Parametric study of condensation at heating, ventilation, and air-conditioning duct's external surface

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ABSTRACT

The compression of insulation causes around a heating, ventilation, and air-conditioning duct usually resulted in dew formation around the outer surfaces because of low temperature, which causes significant energy and financial losses. The parameters such as supply airflow rate, supply air temperature, ambient airspeed, and the convective heat transfer coefficient (ho) plays significant role in dew formation. In this paper, the parametric study is performed to investigate the effects of these parameters on the external surface temperature of the duct to avoid condensation. A mathematical model is developed to quantify these effects using preliminary data obtained from the heating, ventilation, and air-conditioning system of a pharmaceutical company. The results reveal that external surface temperature increases with an increase in insulation thickness and supply air temperature, whereas it decreases with higher supply air flow rate. It is estimated that the minimum insulation thickness at joint and bend should be maintained between 15–55 and 15–35 mm, respectively, with a variation in ho between 6 and 22W/m^{2}K to avoid condensation. Additionally, it is estimated that air flow rate should be greater than 1.4m^3/s at 10W/m^2 K and 2.2m^3/ s at 22W/m^2 K. Similarly, the ambient air speed should be greater than 2.8 m/s at 6W/m^2 K, respectively.

Practical application: Building services engineers have a paucity of information on the effects of the compression of heating, ventilation, and air-conditioning duct thermal insulation. It can cause condensation that will adversely affect the insulation material, thereby increasing the maintenance cost as well increasing the heat loss from the duct so affecting the conditions of supply air. Proper insulation thickness and operating parameters are important for building owners and operators to control ongoing expenses of buildings. This paper seeks to quantify the effect of insulation compression to improve understanding so that this important area may be properly considered by the building services engineer.

Keywords: insulation thickness, air flow rate, supply air temperature, air speed and condensation.

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Introduction

Energy consumption (EC) is escalating throughout the globe due to rapid population growth; improved living standard; and lifestyle, urbanization, migration toward large cities, and developed technologies. The excessive use of fossil fuels for various energy products has raised the concern worldwide on account of non-sustainability of the fuels, and higher environmental impact and costs of the products. During the last two decades, EC enhanced by 49%, whereas CO2 emission is enhanced by 43% at an average annual rate of 2 and 1.2%, respectively, in developed countries. The CO2 emission has raised the mean earth temperature by 2–3 ºC. The energy demand of most emerging economies such as Southeast Asia, Middle East, South America, and Africa is increasing at an annual rate of 3.2% that precedes the energy demand of the developed countries by 2020.[1] Therefore, energy conservation measures are required to reduce EC.[2] In fact, energy conservation is fruitful from both economic and environmental point of view for any country.[3]

In developed countries, buildings are responsible for 40% of total EC and heating, ventilation, and air-conditioning (HVAC) system consumes around 10–20%.1 The HVAC system is responsible for 60% of total EC in buildings.[4] Also, the EC in buildings contributes to around 40% of total CO2 emissions.5,6 However, EC in buildings can be reduced using insulation in particular one for HVAC system. Kumar et al.[5,6] estimated that around 16 and 14% of energy is lost in air distribution system of HVAC system due to compression of insulation considering duct as flat plate and circular. In addition, they investigated that greenhouse gases decrease from 4.2 to 2.3 kg/kW by increasing the insulation thickness from 10 to 40mm at selected points of the duct. Soponpongpipat et al.[7] reported that the energy savings obtained using double layer insulation (rubber and glass wool) in an air-conditioning increases as ho increases from 6 to 22W/m2K and wind speed increases from 0.2 to 7 m/s.[8]

The loss of energy in an HVAC system is significant due to the heat gain from the surrounding. One of the main problems of duct insulation is its compression at bends and joints (tip and bracing) as shown in Figure 1(left) that may cause condensation (Figure 1(right)). The condensation at the joint of the duct is avoided because it grows the fungal bio-contaminates which deteriorate thermal insulation and impairs its thermal performance. However, the duct is still susceptible to fungus since the outer aluminum foil expands to create voids due to compression. This may result in premature failure of insulation material and corrosion of galvanized iron sheet. Therefore, increasing the insulation thickness may decrease EC and environment degradation.[9] Holme[10] reported that condensation causes the risk of mold growth on outer surface of an insulated wall and it decreases as insulation thickness increases. In order to avoid condensation on building wall in a cold climate, the duct surface temperature of wall should be greater than dew point temperature of the ambient air.[11] A multiscale approach found that the EC of vertical open refrigerated display cabinet (VORDC) of HVAC system was reduced by 6.4% with a reduction of SAT from 19 to 16 C. Additionally, reduction in supply airspeed at diffuser from 3.5 to 2.0 m/s decreases EC of VORDC by 23.4%.[12] Another, study reveals that the performance of HVAC system can be improved by reducing the average air flow rates of condition air at the zone.[13]

This study investigates the effect of thermophysical parameters on condensation of external surfaces. The effects of insulation compression, air flow rate, SAT, ambient air speed, and different values of convective heat transfer coefficient on condensation are estimated. The preliminary data regarding operating and design parameters are obtained from HVAC system installed at a renowned pharmaceutical company GSK Pharma Pvt. Ltd, in Hyderabad, Pakistan. The problem of insulation deterioration caused by generation of fungal growth due to accumulation of water vapor at insulation compression is observed in installed HVAC system's air distribution system. Therefore, parametric study is performed to estimate the critical operating thermophysical parameters to avoid condensation at external surface of the duct considering constant ambient conditions. A relevant mathematical model was developed to investigate the effects of operating thermophysical parameters on condensation. The results obtained using the developed mathematical model were discussed. Finally, conclusions were drawn accordingly.

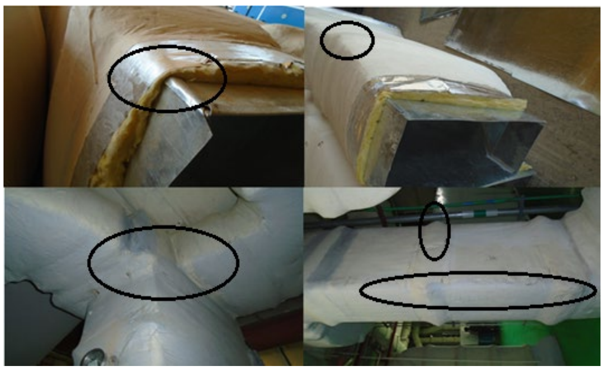


Figure1: Compression of duct insulation (left) and accumulation of water vapor at the point of compression (right).

Selected air distribution system

Figure 2 illustrates the different components of an air distribution system. The outdoor air is primarily filtered by louver. Then the outer door air is mixed with return air in mixing chamber. The mixture of fresh and return air is conditioned to zone parameters through an AHU. The fan then supplies this conditioned air to different zones. The damper controls the conditioned air flow and distributes it to different zones via the splitter. The fan exhausts return air proportional to fresh air and recirculates the remaining quantity air. This study considers the supply air duct (SAD) and its ducting layout is illustrated in Figure 2. The SAD has divided into five different parts named from A to E according to its crosssectional area. The design parameters for all the parts of SAD are given in Table 1, while Table 2 exhibits the values of thermal conductivity for duct, joints and insulation material.

