W_4 - W_3) =$ **0..5 g/s**

= 0.055 kg/s (20.0 - 11.2) g/kg$_{da}$ = 0.029 kg/min = 1.74 kg/h

$\mathbf{mf_{we.dhh}}$ $= mf_{ia}\,(W_4 - W_A) =$ **0.4 g/s**

= 0.055 kg/s (20.0 - 13.6) g/kg$_{da}$ = 0.021 kg/min = 1.26 kg/h

$\mathbf{mf_{we.iec}}$ $= mf_{ia}\,(W_A - W_3) =$ **0.1 g/s**

= 0.055 kg/s (13.6 - 11.2) g/kg$_{da}$= 0.008 kg/min = 0.48 kg/h

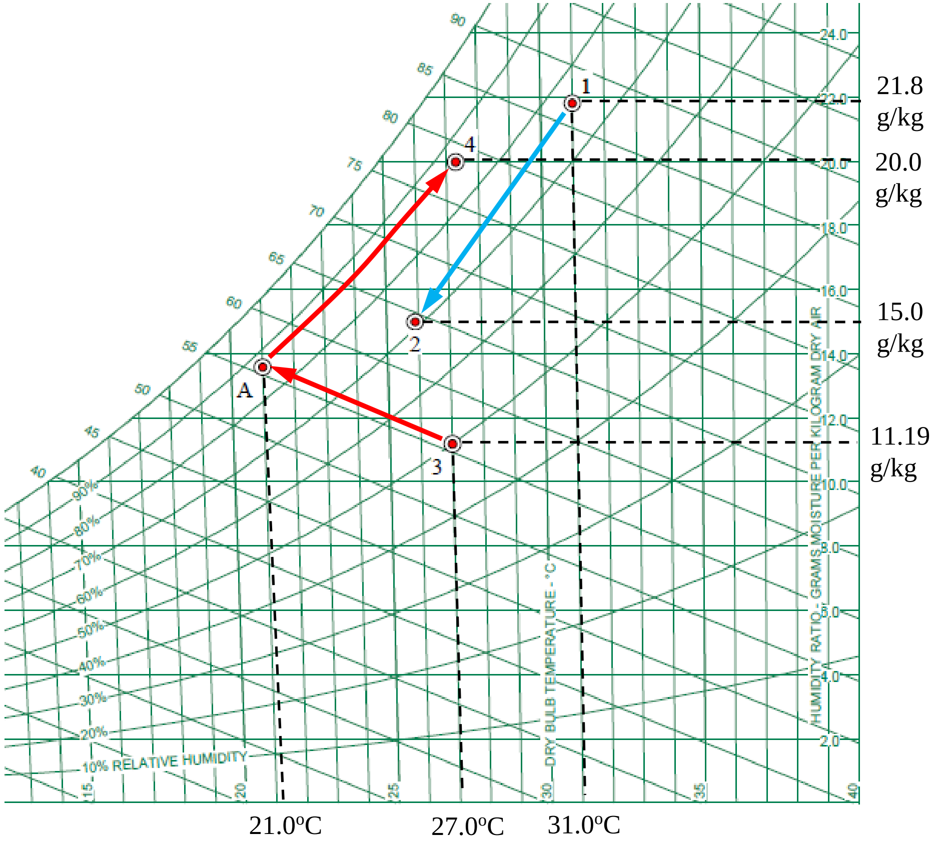


Figure 1: Pschychrometric Analysis of DOAS Configuration in Mumbai

3. THEORETICAL ANALYSIS OF SELECTED CONFIGURATION

Based study of different configurations which can be integrated with VRF systems and psychrometric analysis on Page 4, , LDD followed by IEC of OA is identified as best possible DOAS option for warm and humid climates. Here during Monsoon LDD + IEC of OA can be used and in summer only IEC component can be used. For low internal heat load, a direct evaporative cooling option can be added at end of process. In this section the theoretical analysis for heat and mass transfer has been carried out based on patented nine enhanced passage aluminium extrusion (Rane & Chavan, 2014) to investigate possibility of using it for experiment in Mumbai.

