BLOWING (SUPPLY) AIR DIRECTLY INTO AN ICE STORAGE FOR COLD AIR DISTRIBUTION

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ABSTRACT

Using cold air distribution in thermal storage systems is one of the major areas of energy conservation in HVAC systems.
In this paper, blowing of supply air directly through an ice storage system has been investigated.
Heat transfer calculations have been performed and by obtaining the leaving air temperature. A comparison between this system and the conventional air handling units using cold water of ice storage, as working fluid, has been considered.
In other words, this paper concerns the possibility of blowing supply air directly into a charged ice storage tank, in order to omit water circuit between the storage and the air-handling unit and its coils as well. In this method after ice has been formed in the storage, water is evacuated from the storage and air is blown through the tank.
It is found that for the same amount of made ice, due to omitting the water circuit (and coils of air-handling unit), a considerable lower temperature could be achieved. This results in reduced required air flow, size of ductworks, fan energy consumption and generated noise in supply air distribution.
It is also shown that escaped air is reduced by decreasing free bypass area inside the storage which leads to increase the efficiency of the method.

INTRODUCTION

Using cool thermal storage has done great improvements in reducing electricity peak demand and saving energy in many countries.
Cold air distribution has been widely used in many applications in order to reduce the required air flow, size and cost of ductworks, fan energy consumption to a great deal and after all, reducing the amount of generated noise while the supply air is being distributed, especially in places where noise should not exceed a specified critical limit according to standards. (i.e. hospitals, libraries, convention centers and ...).
The point here is that colder air distribution would cause lower room relative humidity, because colder air can carry out less moisture.
It has been observed in recent DOE (Department Of Energy) studies that occupants in cold air systems will raise thermostat settings by 2 to 3°F because the lower humidity levels make people feel more comfortable even with a slightly higher temperature.
Designers of HVAC systems are aware of considerable effect of 2 to 3°F increasing the design temperature, on cooling costs.
After all, lower relative humidity would have less damaging effects on furniture, especially on wood products.

Various aspects and difficulties

In systems which do not use thermal storage, [entering and leaving water temperatures to the coils are consequently 45°F (7°C) and 55°F (13°C)] **which allow supply air to be generated at temperatures of about 55°F (13°C).** [1] [And even colder to 52°F (11°C).]
In storage systems (ice storage), storage discharge temperature of 34 to 36°F (1 to 3°C) [entering and leaving temperatures of about 35°F (2°C) and 45°F (7°C) to the coils] **allow supply air to be generated at temperatures of about 42°F (6°C)**. [1]
Reduction of about 10 to 13°F (6 to 7°C) in supply air temperature reduces the required air flow (CFM) so that the fan energy consumption reduces by 40%.
The reason for possibility of reducing the required CFM is that in order to provide a place with enough cooling, temperature difference between the supply air temperature and room air temperature can generate the required cooling load.

Here is a little example:

$$Q = 1.08\ CFM\ (T_2 - T_1)$$

$$30\ \text{ton} * 12000\ \text{Btu/hr.ton} = 1.08\ (CFM)\ (80°F - 55°F)$$
$$\Rightarrow CFM = \mathbf{13300} \quad (1)$$

$$30\ \text{ton} * 12000\ \text{Btu/hr.ton} = 1.08\ (CFM)\ (80°F - 42°F)$$
$$\Rightarrow CFM = \mathbf{8770} \quad (2)$$

$$30\ \text{ton} * 12000\ \text{Btu/hr.ton} = 1.08\ (CFM)\ (80°F - 32°F)$$
$$\Rightarrow CFM = \mathbf{6940} \quad (3)$$

So the lower the supply temperature is, the less air flow (CFM) is required for the same total cooling load.

But making lower supply temperature is not as easy as it seems at first. Generating lower supply temperature would require lower coil inlet and outlet temperatures and this low temperature water or Ethylene Glycol solution, (In storage or non-storage system) would need a chiller working at low evaporating temperatures. It is known that lowering the evaporating temperature of a chiller would be gained in expense of total efficiency or COP (Coefficient Of Performance) of a chiller. In other words, with lower chiller COP's, much energy (kW) is consumed for the same amount of cooling load. This can neutralize the cost reduction obtained of cold air distribution effect. So the lower supply air temperature is required, the lower water or Glycol temperature is needed. In storage systems this water or Glycol temperature is limited to discharge temperature of the storage. (For ice storage systems, ordinarily between 34 to 36°F (1 to 3°C).) (As mentioned before, a 42°F (6°C) supply air could be generated by these temperatures in coils.)

