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OPTIMIZING THE DESIGN OF BARE TUBE COOLING COIL IN COLD STORAGE APPLICATION

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ABSTRACT
In this paper, the effects of variations in coil face velocity, coil array and number of rows on the rate of heat transfer, power consumption by the fan and the ratio of heat transfer and fan power are discussed for a cooling coil used in cold storage applications. A bare (unfinned) tube cooling coil is considered in this study The ratio of rate of heat transfer and fan power is formulated in a non–dimensional form. A wide range of tube spacing, defined in terms of longitudinal pitch and transverse pitch and also a wide range of face velocities are considered for the study. The highest value of this non-dimensional ratio indicates the optimum coil array that gives maximum heat transfer per unit fan power consumption. For inline tube arrangement it is observed that four to eight transverse rows give the maximum of this ratio depending upon the coil array. The optimum number of rows decreases as the compactness of the coil increases.

Keywords: Cooling coil, Inline arrangement, Heat transfer, Fan power.

1. INTRODUCTION
In most cases, design of cooling coil for any air -cooling application is done keeping in mind the desired compactness of the system. The power consumption by the fan to make the air flow across the cooling coil has seldom been a criterion for the design. For cold storage application where space usually does not impose any limitation, the power consumption of the fans should be one of the major criteria for designing the cooling coils as the fans are running continuously. The fan, which is placed after the cooling coil, induces the air across the cooling coil and the cooled air is fed to the storage space through ducting. The power consumption of the fan is directly proportional to the pressure drop across the cooling coil and the pressure drop depends upon the face velocity and coil array. For example, a 3000 Metric ton capacity potato cold storage where the compressor consumes about 40 kW of power, the standard cooling coil fan consumes an almost equivalent amount of power. The high power consumption of the fan is due to the large static pressure drop of the air stream across the cooling coil. Suitable design of the cooling coil array may reduce the pressure drop of the air stream substantially resulting in reduced fan power consumption for the same refrigerating capacity.

The simplest cooling coil is a bare (unfinned) tube cooling coil made from standard size tubes. Heat transfer coefficient and air-side pressure drop can be estimated for various tube arrays using available heat transfer and flow correlations [1,2]. For every arrangement of the tube array, an optimum number of transverse tube rows is expected to give maximum heat transfer per unit fan power consumption for a particular air face velocity. The tube diameter, the free stream velocity, the tube array and the Reynold's number characterize the heat transfer and fluid flow past a tube bundle. .

2. MATHEMATICAL FORMULATION.
In this section, the heat transfer, fan power and their non dimensional ratio for the flow of air through the unfinned bare tube cooling coil are formulated. In this study the refrigerant considered to be evaporating inside the tube is ammonia (NH_3).

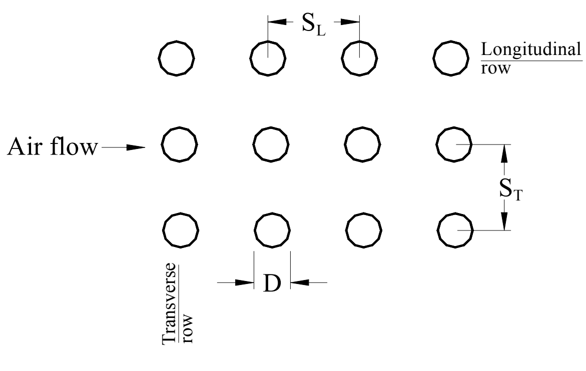


Fig 1. In-line arrangement of a bare tube cooling coil

In cold storage application, the usual process is the cross flow of air over a tube bundle. The fan is situated after the cooling coil.

The geometry of tube bank is characterized by the transverse pitch (S_T) and the longitudinal pitch (S_L) between the tube centers. For this study an inline configuration is considered as shown in fig. 1.

2.1 External Heat Transfer Coefficient.

To determine the external heat transfer coefficient the inline tube array is considered The Reynolds number for the flow across the tube bank is based on the maximum velocity occurs inside the tube bundle. For this configuration, the maximum velocity occurs on the minimum free flow area available for air flow.

In this case, the maximum velocity is given by

$$V_{\max} = V \frac{S_T}{S_T - D} \tag{1}$$

Where, V is the face velocity of air at coil entrance.