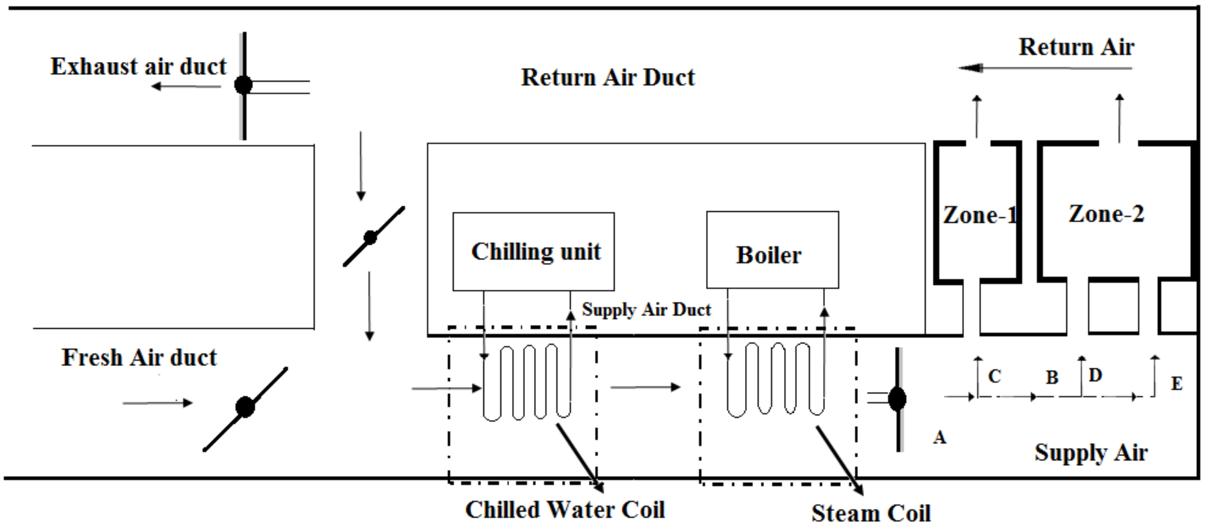


Figure 2: Layout of selected air distribution system.

Table 1: Design parameters of selected air distribution system.

Portions	Duct Size			Thickness					
	Width (W)	Height (H)	Length (L)	x#	x_{ins}	$x_{ins,c}$	bt	n#	b#
	m	m	m	mm	mm	mm	mm	No.	mm
SAD(A)	1.02	0.3	1.37	0.85	38	12	3	3	32
SAD(B)	0.91	0.3	0.3	0.85	38	12	3	2	32
SAD(C)	0.41	0.36	18.59	0.7	38	12	3	3	25
SAD(D)	0.41	0.3	12.02	0.7	38	12	3	3	25
SAD(E)	0.41	0.3	14.61	0.7	38	12	3	3	25

Note: H: height; W: width; L: length; x, b_t ,b#, $t_{gwc}$and t_{gw}: thickness of duct sheet, bracing, bracing tip, insulation and another part respectively.

Mathematical Modeling

Assumptions and operating parameters

For developing a mathematical model, following assumptions have been made:

1. The HVAC duct has rectangular cross section and it is assumed as circular using hydraulic diameter.
2. Air flow rate is uniform throughout the duct.
3. Kinetic and potential energy changes of conditioned air are negligible.
4. The conditioned air temperature at the inlet and exit of the duct is considered as 291 and 294K respectively.[5,6,14]
5. The temperature difference between conditioned air and ambient air is assumed constant for given portion.
6. The dry bulb (T_{db}) and wet bulb temperatures (T_{wb}) of air in the immediate surrounding is 307K and 301K.[5,6,15]
7. The ambient air speed varies from 1.9 to 7.7m/s at Jamshoro, Pakistan.[16]
8. The design pressure in the duct is 1kPa (gauge)

The supply air duct is considered in five parts as per cross-sectional areas given in Table 3.

Table 2: The properties of duct, Joint and insulation material.

	Material	Properties
Duct	Galvanized iron sheet	k_{gi}=18.18 W/m K
Joint	Mild Steel	k_{ms}=54 W/m K
Thermal insulation	Fiberglass wool	k_{ins}=0.037 W/m K

Table 3: Operating parameters of selected air distribution system.

Portion	Average Pressure	Average Temperature	Volume flow rate	Density	Cross sectional Area	Speed
Unit	kPa (g)	K	m^3/s	Kg/m^3	m^2	m/s
A	102.3	291	1.534	1.212	0.3097	4.953
B	102.2	291.2	1.038	1.214	0.2323	4.471
C	102	293.5	0.4954	1.207	0.1445	3.428
D	101.5	293.3	0.5192	1.204	0.1239	4.191
E	101.7	293.4	0.5192	1.200	0.1239	4.191

Moreover, several operating parameters are measured using pressure gauge, digital pyrometers, air flow meter and hygrometer. The average pressure and temperature of conditioned air inside the duct are given in Table 3. The variation in the average pressure in different portions of SAD is due to pressure drop in duct fittings and vertical drops whereas average temperature varies due to difference in heat gain. The volume flow rate is the highest in portion A of SAD because it is connected with AHU and distributes air to different zone.

Modeling Equations

The analysis is based on two basic laws, i.e., conservation of mass and conservation of energy, as defined below

$$\sum \dot{m}_{in} = \sum \dot{m}_{e} \tag{1}$$

$$\dot{Q}_{in} + \dot{W}_{in} + \sum_{in} \dot{m}\left(h + \frac{V^2}{2} + gz\right) = \dot{Q}_{exit} + \dot{W}_{exit} + \sum_{exit} \dot{m}\left(h + \frac{V^2}{2} + gz\right) \tag{2}$$

Energy Loss. The energy loss in selected air distribution system is due to conduction heat transfer takes place radially, and it is calculated as[18,19]

$$\dot{Q} = \frac{(T_o - T_a)}{R_T} \tag{3}$$

Where, R_T is the total thermal resistance offered by heat transfer coefficient of air inside and outside the duct, compressed and incompressed layer of insulation material, duct, and joint material and is calculated by Eq. (11). T_o and T_a is the temperature of ambient and supply air.

