INVESTIGATION AND COMPARISON OF FINNED, TUBULAR HEAT TRANSFER SURFACES OVER A WIDE RANGE OF REYNOLDS NUMBERS

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The present work is devoted to a comparison, from the energy characteristics, of the most prevalent types of finned, tubular surfaces over a wide range of values of Re. The purpose of the comparison is the development of an efficient type of surface and determination of the direction of research for creating a series of surfaces having the optimum geometric parameters.

A list and the principal parameters of the surfaces investigated are given in Table 1. The experimental setup, method of investigation, and interpretation of the experimental data are described in [1]. Data on the heat transfer and aerodynamic resistance of the investigated surfaces are given in Tables 2 and 3.

Comparison of the convection surface studied was carried out on the basis of the energy coefficient proposed by V. M. Antuf'ev [2]:

$$E = \frac{\alpha_g 1}{AN_0},$$

where α_g is the heat-transfer coefficient in W/m$^2\cdot$degrees; AN_0 is the energy expended in 1 h for circulating the heat transfer fluid and exerted per 1 m^2 of surface (A = 1.163 W$\cdot$h/kcal) in kcal/h$\cdot$m^2.

The energy expenditures for overcoming pressure drop can be expressed in the following manner:

$$AN_0 = 3600\, A\, \Delta pw \frac{f}{F},$$

where Δp is the loss of head in the tube sheet in newtons/m^2; w is the velocity of the air stream in m/sec; f is the free cross section of the air stream in m^2; and F is the heat transfer surface in m^2.

Because the comparison of the investigated surfaces is carried out for AN_0 = idem,

$$\frac{E_i}{E_k} = \frac{\alpha_i}{\alpha_k},$$

where E_i, E_k, α_i, and α_k are, respectively, the energy coefficients and heat-transfer coefficients of the surfaces being compared.

As is seen from Fig. 1, in the regime for cooling of air, surfaces No. 3 and 7 have the best characteristics with respect to thermal efficiency in the whole range of values of AN_0. This is explained by the fact that the diameter of the support tube of surface No. 7 is small (d_c = 12.3 mm) and the relative height and thickness of the fin are also not great ($h/d_c = 0.5$, $\delta/d_c = 0.032$). The tubes are arranged in a tight bundle. Surface No. 3 has a small tube mid-section (d_c = 12 mm) and a small relative fin thickness ($\delta/d_c = 0.041$), so that the fin has good contact with the support tube and the tubes are arranged in a tight bundle.

In the initial section where AN_0 = 6-40 kcal/h$\cdot$m^2 the characteristics of surface No. 6 adjoin the characteristics of surfaces No. 3 and 7, but because of the rapid drop in the efficiency of the wire fins it departs sharply from

TABLE 1.

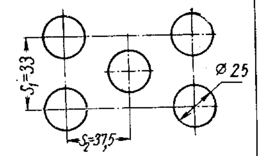
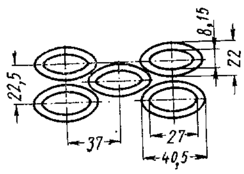
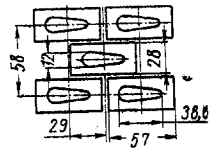
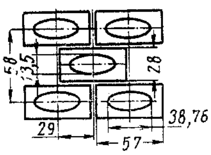
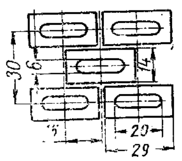
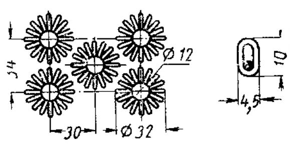
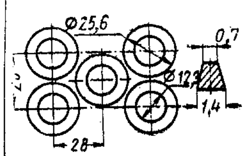
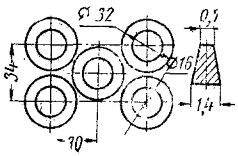
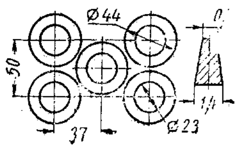
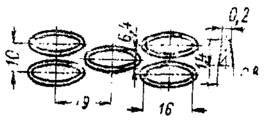
No. of tubular surface	Tube type	Sketch	Fin thickness δ in mm	Distance between S_3 in mm	Ratio of pitch to diam. $\frac{S_1}{d_c}$	$\frac{S_2}{d_c}$	Finning coeff. φ	Compactness coeff. π	Heat transfer surface in m^2	Wt. of 1 m of tube in kg	Surface of 1 m of tube in m^2	Quantity of fins in 1m of tubes	Quantity of tubes in a bundle	Material: tubes	Material: fins
1	Round, smooth		—	—	1,32	1,5	—	63,5	0,691	0,93	0,0785	—	44	Brass	—
2	Elliptical with rolled-on band fins		0,3	3,0	2,76	4,54	8,51	584	6,6	1,1	0,485	330	68	"	Brass
3	Drop-shaped, finned with rectangular plates		0,5	2,8	4,83	2,33	10,8	576	5,42	1,56	0,967	357	68	"	Copper
4	Oval, finned with rectangular plates		0,5	2,7	4,3	2,15	11,4	600	5,65	1,8	0,01	370	28	Copper	"
5	Flat oval, finned with rectangular plates		0,2	1,5	5,0	2,5	8,9	926	4,33	0,715	0,394	666	52	"	"
6	Round, finned with wire		—	4,5	2,83	2,5	13,4	496	4,45	1,235	0,505	—	44	Cupro-nickel	"
7	Round aluminum with seamless rolled fins		—	3,5	2,12	2,28	7,05	375	3,27	0,363	0,272	286	60	Aluminum ADM-1	

