Dimensioning Heat Exchangers for Existing Dry Cooling Towers

by

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A method is proposed whereby the optimum dimensions of finned tube bundles for installation in an existing cooling tower are determined, subject to certain practical constraints and for a given cost structure. The influence of variations in the independent parameters is quantified.

Nomenclature

A area m^2
C cost factor
c_p specific heat J/kg °C
d diameter m
E Euler number
F temperature correction factor
f friction factor
G mass velocity kg/s m^2
g gravitational acceleration m/s^2
H height m
K loss coefficient
k thermal conductivity W/m °C
L length m
m mass flow rate kg/s
n number
P pitch m, or power W
p pressure N/m^2
$\triangle p$ pressure differential N/m^2
Pr Prandtl number
Q heat transfer rate W
R_c thermal contact resistance m^2 K/W
Re Reynolds number
T temperature °C
$\triangle T_{\ell m}$ logarithmic mean temperature difference °C
U overall heat transfer coefficient W/m^2 °C
W width m
v velocity m/s

Greek letters

ε effectiveness
η efficiency
ρ density kg/m^3
μ dynamic viscosity kg/ms
σ contraction ratio
θ angle °

Subscripts

a air
av average
b bundles
ct cooling tower
e heat exchanger or effective
f fin
fr frontal
i inlet or inside
ℓ longitudinal
o outlet or outside
p passes
r tube rows or root
t total, transversal or tube
w water

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Introduction

Although dry air-cooled condensers have been used in relatively small power plants for many years, an increasing number of larger units have been brought into operation more recently. At present plants having outputs of almost 4 000 MWe are under construction.

In this study the possibility of replacing the finned tube bundles of air-cooled systems with optimally dimensioned units is investigated. Replacement may be required where the original heat exchangers are damaged or have become ineffective due to corrosion, erosion or fouling. In addition to the fact that new types of finned tubes may be developed, the relative costs of materials, labour and energy change continually with time and location, resulting in correspondingly different optimised designs.

For purposes of illustration the proposed design procedure is applied to a hyperbolic natural draught cooling tower.

Analysis

Consider the example of a large natural draught dry cooling tower as shown schematically in figure 1.

External to the tower the pressure difference in the stagnant

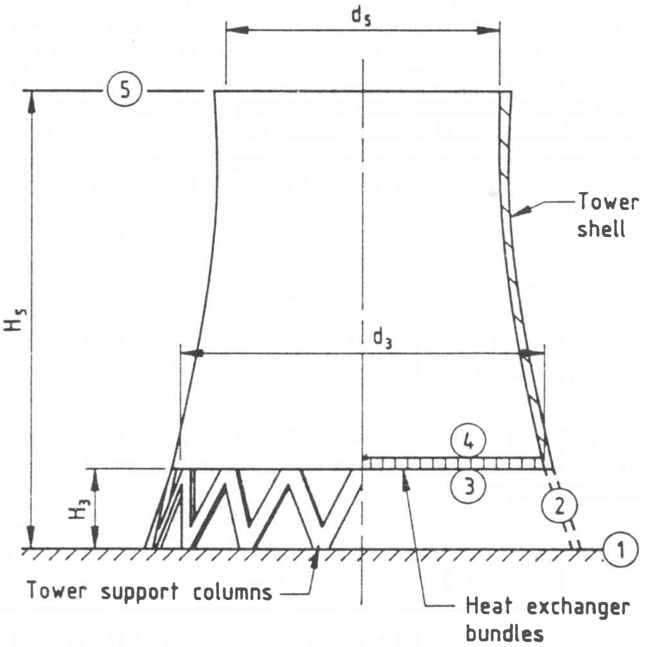

Fig. 1. Natural draft cooling tower

air between ground level and the top of the tower, assuming constant density, is

$$p_{a1} - p_{a5} = \rho_{a1}\, g\, H_5 \tag{1}$$

Based on the assumption of the standard atmosphere, Montakhab [1] shows that this approximate equation deviates by less than 2% from the true pressure difference over a height of 213 m.

The air accelerates from stagnant ambient conditions at 1 to plane 4 after the heat exchanger bundles with a resultant change in pressure due to friction, form drag, inlet and outlet losses and flow acceleration. If $H_3 \approx H_4$ a total pressure balance yields:

$$p_{a1} - (p_{a4} + 0{,}5\, \rho_{a4}\, v_{a4}^2) = (K_{e3} + K_{ct})\, 0{,}5\, \rho_{a1}\, v_{a3}^2 - 0{,}5\, \rho_{a4}\, v_{a4}^2 + \rho_{a1}\, g\, H_3 \tag{2}$$

where K_{e3} is the static pressure loss coefficient across the heat exchanger and K_{ct} is the tower pressure loss coefficient, both based on the tower cross-sectional inlet area A_3.