Thermal resistance of duct surface is calculated by

$$R_s = u\left(\frac{\ln\left(\frac{r_3}{r_2}\right)}{k_{gw}} + \frac{1}{h_o\ r_3}\right) \tag{4}$$

$$R_c = v\left(\frac{\ln\left(\frac{r_{3,c}}{r_2}\right)}{k_{gw}} + \frac{1}{h_o\ r_{3,c}}\right) \tag{5}$$

Where u and v represent the proportion of the uncompressed surface area to total surface area of duct without tip and bracing, and v represent the proportion of the compression surface area of the corner to total surface area of the duct both can be found from

$$u = \frac{2\,\pi\,(r_h + x\#)}{2\,\pi\,(r_h + x\# + L_{gwc})} \tag{6}$$

$$v = \frac{2\,\pi\,L_{gw,c}}{2\,\pi\,(r_h + x\# + L_{gwc})} \tag{7}$$

Total thermal resistance of duct surface except bracing and tip region is calculated as

$$R_s = \left(\frac{1}{a}\right)\left(\frac{1}{h_i\ r_1} + \frac{\ln\left(\frac{r_2}{r_1}\right)}{k_{gi}} + \frac{R_s\,R_c}{R_s + R_c}\right) \tag{8}$$

Thermal resistance at bracing of duct is calculated as

$$R_b = \left(\frac{1}{b}\right)\left(\frac{1}{h_i\ r_1} + \frac{\ln\left(\frac{r_2}{r_1}\right)}{k_{gi}} + \frac{\ln\left(\frac{r_{3,b}}{r_2}\right)}{k_{ms}} + \frac{\ln\left(\frac{r_{4,b}}{r_{3,b}}\right)}{k_{gw}} + \frac{1}{h_o\ r_{4,b}}\right) \tag{9}$$

Thermal resistance at the tip of M.S angle of duct is calculated as

$$R_t = \left(\frac{1}{t}\right)\left(\frac{1}{h_i\ r_1} + \frac{\ln\left(\frac{r_2}{r_1}\right)}{k_{gi}} + \frac{\ln\left(\frac{r_{3,t}}{r_2}\right)}{k_{ms}} + \frac{\ln\left(\frac{r_{4,t}}{r_{3,b}}\right)}{k_{gw}} + \frac{1}{h_o\ r_{4,t}}\right) \tag{10}$$

Total thermal resistance is calculated as

$$\frac{1}{R_T} = \frac{1}{R_s} + \frac{1}{R_b} + \frac{1}{R_t} \tag{11}$$

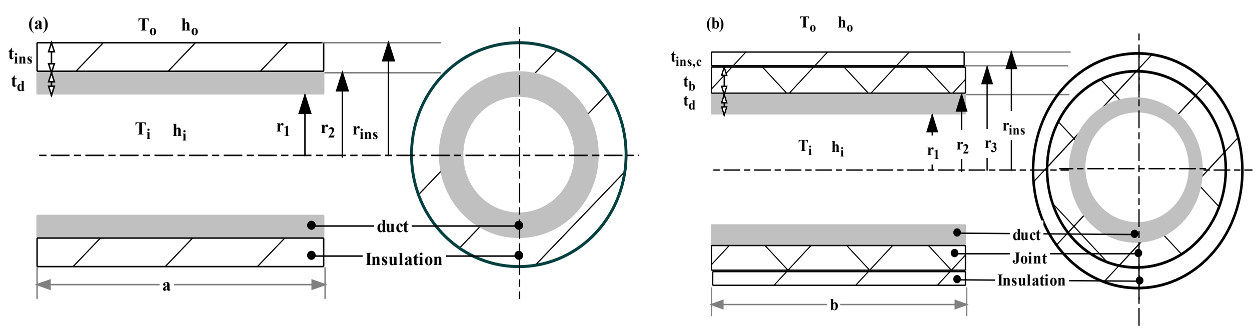


Figure 3: Compression of duct insulation (left) and accumulation of water vapor at the point of compression (right).

Figure 3 exhibits an insulated duct and it's joint. The layer of duct and insulation material is shown by gray and yellow color. In figure $\mathrm{a} = 2\,\pi(\mathrm{L}_1 - \mathrm{n\#}_1(\mathrm{b\#}_1 + \mathrm{b}_\mathrm{t}))$, $\mathrm{b} = 2\,\pi(\mathrm{n\#}_1\mathrm{b\#}_1)$ and $\mathrm{t} = 2\,\pi(\mathrm{n\#}_1\mathrm{b}_\mathrm{t})$ represent the unit length of duct without insulation compression and joint (bracing and tip). The length and radius of the interior and exterior duct are denoted by L, r_1 and r_2. The radius of joint's bracing and tip is $r_{3,b} = r_2 + b_t$ and $r_{3,t} = r_2 + t_{gwc}$. The radius of insulation at corner, bracing and tip of the insulated duct are $r_{3,c} = r_2 + t_{gwc}$, $r_{4,b} = r_{3,b} + b_t$ and $r_{4,t} = r_{3,t} + t_{gwc}$. The h_i [8] and h_o[19] are calculated as

$$h_i = \frac{0.023 \cdot Re^{0.8} \cdot Pr^{0.4} \cdot k_a}{D_h} \tag{12}$$

$$h_o = 11.58 \cdot \left(\frac{1}{D_h}\right)^{0.2} \cdot \left\{\left(\frac{1}{T_s + T_o}\right) - 546.3\right\}^{0.181} \cdot (T_s - T_o)^{0.266} \cdot (1 + 2.86V_o)^{0.5} \tag{13}$$

Where, Re is Reynolds number is calculated by using Eq. 6, Pr is a Prandtl number; k_a is thermal conductivity of supply air, $D_h = \frac{4 \cdot \mathrm{A}_c}{\mathrm{p}}$ is duct's hydraulic diameter, T_s is duct's surface temperature and T_o and V_o are ambient air temperature and velocity.

$$Re = \frac{V_{SA} \cdot D_h}{\vartheta_{SA}} \tag{14}$$

Where, V_{SA} and ϑ_{SA} represents the velocity and kinematic viscosity of supply air.

Surface temperature of the duct. The surface temperature of outer layer at selected points of the duct is calculated by keeping constant rate heat transfer from that point while total thermal resistance decreases because the thermal resistance of ambient air is eliminated. The surface temperature of duct mathematically can be calculated by

$$\mathrm{T_s} = \mathrm{T_a} + \dot{\mathrm{Q}}\,\mathrm{R_s} \tag{15}$$

Where, T_s represents the surface temperature at the different point of the duct, T_a represents the average temperature of the conditioned air and R_s represents the sum of the thermal resistance of the various layers except ambient air at selected points of the duct is calculated by

$$\mathrm{R_{sur,1}} = \mathrm{u}\left(\frac{\ln\left(\frac{\mathrm{r_3}}{\mathrm{r_2}}\right)}{\mathrm{k_{gw}}}\right) \tag{16}$$

$$R_{c,1} = v\left(\frac{\ln(\frac{r_{3,c}}{r_2})}{k_{gw}}\right) \tag{17}$$

The total thermal resistance of different layers of duct at surface region is calculated by

$$R_{s,1} = (\frac{1}{a,1})\left(\frac{1}{h_i\ r_1} + \frac{\ln(\frac{r_2}{r_1})}{k_{gi}} + \frac{R_{sur,1}R_{c,1}}{R_{sur,1} + R_{c,1}}\right) \tag{18}$$

