Weight of duct compressing insulation at duct support angle – a potential source of condensation

Stucco aluminium cladding gives a good finish to insulation

Calculating Energy Losses in Pipe & Duct Insulation

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Introduction

In the HVAC industry, insulation material is used to conserve energy spent on the cooling mediums (refrigerant, air and water). Proper use of thermal insulation material is essential to ensure minimum heat loss and to prevent condensation problems. Condensation can lead to deterioration of insulation and further energy loss over time.

All heat gained by the chilled water during its passage through the pipes or by treated air passing through ducts is a loss of energy and reduces the efficiency of the cooling process. It is not possible to reduce this energy loss to zero. Nor is it possible to control the temperature on the two sides (chilled water or treated air on one side and ambient conditions on the other side) by adding insulation. However, by proper selection of insulation material and insulation thickness calculations, it is possible to ensure minimum losses at optimal costs. It is also necessary to check that the insulation thicknesses are adequate to prevent condensation on duct or pipe surfaces. Sweating or condensation can adversely affect the properties of some types of insulation material, which can have a cascading detrimental effect on the efficiency of insulation.

Generally, in humid climates, thickness of insulation is governed more by condensation considerations than by energy losses. Some industries do accept condensation on insulation as an inevitable but a tolerable condition especially if its occurrence is restricted to a few hours in a year and is confined to uninhabited areas like trenches, shafts etc. However, in a standard AC installation, where chilled water pipes and AC ducts are routed through false ceiling void spaces and accessible shafts, condensation is, more often than not, unacceptable.

This article will try to present simplified ways of using available and existing heat transfer equations for calculating the energy loss in ducting and piping systems. It will also present ways of checking on

About the Author

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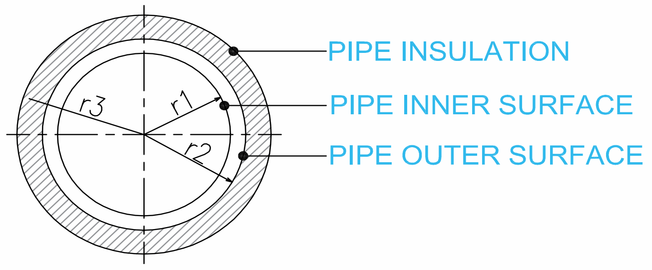

Figure 1: Insulated chilled water piping

condensation risks by considering different ambient conditions.

The Heat Transfer Process

The heat transfer process from the outer surface of the duct or pipe to the cold medium inside occurs in the following modes:

- By convection across the outer air film layer
- By radiation at the outer surface, which depends on the emissivity of the surface and its temperature
- By conduction across the insulation material and by convection across air gaps within insulation, if any
- By conduction across the pipe or duct material
- By convection across the inside air or water film layer

The Fourier equation for steady state one-dimensional heat flow is

$q = k.A.\Delta T/x$

where

q = rate of heat flow in watts, W

A = cross sectional area perpendicular to the heat flow direction, m^2

K = thermal conductivity of material, W/m°K

ΔT = Temperature difference, °K

x = thickness of medium across which heat transfer occurs

Chilled Water Piping

For cylindrical surfaces like piping, thickness x is the equivalent thickness, which is defined as thickness of the medium on a flat surface, which gives the same heat flow as dimension x would give on a cylindrical surface.

x for cylindrical surfaces is $= r_2.\ln(r_2/r_1)$

Therefore, q (for cylindrical surfaces) $= k.A_2.(T_1-T_2)/r_2.\ln(r_2/r_1)$

where r_2 is the outer radius of the cylindrical surface

r_1 is the inner radius of the cylindrical surface

A_2 is the area of the outer surface

T_1 and T_2 are the inside and outside temperatures

For an insulated chilled water pipe with numerous layers of component materials and medium spaces, a factor called Overall Heat Transfer Coefficient comes into play and the governing heat transfer equation becomes

$Q = U.A_{inner}.\Delta T$

where

Q = Rate of heat flow, Watts

U = Overall heat transfer coefficient, $W/m^2{}^\circ C$

A_{inner} = Area based on inner surface of pipe or insulation in sq m

The overall heat transfer coefficient is obtained as the reciprocal of the sum of resistances of all the component layers.

$$\text{Therefore, } U = \frac{1}{R_1 + R_2 + R_3 + \ldots\ldots R_m}$$

$$\text{Therefore, } \frac{1}{U} = R_1 + R_2 + R_3 + \ldots\ldots R_m$$

For an insulated chilled water pipe with insulation,

$$\frac{1}{U} = \frac{1}{h_i} + \frac{r_1 . \ln(r_2/r_1)}{k_{pipe}} + \frac{r_1 . \ln(r_3/r_2)}{k_{insulation}} + \frac{r_1}{h_o .\ r_3}$$