In this article we are looking for a method to lower supply air temperature even below 42°F (6°C), but the main point here is that we are looking for a method of reaching this temperature without lowering chiller COP.

MAIN IDEA

Let us limit our discussion to a special technology of storage systems, "External melt Ice-on-coil system". In this system, water flows outside tubes which are covered with a layer of ice for an evaporating refrigerant or a Glycol solution is passing inside tubes. The cooled water with temperatures of 34 to 36°F (1 to 3°C) is pumped to air-handling unit coils to generate cold supply air at 42°F (6°C). (Figure 1)

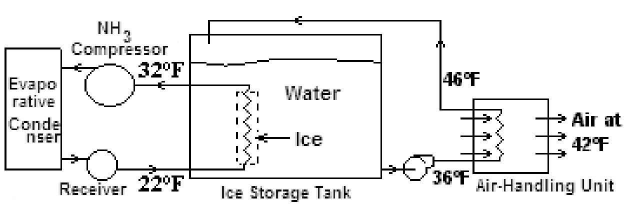


Figure 1. Schematic diagram of components and temperatures in a storage system with cold air distribution

(The diagram of figure 1 is a schematic configuration of components. More components may be added to this system, or piping configuration may change in actual system. But as long as emphasis is on something else here, more complex configurations are not discussed.)

The water circulated here between the storage and air-handling unit is only a working fluid. Considering pumping effects, there is a 2 to 4°F temperature loss in water when reaching the coils of air-handling unit.

The idea is Instead of passing water, it is better to send the supply air directly into the ice storage to be cooled.

When charging phase of ice storage is finished, ice has been formed on the tubes of the tank to the desired thickness. So, if the remained water is taken out and the tank has been designed so that air can be blown between tubes, cold air can be easily maintained at low temperatures of about 32°F (0°C).

(Tanks in these applications are all atmospheric tanks.)

In figure 2, only 5 branches of about 50 branches, which are sat beside each other, have been shown. (Just for having a better vision).

The water inside the tank can be easily stored in an ordinary tank to be used at the next time of charging the storage (Figure 4). Moreover because it is a cold water at an approximately 34 to 36°F (1 to 3°C) temperature, it can be used for other cooling purposes. And if adding a pump, the tank can be used as a fire-protection water tank in emergencies.

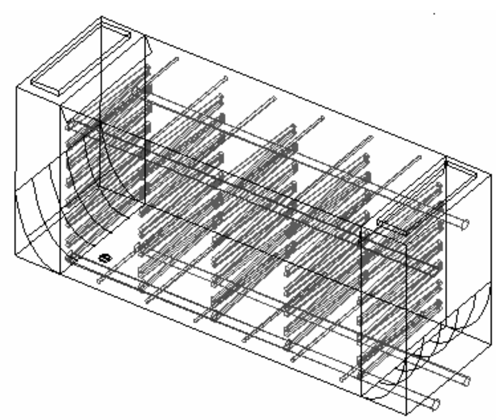


Figure 2. External melt ice storage tank with a special inlet and outlet section to guide air flow in discharge time.

Air can be blown till the whole formed ice in the storage is over. It is obvious that in the previous case - passing water through the coils, cooling could be offered till all of ice in the tank was over. In the present case, blowing air into charged storage can be continued till all of ice in the storage is over. So in both cases the same amount of cooling can be provided and it only depends on (limited to) the amount of ice formed.

Moreover in blowing air method, there are no heat transfer losses by the water as a working fluid where air directly rejects its heat to the ice.

After all, there is no need to pump a working fluid (usually water) to cooling coils of air handler unit. Cooling coil section of air-handling unit will be omitted too.