Thermal resistance at bracing of duct is calculated from

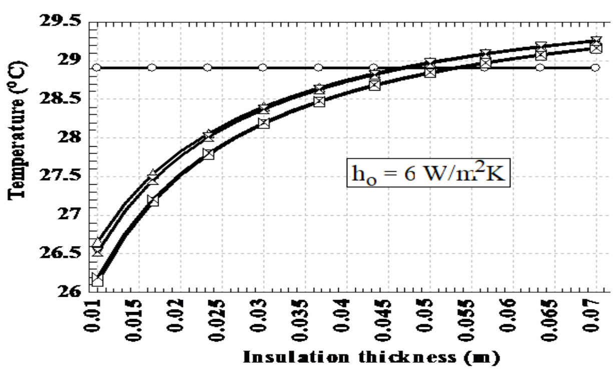
$$R_{b,1} = (\frac{1}{b,1})\left(\frac{1}{h_i\ r_1} + \frac{\ln(\frac{r_2}{r_1})}{k_{gi}} + \frac{\ln(\frac{r_{3,b}}{r_2})}{k_{ms}} + \frac{\ln(\frac{r_{4,b}}{r_{3,b}})}{k_{gw}}\right) \tag{19}$$

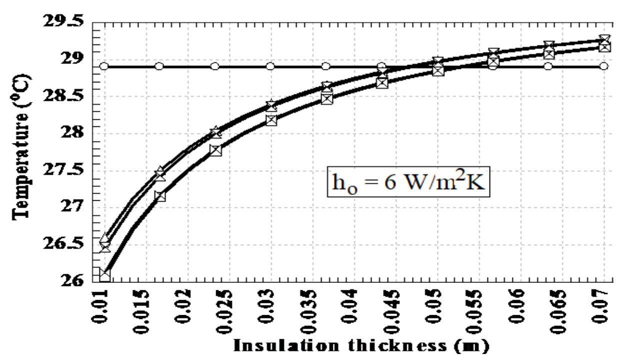
Thermal resistance at the tip of M.S angle of duct is calculated from

$$R_{t,1} = (\frac{1}{t})\left(\frac{1}{h_i\ r_1} + \frac{\ln(\frac{r_2}{r_1})}{k_{gi}} + \frac{\ln(\frac{r_{3,t}}{r_2})}{k_{ms}} + \frac{\ln(\frac{r_{4,t}}{r_{3,b}})}{k_{gw}}\right) \tag{20}$$

Results and discussion

Effect of insulation thickness on the surface temperature at the duct joints.





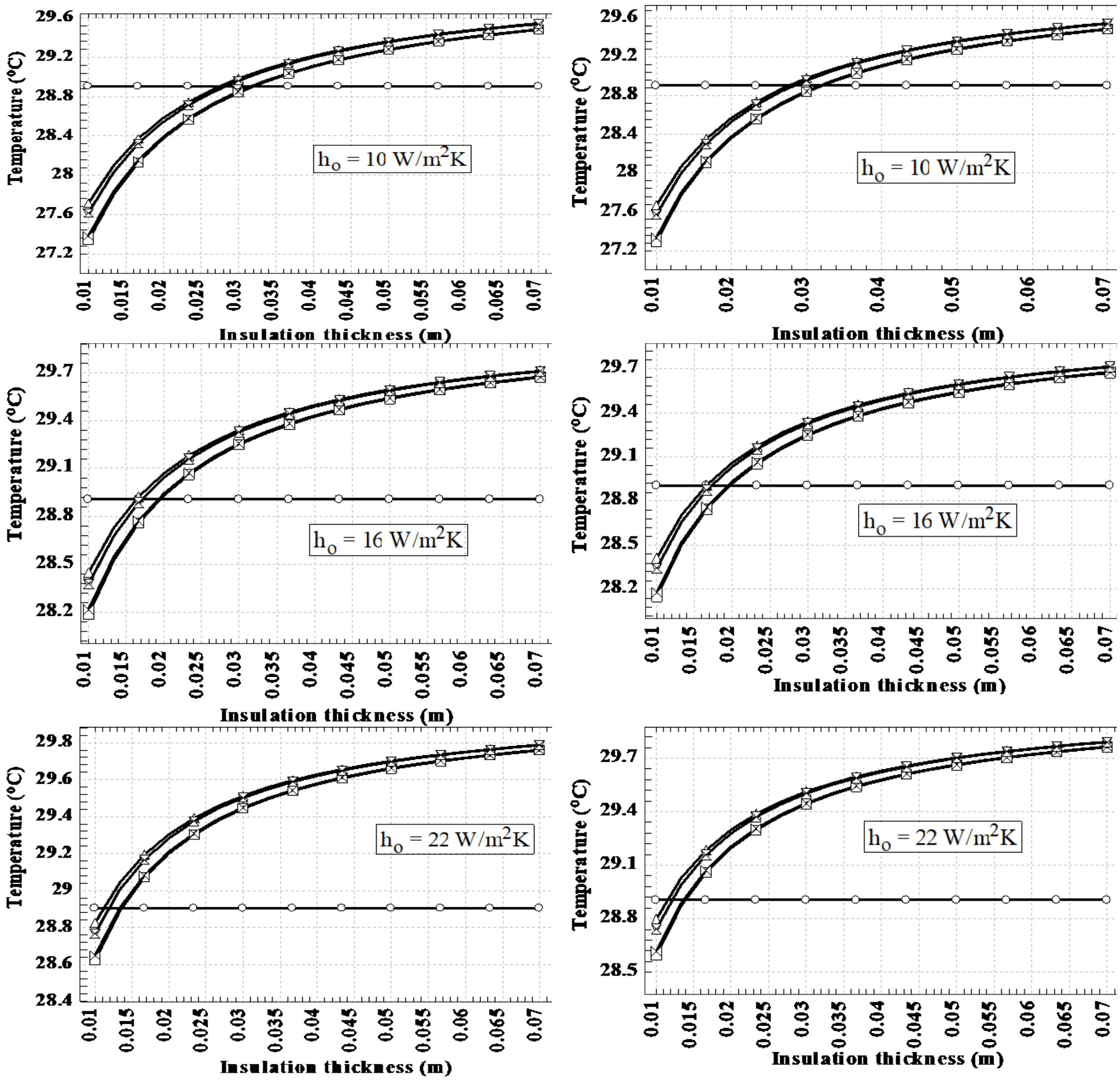


Figure 4: Effect of compression of insulation at joint's bracing (right) and joint's tip (left) on the surface temperature of duct.

The effects of insulation thickness on the duct surface temperature at joint's bracing and tip are illustrated in Figure 4 for different outside heat transfer coefficient. It is evident that the external surface temperature increases with an increase in insulation thickness for all heat transfer coefficients. To avoid deposition of condensate at the insulation, surface temperature must be higher than the dew point temperature of the atmospheric air, which requires corresponding rise in the insulation thickness if dew point temperature increases. The trends in variation of surface temperature for both bracing and tip are similar mainly due to similar thermal resistance offered by MS-angle at bracing and tip. Also, the surface temperature increases when insulation thickness increases during initial course of increments. The duct's surface temperature increases due to increment in the product of thermal resistance and heat gain. The surface temperature at the joint at A & B is lower than that at C and E of SAD because the SAT is lower in the former parts of the duct. As seen in Figure 4 that the difference in surface temperatures at different parts of SAD is low. Therefore, the value of surface temperature above the dew point temperature among different portion of the SAD will be chosen at different convective heat transfer coefficient to avoid condensation at the external surface of the duct. It is shown in Figure 4, the surface temperature at the joint of the duct is greater than dew point temperature at insulation thickness and convective heat transfer coefficient 55 mm and 6W/m^{2}K, 35mm and 10W/m^{2}K, 25mm and 16W/m^{2}K and 15mm and 22W/m^{2}K. The insulation thickness at which the surface temperature of the duct is greater than dew point temperature is known as critical insulation thickness. The results show that in order to avoid condensation at external surface of different parts of SAD the critical thickness should be above 55, 35, 25 and 15mm, respectively. This value probably will reduce the maintenance cost and assure energy saving.

