108. Analysis of Energy loss due to compression of thermal insulation in HVAC duct

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Abstract

This paper estimates the heat gain due to compression of thermal insulation at corner, bracing and the tip of air distribution system of HVAC duct. The design and operating parameter data of a renowned pharmaceutical company i.e., Novartis Pharma Pvt. Ltd. Jamshoro, Pakistan is used. The mathematical model is developed and simulation is carried out in Engineering Equation Solver (EES) for energy analysis. The energy analysis yields that the compression of insulation causes around 7.5 % loss of cooling. It is analyzed that the total heat gain from the surrounding can be reduced from 75 W/m^2 to 42 W/m^2 with optimum thickness of insulation at selected points of the duct. It is determined that with an optimum thickness of around 3.8cm quantity of thermal insulation can be increased by 12%. It is noted that electricity consumption can be reduced by around 15% and exit temperature of air from the duct is reduced by around 0.4°C.

Keywords: Heat gain, thermal insulation, HVAC system and energy conservation.

Notations

A	=	Area (m^2)	B	=	Width of the Duct (m)
D	=	Diameter (m)	$\dot{E}$	=	Rate of Energy Transfer (W)
f	=	Friction Factor (-)	H	=	Height of the duct (m)
K	=	Conductive Heat Transfer Co-efficient (W/m °C)	L	=	Length of the Duct (m)
$\dot{Q}$	=	Rate of Rate of heat gain or Loss (W)	R	=	Thermal Resistance (°C W^{-1})
T	=	Temperature (°C)	U	=	Overall Heat Transfer Co-efficient (W m^{-2} °C^{-1})
$\dot{V}$	=	Volume Flow Rate (m^3/s)	V	=	Velocity (m/s)
h	=	Convective Heat transfer Co-efficient (W/m^2 °C)	$\dot{m}$	=	Mass Flow Rate (kg/s)
p	=	Pressure (kPa)	r	=	Radius (m)
x	=	Height of Vertical Drop (m)	z	=	Enthalpy of conditioned air (kJ/kg)
Lgw	=	Thickness of thermal insulation (m)	Lgwc	=	Thickness of thermal insulation at compression (m)
Nu	=	Nusselt Number (-)	Pr	=	Prandtl Number (-)
Re	=	Reynolds Number (-)	b_t	=	Thickness of bracing of the duct (m)
b#	=	Thickness of tip of the duct (m)	n#	=	Number of Bracing (No.)
t,1	=	Length of tip region of the duct (m)	x#	=	Thickness of Galvanized Iron Sheet (mm)

1. Introduction

1.1. Heating, Ventilation and Air Conditioning System (HVAC)

Heating, Ventilation and Air Conditioning System (HVAC) is designed to maintain and control the conditioned space parameters to an acceptable and healthy indoor environment condition of conditioned space for occupant and product processing requirement [1].Rapid development in technology and intensification of the world's population enhance the energy consumption throughout the world [2]. Conservation of energy is one of the major issues of the recent research [3]. It is estimated that about 40% of total energy consumption (EC) of the world is consumed in building sector (commercial and residential) and about 60% of that consumed by HVAC system [5, 8, 13, 25]. However, conservation of energy in HVAC system can be achieved by the optimum design of HVAC system [6].The energy consumption of HVAC system could approximately be reduced by 12% by decreasing average air flow around 57.6m^3/h [7].
In this research work energy consumption in HVAC system of Novartis Pharma (Pakistan) Ltd. Jamshoro is studied and energy losses associated with its HVAC system are investigated. Heat gain due to compression of thermal insulation at the points of compression of the duct is estimated. The effect of compression of thermal insulation on electricity consumption and duct exits temperature is analyzed.