TABLE 1. (Continued)

	Tube type	Sketch	Fin thickness δ in mm	Distance between fins S_3 in mm	Ratio of pitch to diam. $\frac{S_1}{d_c}$	$\frac{S_2}{d_c}$	Finning coeff. φ	Compactness coeff. π	Heat transfer surface in m²	Wt. of 1 m of tube in kg	Surface of 1 m of tube in m²	Quantity of fins in 1 m of tube	Quantity of tubes in a bundle	Material: tubes	Material: fins
8	Round, bimetallic with seamless rolled fins, $d_r/d_c = 32/16$		—	3,5	2,12	1,87	8,04	396	3,55	0,887	0,403	286	44	Copper	Aluminum ADM-1
9	Round, bimetallic with seamless rolled fins, $d_r/d_c = 44/23$		—	3,5	2,17	1,605	9,85	385	3,99	1,368	0,713	286	28	"	"
10	Oval, finned with wire of trapezoidal cross section		—	3	1,56	2,97	1,74	370	2,19	0,372	0,0704	333	156	"	Copper

them and at the end of the investigated range AN_0 is located in the group of thermal characteristics of the low-frequency surfaces No. 4, 9, and 10. The characteristics of the latter demonstrate their low thermal efficiency in the whole range of AN_0 investigated, which is explained by the following circumstances. Surface No. 9 has a relatively large diameter support tube (d_c = 23 mm) and an oversized relative fin height (h/d_c = 0.6). The method of solder "bathing" used for the fins of surfaces No. 4 and 10 causes, as was shown by experiments [4], a substantial increase of thermal resistance at the contact site.

The thermal characteristics of surfaces No. 8, 1, 2, and 5 in the initial section differ little from one another and are located in the middle between efficient and low-efficiency surfaces. In the range AN_0 = 20-10,000 kcal/h · m² the characteristics of surfaces No. 5 and 8 are improved and after a discontinuity at AN_0 = 800 kcal/h · m² they unite in a single line running parallel to and slightly below the characteristics of surface No. 7. The nature of the thermal efficiency of surfaces No. 5 and 8 is explained by the following reasons. Surface No. 5 has a tube midsection of d_c = 6 mm and a low profile, thin fin. Surface No. 8 has monolithic fins and small relative height and fin thickness (h/d_c = 0.5; δ/d_c = 0.0313). Tubes of both surfaces were arranged in tight bundles, which also has a positive effect on the thermal efficiency.