Effect of insulation thickness on the surface temperature at the duct bends

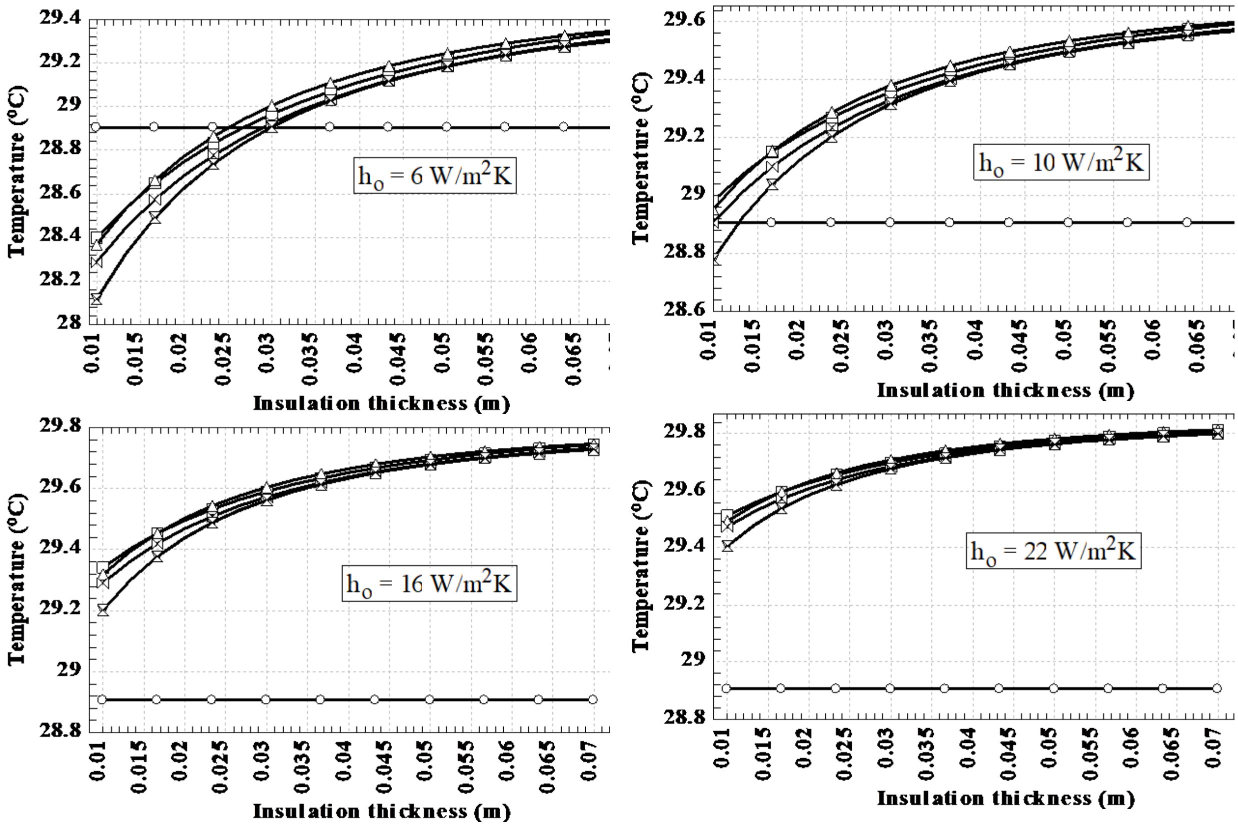


Figure 5: Effect of compression of thermal insulation on surface temperature at the bends of the duct under different convective heat transfer coefficient.

Figure 5 shows effect of insulation compression, at bends of the duct, with different convective heat transfer coefficients. The results show the variation in surface temperature as obtained for the joint of the duct (Figure 4). However, the chances of condensation at the bends are lower than that on the joints of the duct. As seen in Figure 5, the surface temperature at h_o=10W/m^{2}K is lower than dew point temperature; therefore, the chance of condensation at bends of the duct at a convective heat transfer coefficient below 10W/m^{2}K is higher. Moreover, the results reveal that critical insulation thickness at bends is above 35mm at 6W/m^{2}K, 20mm at 10W/m^{2}K to subside condensation and there is no condensation at 16-22W/m^{2}K.

Effect of air flow rate of conditioned air on surface temperature of the duct

The surface temperature is related with the temperature of air inside the duct, thermal resistance and heat gain. The air flow rate increases the heat gain while decreasing total thermal resistance.

The effect of air flow rate on the surface temperature is shown with different convective heat transfer co-efficient in Figure6.