1.2 Analysis of heat transfer [1, 18, 20]

The air distribution system of HVAC system considered in this research work is divided into five portions. The first portion connects the air distribution system to the air-handling unit having length L_1, the second portion connects the main duct to the three different branches, and they transport the conditioned air to three different zones. However, the cooling load of these three zones is considered same but their distribution ducts are of different lengths i.e. L_3, L_4, and L_5. The schematic layout of air distribution system of air-handling unit 9 is shown in Fig. 1. The air-handling unit capacity is 30.66kW which is equally distributed among the given different zones.
Following assumption are made in heat transfer analysis of HVAC duct work:

(a) Steady-state and ideal conditioned are considered.
(b) Dynamic losses and Stack effect are neglected.
(c) Inlet and exist temperature of duct is considered constant T_{in} =20ºC & T_e =23.6ºC.
(d) Ambient temperature of the ductwork is assumed as indoor temperature of building because ductwork installed inside the building T_a=34 ºC
(e) Convective heat transfer co-efficient of ambient air is assumed as h_o=10 W/m^2K. [27]
(f) Effect of duct fittings is assumed negligible.
(g) Design pressure of duct is considered medium pressure i.e. (1kPag)
(h) Volume flow rate of system is considered as 3200CFM.
(i) Leakage losses are neglected.

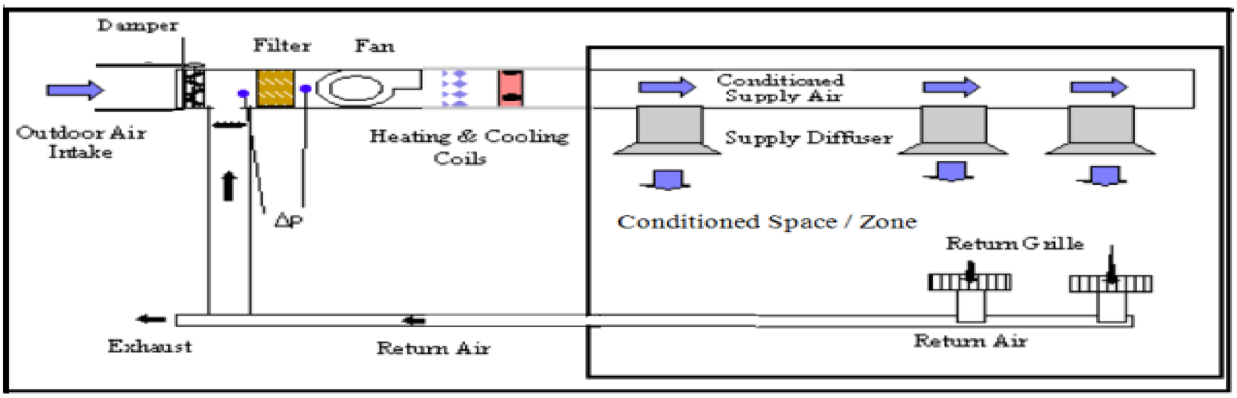

Fig. 1: Schematic layout of simple air-distribution system.

The thermo-physical properties of air are calculated using EES (Engineering Equation Software). The design parameters of HVAC duct system are given in Table 1 and the operation parameters are given in Table 2.

Table 1: Design parameters of HVAC duct

Duct Size			Thickness				Shedule
W	H	L	x#	L_{gw}	L_{gwc}	b_t	b#
cm	cm	m	mm	mm	mm	mm	mm
122	30	1.4	8.5	38	19	3	4
112	30	0.31	8.5	38	19	3	4
40	30	11.1	7.0	38	19	3	2.5
40	30	11.7	7.0	38	19	3	2.5
40	30	12.9	7.0	38	19	3	2.5

Table 2: Thermo-physical properties of air, galvanized iron sheet, mild steal angle and fiber-glass wool insulation.

Pressure	Average Temp.	Density	Specific Heat	Kinematic Viscosity	Prandlt Number	Thermal Conductivity			
P_a	T_a	ρ_{air}	cp_{air}	υ_{air}	Pr_{air}	k_{air}	k_{gw}	k_{gi}	k_{ms}
kPa g	K	m^3/kg	kJ/kg.K	cm^2/sec	(-)	W/m K	W/m K	W/m K	W/m K
1	293	1.652	1.004	0.1105	0.7293	0.02513	0.0376	18.18	54
1	293	1.65	1.004	0.1106	0.7293	0.02514	0.0376	18.18	54
1	294	1.649	1.004	0.1107	0.7292	0.02516	0.0377	18.18	54
1	294	1.644	1.004	0.1114	0.7290	0.02523	0.0379	18.18	54
1	294	1.644	1.004	0.1114	0.7290	0.02523	0.0379	18.18	54