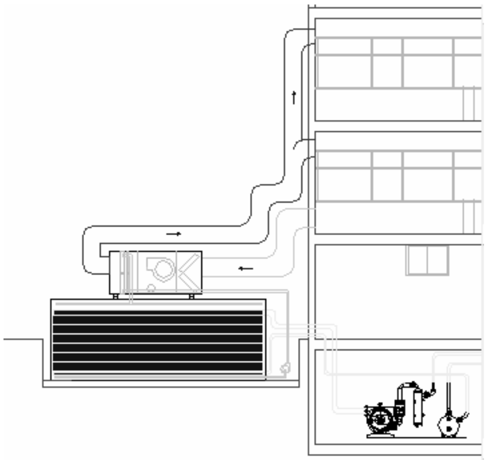


Figure 3. An ordinary external melt ice-on coil storage with an air-handling unit and its chiller. (Common configuration)

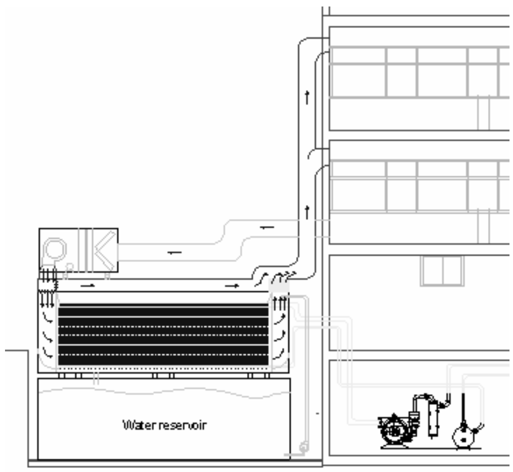


Figure 4. A new method of blowing air directly into the charged ice storage while its water has been evacuated into another tank (Bottom tank).

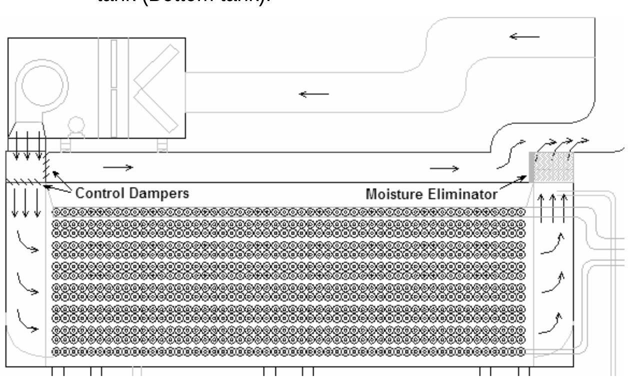


Figure 5. Detail of tank configuration.

With blowing air directly into the storage, controlling of cooling load offered by the air-handling unit is the problem. In previous manner, a three-way valve on the water passage was used to control the flow rate of cold water entering to the air-handling unit cooling coil. This time, the flow rate of air should be controlled, so a bypass damper will be set at the entrance of air to the storage. With installing motor dampers, the final mixture of the air through the iced coils and bypass channel will be at desired temperature (Figure 5).

Because air carries tiny droplets and particles of water (melted ice) with itself, as shown in figure 5, an eliminator has been place at air exit.

Calculations of heat transfer is necessary to find out that whether by omitting the coils and using these fin-less tubes covered with ice as heat transfer surfaces, it is possible to have a cold enough supply air or not.

HEAT TRANSFER CALCULATIONS

The problem is passing a certain amount of air through a bank of tubes covered with ice. The tubes that have been covered with ice do not have a complete cylindrical geometry and there are thicker and thinner points on the tubes but the average summary of the whole geometry of ice formed around the tubes can be assumed cylindrical by a valid approximation. It might have impact on the final results but as long as the overall results do not have or do not need such a precision, this impact can be negligible. This will be more reasonable after observing calculations and results.

The problem is modeled as a flowing air stream though the ice covered tubes.

A sample bank of tubes is to be analyzed. (This used to be a practical size of a storage bank performed with ordinary external melt ice-on-coil method. i.e. water used as the working fluid between storage and coils). It is desired to analyze if it is possible to change this storage to be used directly with air, with the same size and capacity of storage, or not.