The Reynolds number based on the maximum air velocity occurring within the tube bank is given by

$$\mathrm{Re} = \frac{\rho\, V_{\max}\, D}{\mu} \tag{2}$$

Different correlations have been proposed and used by different investigators [3,4,5,6] for evaluating the heat transfer coefficient for flows over tube bundles.

Grimison [3] has proposed a correlation for Nusselt number for a bare tube (unfinned) bank with 10 or more transverse rows as follows:

$$Nu = 1.13\ C_0 (\mathrm{Re})^n (\mathrm{Pr})^{\frac{1}{3}} \tag{3}$$

$$\begin{bmatrix} N \geq 10 \\ 2000 < \mathrm{Re} < 40000 \\ \mathrm{Pr} \geq 0.7 \end{bmatrix}$$

Where, the values of the constants C_0 and n have been suggested by Grimison [3] for different tube array.

Zukaukas [4] has proposed another correlation to estimate Nusselt number for a unfinned bare tube bank with 20 or more number of transverse rows as:

$$Nu = C_2 (\mathrm{Re})^m (\mathrm{Pr})^{0.36} \tag{4}$$

$$\begin{bmatrix} 0.7 \text{ p } \mathrm{Pr} \text{ p } 0.7 \\ 10 < \mathrm{Re} < 10^6 \end{bmatrix}$$

Where, for inline arrangements the value of constant C_2 varies from 0.21 to 0.8 and that of exponent m from 0.4 to 0.84, depending on the Reynolds number of the flow [4].

If the number of rows (N) is less than twenty, the Nusselt number can be found from the relation [3]:

$$Nu \big|_{(N \prec 20)} = C_3\ Nu \big|_{(N \succ 20)} \tag{5}$$

Where, the values of the correction factor C_3, as proposed by Zukaukas, are listed in table 1.

Table 1: Correction factor C_3 for an inline arrangement for Eqn.5 given by Zukaukas [4].

No of rows (N)	**Correction factor** (C_3)
2	0.8
4	0.9
6	0.94
8	0.96
10	0.97
12	0.98
14	0.985
16	0.99
18	0.995
20	1.0

The present analysis is based on Zukaukas model which is a more recent correlation developed as a result of comparison of the experimental data over a wide range of flow rates, Prandtle numbers and tube arrangements. The agreement with the experimental data was shown to be very good [1].

Now, Eq. (4) can be written as:

$$Nu = 0.27\ (\mathrm{Re})^{0.63} (\mathrm{Pr})^{0.36} \tag{6}$$

The external heat transfer coefficient is then expressed by

$$h_o = Nu \frac{K}{D} \tag{7}$$

2.2 Internal Heat Transfer Coefficient

The internal heat transfer coefficient for the flow of refrigerant inside the tube can be estimated using well established Dittus-Boelter [7] equation that gives the value of Nusselt number from the relation:

$$Nu_i = 0.023\ (\mathrm{Re})^{0.8} (\mathrm{Pr})^{0.4} \tag{8}$$

Where, the properties are evaluated at mean refrigerant temperature between the tube inlet and outlet [1].

The internal heat transfer coefficient is then given by

$$h_i = Nu_i \frac{K_R}{d} \qquad (9)$$

2.3 Overall Heat Transfer Coefficient.

The overall heat transfer coefficient U can be expressed in the following form.

$$\frac{1}{UA_0} = \frac{1}{h_0 A_0} + \frac{\ln \frac{D}{d}}{2\pi K_P L} + \frac{1}{h_i A_i} \qquad (10)$$

For a bare tube cooling coil, air flowing outside and the refrigerant evaporating inside, the overall heat transfer is given by

$$Q_S = m_a C_P (t_i - t_o) \qquad (11)$$

and

$$Q_S = UA_S \, LMTD \qquad (12)$$

Equating equations (11) and (12) we get

$$UA_S \, LMTD = m_a C_P (t_i - t_o) \qquad (13)$$

or

$$(t_i - t_o) = (t_i - t_e)(1 - e^{-\frac{UA_S}{m_a C_P}}) \qquad (14)$$

From Eq. (11) we get

$$Q_S = m_a C_P (t_i - t_e)(1 - e^{-\frac{UA_S}{m_a C_P}}) \qquad (15)$$

For inline tube bank, face area is given by

$$A_f = W \text{ x } L \qquad (16)$$

Now, the expression ,

$$\frac{UA_S}{m_a C_P} = \frac{UW \pi DLN}{\rho WLV_f C_P S_T} \qquad (17)$$

or

$$\frac{UA_S}{m_a C_P} = \frac{Nu \pi N}{\text{Re Pr}(S_T / D)} \qquad (18)$$

From equation (15) we get

$$Q_S = \rho V_f A_f C_P (t_i - t_e)(1 - e^{-\frac{Nu \pi N}{\text{Re Pr}(S_T / D)}}) \qquad (19)$$

