Optimum Dimensions and Operating Conditions for Finned Heat Exchanger Tubes

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An iterative procedure based on empirical relations is presented with which both the cost optimized dimensions and operating conditions for circular transverse finned heat exchanger tubes are simultaneously determined. The fin characteristics are found to be dependent on a large number of parameters and may vary significantly for different applications. Similarly optimum operating conditions including fluid velocities, deviate from those generally found in practice.

Nomenclature

A	area
C_p	specific heat
C	cost
D	fin or tube diameter
D_h	equivalent diameter
e	fin efficiency
Eu	Euler number
G	mass flow per unit area
I	modified Bessel function
i	rate of interest per unit time
K	modified Bessel function
n	number of fins per unit tube length
n_R	number of tube rows
Nu	Nusselt number
Pr	Prandtl number
p	pressure
$\triangle p$	static pressure drop
Q	heat transfer rate
R	fouling factor
Re	Reynolds number
S_L	longitudinal tube pitch
S_T	transverse tube pitch
s	fin spacing
T	temperature
t	fin thickness
V	volume
v	velocity
$\dot{W}$	power
α	heat transfer coefficient
ϑ	dynamic viscosity
λ	thermal conductivity
ρ	mass density
τ	time

Subscripts:

c	capital
e	energy
F	fluid
f	fin or free
i	inside tube
m	material
o	outside or based on outside tube diameter
s	static
T	thermal
t	tube
1	conditions before finned tube bank
2	conditions after finned tube bank

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Introduction

With the increasing reliance being placed on the atmosphere as a heat sink, the development of ever larger finned tube heat exchangers is inevitable. The potential dimensions of such units in air conditioning and refrigeration installations, power generating plants and in the process industries, justify attempts at optimizing not only the operating conditions at which these heat exchangers will function, but also the dimensions of the finned tubes employed under those particular conditions. It should be noted that these characteristics are interdependent. Previous attempts at optimizing the design of such heat exchangers were all based on commercially available tubing [1], [2], [3], [4]. [5] and often included limitations as in the case of Joyce [1] who does not take into consideration the cost of power or the number of operating hours per day, or specifications such as in the case of Schmiechen [2] who prescribed the frontal area of the heat exchanger and Kern [3] who prescribes the number of tube rows. The present optimization procedure takes into consideration all parameters that may affect the capital or operating cost of the heat exchanger, and does not, a priori, prescribe or limit any of them.

Analysis

A number of general relations for the heat transfer coefficient and pressure drop during flow across finned tube arrangements in terms of various geometrical and flow parameters are found in the literature [6], [7], [8], [9], [10]. Some of these are listed in a recent publication by Mircović [11] whose correlations for Nusselt and Euler numbers form the basis of this analysis.

$$Nu_o = \frac{\alpha_o D_{hT}}{\lambda_o} = 0{,}224\left(\frac{S_T - D_o}{D_o}\right)^{0,1}\left(\frac{D_o}{S_L - D_o}\right)^{0,15} \times \left[\frac{n(D - D_o)}{2(1 - nt)}\right]^{-0,25} Re_T^{0,662} Pr^{0,33} \quad (1)$$

and

$$Eu_o = \frac{\triangle p_{so}\rho_o}{G_o^2 n_R}\left[1 + \frac{G_o^2 n_R}{\triangle p_{so}\rho_o}\left(1 - \frac{\rho_{o1}}{\rho_{o2}}\right)\right]\left(1 - \frac{\triangle p_s}{2p_o}\right) = \frac{3{,}96}{Re_F^{0,31}}\left(\frac{S_T - D_o}{D_o}\right)^{0,14}\left(\frac{D_o}{S_L - D_o}\right)^{0,18}\left[\frac{n(D - D_o)}{2(1 - nt)}\right]^{0,2} \quad (2)$$

where $R_e = G_o D_h/\eta_o$ and $G_o = \rho_o v$ is the mass flow per unit area at the minimum cross section. The equivalent diameter on which the thermal characteristics and the Reynolds number Re_T are based is defined as follows:

$$D_{hT} = \frac{A_o}{\pi(D - D_o + s)} \quad (3)$$

Similarly fluid dynamic characteristics and Re_F are based on

$$D_{hF} = 4\, V_f/A_o \tag{4}$$

Equations (1) and (2) are based on experimental results obtained with air flowing through tube banks, eight rows deep. The Euler number in equation (2) refers to a row of these tubes. Since for most practical cases $G_o^2\,(1-\rho_{o1}/\rho_{o2})/\triangle p_{so}\rho_o << 1$ and $\triangle p_{so}/2p_o << 1$ the above-mentioned relation may be simplified to:

$$\frac{\triangle p_{o1}\rho_o}{G_o^2} = \frac{3{,}96}{Re_F^{0,31}}\left(\frac{S_T-D_o}{D_o}\right)^{0,14}\left(\frac{D_o}{S_L-D_o}\right)^{0,18} \times \left[\frac{n(D-D_o)}{2(1-nt)}\right]^{0,20} \tag{5}$$

where $\triangle p_{o1}$ is the static pressure drop across a single tube row, $\triangle p_{o1} = \triangle p_{so}/n_R$. Substitute the expression for the Reynolds number in terms of G_o into equation (1) and find

$$G_o = \frac{9{,}58\alpha_o^{1,51}D_{hT}^{0,51}}{\lambda_o^{1,01}}\left(\frac{\eta_o}{C_{po}}\right)^{0,50}\left(\frac{D_o}{S_T-D_o}\right)^{0,15} \times \left(\frac{S_L-D_o}{D_o}\right)^{0,23}\left[\frac{2(1-nt)}{n(d-D_o)}\right]^{-0,38} \tag{6}$$

Upon substitution of equation (6) in equation (5) the following relation for the static pressure drop in terms of the heat transfer coefficient is obtained:

$$\triangle p_{o1} = \frac{180\, D_{ht}^{0,86}\eta_o^{1,16}\alpha_o^{2,55}}{\rho_o\lambda_o^{1,71}C_{po}^{0,84}D_{hF}^{0,31}}\left(\frac{D_o}{S_T-D_o}\right)^{0,12} \times \left(\frac{S_L-D_o}{D_o}\right)^{0,20}\left[\frac{2(1-nt)}{n(D-D_o)}\right]^{-0,84} \tag{7}$$

The pumping power per unit volume of the heat exchanger may be obtained by considering a control volume between two consecutive fins on a finned tube located in a tube bank across which the pressure drop given by equation (7) is maintained:

$$\frac{\dot W_o}{V} = \frac{\triangle p_{01}v_o}{S_L} \tag{8}$$

where from conservation of mass considerations

$$\rho_o v_o S_T(s+t) = G_o[S_T(s+t)-Dt-D_os]$$

or

$$v_o = \frac{G_o}{\rho_o}\,\frac{[S_T(s+t)-Dt-D_os]}{S_T(s+t)} \tag{9}$$

Substitute equation (6) in (9). The resultant expression for v_o, together with equation (7) are substituted in equation (8) to give

$$\frac{\dot W_o}{V} = \frac{740\alpha_o^{4,06}D_{hF}^{1,37}\eta_o^{1,66}}{\rho_o^2\lambda_o^{2,72}C_{po}^{1,34}D_{hf}^{0,31}S_L} \times \left(\frac{D_o}{S_T-D_o}\right)^{0,27}\left(\frac{S_L-D_o}{D_o}\right)^{0,43} \times \left[\frac{1-nt}{n(d-D_o)}\right]^{-1,22}\frac{[S_T(s+t)-Dt-D_os]}{S_T(s+t)} \tag{10}$$

Referring to the fluid flowing turbulently through the inside of the finned tube, the corresponding Nusselt number is according to Sieder and Tate [12]:

$$Nu_i = \frac{\alpha_iD_i}{\lambda_i} = 0{,}027\left(\frac{G_iD_i}{\eta_i}\right)^{0,80}\left(\frac{\eta_iC_{pi}}{\lambda_i}\right)^{0,33} \tag{11}$$

when physical properties are assumed to remain constant.