From 4 to 5 the air flows isentropically and with approximately constant density such that

$$(p_{a4} + 0{,}5\, \rho_{a4}\, v_{a4}^2) - (p_{a5} + 0{,}5\, \rho_{a4}\, v_{a5}^2) = \rho_{a4}\, g\, (H_5 - H_3) \tag{3}$$

Add equations (2) and (3) and equate the sum to equation (1)

$$(\rho_{a1} - \rho_{a4})\, g\, (H_5 - H_3) = (K_{e3} + K_{ct})\, 0{,}5\, \rho_{a1}\, v_{a3}^2 + 0{,}5\, \rho_{a4}\, v_{a5}^2 - 0{,}5\, \rho_{a4}\, v_{a4}^2 \tag{4}$$

For $d_3 = d_4$ this equation may be written in dimensionless form by dividing throughout by $0{,}5\, \rho_{a1}\, v_{a3}^2$ and by satisfying the continuity relation

$$\frac{(\rho_{a1} - \rho_{a4})\, g\, (H_5 - H_3)}{0{,}5\, \rho_{a1}\, v_{a3}^2} = (K_{e3} + K_{ct}) + \frac{\rho_{a1}}{\rho_{a4}} \left[\left(\frac{d_3}{d_5}\right)^4 - 1 \right] \tag{5}$$

Equation (5) is known as the draught equation for the natural convection cooling tower.

In order to make more effective use of the available tower base area, heat exchanger bundles may be arranged in the form of V-arrays or deltas. The air stream leaving this heat exchanger has a highly turbulent distorted velocity distribution in which further losses in total pressure occur above the outlet of the heat exchanger. These and other losses through the inclined heat exchangers are included in the total pressure drop given by Mohandes [2],

$$\triangle p_{et\theta} = 0{,}5 \left(\frac{m_a}{A_{fr}}\right)^2 \left[\frac{K_{et}}{2} \left(\frac{1}{\rho_{a1}} + \frac{1}{\rho_{a4}} \right) + \frac{1}{\rho_{a1}} \left(\frac{1}{\sin\theta_{av}} - 1 \right) \left\{ \left(\frac{1}{\sin\theta_{av}} - 1 \right) + 2\, K_{i\theta=90^\circ}^{0,5} \right\} + \frac{K_{dt}}{\rho_{a4}} \right] \tag{6}$$

where the total pressure loss coefficient through the heat exchanger under normal flow conditions can in general be expressed as [3]

$$K_{et} = \frac{2}{\sigma^2} \left[Eu + \left(\frac{\rho_{a1} - \rho_{a4}}{\rho_{a1} + \rho_{a4}} \right) \right] \tag{7}$$

Robinson and Briggs [4] propose the following correlation for the Euler number for bundles of staggered radially finned tubes:

$$Eu = \frac{\triangle p_e\, \rho_a}{G_{ca}^2} = \frac{18{,}93\, n_r}{Re_a^{0,316}} \left(\frac{d_r}{P_t}\right)^{0,927} \left(\frac{P_t}{P_\ell}\right)^{0,515} \tag{8}$$

where $Re_a = G_{ca}\, d_r/\mu_a = m_a\, d_r/\sigma\mu_a A_{fr}$

A static pressure drop coefficient for the V-array of bundles in the cooling tower, based on conditions at the inlet plane 3 is defined as

$$K_{e\theta 3} = \frac{\triangle p_{et\theta} - 0{,}5\, \rho_{a1}\, v_{a3}^2 + 0{,}5\, \rho_{a4}\, v_{a4}^2}{0{,}5\, \rho_{a1}\, v_{a3}^2} = \frac{\triangle p_{t\theta}}{0{,}5\, \rho_{a1}\, v_{a3}^2} + \left(\frac{\rho_{a1}}{\rho_{a4}} - 1 \right) \tag{9}$$

such that with equation (6) and the continuity relation

$$K_{e\theta 3} = \left(\frac{\pi d_3^2}{4\, A_{fr}}\right)^2 \left[\frac{K_{et}}{2} \left(1 + \frac{\rho_{a1}}{\rho_{a4}} \right) + \left(\frac{1}{\sin\theta_{av}} - 1 \right) \left\{ \left(\frac{1}{\sin\theta_{av}} - 1 \right) + 2\, K_{i\theta=90^\circ}^{0,5} \right\} + \frac{\rho_{a1}}{\rho_{a4}} K_{dt} \right] + \left(\frac{\rho_{a1}}{\rho_{a4}} - 1 \right) \tag{10}$$