(As the considered case was desired to be a practical and real one, a case study of a real storage system, with the same capacities and loads has been used.)

It is assumed that for a special place, load calculations have been performed and it was obtained that <u>272 ton-hour cooling capacity is required.</u>

When such an amount of cooling is required, there should be a greater amount of ice, to overcome the losses and eventually to include a safety margin.

So a factor of 1.2 is assumed for this purpose:

$$272 \text{ ton-hour} * 1.2 = 326 \text{ ton-hour}$$

$$326 \text{ ton-hour} = 3{,}921{,}000 \text{ Btu} = 4{,}127{,}160{,}000 \text{ J}$$

$$\left.\begin{array}{l}\text{Required energy} = 4{,}127{,}160{,}000 \text{ J} \\ \text{Latent heat of Ice} = 335000 \text{ J/kg}\end{array}\right\}$$

$$4{,}127{,}160{,}000 \text{ J} \;/\; 335{,}000 \text{ J/kg} = 12320 \text{ kg of ice}$$

By forming ice layer with 35mm thickness (a common design) on 1" diameter tubes, the whole length required for the storage of required capacity would be:

1 inch nominal diameter tube:

(Outer diameter =1.315") (Inner diameter = 1.049")

$$A = \pi/4 \left((4.071)^2 - (1.315)^2 \right) = 11.658 \text{ in.}^2 = 0.00752 \text{m}^2$$

$$M_{ice} = \rho_{ice} A L = (920 \text{ kg/m}^3)(0.00752 \text{ m}^2)(L) = 12300 \text{ kg}$$

$$\Longrightarrow L = 1778 \text{ m}$$

For making 1778m tube length, different arrangements can be used. A configuration of 54 branch of 33 meter length is selected for this. For making a 33 meter branch, 14 branches of 2.35 meters are joined by 90° bents and joints. (Figure 6)

54 * 14 * 2.35 m = 1776.6 m

To prevent the tubes from having contact with each other in charging time, a gap of 0.25”(6 mm) free space is assumed between them in fully charged condition.
The whole bank configuration can be observed in figures 2 and 5.
For heat transfer calculation, modeling of bank is carried out into two parts; Horizontal tubes and slopped tubes.

Part 1 Horizontal Tubes (Tubes without slope):
In horizontal tubes modeling, there would be little space between fully charged tubes and as shown in figure 6, there is only 6.6 mm space (design distance) between charged tubes. By passing air through this free space, ice melts continuously and the diameter of the iced tubes will begin to decrease from 10.34 cm (Design fully charged diameter).

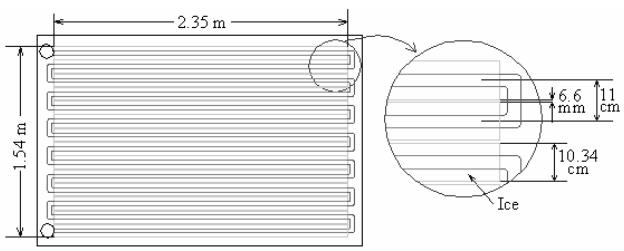
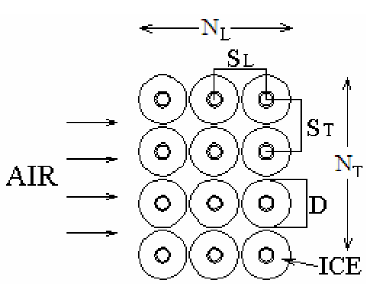


Figure 6. A branch of ice storage with horizontal tubes in fully charged condition.

Tube bank:
Length = 6 m, Height = 2.35 m, Width = 1.54 m

$D = 0.1034\,m \qquad Air\ Flow = 7000\ CFM$

$T_{Air-inlet} = 27\,°C \qquad T_s = 0\,°C$



$S_T = 0.11$

$S_L = 0.11$

$N_T = 14$

$N_L = 54$

$N = N_T \times N_L$

Figure 7. Tube bank configuration.