TABLE 2

No. of tubular surface	Calculation formulas for heat transfer	
	for cooling of air	for heating of air
1	$Nu = 0,270\,Re^{0,6}$ ($Re = 10^4 - 2 \cdot 10^5$); $Nu = 0,008\,Re^{0,886}$ ($Re = 2 \cdot 10^5 - 3,2 \cdot 10^5$)	$Nu = 0,296\,Re^{0,6}$ ($Re = 10^4 - 10^5$); $Nu = 0,0108\,Re^{0,886}$ ($Re = 10^5 - 3 \cdot 10^5$)
2	$Nu = 0,0992\,Re^{0,673}$ ($Re = 4 \cdot 10^3 - 7,4 \cdot 10^3$); $Nu = 1,85\,Re^{0,345}$ ($Re = 7,4 \cdot 10^3 - 2,4 \cdot 10^4$); $Nu = 20,0\,Re^{0,103}$ ($Re = 2,4 \cdot 10^4 - 3,9 \cdot 10^4$)	—
3	$Nu = 0,09\,Re^{0,715}$ ($Re = 3,6 \cdot 10^3 - 5,4 \cdot 10^4$)	$Nu = 0,0154\,Re^{0,976}$ ($Re = 2,5 \cdot 10^3 - 2 \cdot 10^4$)
4	$Nu = 0,103\,Re^{0,665}$ ($Re = 6 \cdot 10^3 - 2,3 \cdot 10^4$); $Nu = 0,563\,Re^{0,495}$ ($Re = 2,3 \cdot 10^4 - 5,2 \cdot 10^4$)	—
5	$Nu = 0,0263\,Re^{0,842}$ ($Re = 2,5 \cdot 10^3 - 1,05 \cdot 10^4$); $Nu = 0,392\,Re^{0,548}$ ($Re = 1,05 \cdot 10^4 - 3 \cdot 10^4$)	—
6	$Nu = Re^{0,456}$ ($Re = 10^4 - 8 \cdot 10^4$)	$Nu = 0,0665\,Re^{0,79}$ ($Re = 5 \cdot 10^3 - 3 \cdot 10^4$)
7	$Nu = 0,225\,Re^{0,633}$ ($Re = 6 \cdot 10^3 - 8 \cdot 10^4$)	$Nu = 0,338\,Re^{0,63}$ ($Re = 4 \cdot 10^3 - 2 \cdot 10^4$); $Nu = 1,1 \cdot 10^{-3}\,Re^{1,227}$ ($Re = 2 \cdot 10^4 - 6 \cdot 10^4$)
8	$Nu = 0,0426\,Re^{0,774}$ ($Re = 9,6 \cdot 10^3 - 3 \cdot 10^4$); $Nu = 0,264\,Re^{0,595}$ ($Re = 3 \cdot 10^4 - 1,4 \cdot 10^5$)	$Nu = 0,0512\,Re^{0,775}$ ($Re = 5,6 \cdot 10^3 - 4,4 \cdot 10^4$); $Nu = 6,44 \cdot 10^{-3}\,Re^{0,97}$ ($Re = 4,4 \cdot 10^4 - 1,5 \cdot 10^5$)
9	$Nu = 0,0971\,Re^{0,662}$ ($Re = 1,4 \cdot 10^4 - 1,8 \cdot 10^5$)	$Nu = 0,021\,Re^{0,841}$ ($Re = 1,3 \cdot 10^4 - 1,4 \cdot 10^5$)
10	$Nu = 0,213\,Re^{0,79}$ ($Re = 4 \cdot 10^3 - 2,3 \cdot 10^4$); $Nu = 2,84\,Re^{0,531}$ ($Re = 2,3 \cdot 10^4 - 4 \cdot 10^4$)	—

The line designating the thermal characteristics of surface No. 2 has a break downward at $AN_0 = 160$ kcal/h $\cdot$ m^2 and, passing through a group of inefficient surfaces, is located below all the remaining lines. This is explained by the poor contact of the fin with the support tube.

As is seen from Fig. 1b, the thermal characteristics of the surfaces during the heating of an air stream are analogous to the characteristics obtained during cooling. Surfaces No. 3 and 7 are the most efficient. After their intersection at $AN_0 = 50$ kcal/h $\cdot$ m^2 the characteristics of surface No. 7 become better than the characteristics of surface No. 3.