Following assumptions were made for carrying out the theoretical analysis.

a. Air flow rate is assumed as 100 cubic feet per minute.
b. Properties of air is taken at 30oC for inlet air and 25oC for indoor air

Total internal surface area as shown in Equation (1)

$$A_{.t.s.e.i} = n_{row}\, A_{.s.e.i}\, l_{ext} = 12 \text{ x } 6022 \text{ cm}^2\text{/m x } 1.22 \text{ m} = 8.82 \text{ m}^2 \quad (1)$$

Total external surface area as shown in Equation (2)

$$A_{.t.s.e.o} = n_{row}\, A_{.s.e.o}\, l_{ext} = 12 \text{ x } 4949.8 \text{ cm}^2\text{/m x } 1.22 \text{ m} = 7.24 \text{ m}^2 \quad (2)$$

Total internal cross section area as shown in Equation (3)

$$A_{.t.cs.e.i} = n_{row}\, A_{.cs.e.i} = 12 \text{ x } 764 \text{ mm}^2 = 0.0092 \text{ m}^2 \quad (3)$$

Length of patented Al extrusion is 1.22 m average surface area, A_{avg} is 8.03 m^2. Taking k_{Al} as 237 W/mK.

Overall heat transfer coefficient is calculated by Equation (4)

$$\frac{1}{U_i\ A_{t.s.e.i}} = \frac{1}{h_{oa}\ A_{t.s.e.i}} + \frac{1}{h_{ia}\ A_{t.s.e.o}} + \frac{l_{ext}}{k_{al}\ A_{avg}} \quad (4)$$

$$U_i = 11.42 \text{ W/m}^2\text{.K}$$

Overall heat transfer coefficient is 11.42 W/m^{2}K referring to internal surface area

The salient feature of patented 9 channel enhanced passage Al extrusion is that the hydraulic diameter of the passage and extrusion is 5.08mm and 6mm which gives lesser pressure drops with increased heat transfer and thereby less costs.

Diffusion coefficient of moist indoor air as show in Equation (5)

$$D_{v.ia} = \frac{0.926}{P}\left(\frac{T^{2.5}}{T+245}\right) \quad (5)$$

$$= \frac{0.926}{101.325 \text{ kPa}}\left(\frac{298 \text{ K}^{2.5}}{298 \text{ K} + 245}\right) = 2.58 \text{ x } 10^{-5} \text{ m}^2\text{/s}$$

Schmidt number as show in Equation (6)

$$Sc_{ia} = \frac{\mu_{ia}}{\rho_{ia}\ D_{v.ia}} \tag{6}$$

$$= \frac{18.85 \text{ x } 10^{-6}\ \text{N.s/m}^2}{1.185\ \text{kg/m}^3 \text{ x } 2.58 \text{ x } 10^{-5}\ \text{m}^2/\text{s}} = 0.613$$

For laminar flow Sherwood number is equal to Nusselt number (Hui, 2007)

Convective mass transfer coefficient is calculated by Equation (7)

$$h_{m.o} = \frac{Sh_{ia}\ D_{v.ia}}{D_{h.i}} \tag{7}$$

$$h_{m.o} = \frac{8.23 \text{ x } 2.58 \text{ x } 10^{-5}\ \text{m}^2/\text{s}}{6.06 \text{ x } 10^{-3}\ \text{m}} = 0.035 \text{ m/s}$$

The log mean temperature difference for the counter heat and mass flow exchanger is calculated to be 4.27°C. Water evaporated on indoor air side and water absorbed in outdoor air side is obtained as 0.34 g/s and 0.37 g/s respectively.

Referring to Figure 1 psychrometric properties of inlet air is 31.0°C DBT and 27.4°C WBT.