Calculations:
All the parameters with mean subscript have been calculated at mean temperature ((27+0)/2 = 13°C).

$$A_{face} = Tube\,bank\ width\ {*}Tube\,bank\ height = 3.619\ m^2$$

$$V_{face} = \left(\frac{1}{60}\frac{CFM}{(3.281)^3}\right)\Big/ A_{face} = 0.913\ m/s$$

$$Continuity: V_{face} S_T = V_{max}\left(S_T - 2(\frac{D}{2})\right) \qquad V_{max} = 15.212\ m/s$$

$$\mathrm{Re}_{D\_max} = V_{max} D / \upsilon_{mean} = 1.074 \times 10^5$$

Now using Zhukauskas correlation for obtaining Nu number for a bank of tubes: (Ref. 2)

$$Nu_D = C_1\,\mathrm{Re}_{D\_max}{}^{m}\,\mathrm{Pr}_{mean}{}^{0.36}\,\mathrm{Pr}_{mean}/\mathrm{Pr}_{surface}$$

This correlation is valid when

$$\begin{cases} N_L \ge 20 \\ 0.7 < \mathrm{Pr}_{mean} < 500 \\ 1000 < \mathrm{Re}_{D\_max} < 2\times 10^6 \end{cases}$$

which are all satisfied with our problem.
According to the table offered for this correlation, C_1 and m for this range of Re number is consequently (0.27), and (0.63). (Ref. 2)

$$Nu_D = 352.546$$

$$h = Nu_D \frac{k_{mean}}{D} \qquad h = 85.92 \qquad \frac{W}{m^2 K}$$

using the relation offered in Ref 2.:

$$T_{Air\_Out} = T_s - (T_s - T_{Air\_Inlet})\left[\exp\left(\frac{-\pi D N h}{\rho_{mean} V_{face} N_T S_T C_{p\_mean}}\right)\right]$$

$$\boxed{T_{Air\_out} = 1.429 \times 10^{-4}\ \text{°C} \quad (\approx 0\ \text{°C})}$$

It is seen that the final outlet air temperature reaches 0°C (32°F) when storage is fully charged. This can provide the place with (0°C) 32°F cold air instead of (6°C) 42°F air.

This reduction in air temperature has been obtained without lowering the COP of refrigeration cycle because the evaporating temperature is constant and the same amount of ice, relative to previous manner (water as working fluid), has been built. Moreover no losses occur by the water as the working fluid, between the storage and air handling unit, and air directly rejects its heat to the ice bank.
Attention that there is no need to calculate the amount of heat transfer here, because as shown in relations 1 to 3 in page 2, if 30 tons of refrigeration is assumed to be needed for this place, a storage of 270 ton-hour capacity would provide the place with enough cooling for about 9 hours (if 30 tons of refrigeration is to be needed continuously) and according to the relations 1 to 3 in page 2, it makes no difference that the load is supplied by 9000 CFM of 42°F air in common method (with chilled water of ice storage pumped to cooling coils) or 7000 CFM of 32°F (with blowing air directly into the storage). However, same amount of cooling could be offered and this total required cooling load (270 ton-hour), depends only on the amount of ice built, which is the same in both manners (12300 kg).
The point here is that the rate of instant cooling can be offered by this new method, which directly depends on

difference between entering and leaving temperature. Entering temperature is assumed to be constant at 80°F (27°C) so the only main parameter is the leaving air temperature. So it should be checked that how the continuous ice melt and storage discharge would affect the leaving air temperature.

The ice on tubes, located at the place where air enters the bank, is melted first. Calculations show that (the same method mentioned above) in fully charged situation, after air passes 15 rows of 54 rows, its temperature drops below 1°C. After this, it is obvious that there is little heat transfer between this 1°C air and 0°C ice on the tubes. So the ice formed on the rear rows does not melt till the ice on primary rows melts.

The velocity and the Reynolds number of passing air reduces as the primary rows get thinner in diameter. Besides, the reduction in the surface of heat transfer results in reduction of overall heat transfer. For the reasons outlined above, the air reaches the rear tube with higher temperature. At that time, the rear tubes begin to melt too. So the primary tubes melt sooner than the rear tubes.

With a simulation of thinner primary tubes and thicker rear tubes, as steadily increasing thickness through the bank length, calculation of air outlet temperature has been done with the same method mentioned above in different situations of the bank.