Therefore, the rate of heat transfer per unit face area per unit temperature difference becomes

$$q = \rho V_f C_P (1 - e^{-\frac{Nu \pi N}{\text{Re Pr}(S_T / D)}}) \qquad (20)$$

2.4 Fan Power Consumption.

The pressure drop across the tube bank in cross flow is given by

$$\Delta P = f \frac{\rho^2 V_{max}^2}{2} N Z \qquad (21)$$

Where f is the friction factor and Z is the correction factor that depends on tube bank configuration and Z =1 for square tube arrangements. The values of f and Z for other arrangements are given in table 4 and 5 respectively.

Table 2: Values of friction factor f for an inline arrangement for Eq. (21) [4].

	Velocity (V)			
	1.5	2.0	2.5	3.0
S_L/D	**f**	**f**	**f**	**f**
1.5	0.35	0.36	0.35	0.35
2.0	0.24	0.25	0.25	0.25
2.5	0.18	0.18	0.18	0.18

Table 3: Values of correction factor Z for an inline arrangement for Eq. (21) [4].

	S_T/D			
	1.5	2.0	3.0	4.0
S_L/D	**Z**	**Z**	**Z**	**Z**
1.5	1.0	0.6	0.4	0.3
2.0	1.8	1.0	0.6	0.5
2.5	3.0	1.3	0.9	0.6

Now fan power is given by

$$P = AV_{max} \, \Delta P \, x10^{-3} / \eta \qquad (22)$$

Fan power per unit face area is given by:

$$p = V_{max} \, \Delta P \, x10^{-3} / \eta \qquad (23)$$

From equations (21) & (23) we get

$$p = \frac{\rho f V_{max}^3 \left(\frac{S_T}{S_T - D} \right)^2}{2\eta} N Z \, x10^{-3} \qquad (24)$$

To find out optimum rows of tube banks for which heat transfer rate per unit area per unit temperature difference (q) is maximum, the ratio of heat transfer (q) to fan power (p) in non-dimensional form can be obtained as

$$R = \frac{2}{fNZ} (1 - e^{-\frac{Nu \pi N}{\text{Re Pr}(S_T / D)}}) \qquad (25)$$

The equations (20) , (24) , and (25) give the heat transfer , fan power , and their non-dimensional ratio for a particular set of values for V, ST and SL.

3. RESULTS AND DISCUSSION

The performance of the cooling coil in terms of the non dimensional ratio of heat transfer to fan power is studied to ascertain the effect of variations in some design parameters such as number of transverse rows of cooling coil, face velocity of air across cooling coil and longitudinal and transverse pitch of the coil array.

In this study 25 mm nominal bore seamless steel pipe of outside diameter 26.9 mm and thickness 3.25 mm is considered. For longitudinal pitch S_L/D = 1.5, 2.0, 2.5 and for transverse pitch S_T/D = 1.5, 2.0, 3.0, 4.0 are chosen. Face velocities of 1.5 m/s, 2.0 m/s, 2.5 m/s and 3.0 m/s is considered. Properties related to heat transfer are found at mean bulk temperature.

Fig. 2 shows the variation of non-dimensional ratio with number of transverse rows for different face velocities for S_T/D = 3.0 and S_L/D = 1.5. The non dimensional ratio R optimizes at a particular value of N for a given face velocity as is apparent from fig. 2.

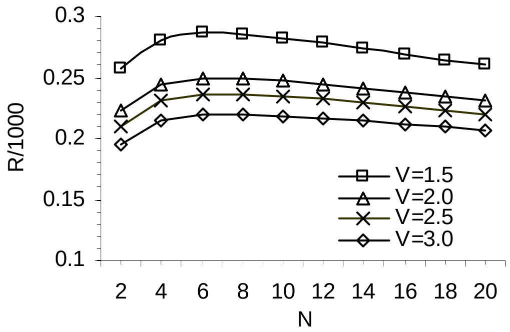


Fig 2. Variation of non dimensional ratio R with no of rows S_T/D = 3.0 and S_L/D = 1.5