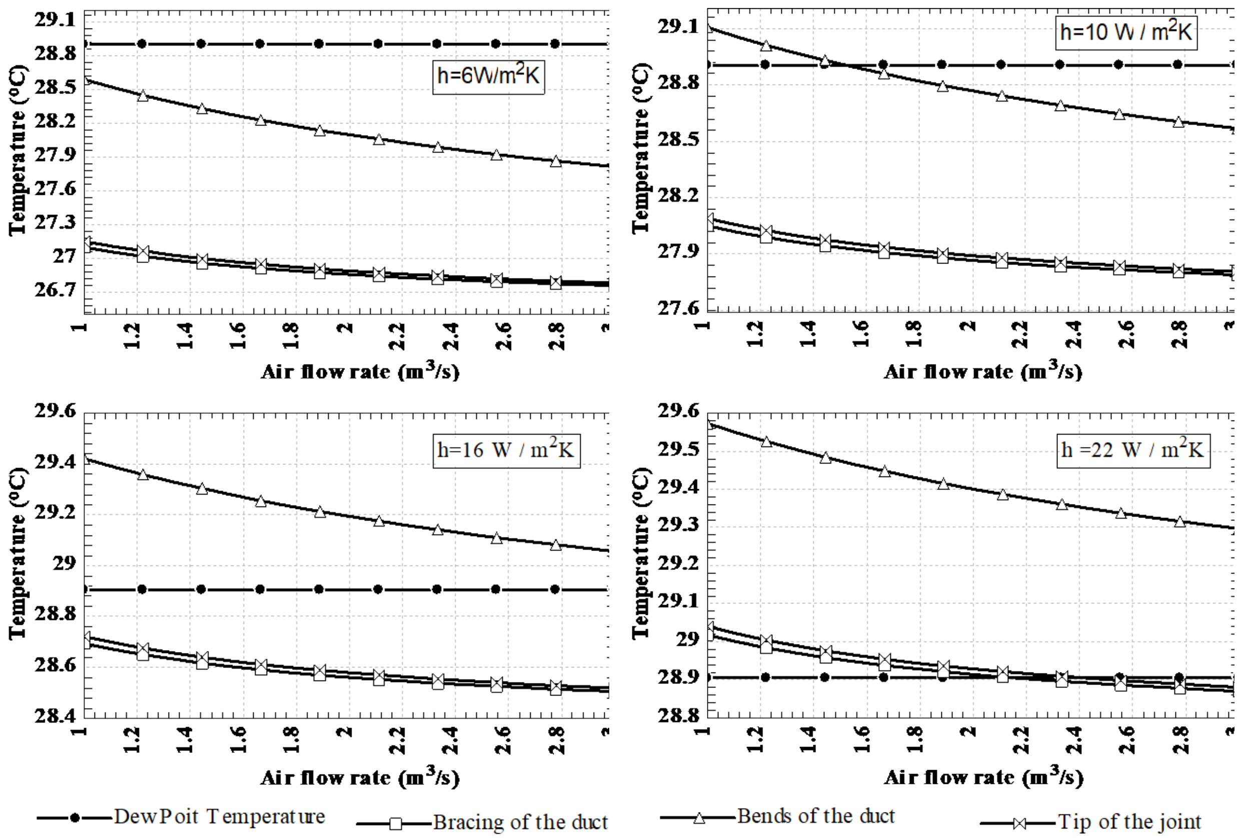


Figure 6: Effect of air flow rate on surface temperature with compression at different convective heat transfer coefficient.

Figure6 shows that the higher air flow rate decreases the surface temperature at joint and bends of the duct hence increases the chances of condensation. The surface temperature decreases linearly as air flow rate increases because convective heat transfer coefficient of conditioned air increases. It is clear from **Eq. (16)** that convective heat transfer coefficient is a function of Nusselt number and Nusselt number is a function of Reynolds number as given in Eq. 21.The air flow rate increases the speed and Reynolds number of conditioned air. Reynolds number increases the Nusselt number and convective heat transfer coefficient of air. Therefore, total thermal resistance decreases with air flow rate. Additionally, it increases the heat gain in the duct. The temperature at the exterior surface of the duct is a function of SAT, heat gain, thermal resistance, duct material, and convective heat transfer coefficient. Therefore, increment in heat gain increases the external surface temperature and the chance of condensation at joint and bends increases. As seen in Figure 5, the chance of condensation at joints (bracing & tip) is higher than that on bends of the duct with compression of insulation. The chances of condensation are also higher if convective heat transfer coefficient of ambient air is below 10W/m^{2}K with the lowest speed of supply air flow. The chances of condensation at the bends of the duct are subsided by keeping the air flow rate below 1.6-3.0 m^3/s at 10-16W/m^{2}K. The air flow rate with specified range does not cause condensation at convective heat transfer coefficient above 16W/m^{2}K. In contrast, the chances of condensation at joints (Tip and Bracing) are higher at 6-16W/m^{2}K but it can be avoided keeping air flow rate below 2.2m^3/s with 22W/m^{2}K. Finally, the air flow rate should be lower than 2.2m^3/s at 22W/m^{2}K in order to avoid condensation at joint whereas bends of the duct is insusceptible to the condensation. It is concluded that to avoid condensation the convective heat transfer coefficient of ambient air should be greater than 16W/m^{2}K and air flow rate should be greater than 1.4m^3/s with compression of thermal insulation.

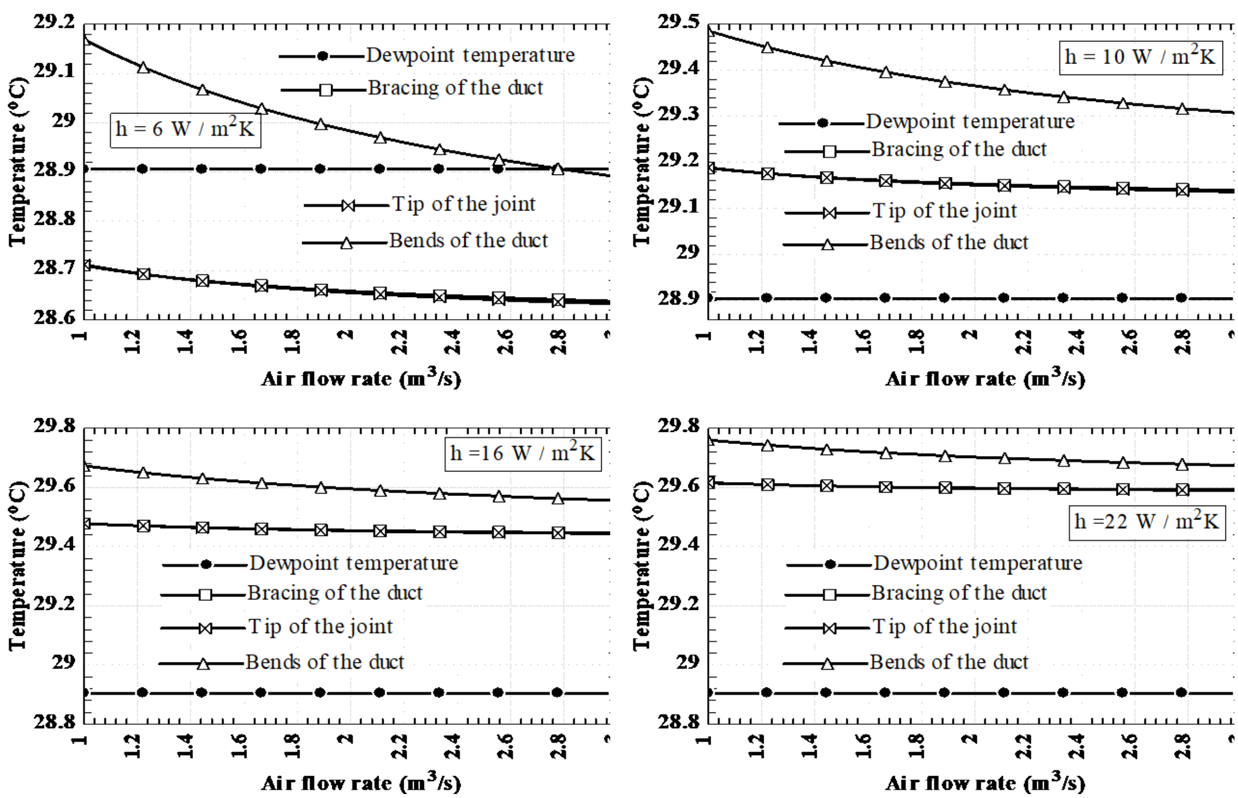
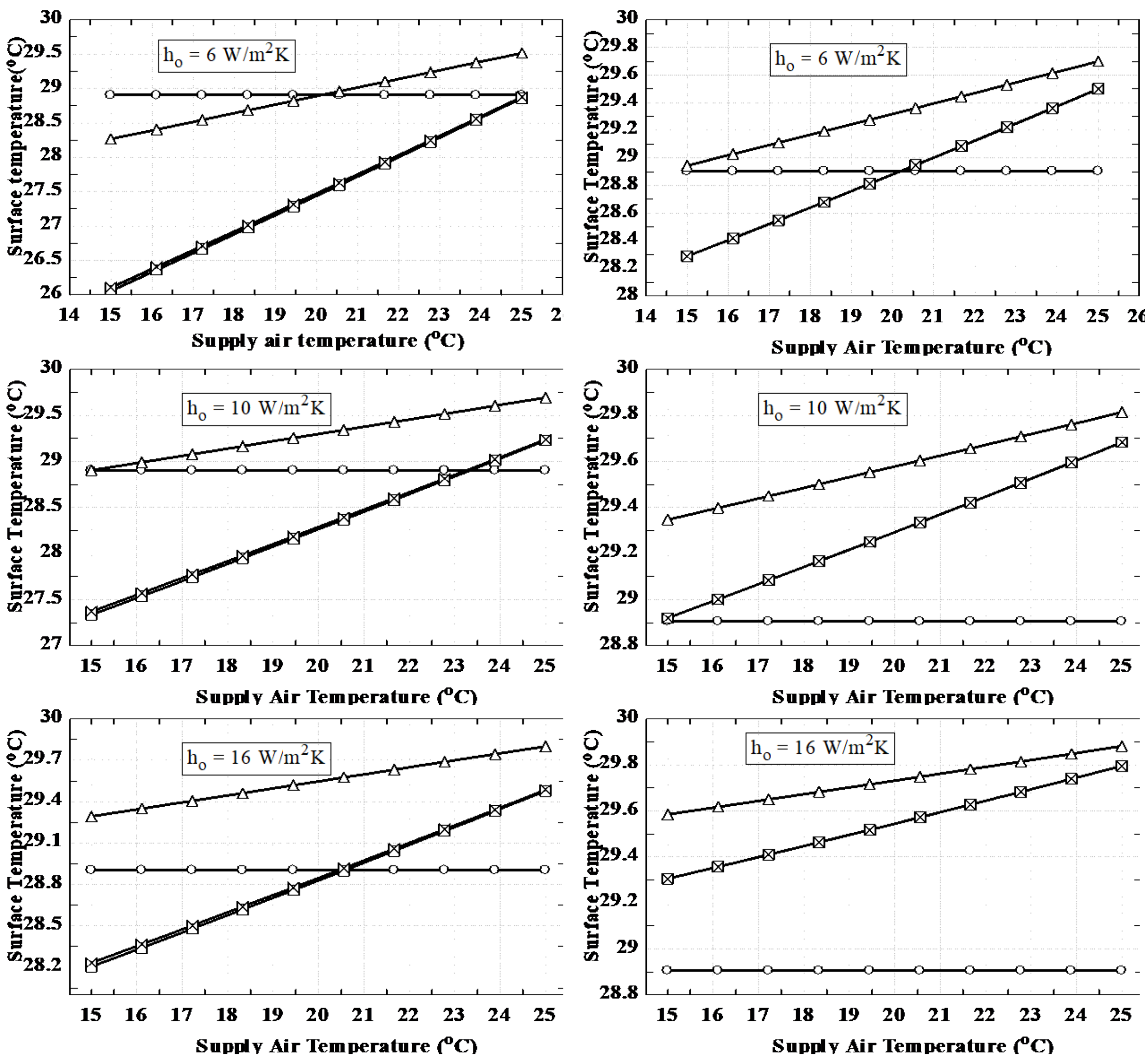