In this analysis, the temperature is obtained after each two rows and the air entering the remainder of the tubes has been considered by the modified new temperature. In other words every two rows have been considered with a different D, which has been defined in regard to local Nu and h at that couple of rows with the assumption of steady increasing D. The results can be observed in table 1. ("Horizontal Tubes" column)

Table 1. Air Leaving Temperature for three bank configurations

Percent of Ice Melted (%)	Air Leaving Temperature (°C)		
	(Horzontal Tubes)	(Two branches)	(Four branches)
0	0 (0.0001)	1.2	0.2
10	0 (0.0005)	1.6	0.4
20	0 (0.01)	1.9	0.7
30	0 (0.02)	2.5	1.1
40	0.1	3.0	1.5
50	0.2	4.5	2.3
60	0.5	5.5	3.7
70	1.5	8.2	6.2
80	4	11.6	10.4
90	10	15.5	15.8
100	27	27	27

As it can be seen in figure 8, by storage discharge and reduction in number of iced tubes in air passage, temperature of air increases. This is just like the previous manner of ice storage system (water used as working fluid). In that case, also, the temperature of leaving water increases as it reaches the end of discharge process. So this is not a disadvantage of this new method.

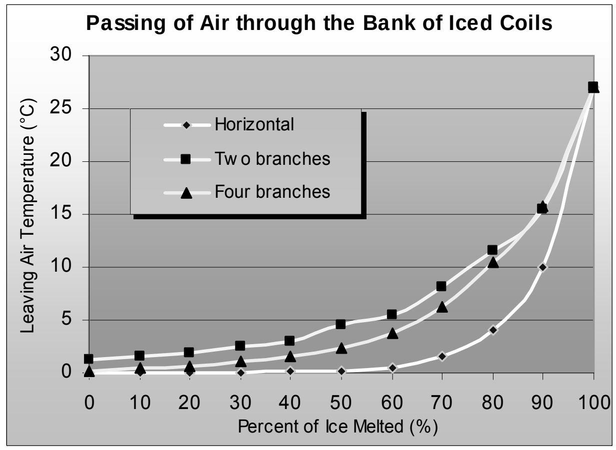


Figure 8. Air leaving temperature passing through the bank of iced tubes. Air entering temperature is 27°C (80°F)

The main point here is that the designer should consider this increase in designing and time scheduling of ice storage against the load profile of the project. In other words, required load should be supplied before the storage reaches the end points of its reservoir.

Although this arrangement with obtained results seems ideal for heat transfer purpose, but because the tubes are so near to each other (Figure 6) and there is little free space for passing air in fully charged condition, the air pressure drops dramatically through these ice coils. According to calculations based on method mentioned in ref. [2], passing a fluid through a bank of tubes, there would be about 8 inch of water pressure drop for air passing through coils in fully charged condition. Of course with melting of ice, even in primary stages, this amount of pressure drop decreases immediately; but it would be impossible to provide this static head pressure in primary stages for a commercial air-handling unit, which is also expected to reduce its power consumption (by using cold air distribution) too.

A solution is to be offered as follows.

In the external melt ice-on-coil storage, the tubes are not exactly horizontal and they have a little slope in order to lead the evaporated refrigerant, in its way up to the end point of evaporator.

Different slopes may be defined for an evaporator but here, it is considered to study a 4% slope (4.25%) in all horizontal tubes.

Part 2 (Tubes with 4% slope)

a) Using two branches (Two entrance headers):

In this part all of tubes have a slope of 4.25% upward to lead the vapor.

Because the branches are 2.35m long, every tube should rise 10cm. But this would create a great free space between tubes and by increasing bypass area and reducing velocity and Reynolds number, Nusselt number (Nu) decreases and so it has a negative effect on overall heat transfer.