Evaluation of the investigated surfaces with respect to their dimensions was carried out for the same heat extraction and for the same specific energy consumptions for overcoming pressure drop. The dimensions of the surfaces are determined by the degree of efficiency and compactness. The degree of efficiency is determined from Fig. 1 and the efficiency coefficient from the formula

$$\psi_i = \frac{\alpha_i}{\alpha_k}.$$

Surface No. 1 was taken as a standard surface. The efficiency coefficients of the surfaces compared were obtained by dividing the heat-transfer coefficients of the investigated surfaces by the heat-transfer coefficient of surface No. 1. We will use the compactness coefficient II for the characteristics of the volume occupied by the heat

TABLE 3

No. of tubular surface	Calculation formulas for aerodynamic resistance
1	$Eu = 16{,}2\,Re^{-0{,}255}$ $(Re = 10^4 - 10^5)$; $Eu = 0{,}86$ $(Re = 10^5 - 2\cdot 10^5)$; $Eu = 0{,}65\cdot 10^{-4}\,Re^{0{,}78}$ $(Re = 2\cdot 10^5 - 3\cdot 10^5)$
2	$Eu = 4{,}2\,Re$ $(Re = 4\cdot 10^3 - 9\cdot 10^3)$; $Eu = 36{,}8\,Re^{-0{,}24}$ $(Re = 9\cdot 10^3 - 7\cdot 10^4)$
3	$Eu = 17{,}5\,Re^{-0{,}227}$ $(Re = 2\cdot 10^3 - 7\cdot 10^4)$
4	$Eu = 192\,Re^{-0{,}461}$ $(Re = 6\cdot 10^3 - 1{,}3\cdot 10^4)$; $Eu = 37{,}2\,Re^{-0{,}287}$ $(Re = 1{,}3\cdot 10^4 - 6\cdot 10^4)$
5	$Eu = 24{,}5\,Re^{-0{,}187}$ $(Re = 2{,}5\cdot 10^3 - 3\cdot 10^4)$
6	$Eu = 10{,}5\,Re^{-0{,}09}$ $(Re = 5\cdot 10^3 - 8\cdot 10^4)$
7	$Eu = 17{,}3\,Re^{-0{,}2}$ $(Re = 4\cdot 10^3 - 8\cdot 10^4)$
8	$Eu = 18{,}5\,Re^{-0{,}23}$ $(Re = 6\cdot 10^3 - 7\cdot 10^4)$; $Eu = 1{,}4$ $(Re = 7\cdot 10^4 - 1{,}6\cdot 10^5)$
9	$Eu = 20{,}6\,Re^{-0{,}24}$ $(Re = 1{,}3\cdot 10^4 - 1{,}8\cdot 10^5)$
10	$Eu = 1{,}99\,Re^{-0{,}077}$ $(Re = 4\cdot 10^3 - 6\cdot 10^4)$

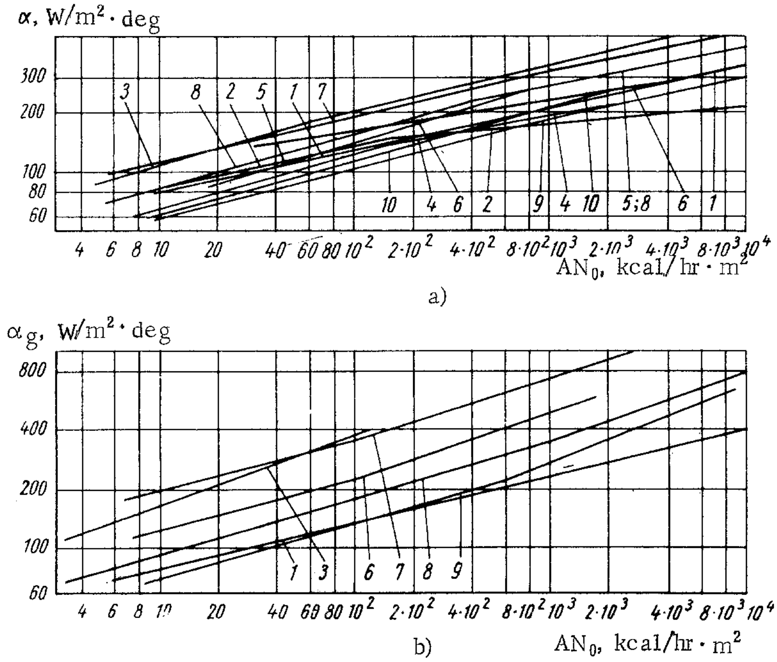


Fig. 1. Plot of the heat-transfer coefficients α_g of various tubular surfaces versus the energy AN_0 expended for circulating the heat-transfer fluid. The designations of the tubular surfaces are the same as in Tables 1-3; a) cooling of air; b) heating of air.