1.3 Continuity Equation

The continuity equation reveals that during steady flow process the total mass flow rate of conditioned air at inlet of HVAC duct is equal to mass flow rate of conditioned air at exit of the duct. Thus, mass flow rate will be constant throughout the HVAC duct and will be calculated from Eq. 1

$$\dot{m} = \rho \dot{V} \tag{1}$$

Mass flow rate of air through duct of portion L_1 is $\dot{m}_1$, L_2 is $\dot{m}_2$, L_3 is $\dot{m}_3$, L_4 is $\dot{m}_4$ and L_5 is $\dot{m}_5$.

$$\dot{m}_1 = \dot{m}_{air} = \dot{m}_3 + \dot{m}_4 + \dot{m}_5 \tag{2}$$

$$\dot{m}_2 = \dot{m}_1 - \dot{m}_3 \tag{3}$$

The psychometric properties of air at inlet and exist of HVAC duct are given i.e. dry bulb temperature $T_{db,in}$ = 293 K and relative humidity $R.H_{in}$ = 45 % these are obtained from company with portable digital hygrometer and the enthalpy is calculated by using psychrometric chart h_{in} & h_e.

Table 3 system operation parameters corresponding to thermodynamic parameters.

Portion	Dry bulb Temperature		Relative Humidity		Enthalpy	
Unit	ºC		%		kJ/kg	
	$T_{db,in}$	$T_{db,ext}$	$R.H_{in}$	$R.H_{ext}$	h_{in}	h_{ext}
1	20	20.1	45	44.7	35.3	35.7
2	20.1	20.15	44.7	44	35.7	35.81
3	20.1	22	44	42	35.7	37.29
4	20.2	22.6	44	43	35.8	38.97
5	22.2	23	44	41	35.8	39.15

2. Energy Balance Equation

The energy balance equation reveals that during steady flow of conditioned air inside the duct the total energy of the conditioned air at the inlet ant exit of the duct remains same. It is mathematically can be written as

$$\dot{E}_{in} = \dot{E}_e \tag{4}$$

Where, $\dot{E}$ represents the total energy transfer through HVAC duct. It is sum of the energy transfer due heat transfer mechanism and energy contained by conditioned air at the inlet of the duct. the energy transfer due to work transfer and leakage loss is neglected. Therefore, energy balance equation can be written as

$$\dot{Q} + (\dot{m}\, z)_{in} = (\dot{m}\, z)_{e} \quad (5)$$

Where, $\dot{Q}$ represents the energy wasted due to heat gain into the duct from surrounding and z represents the enthalpy of conditioned air.

The optimum thickness of thermal insulation at selected points of HVAC duct is calculated by considering uniform heat gain into the duct from surroundings. Thermal conductivity of thermal insulation, G.I sheet and M.S angle is chosen and their corresponding values of thermal conductivity are given in Table 2.
The heat gain into the duct due to conduction is given as

$$\dot{Q} = U\, A\, (T_{amb} - T_{avg}) \quad (6)$$

Where U is the overall heat transfer coefficient of different layers of HVAC duct and is calculated by Eq.7, ΣA is the total surface area of the duct, T_a=30ºC is the surrounding temperature of duct inside the building and T_{avg} is average temperature of conditioned air inside the duct which is given in Table 2 respectively.

$$U = \frac{1}{\Sigma RA} \quad (7)$$

Where ΣR is total thermal resistances, which are calculated by Eq. 12

Thermal resistance of side of duct is calculated by Eq.8

$$R_s = \frac{\left(\frac{1}{h_i} + \frac{x\#}{k_{gi}} + \frac{L_{gw}}{k_{gw}} + \frac{1}{h_o}\right)}{A_{s1}} \quad (8)$$

Thermal Resistance at the corner of duct is calculated by Eq. 9

$$R_c = \frac{\left(\frac{1}{h_i} + \frac{x\#}{k_{gi}} + \frac{L_{gwc}}{k_{gw}} + \frac{1}{h_o}\right)}{A_c} \quad (9)$$