To compensate the reduction of overall heat transfer, it is better to divide every long horizontal branch into two parts. In other words, instead of 1 * 2.35m tube (figure 4), it is better to use 2 * 1.17m tubes (figure 9).
(Tube bank width = 2.35 m) (the same)
(Tube bank height = 2.24m)

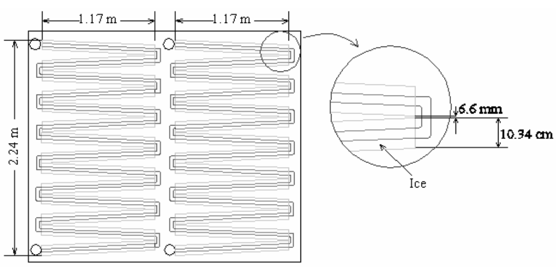


Figure 9. Bank of coils when two branches are used

The method offered for heat transfer calculation of a bank of tubes, are for parallel tubes but here the distance between tubes is not constant. For this reason there should be a simulation to approximate the behavior of air stream passing through this tube arrangement.
Based on the same free area between tubes, this arrangement can be simulated by a parallel bank of tubes with S_T = 16cm (mean S_T) (i. e. center to center of two different tubes). With this assumption the velocity of air passing through tubes would be the same, (as slopped tubes), and with having the same Re number, by a valid approximation it can be deduced that the Nu number and overall heat transfer would be approximately equal to the real slopped tube bank.
With the new arrangement, calculations would be repeated just like before with this new value of S_T and "Tube bank height". The other parameters are the same as before. Because the calculation process is just like before, as for horizontal tubes, only the results are mentioned here.

$$\mathrm{Re}_{D\_\max} = \frac{V_{\max} D}{\nu_{mean}} = 1.253 \times 10^4$$

$$Nu_D = 91.044 \qquad h = 22.189 \ \text{W/m²K}$$

$$\boxed{T_{Air\_out} = 1.171 \ \text{°C}}$$

This time, air temperature reaches 1.2°C in fully charged condition. This increase in air outlet temperature is only because of slopes that have provided free areas between tubes which results in reduction of Re, Nu number and overall heat transfer.
With discharging of storage just like described before, primary rows begin to melt first. With a simulation of thinner primary tubes and thicker rear tubes, steadily increasing thickness through the bank length, calculation of air outlet temperature has been performed with the same method mentioned above. The results can be observed in table 1, ("Two branches" column) and in figure 8.

b) Using four branches (Four entrance headers):

As mentioned before, slopping of tubes has a negative effect on overall heat transfer, as it makes additional free spaces between tubes. The less this free spaces are, the better heat transfer will occur. For this reason, it is better to divide every long horizontal branch into moreover parts. So instead of 2 * 1.17m tube (part a) it is better to use 4 * 0.588m tubes.(figure 10)
(Tube bank width = the same 2.35m)
(Tube bank height = 1.94m) (S_T = 0.1334m) (mean S_T)

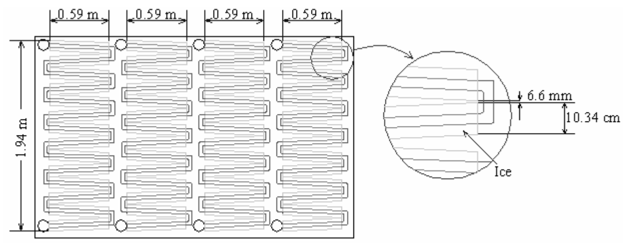


Figure 10. Bank of coils when four branches are used

Other conditions are just like part (a).
With the same method, calculation has been performed and result is shown in table 1 ("Four branches" column). The plot can be observed in figure 8.
It is seen that the results are not as satisfactory as the results of the horizontal (without slope) tubes, but show improvements to results of part (a).
The final result is: "It would be better to divide the long slopped branches into smaller pieces as much as possible." This will reduce free bypass area between tubes and will lead to a better overall heat transfer.
Another point which should be mentioned here is the calculated air flow (CFM) which is designed upon leaving air temperature of 0°C (32°F) in (3) (page 2).
In real storage, as it can be seen in figure 8 (four branches), leaving air temperature is about 0°C when the storage is fully charged and by discharging of storage, this temperature will increase. To correct this deviation, one way is to increase air CFM 10% or 20% to provide the place with necessary cooling load. Calculations show that increasing CFM by 10% or 20% will not make a great impact on the leaving air temperature or at least the effect of increasing CFM is much more than effect of increasing temperature due to this CFM increase. So the required cooling load could be obtained. Moreover in so many applications, especially for residential applications which are the main part of cold air distribution purpose, there is no need to be worried about increasing leaving air temperature. Because by discharging of storage and increasing air temperature, the peak of cooling load will be passed and the need for cooling load decreases as the time goes on. So it depends on the design engineer