Fig. 2 also suggests that lower face velocity gives higher value of this optimum R. This due to the fact that fan power is directly proportional to the square of the face velocity whereas heat transfer is proportional only to the face velocity. It may, therefore inferred that a lower face velocity (1.5 m/s in the current study) is always desirable

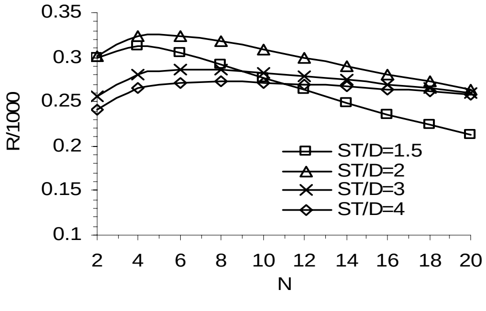


Fig 3. Variation of non dimensional ratio R with number of rows for V=1.5 and S_L/D = 1.5

Fig. 3 shows the effect of varying transverse pitch on R. for a given face velocity and longitudinal pitch. It is found that S_T/D = 2.0 gives the maximum value of R throughout the considered range of N. Values higher than 2.0 (S_T/D = 3.0, S_T/D = 4.0) tend to decrease R particularly at lower values of N .

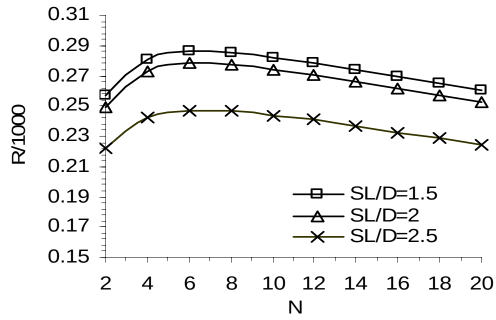


Fig 4. Variation of non dimensional ratio R with no of rows for V=1.5 and S_T/D = 3.0

S_T/D = 1.5 gives intensifying results showing high R at low N values and rapidly decreasing as N increases. This is due to the fact that compact coil increases pressure drop across coil more rapidly hence fan power as N increases.

In fig. 4 non dimensional ratio R is plotted with N for different S_L/D for velocity 1.5 m/s and S_T/D = 3.0 . The plot reveals that R increases as S_L/D decreases. For the assumed S_T/D value, R maximizes at about N=4 for all S_L/D values signifying that SL/D has no significant effect on the optimum number of rows provided other parameters remain unchanged.

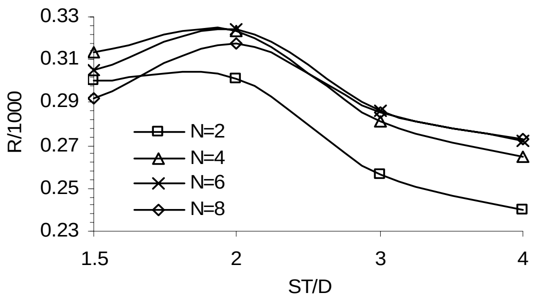


Fig (a)

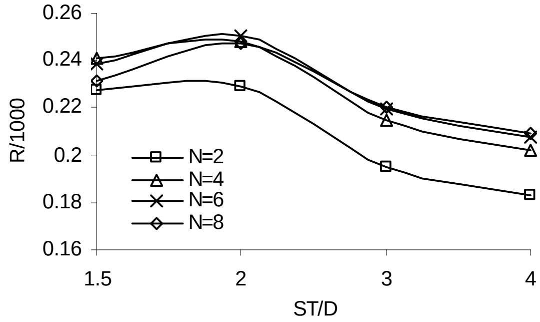
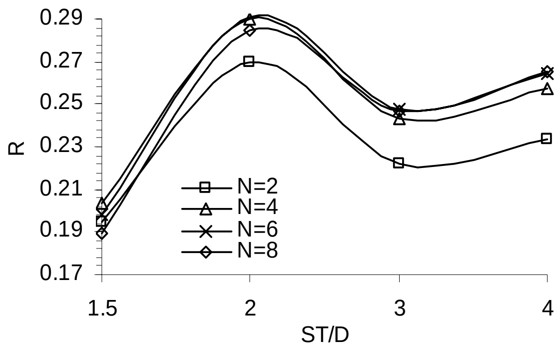