Thermal resistance at bracing of duct is calculated by Eq. 10

$$R_b = \frac{\left(\frac{1}{h_i} + \frac{x\#}{k_{gi}} + \frac{b_t}{k_{ms}} + \frac{L_{gwc}}{k_{gw}} + \frac{1}{h_o}\right)}{A_b} \quad (10)$$

Thermal resistance at the tip of M.S angle of duct is calculated by Eq. 11

$$R_t = \frac{\left(\frac{1}{h_i}+\frac{x\#}{k_{gi}}+\frac{b\#}{k_{ms}}+\frac{L_{gwc}}{k_{gw}}+\frac{1}{h_o}\right)}{A_t} \tag{11}$$

$$\Sigma R = R_s + R_b + R_t + R_c \tag{12}$$

The convective heat transfer inside HVAC duct is calculated as

$$h_i = \frac{K_{air}\,Nu}{D_h} \tag{13}$$

Where, k_{air} is thermal conductivity of conditioned air inside duct, D_h is hydraulic diameter of duct which is calculated by Eq. 14 and Nu is Nusselt number which is calculated by Eq. 16 or 17 according to flow conditions.

$$D_h = \frac{4\,A_c}{p} \tag{14}$$

Where A_c is cross section area of duct and P is perimeter of duct i.e. P = 2 (L+W)

Reynolds Number of air flowing inside the duct is calculated by Eq.15

$$Re = \frac{V\,D_h}{\vartheta} \tag{15}$$

Where V is velocity of air and υ is kinematic viscosity of air inside duct.

If $Re \leq 2300$ and $Pr \leq 0.7$then flow through duct will be laminar then

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$$Nu = 4.31 \tag{16}$$

If $Re \geq 5x10^5$ and $Pr \leq 200$ then flow through duct will be turbulent then

$$Nu = \frac{\frac{f}{8}(Re-1000)\,Pr}{(1+12.7\,(Pr^{2/3}-1)\,(\frac{f}{8})^{1/2}} \tag{17}$$

Where f is the friction factor of inside duct surface it is calculated by Eq. 18 and Pr is Prandlt number of air inside duct.

$$\frac{1}{\sqrt{f}} = -2.0\log\left(\frac{\frac{\epsilon}{D_h}}{3.7}+\frac{1}{Re\sqrt{f}}\right) \tag{18}$$

Where ε is relative roughness of duct surface and is calculated by Eq. 19

$$\varepsilon = \left(\frac{1.5x10^{-4}}{D_h}\right) \tag{19}$$

The enthalpy of air at exit of duct if optimum thickness of thermal insulation at selected points of duct is achieved and it is calculated by Eq. 20 and Effect of thickness of thermal insulation on exit temperature of the duct is calculated by Eq. 21.

$$h_e = \dot{Q}_{opt} + h_{in} \tag{20}$$

$$T_e = \frac{\dot{Q}}{\dot{m}\, c_p} + T_{in} \tag{21}$$

3. Result and discussion

Table 4 and Fig. 2, shows that compression of thermal insulation increases heat gain into the duct from surrounding. The simulation results of mathematical model of HVAC duct result yields that around 75 W/m^2 of total heat gain in the duct from the surrounding with having the compression of thermal insulation at selected points of HVAC duct. Thus, it is about 15% of total cooling capacity of selected air distribution system of the HVAC system. This heat gain can be reduced to 42 W/m^2 if the optimum thickness of thermal insulation at the point of the compression of thermal insulation is obtained with a critical thickness of 3.8 cm. The optimum thermal insulation decreases the heat gain by the duct to around 42 W/m^2 which is about 10.4% of total cooling capacity of the given HVAC system. The optimum thickness of thermal insulation can be obtained by increasing the amount of thermal insulation by 12%of existing thermal insulation.

Table 4, shows the effect of compression of insulation on the total heat gain into the HVAC duct and percentage of total cooling lost from HVAC duct.