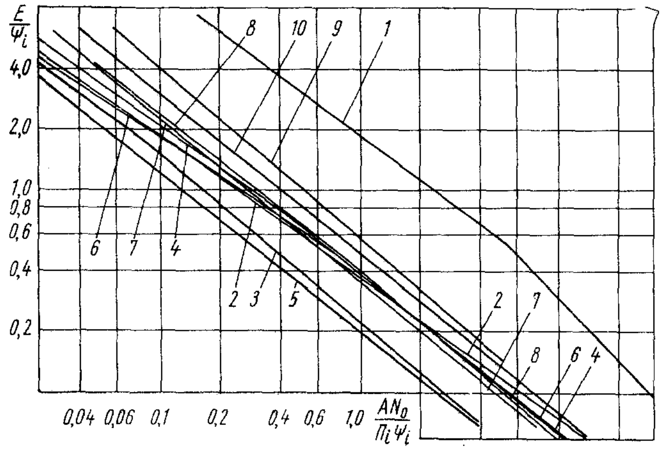


Fig. 2. Plot of the dimensional characteristics $AN_0/\Pi_i\psi_i$ versus the reduced energy coefficient E_i/ψ_i of various tubular surfaces during the cooling of air. Designations of the tubular surfaces are the same as in Tables 1-3.

transfer surface; Π is the magnitude of the surface in 1 m^3 of volume. For realization of the condition of identical heat extraction at AN_0= idem, it is necessary to divide the values of $E_1, E_2, E_3, \ldots, E_{10}$ by the coefficients $\psi_1, \psi_2, \psi_3, \ldots, \psi_{10}$. By taking surface No. 1 as the standard we will obtain values of E reduced to the same heat extraction:

$$E_1 = \frac{E_2}{\psi_2} = \frac{E_3}{\psi_3} = \ldots = \frac{E_{10}}{\psi_{10}}.$$

If the quantity of heat transferred by the heat transfer surface of the apparatus is Q = idem, and the temperature difference equals 1°C, then

$$\alpha_i F_i = \alpha_k F_k,$$

whence

$$\frac{F_i}{F_k} = \frac{\alpha_k}{\alpha_i} = \frac{1}{\psi_i}; \quad F_i = \frac{F_k}{\psi_i},$$

The volume of the i-th surface is expressed by the formula

$$V_i = \frac{F_i}{\Pi_i} = \frac{F_k}{\Pi_i \psi_i}.$$

$F_k = 1\ m^2$ is taken as the standard for evaluating the dimensions. The expression $1/\Pi_i\psi_i$ is a typical dimensional characteristic taking into account the geometric compactness and thermal efficiency of the heat transfer surface.

From Fig. 2, it is evident that the dimensional characteristics are clearly divided into three groups.

The most compact surfaces No. 3 and 5 fall into the first group, which is explained by the large values of the product $\Pi_i\psi_i$, although there are also other reasons. The less compact surfaces No. 6, 7, 4, 2, and 8 fall into the second group.

Despite the fact that these surfaces have different thermal efficiencies and compactness coefficients, they are practically equivalent with respect to their dimensional characteristics because they have close values of the product $\Pi_i\psi_i$. The least compact surfaces No. 9, 10, and 1, combining low efficiency coefficients ψ_i with low compact- Π_i, fall into the third group. The dimensional characteristics obtained during the heating of an air stream are analogous and are not cited in this work.

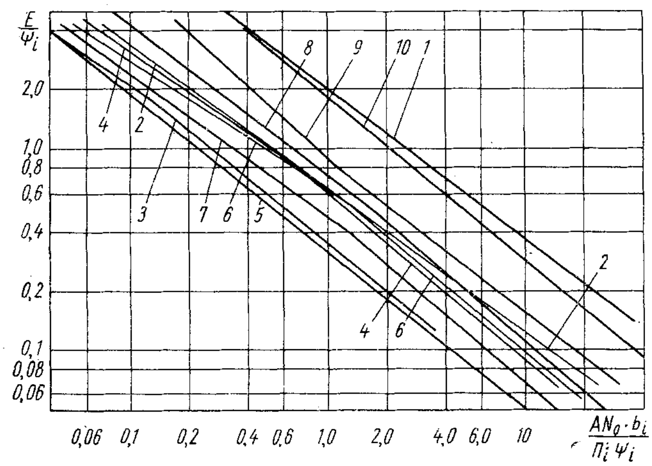


Fig. 3. Plot of the weight characteristics $AN_0b_i/\Pi_i\psi_i$ versus the reduced energy coefficient E/ψ_i for various tubular surfaces during the cooling of air. Designations of the tubular surfaces are the same as in Tables 1-3.