With respect to the outer surface of pipe insulation, the overall heat transfer coefficient is expressed as

$$\frac{1}{U} = \frac{r_3}{r_1 . h_i} + \frac{r_3 . \ln(r_2/r_1)}{k_{pipe}} + \frac{r_3 . \ln(r_3/r_2)}{k_{insulation}} + \frac{1}{h_o} \quad \ldots\ldots\ldots (1)$$

where

h_i = Convection heat transfer coefficient at the inner surface of the pipe in W/m^2 .°K

h_o = Radiation heat transfer coefficient at the outer surface of pipe insulation in W/m^2 .°K

r_1 = inner pipe radius

r_2 = outer pipe radius

r_3 = outer pipe insulation radius

k_{pipe} = Thermal conductivity of pipe material, W/m °K

$k_{insulation}$ = Thermal conductivity of pipe insulation material, W/m °K

See *Figure 1*.

The term h_i is the water film coefficient at the inner pipe surface that exists as water flows inside the pipe, and is directly proportional to water velocity. In this exercise, the empirical formula given in *ASHRAE Fundamentals (2009) Handbook* is used to calculate h_i.

$$h_i = (1431 + 20.9 *t)V^{0.8} / D^{0.2} \quad \ldots\ldots\ldots (2)$$

where t is the water temperature in °C

V is the water velocity in m/sec

D is the pipe diameter in m

The term h_o is the heat transfer coefficient of the air film at the outer surface of pipe insulation. As both convection and radiation occur at the outer surface, this term can be expressed as

$h_o = h_{co} + h_{ro}$

where

h_{co} = Convection heat transfer coefficient of the air film at the outer surface in W/m^2.°K

h_{ro} = Radiation heat transfer coefficient at the outer surface in W/m^2 .°K

h_{co} ranges from 8 W/m^2 °K for indoor air free convection to 33.3 W/m^2 °K for external applications, where wind conditions prevail. (Example: Chilled water piping in tunnels or trenches with forced ventilation.)

As per Stefan-Boltzman law, heat loss from the surface of a pipe by radiation is equal to