L_{gwc}	$\dot{Q}$	$\dot{Q}_{cap}$	Percentage
m	W	kW	$(\dot{Q}\ /\ \dot{Q}_{cap})$ 100
0.01	5.586	30.66	18.22
0.02	4.195	30.66	13.68
0.03	3.564	30.66	11.62
0.04	3.204	30.66	10.45
0.05	2.972	30.66	9.69
0.06	2.809	30.66	9.16
0.07	2.690	30.66	8.77
0.08	2.598	30.66	8.47

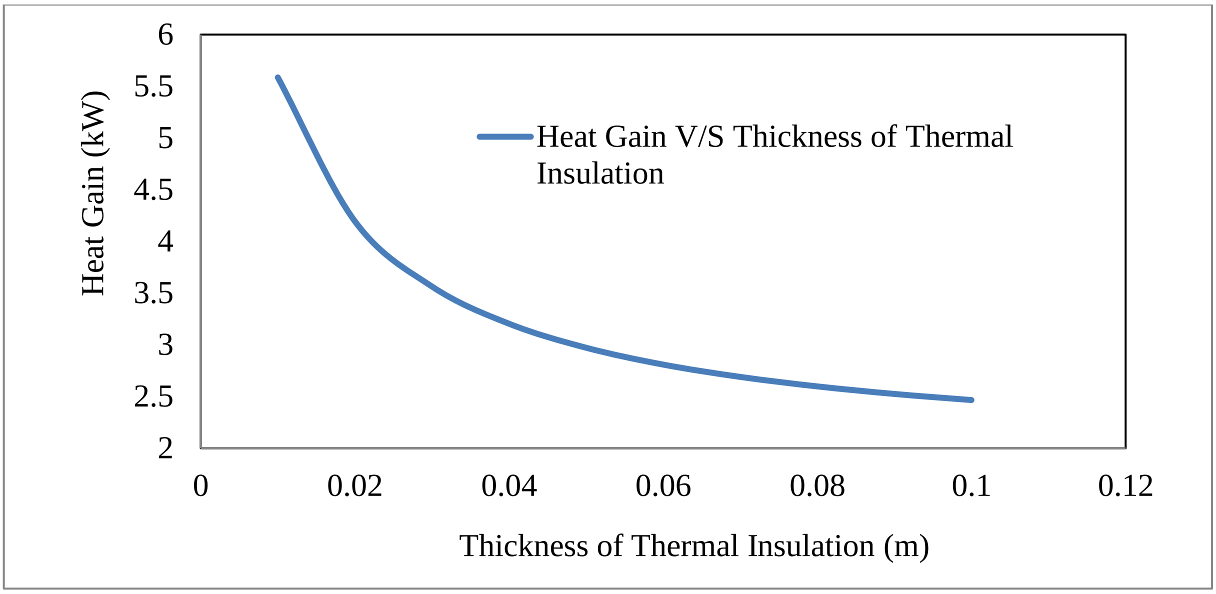

Fig. 2 Effect of thickness of insulation on the heat gain into the HVAC duct from ambient environment.

The HVAC system in this research work has cooling load of around 8.8 ton consuming 5.67 kW of electricity. However, decrease in the cooling load of HVAC system with optimum thickness of the insulation decreases the electricity consumption of the HVAC system from 7.32 kW to 6.2 kW, which is about 15% of the total consumption of HVAC system.

Table 5: Effects of thickness of thermal insulation on the exit temperature.

L_{gwc}	$T_{ext,na}$	$T_{ext,nb}$	$T_{ext,nc}$	$T_{ext,nd}$	$T_{ext,ne}$
cm	ºC	ºC	ºC	ºC	ºC
1.00	20.1	20.1	21.9	22.3	22.3
1.31	20.1	20.1	21.9	22.2	22.2
1.62	20.1	20.1	21.8	22.1	22.1
2.24	20.1	20.1	21.7	22	22
2.87	20	20.1	21.7	21.93	21.9
3.80	20	20.1	21.7	21.85	21.8