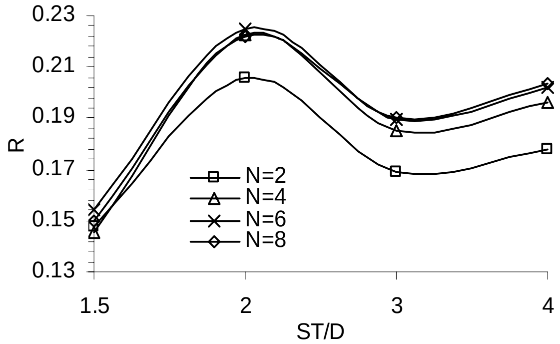
Fig (b)

Fig 5. Variation of non dimensional ratio R with transverse pitch for pitch S_T/D for (a) S_L/D =1.5, V=1.5 (b) S_L/D =1.5, V=3.0

In Fig.5, non dimensional ratio R is plotted against S_T/D for N=2, 4, 6 and 8 (since maximum R varies in between N=2, to N=8. as found in Figs. 3 and 4). For Fig.5(a) V=1.5 and for Fig.(b) V=3.0 while S_L/D value remains same for both of them (1.5).



(a)



(b)

Fig 6. Variation of non dimensional ratio R with transverse pitch for pitch S_T/D for (a) S_L/D =2.5, V=1.5 (b) S_L/D =2.5, V=3.0

It is seen that at lower S_T/D (<2), optimum value of N is 4 and at higher S_T/D (>2) both N=6 and N=8 give similar results showing maximum values of R. The trend is same for both V=1.5 and V=3.0. However, interesting results are seen if the analysis is carried out for a higher value of S_L/D, say 2.5. In Fig.6, non dimensional ratio R is plotted against S_T/D for N=2, 4, 6 and 8 with S_L/D fixed at 2.5 instead of 1.5 as considered earlier. Fig .6(a) is plotted for V=1.5 and Fig.(b), for V=3.0.It is observed that at S_T/D 1.5 and 3 ,R is comparatively less while it is more at S_T/D 2 and 4. for all number of rows. In both cases (a) and (b), S_T/D =2 gives the optimum value of R. For all values of S_T/D excepting 1.5, R is least for N=2 and at S_T/D=1.5, N does not have any significant effect on R.

4. CONCLUSIONS

In this paper the effect of variations in coil face velocity and tube array on heat transfer rate, power consumption of fan and their non-dimensional ratio is studied. From this study it is seen that the non-dimensional ratio of heat transfer rate to fan power consumption is greatly influenced by number of transverse rows, tube array and coil face velocities. It is seen that for every tube array and face velocity there is an optimum number of transverse rows which results in maximum heat transfer per unit fan power consumption. It is observed that lower face velocities give better heat transfer to fan power ratio for any tube array. It is also observed that for compact tube bundles (lower values of ST/D and SL/D) the optimum number of rows is 4 whereas for less compact bundles six to eight rows will give optimum performance. Compact cooling coils gives better heat transfer but consumes more fan power indicating higher running cost with low installing cost.

5. REFERENCES

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6. NOMENCLATURE

Symbol	Meaning	Unit
A_i	Inside surface area of the tube bank	(m^2)
A_0	Outside surface area of the tube	(m^2)
A_S	Total heat transfer surface area	(m^2)
C_p	Specific heat of air	(kj/kgK)
D	Outside diameter of tube	(m)
d	Iutside diameter of tube	(m)
f	Friction factor	
h	Heat Transfer co-efficient	(W/m^2K)
hi	Internal heat transfer co-efficient	(W/m^2K)
ho	External heat transfer co-efficient	(W/m^2K)
K	Thermal conductivity of air	(W/mK)
K_P	Thermal conductivity of tube material	(W/mK)
L	Length of tube	(m)
m_a	Mass flow rate of air	(kg/s)
N	Number of transverse rows	

Nu	Nusselt number	
P	Fan power	(kW)
p	Fan power per unit area	(kW/ m^2K)
Pr	Prandtle number	
Q	Rate of heat transfer	(kW)
q	Rate of heat transfer per unit area	
R	Non dimensional ratio of heat transfer to fan power	
Re	Reynolds number	
t_e	Evaporating temperature of refrigerant	(K)
t_i	Coil inlet temperature of air	(K)
t_o	Coil outlet temperature of air	(K)
V	fFace velocity of air	(m/s)
V_{max}	Maximum velocity of air	(m/s)
Z	Correction factor	
μ	Absolute viscosity	(Ns/m^2)
ρ	Density of air	(kg/m^3)