$$\sigma . \epsilon . A . (T_a^4 - T_s^4) \quad \ldots\ldots\ldots A$$

where

σ = Stefan-Boltzman constant = 5.673×10^{-8} $W/m^2 . K^4$

ϵ = Emissivity of the surface

T_a = Ambient temperature, °K

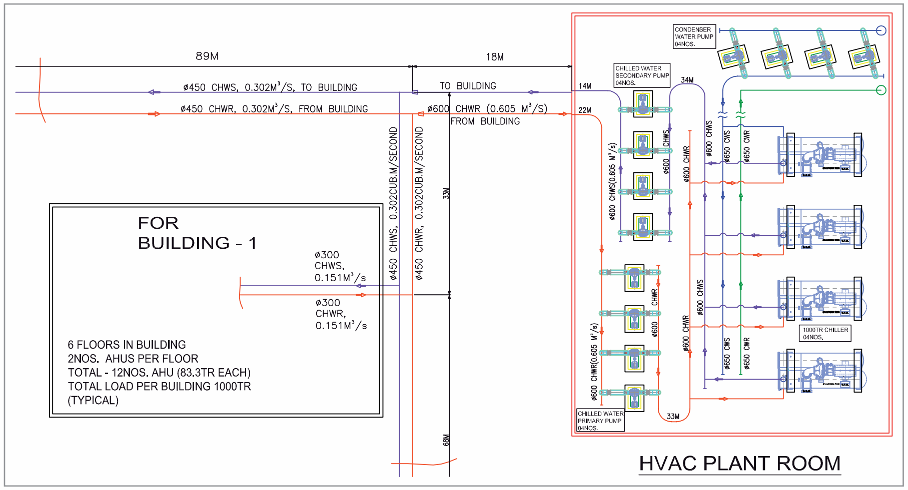

Figure 2: Schematic chilled water piping

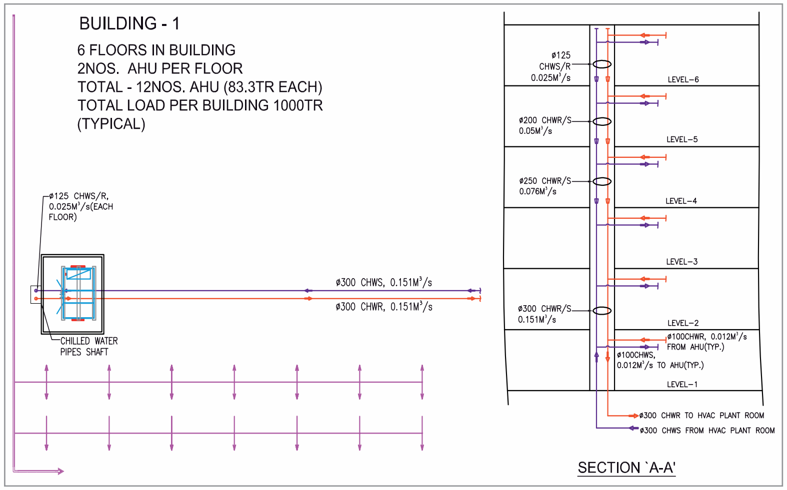

Figure 3: Schematic ductwork

Table 1: Sample calculations for temperature rise in supply/ return chilled water piping

Insulation material	Fibreglass	K value	0.033 W/m°K	Chilled water supply temperature	5.55 °C
Insulation density			80 Kg/m³	Chilled water return temperature	11.11 °C
Thermal conductance steel pipe			49 W/m°K	Check for dew point at	35 °C 85% RH

Item	Nominal dia, mm	Inner dia, m	Pipe thickness, m	Insulation thickness, m	Pipe length, m	Flow, gpm	Flow, l/sec	Water velocity, m/sec	Ambient temp., °C	h_i, W/m² °K (Eqn. 2)	h_{co}, W/m² °K	h_{ro}, W/m² °K (Eqn. 3)	U overall (Eqn. 1)	Water inlet temp., °C	Q, Watts (Eqn. 4)	Water outlet temp., °C (Eqn. 5)	Temp. rise across section, °C	Insulation surface temp. °C (Eqn. 6)	DP temp. °C
Suction / discharge lines	300	0.3097	0.0071	0.025	6	2400	151	2.0	35	3675	8	5.93	1.13	11.1100	190	11.110	0.00030	33.06	32.10
Header	600	0.59	0.01	0.025	33	9600	606	2.2	35	3492	8	5.93	1.16	11.1103	1901	11.111	0.00075	33.01	32.10

Table 2: Sample calculations for temperature rise in supply/ return ducting

Area served :	Office floor (half)				
Unit Capacity :	SUPPLY				
	Supply air qty	19.13 m³/sec	Air off coil temp	12.78°C	
	Total static	60 mm wc	Temp. rise across SA fan motor	0.000175656°C	
	Motor bkw	14.1 kW	Temperature at AHU outlet	12.78017566°C	
	Unit type	draw thru	Check for Dew Point at	24°C	90 % RH
Insulation material	Fibreglass	0.033 W/m°k			

	Duct width, mm	Duct height, mm	Duct length, m	Air flow rate, m³/sec	Air velocity, m/sec	Insulation thickness mm	Temp. around duct °C	h_{ro} W/m°K (Eqn. 7)	U value, W/m² °K (Eqn. 6)	Air temp. entering section, °C	Q, Heat loss from section, Watts (Eqn. 8)	Air temp. leaving section, °C (Eqn. 9)	Temp. rise across section, °C	Insulation surface temp. °C (Eqn. 10)	DP temp.
Straight Duct	2500	800	4	19.13	9.57	25	24	5.3180530	1.134	12.78	346	12.80199	0.022	23.045	22.3
Straight Duct	2500	600	8	12.75	8.50	25	24	5.3180530	1.131	12.80	649	12.86335	0.061	23.049	22.3

Editor's Note: Due to space constraints, the above Tables 2 and 3 have been extracted from the Author's excel worksheets running into several pages with painstaking detailed calculations for every section of chilled water piping and ductwork. These tables give readers an idea of the calculations involved.

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T_s = Temperature of the outer surface of pipe, °K

Heat loss from pipe surface **due to radiation** can also be expressed as

$h_{ro} . A . (T_a - T_s)$ .... B

From equations A & B above, we get

$h_{ro} = \sigma . \epsilon . (T_a^2 + T_s^2) . (T_a + T_s)$ .... (3)

However, T_s is an unknown to be calculated. So, for the purposes of calculation, it can be assumed to be 1.5°C less than the ambient temperature.

Typical ϵ values for surfaces encountered in chilled water pipe insulation coverings are as under:

Aluminium cladding

• Polished surface	0.03 to 0.1
• Dull surface	0.1 to 0.4
• Oxidised space	0.6
• Rubber	0.9
• Plastic sleeve	0.9
• Glass cloth with vapour barrier	0.9

Equation (1) can be used to find out the U values at various sections of a chilled water pipe run for a project.