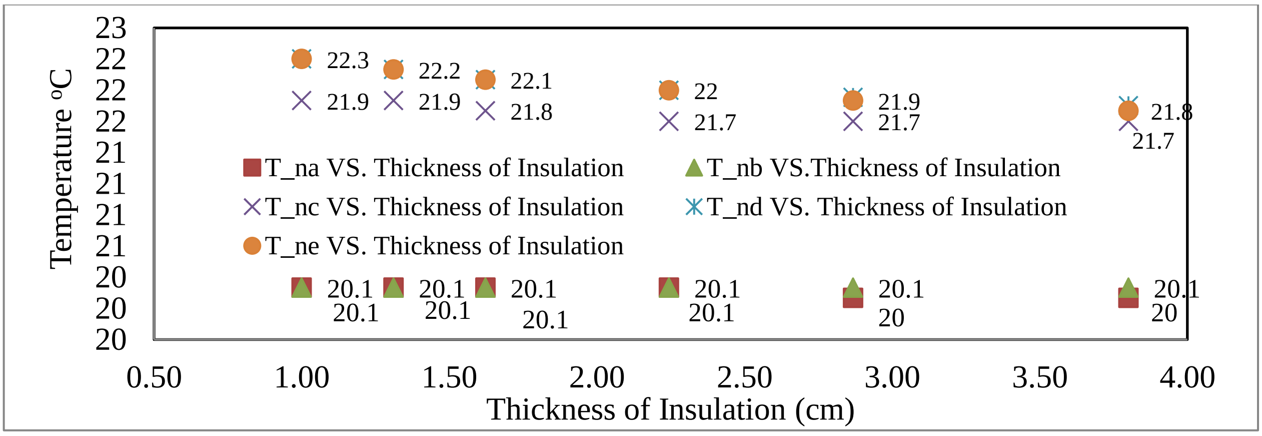

Fig. 3: Effects of thickness of insulation on the exits temperature.

From Table 5 and Fig. 3, it is clearly shown that thickness of insulation has an inverse effect on the exit temperature of HVAC duct. For first two portions of the HVAC system the decrease in temperature with the increase in thickness of thermal insulation is smaller than the remaining three because both first portions of HVAC duct have smaller surface area. Thus, result in lesser heat gain as compared to the remaining three branches along with that these two portions of the duct from the design point of view are of greater size than the remaining three because of that it has lower compression area. Therefore, the effect of compression on heat gain in this point of HVAC duct is negligible. Thus, in this portion of

HVAC duct there is only 0.1oC of temperature reduction and in the second portion observes no effect on the exit temperature of air. The preceding three branches of air distribution system are of small size and there is a considerable proportion of compression area of thermal insulation exists. However, the duct has higher surface area due to a longer length. Therefore, compression area of this point of the duct is also high. Thus, it results in higher heat gain from the surrounding into the HVAC duct. Therefore, the thickness of insulation has a significant impact on the exit temperature of these HVAC ducts. Thus, the exit temperature is reduced by around 0.5, 0.4 and 0.2^{o}C in 5, 4, and 3 branches. The branch 3 has intermediate temperature because it is of smaller length than the remaining two branches. The exits temperature of three branches 3, 4 and 5 is different from one another because heat gain into them from surrounding is different due different surface area.

The decrease in exit temperature of the duct air causes decrease in the inlet temperature of the duct without changing the conditioned space conditions of the HVAC system. This decrease in temperature conditioned air has a significant effect on the performance of whole HVAC system because it reduces the chilled water temperature considerably because the effectiveness of cooling coil unit is low due to heat transfer form water to air. Therefore, cooling load of absorption chiller will be reduced significantly by increasing the thickness of thermal insulation at the point of compression.

4. Conclusion

The optimum thickness of thermal insulation at the different point of compression of thermal insulation yields following results.

Total heat gain from the surrounding to HVAC duct can be reduced to 10.4% of total cooling capacity of the system.

1. The tone of cooling load could be reduced to 15% of the total cooling capacity of the system.
2. The electricity consumption of the HVAC system could be reduced from 7.32 kW to 6.2 kW.
3. The exit temperature of HVAC duct can be averagely reduced to 0.37^{o}C in the three different branches.
4. In order to achieve optimum thickness, thermal insulation should be increased by 12%.

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