The weight characteristics, like the dimensional, were obtained for the surfaces compared during the cooling of air.

As is seen from Fig. 3, all of the weight characteristics are divided into three groups. Into the first, most efficient, fall surfaces No. 3, 5, and 7. The first two of these, having the best dimensional characteristics, possess values of b (b is the weight of 1 m^2 of heat transfer surface) that are close in magnitude and sufficiently low. The thermally efficient surface No. 7 is less compact than surfaces No. 3 and 5, but the size of b_7 is less than that of b_3 and b_5; therefore, its weight characteristic is located closer to the characteristics of surfaces No. 3 and 5 than on the plot of the dimensional characteristics. Surfaces No. 6, 2, 4, and 8, with sufficiently close values of the coefficient $\Pi_i\psi_i$, and b_i, fall into the second group. They are located in a group above the characteristics of the first group. The least efficient surfaces with respect to the weight characteristics, No. 9, 10, and 1, fall into the third group.

For the same reason as the dimensional characteristics, the weight characteristics obtained during the heating of an air stream are not cited in this article.

The grouping characteristics of the surfaces are determined analogously to the weight and dimensional characteristics for the condition Q = idem and N_0 = idem. To maintain this condition the stream velocities of the surfaces compared must be different. The relationships of the velocities for the i-th and k-th surfaces are determined from the equality $N_i = N_k$. In addition,

$$N_i = B_i \rho_i w_i^3; \; N_k = B_k \rho_k w_k^3,$$

where ρ_i and ρ_k are air densities.

For the case when $\rho_i = \rho_k$,

$$B_i w_i^3 = B_k w_k^3; \; \frac{w_i}{w_k} = \sqrt[3]{\frac{B_k}{B_i}}.$$

Frontal sections can be found from the condition $f_i w_i = f_k w_k$ where f_i and f_k are the free cross sections for passage of the working media.

Hence,

$$\frac{f_k}{f_i} = \frac{w_i}{w_k} = \sqrt[3]{\frac{B_k}{B_i}}.$$

For lateral streamlining

$$B_i = 2.69\,(\sigma_1 - 1)\,\mathrm{Eu}_0,$$

where σ_1 is the transverse pitch of the bundle ($\sigma_1 = S_1/d$).

The investigation conducted permits the following conclusions.

Surfaces No. 3 and 7 are the most thermally efficient. In the range of large Reynolds numbers surface No. 5 adjoins them. With respect to dimensional characteristics surface No. 7 leaves the group of most efficient surfaces and surfaces No. 3 and 5 remain in it. With respect to the weight characteristics the most efficient group is comprised of surfaces No. 3, 5, and 7.

Surfaces No. 2, 6, and 8, which differ only slightly in their characteristics, belong to the group that are average with respect to all indexes. These surfaces can be successfully replaced by surface No. 8 which possesses good technological properties.

Surfaces No. 9 and 10 are in the low efficiency group according to all indexes; with respect to thermal characteristics surface No. 4 adjoins them. These surfaces can not be recommended for use in heat exchange apparatus; therefore, it is expedient to take them out of production.

As the experimental investigations showed, surfaces with noncircular profiles (approximating streamlined) do not have, in principle, advantages over surfaces with a circular profile. On the basis of this, surface No. 7 should be considered the most efficient surface in spite of some advantage of surface No. 3 in dimensional characteristics. This choice is predicted by the following considerations.

Rolling mills that prepare tubes with seamless finning are distinguished by efficiency and high productivity; they permit the production of any sizes of fins and tubes required in practice, whereas mechanization of the production of finned tubes having a noncircular profile is rendered difficult because of the large number of manufacturing operations of varying nature. In addition, monolithic joining of the fin to the support tube allows avoidance of the use of expensive solders (comprising up to 30% of the cost of the whole tube) and avoidance of high contact resistances (which increase during use because of vibration) at the point where the fins join the support tube. And finally, the production of tubes having seamless rolled fins permits replacing scarce nonferrous metals with aluminum and steel and at the same time allows the use of bimetallic tubes for operation in corrosive media.

The use of tubes having seamless rolled fins is dependent on centralized production of the optimum surfaces satisfying the broadest range of operating conditions.

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