With $K_{e3} = K_{e\theta 3}$ for the V-array layout, substitute equation (10) in equation (5) and find

$$\frac{(\rho_{a1} - \rho_{a4})\, g\, (H_5 - H_3)}{0{,}5\, \rho_{a1}\, v_{a3}^2} = \left(\frac{\pi\, d_3^2}{4\, A_{fr}}\right)^2 \left[\frac{K_{et}}{2} \left(1 + \frac{\rho_{a1}}{\rho_{a4}} \right) + \left(\frac{1}{\sin\theta_{av}} - 1 \right) \left\{ \left(\frac{1}{\sin\theta_{av}} - 1 \right) + 2\, K_{i\theta=90^\circ}^{0,5} \right\} + \frac{\rho_{a1}}{\rho_{a4}} K_{dt} \right] + K_{ct} + \frac{\rho_{a1}}{\rho_{a4}} \left(\frac{d_3}{d_5}\right)^4 - 1 \tag{11}$$

The average flow incidence angle may be expressed by the following empirical relation:

$$\theta_{av} = 0{,}0019\, \theta^2 + 0{,}9133\, \theta - 3{,}1558 \tag{12}$$

where the apex semi-angle θ is given in degrees.

Furthermore $K_{i\theta=90^\circ}$ is the heat exchanger inlet contraction pressure loss coefficient under normal flow conditions ($K_{i\theta=90^\circ} \approx 0{,}05$ for many industrial finned tube heat exchangers), while the downstream pressure loss coefficient can be expressed approximately by the following empirical relation

$$K_{dt} = \exp(5{,}83479.10^{-4}\, \theta^2 - 0{,}1176997\, \theta + 4{,}2817) \tag{13}$$

According to Hempel et al [5] the cooling tower pressure loss coefficient may be expressed as:

$$K_{ct} = 0{,}19 \left(\frac{d_3}{H_3}\right)^2 - (0{,}38 + 0{,}11\, K_e^{0,7})\, \frac{d_3}{H_3} + 4{,}9 \tag{14}$$

This equation includes a value of $0{,}01\, (d_3/H_3)^2$ which makes provision for the approximate resistance due to tower supports.

By introducing the gas law, equation (11) may be expressed approximately in terms of the ambient pressure, and temperatures before and after the heat exchanger as well as the air flow rate through the tower.

$$\frac{2\, p_{a1}^2}{R^2\, T_{a1}^2} \left(\frac{A_{fr}}{m_a}\right)^2 \left(1 - \frac{T_{a1}}{T_{a4}}\right) g\, (H_5 - H_3) = \frac{K_{et}}{2} \left(1 + \frac{T_{a4}}{T_{a1}}\right) + \left(\frac{1}{\sin\theta_{av}} - 1\right) \left[\left(\frac{1}{\sin\theta_{av}} - 1\right) + 2\, K_{i\theta=90^\circ}^{0,5} \right] + \frac{T_{a4}}{T_{a1}} K_{dt} + \left(\frac{4\, A_{fr}}{\pi\, d_3^2}\right)^2 \left[K_{ct} + \frac{T_{a4}}{T_{a1}} \left(\frac{d_3}{d_5}\right)^4 - 1 \right] \tag{15}$$