At this condition

Enthalpy of outdoor air at inlet $h_{oa.1}$ = 86.9 kJ/kg

Enthalpy of outdoor air at outlet, $h_{oa.2}$ = 61.0 kJ/kg

Mass flow rate of air = 3240 m^3/h

Energy recovered by applying DOAS configuration as shown in Equation (8)

$$Q_{12} = m_{oa}\ (h_1 - h_2) \tag{8}$$

$$= \frac{3240}{3600}\ \text{m}^3/\text{s x } 1.164\ \text{kg/m}^3\ (86.8 - 61.0)\ \text{kJ/kg} = 27.14\ \text{kW}$$

Energy saved is 27.14 units

Assuming 8 hours a day of operation total energy saved in one day is

Q = 8 x 27.14 = 217.12 kW/d

Assuming cost of electricity as 10 INR/unit total operating cost savings is

Cost savings =10 x 271.4 = 2171.2 INR/d

VRF system installed at the site has capacity of 9 TR = 9 x 3.52 = 31.68 kW

Energy savings by employing DOAS configuration with respect to total energy consumption is

$$= \frac{27.14}{31.68 + 27.14} = 46.14\%$$

Since DOAS handles all the space latent load VRF systems operating in parallel to DOAS handle only sensible load. Maximum evaporator temperature due to constraint from compressor is 12^oC. Cost saving analysis given below is based on assumption that evaporator temperature is elevated to 12^oC from 7.2^oC which is standard operating condition. Refrigerant used in VRF system is R410A.

At 7.2^oC evaporator temperature

Enthalpy at inlet to compressor h_1 = 423.31 kJ/kg

At 54.4^oC condenser temperature (standard operating conditions)

Enthalpy at outlet of compressor, h_3 = 295.43 kJ/kg

For isenthalpic expansion $h_3 = h_4$ = 295.43 kJ/kg

From refrigerant chart for R410A, h_2 = 458 kJ/kg

Mass flow rate of refrigerant as shown in Equation (9)

$$m_r = \frac{Q_c}{(h_1 - h_4)} \tag{9}$$

$$= \frac{31.68 \text{ kW}}{(423.31 - 295.43) \text{ kJ/kg}} = 0.248 \text{ kg/s}$$

Work done by the compressor and COP as shown in Equations (10) and (11)

$$W_{comp} = m_r \ (h_2 - h_1) \tag{10}$$

$$W_{comp} = 0.248 \text{ kg/s } (458.0 - 423.3) \text{ kJ/kg} = 8.6 \text{ kW}$$

$$COP = \frac{Q_c}{W_{comp}} \tag{11}$$

$$= \frac{31.68 \text{ kW}}{8.6 \text{ kW}} = 3.68$$

At 12^oC evaporator temperature

Enthalpy at inlet to compressor h_1 = 4234.7 kJ/kg

At 54.4^oC condenser temperature (standard operating conditions)

Enthalpy at outlet of compressor, h_3 = 295.43 kJ/kg

For isenthalpic expansion $h_3 = h_4$ = 295.43 kJ/kg

From refrigerant chart for R410A, h_2 = 450.0 kJ/kg

Mass flow rate of refrigerant as shown in Equation (12)

$$m_r = \frac{Q_c}{(h_1 - h_4)} \tag{12}$$

$$= \frac{31.68 \text{ kW}}{(424.7 - 295.4) \text{ kJ/kg}} = 0.245 \text{ kg/s}$$

Work done by the compressor and COP as shown in Equations (13) and (14)

$$W_{comp} = m_r \; (h_2 - h_1) \tag{13}$$

$$W_{comp} = 0.245 \text{ kg/s } (450.0 - 424.7) \text{ kJ/kg} = 6.2 \text{ kW}$$

$$COP = \frac{Q_c}{W_{comp}} = \frac{31.68 \text{ kW}}{6.2 \text{ kW}} = 5.11 \tag{14}$$

Percentage reduction in compressor work due to elevation of evaporator temperature as shown below

$$= \frac{8.6 - 6.2}{8.6} = 27.9\%$$

Percentage increase in COP due to elevation of evaporator temperature as shown below

$$= \frac{5.11 - 3.68}{3.68} = 38.9\%$$

5. CONCLUSION

Dedicated outdoor air systems, DOAS have evolved as a potential option for meeting IAQ standards while minimizing energy consumption. VRF systems have emerged as energy efficient HVAC equipment in recent years, because of inverter driven variable speed compressors in their outdoor units, which offer good part load efficiency. Indoor units of VRF system either do not have or have a limited provision for supply of fresh air. To meet indoor air quality requirement, VRF integrated with DOAS is recommended which enables increase in OA in system.