Figure 7: Effect of air flow rate on surface temperature without compression of insulation at selected points of the duct under different convective heat transfer coefficient.

Figure 7 exhibits the effects of air flow rate on HVAC duct's external surface temperature without compression of insulation. The external surface temperature of the duct decreases drastically with air flow rate at the lowest value of convective heat transfer coefficient. Whereas air flow rate has insignificant effect on HVAC duct's external surface temperature as convective heat transfer value increases. It is shown in Figure 7 that chance of condensation occurs at external duct surface at h_o=6 W/m^{2}K. The chances of condensation are avoided with different air flow rate when convective heat transfer coefficient varies from 10 to 22W/m^{2}K.

Effects of supply air temperature on the surface temperature of the duct



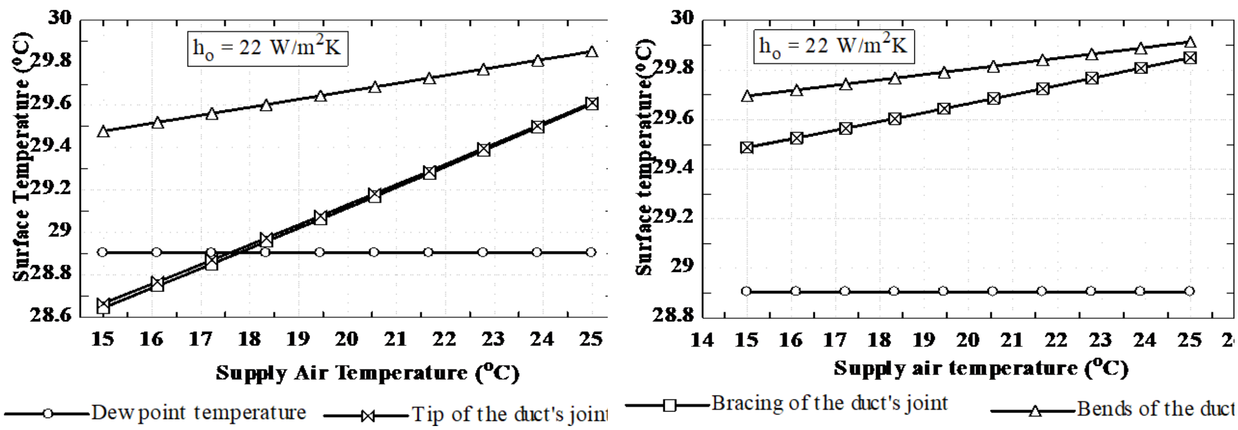


Figure 8: Effect of supply air temperature on surface temperature with (left) and without (right) compression of insulation at different convective heat transfer coefficient.

Figure 8 exhibits the effect of SAT on duct's surface temperature with (left) and without (right) compression of insulation at different convective heat transfer coefficients. It is clear that linear relationship exists between SAT and surface temperature. The surface temperature increases with supply air temperature. Therefore, the chance of condensation at external surface of the duct decreases with increment of SAT. As seen in Figure 8, the chances of condensation at bends are higher than joints with and without compression of insulation at different convective heat transfer coefficients. In case of insulation compression, the condensation could be avoided at bends of the duct if the SAT is higher than 20ºC at h=6W/m^{2}K, 15ºC at 10W/m^{2}K. The chances of condensation at bend s of the duct are negligible at convective heat transfer coefficient above 10W/m^{2}K. Similarly, the SAT should be higher than 25ºC at h=6W/m^{2}K, 23ºC at h=10W/m^{2}K, 20.1ºC at h=16W/m^{2}K and 18ºC at h=22W/m^{2}K to avoid condensation at joint of the duct. In case of without insulation compression, the condensation could be avoided at bends and joint if the SAT is higher than 15 and 20.5ºC at h=6W/m^{2}K and 15ºC at h=10W/m^{2}K whereas condensation does not exist at h=16-22W/m^{2}K. It is seen in Figure 8 that SAT required to avoid condensation at external surface of the duct decreases with insulation thickness.