to schedule the storage in such a time table that fully charged storage is to be as much near as possible to the maximum cooling load demand of the place.
Checking up the pressure drop of air passing through this arrangement of four branches shows that even in fully charge condition it is about 0.5 inch of water and moreover it decreases immediately as the ice melts which is completely satisfactory.

CONCLUSIONS

It is described that the following improvements can be achieved by this new method of passing air directly through a series of ice covered tubes of an external-melt ice storage, designed for this purpose.

1. Reduced CFM
   CFM can be reduced about 20% because of reduction of supply air temperature from approximately 42°F to 32°F. It has many benefits:
   a. Reduced Fan Power consumption
   Reduced CFM would cause a great improvement in fan energy consumption. A pressure drop of about 0.5 inch water would be added to the system but a 0.5 inch water pressure drop would be omitted from the system because of omitting the air handling unit coil. Moreover it is obvious that air flow has a greater effect on fan energy consumption than the static pressure. (Air flow is proportional to the fan power consumption with power 3 while pressure drop has power 2.
   b. Reduced noise (according to reduced CFM)
   c. Reduced size of ductworks
2. Colder supply air would lead to lower room relative humidity and a more comfortable situation.
3. Omitting the losses occurred when using water as a working fluid. (Water circulation pump itself, its energy consumption and heat transfer losses of additional heat exchangers.)
4. No additional cost with this method except a space for a free tank holding water temporarily.
5. Having a fire protection tank ready at hand.

It can be concluded that in the cold air distribution purposes, it is possible to omit the working fluid (usually water) between storage and coils of air-handling unit. By so, the heat transfer losses between the storage and supply air will become minimum and air directly rejects its heat to the ice made on the tubes; moreover cold air at temperature of about 0°C(32°F) could be obtained. By the reduction of required air flow (CFM) in this case, a large amount of energy could be saved.
Another point is that it was determined that in the case of slopped tubes it would be better to break the long branches to smaller pieces. This will reduce the free bypass area and would enhance the overall heat transfer.

NOMENCLATURE

°C	Temperature, Degrees Celsius
°F	Temperature, Degrees Fahrenheit
A	Area, m²
A_{face}	Bank total sectional area, m²
C_1	correlation coefficient, dimensionless
C_{p_mean}	Air specific heat at mean temperature J/kgK
CFM	Air flow rate, cubic foot per minute
COP	Cycle coefficient of performance
D	Tube outer diameter covered with ice, m
h	convection heat transfer coefficient, W/m²K
k_{mean}	Air thermal conductivity at mean temperature, W/mK
L	Length, m
M_{ice}	mass of ice, kg
N_L, N_T	Number of tubes in longitudinal and transverse direction (figure 7)
N	Total number of tubes in the bank
Nu_D	Nusselt number of tube bank
$\Pr_{mean}$	Air Prandtl number at mean temperature
$\Pr_{surface}$	Air Prandtl number at surface temperature
Q	Power (cooling load), kBtu/hr (or ton of refrigeration) (12000 Btu/hr=1 ton)
$\mathrm{Re}_{D\_\max}$	Reynolds number with maximum velocity between tubes
S_L, S_T	longitudinal and transverse pitch of a tube bank, m (figure 7)
$T_{Air-inlet}$	Air entering temperature to the bank, °C
$T_{Air-out}$	Air leaving temperature (supply temp.), °C
T_s	Surface temperature (ice), °C
V_{face}	Air velocity at bank entrance, m/s
$V_{\max}$	Maximum air velocity between tubes, m/s

Greek Letters

ρ_{ice}	Ice density, kg/m³
ρ_{mean}	Air density at mean temperature, kg/m³
υ_{mean}	Dynamic viscosity at mean temperature, m²/s

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