The corresponding pressure drop per unit length of finned tube is [13]

$$\triangle p_i = \left[0{,}0028+\frac{0{,}25}{(G_iD_i/\eta_i)^{0,32}}\right]\frac{G_i^2}{\rho_iD_i} \tag{12}$$

Substitute G_i from equation (11) in terms of the heat transfer coefficient in equation (12) and find

$$\triangle p_i = \left[23{,}37+\frac{492{,}1\lambda_i^{0,27}(\eta_iC_{pi})^{0,13}}{(\alpha_iD_i)^{0,40}}\right] \times \frac{\eta_i^{1,17}\alpha_i^{2,5}}{D_i^{0,50}\lambda_i^{1,67}C_{pi}^{0,83}\rho_i} \tag{13}$$

Selecting a control volume as before, the pumping power per unit volume of the heat exchanger required to maintain the flow inside the tubes is

$$\frac{\dot W_i}{V} = \frac{\pi\triangle p_iD_i^2G_i}{4S_TS_L\rho_i} = 10^3\left[1{,}68+\frac{35{,}3\lambda_i^{0,27}(\eta_iC_{pi})^{0,13}}{(\alpha_iD_i)^{0,40}}\right] \times \frac{\eta_i^{1,75}\alpha_i^{3,75}D_i^{1,75}}{S_TS_L\lambda_i^{2,50}\rho_i^2C_{pi}^{1,25}} \tag{14}$$

The heat transfer rate per degree temperature difference between the two streams per unit volume of the heat exchanger is found by applying an energy balance to the previously mentioned control volume located between consecutive fins.

$$\frac{Q}{V} = [S_LS_T(s+t)]^{-1} \times \left[\frac{1}{\alpha_iA_i}+\frac{R_i}{A_i}+\frac{\ell n(D_o/D_I)}{2\pi\lambda_t(s+t)}+\frac{1}{\alpha_o(eA_f+A_{ot})}+\frac{R_o}{A_o}\right]^{-1} \tag{15}$$

where $A_i = \pi D_I\,(s+t)$, $A_f = \pi(D^2-D_o^2)/2$,

$A_{ot} = \pi D_os$ and $A_o = A_f+A_{ot}$.

The efficiency of a circular rectangular fin is [14]

$$e = \frac{\sqrt{2}/\xi}{(1+D/D_o)}\left[\frac{I_1(a\xi)K_1(b\xi)-I_1(b\xi)K_1(a\xi)}{I_1(a\xi)K_o(b\xi+I_o(b\xi)K_1(a\xi)}\right] \tag{16}$$

where $\xi = [(D-D_o)/2](\alpha_o/\lambda_ft)^{0,5}$,

$a = 2^{0,5}/(1-D_o/D)$ and $b = (D_o/D)a$

The time total cost per unit volume to achieve and maintain the above-mentioned heat transfer rate, primarily consists of the cost of the pumping power, material and assembly and can be expressed as

$$\frac{C}{V} = \left(\frac{\dot W_i}{V}+\frac{\dot W_o}{V}\right)C_e+\left[\frac{i(1+i)^n}{(1+i)^n-1}\,\frac{C_c}{S_LS_T\tau}\right] \tag{17}$$

where C_c is the cost per unit length of the finned tube.

Upon dividing equation (15) by equation (17) the ratio of the time rate of heat transfer per degree temperature difference per unit cost is

$$\frac{Q}{C} = \left[\frac{1}{\alpha_i A_i} + \frac{R_i}{A_i} + \frac{\ell n(D_o/D_i)}{2\pi\lambda_t(s+t)} + \frac{1}{\alpha_o(eA_f + A_{ot})} + \frac{R_o}{A_o}\right]^{-1}$$
$$\times \left[(\dot{W}_i + \dot{W}_o)C_e + \frac{i(1+i)^n C_c(s+t)}{\tau((1+i)^n - 1)}\right]^{-1} \tag{18}$$

Since in the case of most industrial heat exchangers it is desirable to transfer as much heat as possible at the lowest cost, a maximum value of this ratio which is usually subject to certain limitations is sought. The procedure is best illustrated by a numerical example.

Application

Consider the problem where it is desired to cool or heat an air stream by passing it through a closely packed finned tube heat exchanger (S_T = D and S_L = 0,866 D) with water flowing inside the tubes.

Practical considerations usually prescribe certain design parameters such as the tube diameter, wall thickness and material, and the fouling factors.

The finned tube span will to some extent dictate the minimum allowable tube diameter to prevent the tubes from sagging or bending. Similarly the internal fluid pressure may in part be responsible for the determination of the tube wall thickness. If the air stream contains impurities, this will usually limit the minimum fin spacing. The degree of fouling inside the tubes is dependent on the quality of the water and the method of treatment thereof. In areas where the atmosphere or water are very corrosive this will have to be taken into consideration when selecting tube and fin materials respectively.