Having worked out U values, it is possible to work out the heat transfer occurring at every section of the pipe using the equation

$Q_{section}$ = U (from Eqn. 1). A (outer surface area of insulated pipe). $(T_a - T_{water\ inside\ pipe})$ .... (4)

where

T_a is the ambient temperature existing outside the insulated pipe section

From the above heat transfer figure, the temperature drop across the section can be worked out using the mass flow equation for chilled water as under:

$Q_{section} = M . C_p . (T_{in} - T_{out})$ .... (5)

where

Q section is the heat transfer occurring in that particular pipe section, Watts

C_p is the specific heat of water, kJ/kg °K

$T_{in} - T_{out}$ is the temperature drop undergone by chilled water across that pipe section, °C

The temperature on the surface of the insulated pipe can be found from the equation

$$T_{surface} = \frac{T_{in} - (T_{in} - T_a) . U_{surface}}{U} \qquad (6)$$

where $U_{surface}$ is the U value upto the insulation surface.

Equations 1,2,3,4,5 & 6 can be incorporated in an excel sheet to work out the energy loss over the entire chilled water piping system and the temperature drop over the longest length of pipe for any installation.

This, in turn, helps decide the thickness of insulation for the pipe to reduce energy losses to a minimum.

The surface temperature can be compared to the dew point temperature existing at that pipe section to determine if condensation can occur at that section and the insulation thickness that would be required for avoiding the same.

The insulation thickness can also be varied along the piping run based on surrounding space conditions. Examples - thicker insulation in unventilated shafts, thinner thickness for return void space.

The above exercise was carried out for a typical AC layout shown in *Figure 2*. The AC system consists of 4 chillers, all working, each of 1000 TR capacity. There are 4 buildings, each of 300,000 sqft built up area with a peak cooling capacity requirement of 1000 TR.

Each building is a six storeyed structure with 2 AHUs per floor, each AHU of 83.3 TR cooling capacity. Chilled water is generated at 5.55°C (42°F) in the chiller plant. Return water temperature is at 11.11°C (52°F).

Inside conditions are 23°C (73.4°F), 55%RH.

Aluminium cladding is considered for the entire length of piping. Ducting is covered with glass cloth and finished with vapour barrier.

The insulation thickness taken is based on recommendations given in *ASHRAE Fundamentals (2009) Handbook* (25 mm thick for pipes upto 1000 mm dia.).

The step wise sample calculations extracted from the excel sheet are presented in *Table 1*.

It can be seen from the completed excel sheet that

- The temperature drop over the longest length of piping (primary + secondary) works out to 0.034°C.
- At 70% chilled water flow (variable speed operation of secondary chilled water pumps), the temperature rise increases to 0.095°C. (Multiply the chilled water flow rates by 0.7 to get this result). However, the heat gain remains practically the same as before. This is because a smaller flow rate of water picks up heat from the same surface area of piping.
- The total energy loss from the chilled water piping works out to 0.45% of the total energy contained in the cooling system, which is within the acceptable limits of 1%.
- Increasing the header piping length from 65m to 1 km – typical of district cooling systems where AC plants are remote – increases the energy loss to 1.3%.

Increasing the insulation thickness from 25 mm to 50 mm increases capital costs by around Rs.11 lakhs but reduces energy losses to less than half. Calculations show a payback period of a little over a year, and this option is definitely worth considering.

The table also shows surface temperature on the insulated pipe surface. This can be compared with the dew point conditions of the air surrounding the pipe at that location to check if condensation is possible. Thickness of insulation can be decided accordingly.

For a city like Mumbai, RH is higher than 95% for more than 300 hours in a year. Pipes in trenches and shafts could be exposed to such ambient, and insulation thickness in these segments of piping run should be checked for condensation possibilities.

BIS codes for insulation allow a tolerance of 15% on the K value. A check for condensation and heat loss can be carried out by inserting the highest possible K value within the tolerance

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limits and take remedial action, if necessary. For example, at the upper limit of K value tolerance, condensation can occur even at 35°C and 85% RH for the 600 dia. pipe mentioned above.

Ducting

Skipping intermediate steps, the U value for insulated ducting can be expressed as

$$\frac{1}{U} = \frac{1}{h_i} + \frac{x}{k_{insulation}} + \frac{1}{h_o} \quad \text{.......... (6)}$$

where

h_i = Convection heat transfer coefficient at the inner surface of the duct in w/m² .°K

h_o = Heat transfer coefficient at the outer surface of duct insulation in w/m² .°K

x = Insulation thickness over duct

$k_{insulation}$ = Thermal conductivity of duct insulation material, W/m °K

Here the resistance offered by the duct material to air flow is neglected as it is negligible.