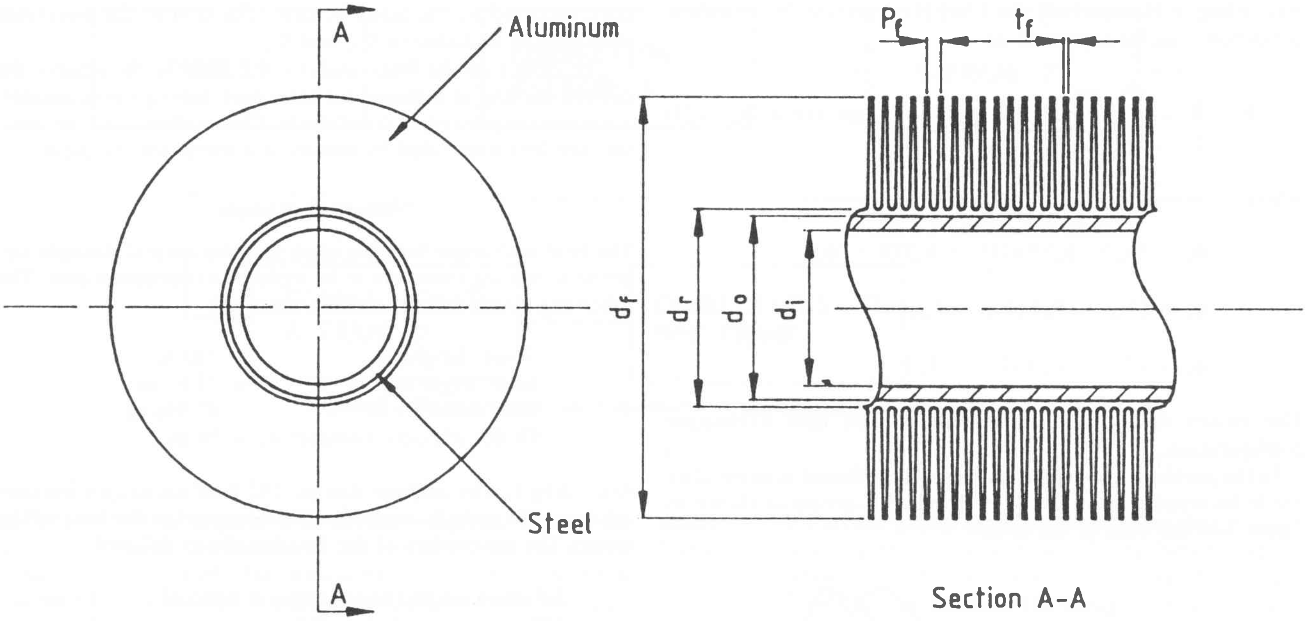

Fig. 2. Bi-metallic finned tube

The ultimate relation between the air flow rate through the tower and the outlet air temperature at given ambient conditions will be determined by the heat transfer characteristics of the finned tube bundles.

The heat transfer rate from the warm water to the air stream is

$$Q = m_a c_{pa} (T_{a4} - T_{a1}) = m_w c_{pw} (T_{wi} - T_{wo})$$

or

$$Q = UA\, F \triangle T_{\ell m} \tag{16}$$

where

$$UA = (1/h_{ae} A_a + 1/h_w A_w)^{-1}$$

For a clean bi-metallic radially finned tube as shown in figure 2,

$$h_{ae} A_a = \left[\frac{1}{h_a \varepsilon_f A_a} + \frac{\ell n\,(d_o/d_i)}{2\pi k_t L_{tt}} + \frac{\ell n\,(d_r/d_o)}{2\pi k_f L_{tt}} + \frac{R_c}{A_o} \right]^{-1} \tag{17}$$

According to Briggs and Young [6] the air-side heat transfer coefficient h_a through bundles of staggered radially finned tubes is given by

$$h_a = \frac{0{,}134\, k_a\, Pr_a^{0{,}33}\, Re_a^{0{,}681}}{d_r} \left[\frac{2(P_f - t_f)}{(d_f - d_r)} \right]^{0{,}2} \left[\frac{(P_f - t_f)}{t_f} \right]^{0{,}1134} \tag{18}$$

where $Re_a = G_{ca}\, d_r/\mu_a = m_a\, d_r/\sigma\, \mu_a\, A_{fr}$

The effectiveness of the finned surface is expressed in terms of the fin efficiency, i.e.

$$\varepsilon_f = 1 - A_f(1 - \eta_f)/A_a$$

According to Schmidt [7] the fin efficiency for radial fins can be determined approximately from

$$\eta_f = \frac{\tanh\,(b\, d_r\, \varphi/2)}{(b\, d_r\, \varphi/2)}$$

where $\varphi = (d_f/d_r - 1)\,[1 + 0{,}35\, \ell n\,(d_f/d_r)]$

and $b = (2\, h_a/k_f t_f)^{0{,}5}$

The water-side heat transfer coefficient is according to Gnielinski [8]

$$h_w = \frac{k_w\, f_w\, (Re_w - 1\,000)\, Pr_w\, [1 + (d_i/L_t)^{0{,}67}]}{8\, d_i\, [1 + 12{,}7\, (f_w/8)^{0{,}5}\, (Pr_w^{0{,}67} - 1)]} \tag{19}$$