Other parameters which are known during the design period, include the envisaged number of hours of operation of the heat exchanger, the cost of the finned tube material as well as the pumping costs. The latter will undoubtedly change during the life of the heat exchanger and a reasonable extrapolation to determine the average value thereof over this period is desirable.

Let us assume conditions to be such that a mild steel tube having an outside diameter of 20 mm and a wall thickness of 2 mm will satisfy the above requirements. Similarly rectangular circular aluminium fins with a minimum spacing of 3 mm are acceptable. A fouling factor of 0,00002 m² °C/W is anticipated inside the tube while fouling on the outside surface is negligible. The projected average cost of power over the 15 year life of the installation is 2,5 c/kWh while the cost of steel tubing with aluminium fins can be approximated by the following equation:

$$C_c = \left[\frac{2D_o}{0{,}02} + \frac{\pi}{4}(D^2 - D_o{}^2)\frac{5414t}{(s+t)}\right] \text{ \$/m} \tag{19}$$

All dimensions are in meters.

Table 1 – Results of Calculations

		Q/C	Q/V	D	t	α_o	α_{oo}	v_o	Δp_o	α_i	α_{io}	v_i	Δp_i
		W/$/h	W/m³	mm	mm	W/°Cm²	W/°Cm²	m/s	N/m²	W/°Cm²	W/°Cm²	m/s	N/m²
(*a*)		393165	2580	90,7	0,087	10,0	364,8	0,67	0,93	3164	2530	0,68	1,42
		466541	5925	64,5	0,116	20,0	438,4	1,16	2,80	3253	2601	0,71	1,53
Outside heat transfer		470900	7539	57,3	0,124	24,3	441,3	1,29	3,60	3253	2601	0,71	1,53
coefficient	α_o	465850	9693	50,1	0,129	30,0	432,4	1,40	4,64	3237	2589	0,70	1,52
		445656	13300	41,6	0,128	40,0	403,2	1,49	6,32	3187	2549	0,69	1,47
		424259	16538	36,1	0,120	50,0	371,8	1,47	7,80	3129	2502	0,67	1,41
(*b*)	15	490313	7217	51,9	0,114	23,4	473,6	1,26	2,94	3548	2600	0,72	2,52
D_o	20	470900	7539	57,3	0,124	24,3	441,8	1,29	3,61	3253	2601	0,71	1,53
mm	25	456451	7637	62,4	0,129	25,0	418,7	1,29	4,23	3068	2575	0,70	1,08
	30	445522	7630	67,4	0,132	25,6	402,0	1,28	4,80	2937	2545	0,70	0,82
(*c*)	1	677119	14673	48,5	0,054	19,0	676,0	1,01	4,81	3487	2789	0,77	0,60
	2	539437	9604	54,0	0,091	22,1	516,0	1,81	4,00	3342	2673	0,73	1,09
s	3	470900	7539	57,3	0,124	24,3	441,8	1,29	3,61	3253	2601	0,71	1,53
mm	4	427421	6354	69,6	0,153	26,1	394,7	1,37	3,36	3188	2550	0,69	1,95
(*d*)	0	515967	8128	59,2	0,134	24,2	466,3	1,35	3,81	3399	2718	0,75	1,69
$R.10^{-4}$	1	492116	7798	58,3	0,128	24,2	452,9	1,32	3,70	3321	2656	0,72	1,60
	2	470900	7539	57,3	0,124	24,3	441,8	1,29	3,61	3253	2601	0,71	1,53
°Cm²/W	3	451845	7271	56,6	0,119	24,3	430,9	1,26	3,52	3193	2553	0,69	1,47
(*e*)	4	108741	13310	49,8	0,147	44,4	598,5	2,49	12,45	5160	4127	1,26	4,21
	8	191930	10709	52,6	0,138	35,2	531,2	1,94	7,73	4319	3454	1,01	2,85
τ	12	267413	9402	54,3	0,132	30,6	496,7	1,67	5,84	3893	3114	0,88	2,27
h	16	338260	8574	55,6	0,129	27,8	473,2	1,50	4,77	3614	2890	0,80	1,93
	20	405834	7979	56,5	0,126	25,8	454,6	1,38	4,09	3410	2727	0,75	1,70
	24	470900	7539	57,3	0,124	24,3	441,8	1,29	3,61	3253	2601	0,71	1,53
(*f*)	2,5	470900	7539	57,3	0,124	24,3	441,8	1,29	3,61	3253	2601	0,71	1,53
C_e	5,0	414049	6012	60,4	0,114	19,3	391,1	1,00	2,22	2715	2172	0,56	1,03
c/kWh	7,5	383796	5307	62,3	0,110	16,9	365,8	0,86	1,70	2445	1956	0,49	0,82
	10,0	363588	4811	63,8	0,106	15,3	347,8	0,77	1,38	2266	1813	0,45	0,70