The term h_i is the air film coefficient at the inner duct surface and is directly proportional to air velocity. It is a function of Reynolds number and is expressed by the following empirical equation:

$$\frac{1}{h_i} = 0.286 \frac{D^{0.25}}{V_{0.8}} \quad \text{for circular ducts}$$

$$= \frac{0.286\ (2wd / (w+d)^{0.25}}{V_{0.8}} \quad \text{for rectangular ducts}$$

where

D is the internal diameter, m

V is the mean air velocity, m/sec

w and d are the width and depth of the duct respectively.

The above empirical equations are taken from *Air Conditioning Engineering* by W P Jones.

The term h_o is the heat transfer coefficient of the air film at the outer surface of duct insulation. As both convection and radiation occur at the outer surface, this term can be expressed as

$$h_o = h_{co} + h_{ro}$$

where

h_{co} = Convection heat transfer coefficient of the air film at the outer surface in w/m² .°K

h_{ro} = Radiation heat transfer coefficient at the outer surface in w/m² .°K

h_{co} ranges from 8 w/m² °K for indoor air free convection to 33.3 w/m² .°K for external applications, where wind conditions prevail.

As before, h_{ro} can be expressed as

$$h_{ro} = \sigma. \epsilon. (T_a^2 + T_s^2) . (T_a + T_s) \quad \text{.......... (7)}$$

However, T_s is an unknown to be calculated. So, for the purposes of calculation, it can be assumed to be 1.5°C less than ambient temperature.

Equation (6) can be used to find out the U values at various sections of an AC duct run for a duct installation.

Having worked out U values, it is possible to work out the heat transfer occurring at every section of the duct using the equation

$$Q = U \text{ (from Eqn. 1). A (outer surface area of insulated duct).} (T_a - T_{air\ inside\ duct}) \quad \text{.......... (8)}$$

where

T_a is the ambient temperature existing outside insulated duct section.

From the above heat transfer figure, the temperature drop across the section can be worked out using the mass flow equation for air as under:

$$Q_{section} = M . C_p . (T_{in} - T_{out}) \quad \text{.......... (9)}$$

where

M = Mass flow rate of air in Kg/sec

$Q_{section}$ is the heat transfer in that particular duct section

C_p is the specific heat of air in kJ/kg°K

$T_{in} - T_{out}$ is the temperature drop undergone by air across that air section, °K

The temperature on the surface of the insulated duct can be found from equation

$$T_{surface} = T_{in} - (T_{in} - T_a).U_{surface} / U \quad \text{.......... (10)}$$

where $U_{surface}$ is the U value upto the insulation surface.

Equations 6, 7, 8, 9 & 10 can be incorporated in an excel sheet to work out the energy loss over the entire ducting run system and the temperature drop over the longest length of duct for an installation.

This, in turn, helps decide the thickness of insulation over the duct to reduce energy losses to a minimum.

The surface temperature can be compared to the dew point temperature existing at that duct section to determine if condensation can occur at that section, and the insulation thickness required for avoiding the same.

The above exercise was carried out for a typical ducting with AHU as shown in *Figure 3*.

Using the same example as before, the ducting system caters to one half of a floor of the example considered before.

The insulation thickness taken is based on recommendations given in *ASHRAE Fundamentals (2009) Handbook*, viz. R-0.62 for supply (m².°K/w), which works out to 25 mm thick fibreglass insulation.

The step wise sample calculations extracted from the excel sheet are presented in *Table 2*.

It can be seen from the completed excel sheet that

- The temperature rise in the longest length of duct is 1.3°C.
- At 70% air flow (VFD operation of AHU motors with VAVs), the temperature rise increases to 1.8°C. (Multiply the air flow rates in sheet 5 by 0.7 to get this result). Heat gain remains practically the same as before. This is because smaller air flow of air collects practically the same amount of heat from the same surface area of duct.
- The heat gain in the system is about 10 kW, which is about 3.4% of the total energy in the system. This is higher than normally acceptable standards. Increasing the insulation thickness to 50 mm reduces heat gain to 2% and is definitely advisable.
- At 70% treated air flow, the total heat gain reduces marginally

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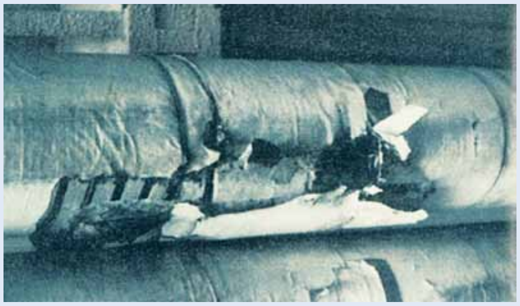
to 9.5 kW. (Multiply the air flow rates by 0.7 to get this result).This is because smaller air flow rate collects practically the same amount of heat from the same surface area of duct.