Effect of ambient air speed on the surface temperature of the duct

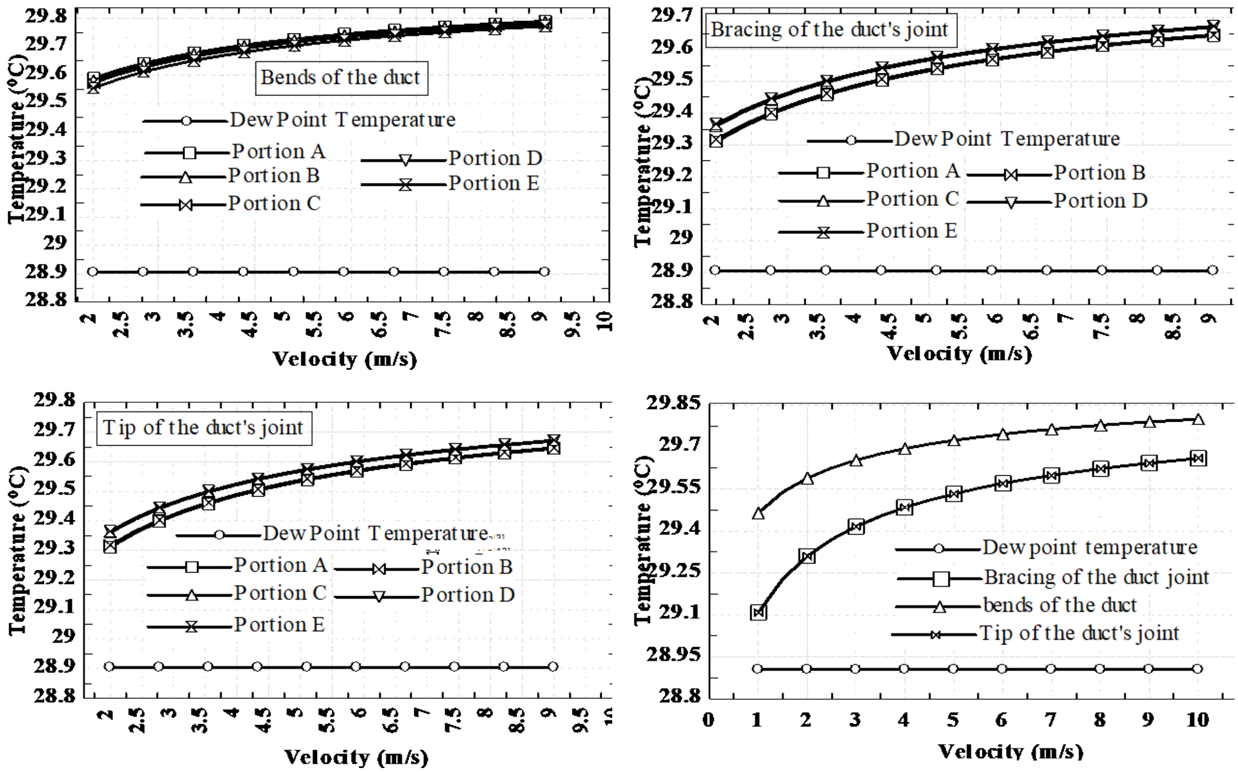


Figure 9: Effect of ambient air velocity on surface temperature with compression of insulation at different convective heat transfer coefficient.

Figure 9 exhibits the effects of ambient air speed on the external surface temperature with compression of insulation at different convective heat transfer coefficients. It is clear that surface temperature increases as ambient air speed increases. Therefore, the chance of condensation at external surface of the duct is avoided by increasing ambient air speed. However, lower ambient air speed increases the chances of condensation at external surface of the duct.

Conclusion

This paper discusses the effects of different thermo-physical parameters i.e., insulation compression, supply air flow rate, supply air temperature, ambient air speed and convective heat transfer coefficient on the external surface temperature of the duct in order to avoid condensation. It is concluded on the basis of obtained results that insulation thickness at the duct's joint should be higher than 55mm at $h=6W/m^2 K$, 35mm at $h=10W/m^2 K$, 20mm at $h=16W/m^2 K$ and 15mm at $h=22W/m^2 K$ to avoid condensation at the external surface of the duct, respectively. Similarly, the insulation thickness at bends of the duct should be higher than 35mm at $h=6W/m^2 K$ and 15mm at $h=10W/m^2 K$ to avoid condensation and the water vapors in external air do not condense at $h=16\text{-}22W/m^2 K$, respectively. In addition to that the air flow rate of conditioned air increases the chances of condensation at external surface of the joint (bracing and tip) and bend of the duct. The air flow rate of conditioned air may be kept higher than $1.4\text{-}2.2m^3/s$ at $10\text{-}22W/m^2K$ to avoid condensation.. Moreover, the effect of SAT and ambient air speed on external surface temperature is investigated. It is determined that external surface temperature increases as SAT and ambient air speed increases.

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NOMANCLATURE

A = Duct's external surface area (m^2) a = unit length of duct except joint (m) b = unit length of bracing (m) D_h = Hydraulic Diameter of the duct(m) H = Height of the duct (m) h = Convective heat transfer coefficient (W/m^2 K) k = Thermal conductivity (W/m.K) L = Length of the Duct (m) $\dot{m}$ = Mass Flow Rate (kg/s) n# = Schedule Nu = Nusselt Number (-) p = Pressure (kPa) v = represents the proportion of the compression surface area of corner to total surface area of the duct (-) W = Width of the Duct (m) Z = Enthalpy of air (kJ/kg) Pr = Prandtl Number (-) $\dot{Q}$ = Rate of heat gain (W) Re = Reynolds Number (-) R = Thermal resistance (K/W) r = Radius (m) T = Temperature (°C) t = unit length of tip (m) x =Thickness (m) x# = thickness of duct sheet U = Overall Heat Transfer Co-efficient ($W/m^2.K$) u = the proportion of the uncompressed surface area to total surface area of the duct without tip and bracing (-)	$\dot{V}$ = Volume Flow Rate (m^3/s) V = Speed (m/s) **Greek Letters** $\mathcal{R}$ = thermal resistance except ambient air (K/W) ϑ = Kinematic viscocity (m^2/s) f = friction factor (-) $\in$ = relative roughness (-) μ = Conversion Factor (kg/kW) ρ = density of conditioned air (kg/m^3) $\mathcal{U}$ = Speed of ambient air (m/s) **Abbreviations** EES = Engineering Equation Solver SAD = Supply Air Duct **Subscripts** amb = ambient a = average b = bracing c = compression d = dynamic e = exit gi = galvanized iron i = inlet ins = insulation ms = mild steel r = riser s = surface s,1 = supply air t = tip T = Total

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