		Q/C	Q/V	D	t	a_o	a_{oo}	v_o	Δp_o	a_i	a_{io}	v_i	Δp_i
		W/\$/h	W/m³	mm	mm	W/°Cm²	W/°Cm²	m/s	N/m²	W/°Cm²	W/°Cm²	m/s	N/m²
(g)	Aluminium	470900	7539	57,3	0,124	24,3	441,8	1,29	3,61	3253	2601	0,71	1,53
	Copper	435173	9733	48,4	0,051	30,1	397,5	1,35	4,32	3190	2551	0,69	1,43
	Mild Steel	392534	10648	43,0	0,259	35,5	334,0	1,28	5,03	3056	2444	0,65	1,39
(h)	Air	470900	7516	57,4	0,124	24,3	441,8	1,29	3,60	3253	2601	0,71	1,53
	Carbon dioxide	459907	7136	58,3	0,121	22,9	429,5	1,19	3,86	3240	2591	0,70	1,52
	Hydrogen	238418	1946	83,7	0,079	5,5	216,9	2,77	1,34	2895	2315	0,61	1,17
	Helium	716428	18813	43,3	0,154	70,0	706,0	2,03	2,74	3471	2776	0,77	1,78

The interest on capital is assumed to be ten per cent per annum. For purposes of illustration all physical properties are assumed to remain constant and are evaluated at a temperature of 20 °C and a pressure of 10^5 N/m². The unit is operational 24 hours per day.

Substitute equations (10) and (14) in (17) and equation (16) in (15), whereupon the ratio of Q/C as given by equation (18) may be found in terms of given values.

A computer program was written with which the maximum value of Q/C under these conditions was determined. The results for a few different values of a_o are tabulated in Table 1 and a graphical prèsentation of the ratio Q/C and the heat transfer rate per unit volume Q/V of the heat exchanger as a function of fin diameter are shown in Figure 1. A maximum value for Q/C (470900 W/\$/h) is achieved for an external heat transfer coefficient equal to 24,3 W/m² °C and an internal coefficient of 3253 W/m² °C. The corresponding air and water velocities are 1,29 m/s and 0,71 m/s respectively. Furthermore the optimum fin has a diameter of 57,3 mm and is 0,124 mm thick. At these conditions the rate of heat transfer per unit volume is 7539 W/m³. A more compact heat exchanger can be constructed by reducing the fin diameter, but this will result in a more costly unit to transfer the same amount of heat.

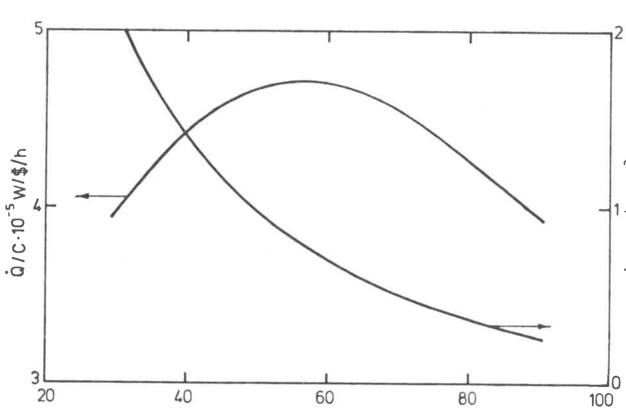

Figure 1 – Heat transfer rate versus fin diameter

A reduction in the outside tube diameter slightly improves performance on a cost basis. To transfer a prescribed amount of heat, the required heat exchanger volume however increases while the fin diameter decreases, with the result that a larger number of smaller tubes is required. This will increase assembly costs and thereby reduce the potential improvement in performance.