The table shows surface temperature on insulated duct surface. By changing the ambient conditions, the insulation surface temperature at different times of the year can be calculated and hence the condensation risk can be checked.

For example, if the supply duct passes through a store room or a toilet crawl space, the temperature around the duct would be 37°C to 38°C with 85%-90% RH on some hot days. Condensation could occur under such conditions. Increasing insulation thickness to 50mm removes the condensation risk.



Wherever dewpoint temperatures are above 4.4°C, water vapour is in the atmosphere. That water vapour penetrates permeable insulation on chilled water systems and condenses within it.

The result? The wet insulation becomes the thermal conductor, totally losing its insulating value.

Condensation Risks

In enclosed spaces without ventilation, temperature as well as humidity builds up. Air at higher temperature has more affinity and ability to retain moisture than at lower temperature. Humidity in such conditions can easily reach 90%. In humid areas like Mumbai, ambient with 95% relative humidity have been recorded even in the months of February and March, leave alone in the monsoon months. For condensation checks, the dew point of this high temperature and humidity air should be considered.

Some typical conditions in HVAC installations, where condensation can occur are described below:

False ceiling voids

Return air is often routed through false ceiling voids. Chilled water pipes are often routed through these voids – operating 24x7 based on occupant requirements. If an office space is switched off or if the office space has no occupant for long, the ceiling void space can become very humid. Sweating can occur on the pipes under such conditions.

A similar situation can arise for pipes or ducts passing through unventilated enclosed shafts.

AHU rooms

Unventilated AHU rooms, with ducted return, are another location where sweating or condensation is common occurrence.

If the AHU room has an outer periphery wall, moisture permeation can occur across the walls especially during the monsoon, increasing the humidity within the room.

The AHU casing itself can be a major source of condensation. AHUs carry air at 13°C whereas the temperature within the AHU room can be 35°C if return air is ducted. The double skinned casing as well as the pentapost framework forms a cold bridge from the inside to the outside of the AHU. This is called Thermal Bridging. This leads to condensation on the AHU casing and framework (see *Figure 4*).

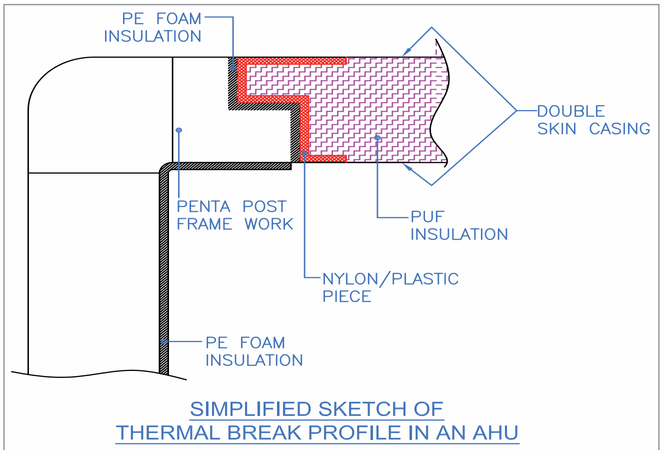


Figure 4: Thermal break profile in an AHU

The EN standard for AHU (*EN 1886*) quantifies this characteristic by defining a Thermal Bridging Factor which is defined as the ratio of the lowest temperature difference between the external surface and internal AHU temperature to the air-to-air temperature difference (AHU inside and outside).

$$\text{Thermal Bridging Factor } K_b = \frac{\Delta t_m}{\Delta t_{air}} = \frac{t_{smax} - t_i}{t_a - t_i}$$

where

t_i is the mean internal air temperature

t_a is the mean external temperature

t_{smax} is the maximum external surface temperature on the unit casing.

Classification of AHU casings as per EN 1886 is as follows:

Unit classification	Thermal Bridging Factor
TB1	$0.75 \leq Kb < 1.0$
TB2	$0.6 \leq Kb < 0.75$
TB3	$0.45 \leq Kb < 0.6$
TB4	$0.3 \leq Kb < 0.45$
TB5	No requirement

Good specifications generally insist on TB2 classification for AHU. However, in India, there are only a couple of EN certified AHU manufacturers. A designer could insist on submission of thermal decoupling characteristics before ordering an AHU.

Taking an example of an AHU room with a TB2 classified AHU and 35°C room temperature, condensation will occur when RH in the room reaches 60%. At 28°C room temperature, condensation occurs when RH reaches 70%. It is advisable to provide some

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minimal cooling to the AHU room to avoid condensation issues.

Duct supports

Another typical case of potential condensation is when the weight of the duct compresses the 50 mm thick blanket insulation above the duct support angle to less than 25 mm. The R value here reduces to less than half causing the metal support to cool down to below dew point temperature.