where

$$Re_w = 4\, m_w\, n_{wp}/\pi\, d_i\, \mu_w\, n_{tb}\, n_b$$

and for a smooth tube [9],

$$f_w = (1{,}82 \log_{10} Re_w - 1{,}64)^{-2} \tag{20}$$

The wetted inside tube surface area is

$$A_w = \pi\, d_i\, L_t\, n_{tb}\, n_b$$

If the water mass flow rate per unit cross-sectional inside tube area is given by

$$G_w = 4\, m_w\, n_{wp}/\pi\, d_i^2\, n_{tb}\, n_b$$

the pressure drop per unit length of finned tube is

$$\triangle p_w = f_w\, G_w^2/2\, \rho_w\, d_i$$

The total water pumping power is thus

$$P_w = \pi\, d_i^2\, \triangle p_w\, G_w\, L_t\, n_{tb}\, n_b/4\, \rho_w \tag{21}$$

The logarithmic mean temperature difference is defined as

$$\triangle T_{\ell m} = \frac{(T_{wi} - T_{a4}) - (T_{wo} - T_{a1})}{\ell n\,[(T_{wi} - T_{a4})/(T_{wo} - T_{a1})]} \tag{22}$$

According to Roetzel [10], the LMTD correction for crossflow conditions can be expressed as

$$F = 1 - \sum_{i=1}^{4} \sum_{k=1}^{4} a_{ik}(1 - \varphi_p)^k \sin(2i \arctan \varphi_a/\varphi_w) \quad (23)$$

where

$$\phi_p = (\phi_a - \phi_w)/\ell n\,[(1 - \phi_w)/(1 - \phi_a)]$$

$$\phi_a = (T_{a4} - T_{a1})/(T_{wi} - T_{a1})$$

$$\phi_w = (T_{wi} - T_{wo})/(T_{wi} - T_{a1})$$

The values of a_{ik} are individual to each heat exchanger configuration.

In the particular tower to be analysed the finned tube bundles are to be arranged radially in the form of V-arrays as shown in figure 3 in the base of the tower.

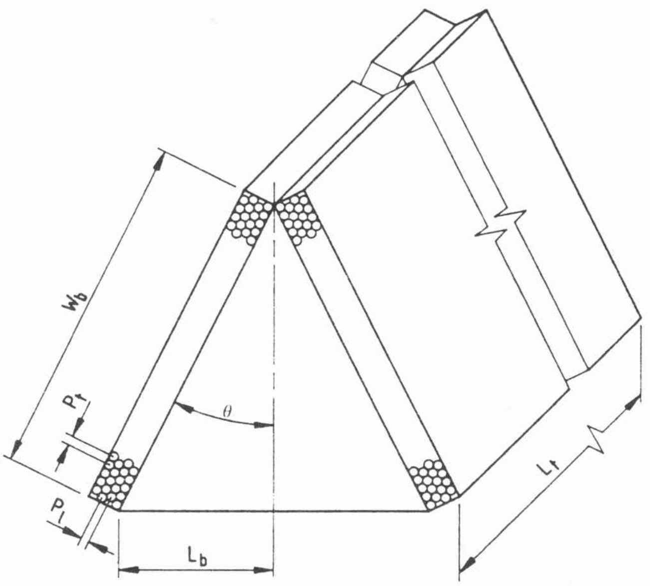

Fig. 3. Heat exchanger bundles

The final "optimised" retrofit finned tube geometry and layout within certain prescribed constraints is a function of material and other costs. For purposes of illustration it will be assumed that the total effective finned tube cost can be expressed as

$$C_{ft} = [C_{wf}(C_t + C_f) + C_{fix}]L_t\, n_{tb}\, n_b \quad (24)$$

where C_{wf} is a weighting factor

The core tube cost per unit length is

$$C_t = \frac{\pi}{4}\rho_t(d_o^2 - d_i^2)\,C_{tm}$$

where C_{tm} is the tube material cost per unit mass.

Similarly the cost of the fin material per unit tube length is

$$C_f = \frac{\pi\,\rho_f}{4\,P_f}[(d_f^2 - d_r^2)\,t_f + (d_r^2 - d_o^2)(P_f - t_f)]C_{fm}$$

where C_{fm} is the fin material cost per unit mass.