Since the internal heat transfer coefficient is considerably larger than that on the outside in this case, a reduction in fin spacing is clearly justified as is shown in Figure 2 if the air stream were clean enough to prevent blockages from occurring between the fins. The decrease in fin spacing results in a decrease in air side heat transfer coefficient and fin diameter but increases the effective external heat transfer coefficient a_{oo} as shown in Table 1c.

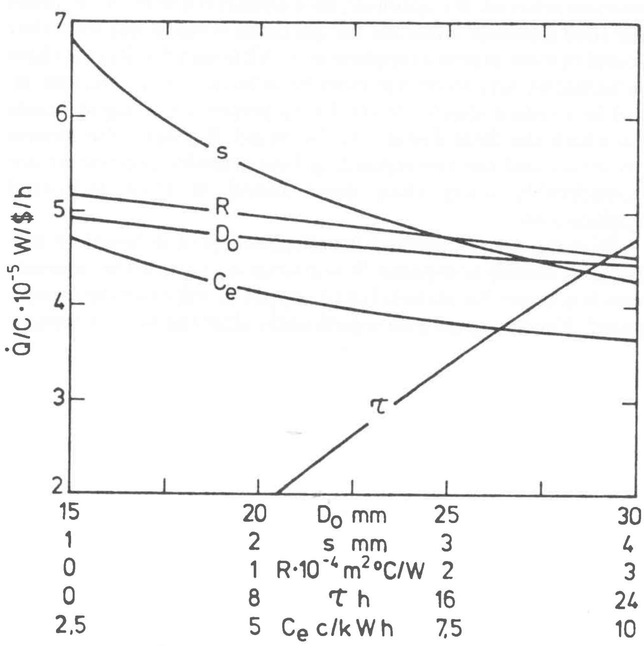

Figure 2 – Effect of some parameters on heat transfer rate

As may be expected any increase in the fouling factor will reduce the effectiveness of the heat exchanger.

The number of hours per day during which the heat exchanger is operational has a significant influence on its performance. For short running periods a high capital outlay becomes more prohibitive, with the result that higher heat transfer coefficients are justified in order to reduce the size of the core. As the operating time is increased, velocities are reduced in order to prevent energy costs from escalating. The performance improves considerably as running times become longer.

The total pumping power consumed during the life of any finned tube heat exchanger is a major cost item. Performance decreases with rising power prices but the optimum fin dimensions are not significantly changed.

Different fin materials strongly affect fin dimensions as shown in Table 1g. Costing of finned tubes was based on the following relations:

$$\text{Copper: } C_c = \left[2+\frac{\pi}{4}(D^2-D_o^2)\frac{17944\,t}{(s+t)}\right] \text{ \$/m} \qquad (20)$$

$$\text{Mild steel: } C_c = \left[2+\frac{\pi}{4}(D^2-D_o^2)\frac{3900\ t}{(s+t)}\right] \$/m \qquad (21)$$

The aluminium fin appears to be the most effective at the specified material prices.

Table 1h shows that gases having widely different physical properties also require finned tubes having different dimensions for the most effective heat exchanger design. The fin diameter in the case of hydrogen gas is more than four times the outside diameter of the tube while in the case of air it is less than three times this value and about twice in the case of Helium. Furthermore the former is very much thinner than the latter notwithstanding its larger diameter.

Conclusions

The proposed procedure for obtaining an optimum design of large finned tube heat exchangers will not only prescribe the most effective operating conditions but also the corresponding optimum tube dimensions. These results may differ considerably from conventional designs as is illustrated by the numerical example of the air-water heat exchanger. For the material cost relation selected, the optimum fin diameter is almost three times the tube diameter while the fin thickness is much less than that found in most practical applications. Although fins having these dimensions, may in certain cases be difficult to manufacture, or not be strong enough, the results do present meaningful values on which the final design can be based. Similarly the stream velocities and the corresponding heat transfer coefficients are considerably lower than those found in most industrial applications.

With the present method the effectiveness of different fin materials is readily compared. It is interesting to note that aluminium is a better fin material than copper in the example investigated. Various gases also significantly alter the heat exchanger dimensions. Fluids other than water inside the tubes may similarly affect the design.

The limitations of all equations employed should be clearly noted. The relations for finned tube prices can be extended considerably to include the detailed cost break-down of assembly, erection, land area, pumps, blowers etc. For purposes of illustration and because local prices may vary considerably, the simplified equations were considered to be adequate.

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