Duct supports should have a hard wood or hard rubber insert between the duct and support member. The aluminium foil facing and the glass cloth with the vapour barrier coat should wrap around this insert piece for a condensation risk free installation.

Workmanship

A similar situation of reduction in insulation thickness occurs at duct corners, when fibreglass blanket insulation is wrapped around rectangular ducts. If the blanket is stretched tight, the reduction in insulation thickness can lead to beads of sweat forming at the duct corners.

Photo 1: Condensation due to air gaps in fibreglass blanket insulation, leaking out of insulation joints

The insulation installation process requires proper application of adhesive on the entire duct/pipe surface – without any air gaps. Condensation occurring in air gaps can destroy the insulation properties of fibrous insulation over time. For cellular insulation, condensate can leak out at insulation joints giving the impression of duct or pipe sweating (see *Photo 1*).

Fibreglass duct insulation is available in rigid board or blanket form. Rigid boards are costlier and installation costs are higher as boards have to be cut to duct size and insulation joints sealed with aluminium tape. Wastage is also higher than in blanket insulation. However, rigid board insulation gives neater appearance.

Material Properties

Insulation Materials

The basic property of insulation material is its low thermal conductivity. As air is a poor conductor of heat and is commonly available, most insulation products in HVAC industry use air as the insulating medium. Insulation materials commonly used in HVAC can be classified as:

- Cellular insulation, and
- Fibrous insulation

Cellular insulation has a closed cell structure containing air or other foaming agents. Common types are expanded polystyrene, elastomeric nitrile rubber, polyurethane foam (PUF), polyisocyanurate (PIR), phenolic foam, polyethylene/ polyolefin foam, etc. PUR and PIR have very low K values and, therefore, have lower thickness requirement as compared to other insulation materials for a particular resistance value. These materials use CFC or HCFC as the foaming agent.

Fibrous insulations are loose filled with fibres or nodules held together by binder with air spaces in between. Common types are fibreglass or mineral wool. Fibreglass is produced by melting silica, with binder material added to hold the fibres in place.

Mineral wool is obtained from molten rock or slag and is manufactured at high temperatures. They can, therefore, be used for high temperature applications. These materials are inorganic in nature (do not contain carbon).

Fibreglass duct insulation (blankets/ slabs) and pipe insulation (rigid or mat form) are normally manufactured with a laminated aluminium foil jacket as a vapour barrier. This is called FSK facing (foil + scrim, a fabric lining + kraft paper)

Thermal conductivity values of these materials are available in published catalogues for different makes. They are expressed as K values in units of W/m°C or Btu-in/hr sqft °F. Thermal conductivity is directly proportional to temperature, as movement of trapped air molecules increases with temperature. The designer should ensure that K values selected are for the average temperature across the insulation thickness.

Closed cell insulation, that uses gas agents in place of air, tends to lose the gas over time by migration. This reduces their resistance to heat transfer with increase in thermal conductivity. This is called thermal drift or ageing. The K value stabilises over time – generally 2 to 5 years, and designers should ensure that the stabilised K value is taken for calculations.

Critical Thickness of Insulation

As the pipe insulation thickness increases, the surface area available for heat transfer increases. This increases the heat loss. However, this is seldom an issue with pipe insulation and is more relevant to current carrying cables and wires.

Economic Thickness of Insulation

This concept calculates the initial installed cost of insulation for a particular thickness as well as the time value of the money

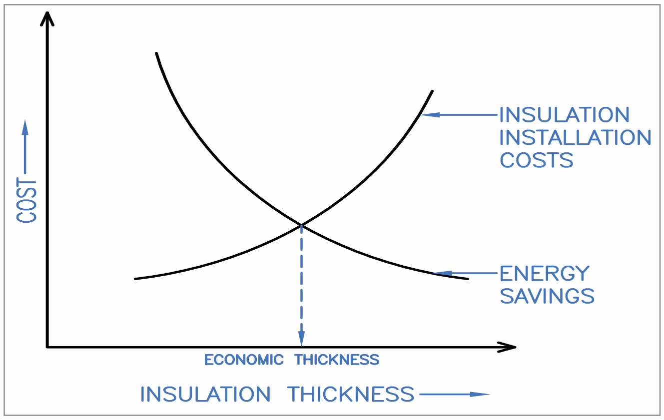

Figure 5: Economic insulation thickness

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spent on initial cost spread over the life of insulation in years.

This value is compared with the value of energy savings again spread over the life of insulation.

The installed costs increase with thickness, and the energy saved reduces with thickness.

The point of intersection of these two variables, as shown in *Figure 5*, gives the economic thickness value.