The fixed cost of the tube per unit length C_{fix}, covers other costs incurred by the manufacturer. The cost of the manifolds may also be included in C_{fix} and C_{wf}.

The object of the final analysis will thus be to achieve the desired cooling at a minimum total cost, taking into consideration certain prescribed constraints. The results of such an analysis are best illustrated by means of a numerical example.

Numerical example

The heat exchanger bundles in an existing natural draught hyperbolic cooling tower are to be replaced at minimum cost. The following tower dimensions are specified:

Tower height H_5	= 120 m
Inlet height H_3	= 13,67 m
Inlet diameter d_3	= 82,958 m
Outlet (throat) diameter d_5	= 58 m.

According to the original design, 142 heat exchanger bundles are arranged radially in the form of V-arrays in the base of the tower. The particulars of the bundles are as follows:

Effective length (finned tube) of bundle	L_t	= 15 m
Effective width of bundle	W_b	= 2,262 m
Bundle semi-apex angle	θ	= 30,75°
Number of water passes	n_{wp}	= 2
Number of tube rows	n_r	= 4

To minimise cost it is recommended that the original bundle base support be employed and that no changes be made to the water pipe layout to and from the bundles or the water pump. The bundle support structure imposes the restriction that $L_b = W_b \sin\theta = 1{,}157$ m.

The existing pump provides a water flow rate of 4 390 kg/s at a pressure drop of 16 302 N/m² through the finned tubes and deviations from these values are not permitted. In effect this also means that the pumping power P_w remains unchanged.

Due to structural considerations the tube wall and the fin root thickness are assumed constant at 1,655 mm and 1,08 mm respectively. The contact resistance R_c is assumed to be negligible.

The cooling tower is required to remove 365,08 MW at an inlet water temperature of 61,45°C, an ambient air pressure at ground level of 84 600 N/m² and a corresponding air temperature of 15,6°C.

In order to achieve an optimum design, the independent parameters t_f, d_f, θ, d_i, P_t, P_ℓ, P_f and n_{wp} may be varied subject to the constraints specified. A fin pitch of 2,35 mm is arbitrarily chosen as the design reference value. Where fouling is of significance very small fin pitches are avoided. The values of C_{wf}, C_{tm}, C_{fm} and C_{fix} are taken as 2, 0,8$/kg, 4,2$/kg and 2$/m respectively.

The cost minimisation problem may be solved numerically with a general purpose algorithm for non-linear constrained optimisation. A Generalised Reduced Gradient algorithm [11] and a Constrained Variable Metric algorithm [12], were investigated and it was found that the Variable Metric algorithm was very efficient when applied to this problem. All results presented in this paper were obtained using the FORTRAN routine VMCWD [13]. Derivatives of the cost function and constraints were calculated numerically. Variables and constraints were carefully scaled, since they differ greatly in magnitude. All calculations were performed in double precision (approximately 16 decimal digits) in order to obtain reasonably accurate values for the derivatives. On average, routine VMCWD called the routines which calculate the cost function and constraints and their gradients, 6 to 10 times for each complete optimisation.

The results of the cost minimisation are presented in table 1. The values of the independent parameters corresponding to the minimum cost are taken as reference values (first line of table 1).

In order to investigate the dependence of the minimum cost on each parameter, one parameter is then perturbed and a minimisation performed with respect to the other parameters. The influence of various parameters on the total cost are also shown graphically in figure 4.

Conclusions

According to the results shown in table 1 the optimum fin thickness is only 0,0903 mm while the fin diameter is 61,95 mm. In practice somewhat thicker fins are preferred resulting in a corresponding increase in system cost as shown in table 1.

According to the analysis the finned tubes are closely packed at optimum conditions if a triangular pattern is assumed. For larger fin thicknesses a more open arrangement is preferable. The influence of the longitudinal pitch is shown for both a triangular tube layout and for the case where P_ℓ is not tied to the transversal pitch. As P_ℓ increases the system cost is reduced. This reduction is however only applicable within certain limits in which the finned tube correlations apply.