Few DOAS configurations in literature have been discussed and best configuration for each climate zone of India have been presented as DOAS is site specific. A novel configuration, with less number of components is proposed and investigated for possible integration with existing air-conditioning systems. The configuration is cooling and LDD of fresh OA by IEC of IA.

Psychrometric analysis of select configuration has been presented for different cities and theoretical analysis for heat and mass transfer has been carried out based on patented nine enhanced passage aluminium extrusion to investigate possibility of using it for experiment in Mumbai. Overall heat transfer coefficient on outdoor air side is calculated as 13.89 W/m2·K. Mass transfer coefficient on outdoor and indoor air side is obtained as 0.03 m/s and 0.04 m/s respectively. Water evaporated on indoor air side and water absorbed in outdoor air side is obtained as 0.34 g/s and 0.37 g/s respectively. LMTD for the heat transfer process is obtained as 4.2^{o}C. Energy and cost saving calculations for a case study suggest savings of 2171 INR/day from heat recovery and 38.6% increase in COP of parallel VRF system.

After analysis complete experimentation for configuration is recommended. Liquid desiccant has to be regenerated and cooled before being supplied again for dehumidification of outdoor air. A regeneration and cooling system for liquid desiccant should be designed before proceeding with experiment. Regeneration of liquid desiccant through condenser heat recovery can be considered. Using solution heat exchangers for cooling strong liquid desiccant solution by exchanging heat with incoming weak desiccant solution is one of the energy efficient method of cooling desiccant after regeneration.

NOMENCLATURE

A	area	(m^2)
DEC	Direct Evaporative Cooling	
DHH	Diabatic Heating and Humidification	
DOAS	Dedicated Outdoor Air System	
HVAC	Heating Ventilation and Air Conditioning	
IA	Indoor Air	
IAQ	Indoor Air Quality	
IEC	Indirect Evaporative Cooling	
k	thermal conductivity	(W/mK)
l	length	(m)
LDD	Liquid Desiccant Dehumdification	
LMTD	Log mean temperature difference	
OA	Outdoor Air	
RP	Radiant Panel	
U	Overall heat transfer Coefficient	
VAV	Variable Air Volume systems	
VRF	Variable Refrigerant Flow system	

Subscript

avg	average
Al	Aluminium
e	extrusion
i	inside
m	mass
o	outside
s	surface
t	total

REFERENCES

Heidarinejad G, Bozorgmehr M, Delfani S and Esmaeelian J (2009). Experimental investigation of two-stage indirect/direct evaporative cooling system in various climatic conditions. *Building and Environment 44:* 2073 to 2079.

Heidarinejad G, & Bozorgmehr M (2008). Modeling, evaluation and application potential of two stage indirect/direct evaporative air coolers. *Proceedings of the 6th international energy conversion engineering conference (IECEC)*, Cleveland, Ohio

Jiang Y & Xie X. (2010). Theoretical and testing performance of innovative indirect evaporative chiller, *Solar Energy, 84* (2010), pp 2041-2055.

Mumma S. A. (2001). Dedicated OA systems, *IAQ applications/Winter 2001*.
http://doas-radiant.psu.edu/IAQ_Winter, pp 20-22

Rane M. V & Chavan S. A. (2014). Indirect Evaporative Precooling of Fresh Air Using Heat Exchangers with Enhanced Flow Passages. *International Conference on Energy and Indoor Environment for Hot Climate*. Doha, Qatar. 12602

Rane M V and Chavan S A, (2014). DOAS with Evaporative Precooling with Exhaust Air Using Rotating Contacting Device. *International Conference on Energy and Indoor Environment for Hot Climate*. Doha, Qatar. 12600

Wu J. M, Huang X & Zhang H. (2009). Theoretical analysis on heat and mass transfer in a direct evaporative cooler. Applied Thermal Engineering 29: 980 to 984.

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