However, this aspect is generally not considered in HVAC installation as the overall aim is not only to reduce energy losses but also to avoid condensation. The latter criteria would generally require more thickness than that dictated by economics.

Vapour Barriers and Permeability

The importance of water vapour permeability in insulation cannot be overemphasized.

Water Vapour Transmission or Water Vapour Permeability of a material is a measure of its capability to allow moisture to pass through. Moisture intrusion into insulation can seriously degrade its thermal conductivity, and in many cases lead to complete insulation failure. Vapour barrier or retarder permeance is typically measured in units called perm, and is defined as the number of grains of moisture per square foot passing through in 1 hour across a pressure differential of 1 inch of mercury.

Perm-inch is a unit of water vapour permeability and is simply a product of the perm value of a material and its thickness in inches. Permeability is useful for specifying insulation materials.

Let us take the example of a chilled water outlet header of 600 mm dia. carrying water at 5.55°C in the layout considered before. At the pipe surface, the temperature will be 5.6°C progressing to 32°C at the insulation surface. At insulation thickness of 5 mm, the temperature will be 25°C. As this is below the dew point, condensation will occur and RH will be 100%.

The vapour pressure at the pipe surface (ambient conditions 35°C, 80% RH) will be 33.7 mm Hg (4499Pa). At 25°C and 100% RH, vapour pressure will be 23.8 mm Hg (3168Pa). Therefore, the vapour pressure difference is 1331Pa.

Considering elastomeric insulation material with permeability of 0.09 perm-inch, the water vapour transmitted per sq m of area over 1 year will be:

$$1331\text{Pa} \times 0.09\ \text{perm - inch} \times 1.453 \times 10^{-12}\ \frac{\text{Kg} \times 1\ \text{m}^2}{\text{m Pa sec}} \times \frac{365\ \text{days}}{0.025\ \text{m}} \times 24\ \text{hr} \times 3600\ \text{sec}$$

= 0.22 Kg

For the 600 mm dia. header in question (33 m length), water vapour transmitted will be @15 Kg per year.

Again, this is a hypothetical study and it is unlikely that the environmental conditions would remain that severe for 12 continuous months. However, it does indicate that even with a perfectly installed insulation system, water vapour can permeate into the insulation. The water vapour permeation can by reduced by increasing the thickness of insulation.

For open cell material like fibreglass, which has otherwise a very high permeability, the permeance of the FSK facing is considered for calculations. The FSK facing alone will not suffice, as the insulation joints, if not properly sealed, will allow moisture permeation.

The joints of FSK foil should be sealed with broad aluminium tape. The duct or pipe insulation should then be wrapped around with glass cloth and finished with 2 coats of anti fungal type vapour barrier compound (second coat after the first coat has dried). The glass cloth serves as reinforcing base. This gives a neat finish in addition to preventing moisture permeation.

No damage should occur to this barrier coat, during cladding or otherwise, for the insulation installation to have a long life. Vapour barriers are installed on the warm side of the insulation to stop the vapour from entering the insulation itself.

Cladding

All duct and pipe insulation is generally provided with aluminium cladding for mechanical and weather protection. The cladding is shaped to suit the insulation surface and matching beading is provided on all corners of adjacent pieces to ensure water tightness. Bands are used to hold the cladding in position. If screws or rivets are to be used, care should be taken to ensure that they do not pierce the vapour barrier on the insulation. Otherwise, this can be the starting point of insulation damage.

Suitable sealant should be used at cladding joints to give a weather proof installation. Use of stucco aluminium sheets gives a good finish to the cladding.

To avoid dents in cladding at support locations, pipe saddles of hard wood or hard rubber of appropriate dimensions should be used. The cladding should wrap round the support and should not be cut at these locations.

The cladding material itself can also affect insulation performance.

The emissivity values of various types of material have been indicated earlier in this article. Shiny aluminium sheets have lower emissivity and high reflectivity values. (For any material, emissivity + reflectivity = 1). So, most of the heat incident on the pipe is reflected back and very little is emitted to the air space in contact with the insulation. This makes the insulation surface cooler and the heat loss from that section of pipe is lower. However, this could lead to condensation possibilities.

It is therefore better to use higher emissivity material so that the surface temperature is closer to ambient and condensation risk is minimised. The increase in heat loss on account of higher emissivity can be nullified by using thicker insulation.

Conclusion

On an overall basis, insulation forms a very small percentage of the total HVAC project cost – about 1.5%. Poor application during design stage can lead to avoidable problems post commissioning. Rectification can cost more both in terms of time and effort.

Each project can have unique features in terms of installation. It is advisable to invest adequate time studying the system and carry out basic calculations for insulation to avoid rework later.

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