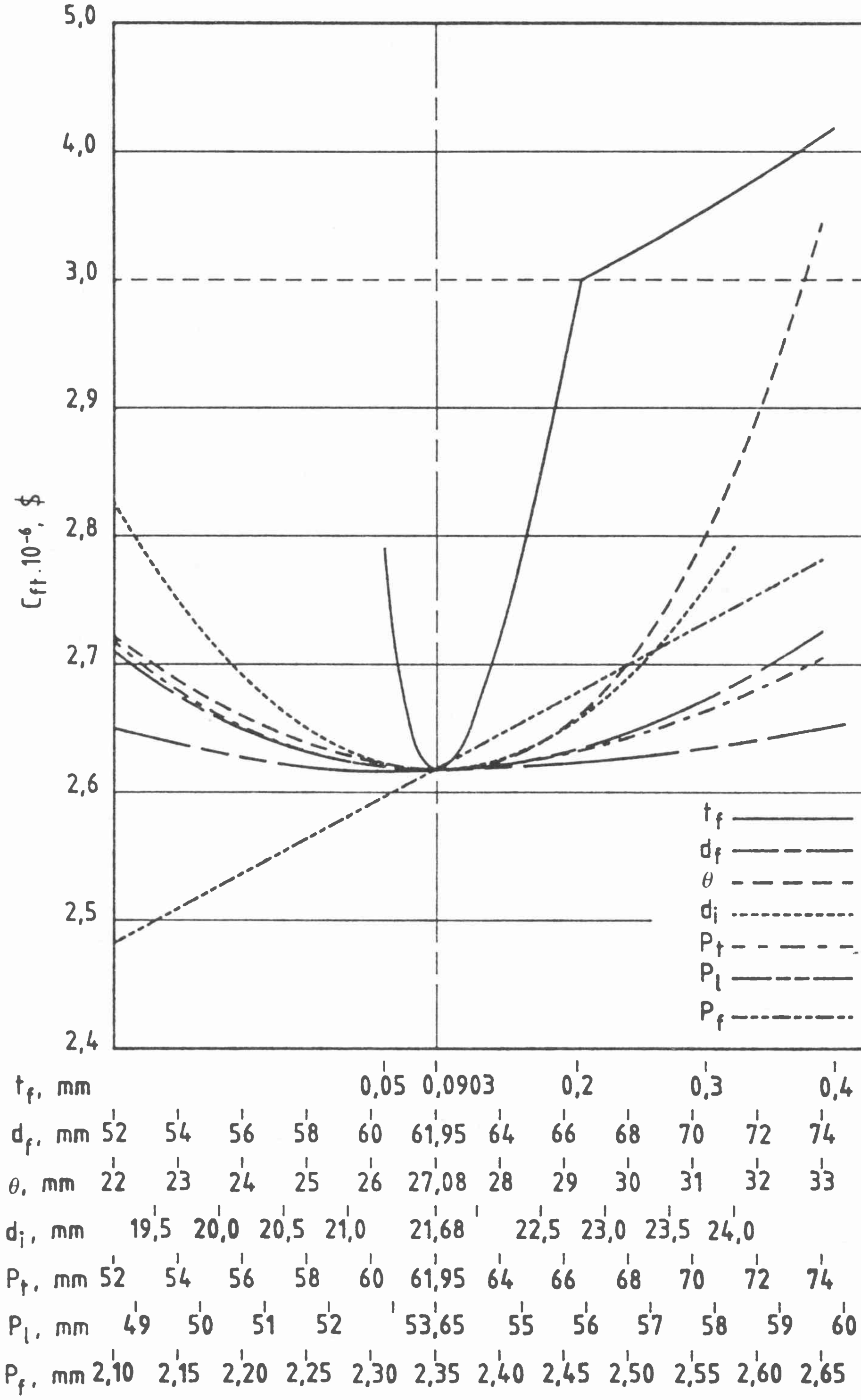

Fig. 4. Influence of changes in independent variables on cost

Table 1 Optimised finned tube bundles ($n_{wp} = 2$)

Parameter	t_f (mm)	d_f (mm)	θ (°)	d_i (mm)	P_t (mm)	P_ℓ (mm)	P_f (mm)	C ($)
(Optimum)	0,0903	61,95	27,08	21,68	61,95	53,65	2,35	2 617 867
	0,05	55,67	22,68	19,57	55,67	48,21		2 789 929
	0,1	62,79	27,88	22,01	62,79	54,37		2 623 508
t_f	0,2	64,71	30,35	23,46	68,86	59,63	2,35	2 984 172
	0,3	64,40	30,61	23,70	70,19	60,78		3 545 850
	0,4	63,22	30,00	23,48	69,75	60,40		4 181 762
	0,0768	50	25,20	19,85	52,40	45,37		2 756 065
	0,0831	55	26,24	20,70	56,37	48,81		2 661 283
d_f	0,0878	60	27,04	21,41	60,00	51,96	2,35	2 621 230
	0,0940	65	27,08	22,07	65,00	56,29		2 625 553
	0,0996	70	26,90	22,64	70,00	60,62		2 668 214
	0,0734	60,38	21	20,60	68,68	59,48		2 762 512
	0,0793	60,78	24	21,13	64,73	56,06		2 659 733
θ	0,0895	61,94	27	21,65	61,94	53,64	2,35	2 617 917
	0,1328	64,69	30	22,82	64,69	56,02		2 702 217
	0,2016	72,15	33	24,56	72,15	62,48		3 044 095
	0,039	44,41	22,16	18	45,46	39,37		3 080 373
d_i	0,079	58,57	26,26	21	58,57	50,73	2,35	2 632 745
	0,139	74,33	29,82	24	74,33	64,37		2 791 658
	0,0732	50,00	26,11	19,75	50	43,30		2 767 791
P_t	0,0878	60,00	27,04	21,41	60	51,96	2,35	2 621 230
	0,1011	67,16	26,07	22,40	70	60,62		2 660 119
	0,0763	51,96	26,39	20,12	51,96	45		2 718 706
P_ℓ	0,0922	63,51	27,09	21,88	63,51	55	2,35	2 619 920
($P_\ell = 0{,}866P_t$)	0,1075	69,86	25,14	22,70	75,06	65		2 719 226
	0,0863	58,89	26,99	21,25	58,89	51		2 626 282
P_ℓ	0,0870	58,89	27,36	21,34	58,89	55		2 602 379
($P_\ell \geq P_{\ell\ min}$	0,0876	58,77	27,71	21,42	58,77	59	2,35	2 580 370
	0,0881	58,66	28,04	21,50	58,66	63		2 559 978
	0,0828	61,91	28,17	22,22	63,82	55,27	2,1	2 482 365
	0,0854	62,07	27,83	22,00	62,83	54,41	2,2	2 537 426
P_f	0,0884	62,09	27,40	21,79	62,09	53,77	2,3	2 591 074
	0,0922	61,81	26,76	21,57	61,81	53,53	2,4	2 644 808
	0,0959	61,53	26,14	21,36	61,53	53,29	2,5	2 699 153
0,5	0,0857	60,66	26,77	21,42	60,66	52,53		2 418 587
C_{tm} 1,0	0,0933	62,78	27,28	21,84	62,78	54,36	2,35	2 748 466
1,5	0,1005	64,74	27,73	22,22	64,74	56,06		3 067 823
3,0	0,1039	65,64	27,94	22,39	65,66	56,84		2 204 230
C_{fm} 4,0	0,0921	62,45	27,20	21,78	62,45	54,08	2,35	2 550 819
5,0	0,0843	60,26	26,67	21,34	60,26	52,19		2 880 734
1,0	0,1118	67,71	28,41	22,79	67,71	58,64		1 630 395
C_{wf} 3,0	0,0818	59,55	26,50	21,20	59,55	51,57	2,35	3 571 350
5,0	0,0744	57,35	25,95	20,75	57,35	49,67		5 449 476
1,0	0,0773	58,21	26,17	20,93	58,21	50,41		2 256 577
C_{fix} 3,0	0,1016	65,04	27,81	22,28	65,04	56,33	2,35	2 950 093
5,0	0,1211	70,07	28,93	23,23	70,07	60,68		3 554 603
$n_{wp} = 4$								
(Optimum)	0,0826	77,39	26,72	36,18	83,87	72,63	2,35	2 611 998
	0,1	78,88	27,63	37,10	86,91	75,27		2 629 576
	0,2	82,08	29,97	39,42	94,67	81,99	2,35	3 051 830
t_f	0,3	81,59	30,10	39,61	95,50	82,71		3 666 459
	0,4	79,70	29,37	39,00	93,77	81,20		4 354 121

Where fouling is not serious, a reduction in fin pitch will generally result in a lower cost system.

The influence of variations in the cost structure are also shown in table 1.

The possibility of installing a four-pass heat exchanger was also investigated. The resultant optimum design is only fractionally less costly than the two-pass system. For the thicker fin the converse is true. Furthermore it is noted that the inside diameter of the tubes is considerably larger to ensure a pressure drop on the water-side that does not exceed that specified for the pump.

The authors have extended the above optimisation method to include the designs of complete cooling systems for new plants.

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