

PAPER • OPEN ACCESS

Calculations of air cooler for new subsonic wind tunnel

To cite this article: A S Rtishcheva 2017 *J. Phys.: Conf. Ser.* **891** 012042

View the [article online](#) for updates and enhancements.

You may also like

- [Features of flow around transport aircraft model with running propellers by modelled engine failure in wind tunnel](#)
V I Chernousov, A A Krutov and E A Pigusov
- [Energy approach to transonic flutter and LCO with shock waves movement](#)
S Kuzmina, F Ishmurov and O Karas
- [Study on a twin-fuselage transport airplane model in a low speed wind tunnel](#)
V I Chernousov, A A Krutov and E A Pigusov



ECS
The
Electrochemical
Society
Advancing solid state &
electrochemical science & technology

DISCOVER
how sustainability
intersects with
electrochemistry & solid
state science research

Calculations of air cooler for new subsonic wind tunnel

A S Rtishcheva

Central Aero Hydrodynamic Institute named after Professor N. E. Zhukovsky (TsAGI)
Russia, 140180 Zhukovsky, Zhukovsky st., 1

Abstract. As part of the component development of TsAGI's new subsonic wind tunnel where the air flow velocity in the closed test section is up to 160 m/sec hydraulic and thermal characteristics of air cooler are calculated. The air cooler is one of the most important components due to its highest hydraulic resistance in the whole wind tunnel design. It is important to minimize its hydraulic resistance to ensure the energy efficiency of wind tunnel fans and the cost-cutting of tests. On the other hand the air cooler is to assure the efficient cooling of air flow in such a manner as to maintain the temperature below 40 °C for seamless operation of measuring equipment. Therefore the relevance of this project is driven by the need to develop the air cooler that would demonstrate low hydraulic resistance of air and high thermal effectiveness of heat exchanging surfaces; insofar as the cooling section must be given up per unit time with the amount of heat $Q=30$ MW according to preliminary evaluations. On basis of calculation research some variants of air cooler designs are proposed including elliptical tubes, round tubes, and lateral plate-like fins. These designs differ by the number of tubes and plates, geometrical characteristics and the material of finned surfaces (aluminium or cooper). Due to the choice of component configurations a high thermal effectiveness is achieved for finned surfaces. The obtained results form the basis of R&D support in designing the new subsonic wind tunnel.

1. Introduction

One of TsAGI scientific research trends is focused on the development of wind tunnel components. Presently, current works in TsAGI include the design of new subsonic wind tunnel whose circuit consists of open and closed test sections with the size of nozzle exit section 7.5×5.5 m. The air flow velocity in the open test section is up to 140 m/sec, in the closed test section – up to 160 m/sec. Preliminary evaluations shows that the power of fan drive is 30 megawatt.

In return-flow wind tunnels the power conveyed to the fan is expanded to energize the working gas so as the kinetic energy maintains air circulation in the tunnel. Due to friction the mechanic energy is converted into thermal energy resulting in air heating and surface heating. When the temperature constantly rises, the inaccuracy in model aerodynamic characteristic measurements occurs and the model together with tunnel circuit components is uncontrollably deformed. Stable operation of measurement equipment requires that the air temperature in the wind tunnel test section can be only below 40 °C therefore the air cooling is necessary.

The air cooler is suggested to be installed into the wind tunnel stilling chamber whose size is 16.5×22.5 m (width×height). This project is challenging because of large linear dimensions and strict



requirements to meet as for low hydraulic resistance of air ($\Delta p \leq 500$ Pa). In its turn it requires such a configuration of heat-exchanging surfaces which would provide their high thermal effectiveness.

Thus, the objective of this study is the aerodynamic design of air cooler, and the definition of its hydraulic and thermal parameters basing on the studies of temperature patterns and heat-exchanging surface effectiveness.

The results obtained during the studies have become the ground of R&D support in designing the components of subsonic wind tunnel.

2. Description of air cooler

On basis of the performed R&D analysis of air cooling systems of wind tunnels in TsAGI, German and Dutch LLF DNW, and Chinese ARI, the air cooler scheme for the new subsonic wind tunnel is proposed; figure 1 shows its 3D model.

The air cooler is rectangular in shape, its size is 16.5×22.5 m (width \times height), its length is $L \approx 0.5$ m, it contains 30 modules set into two rows (figure 2). The dimensions of one module are $8.25 \times 1.5 \times 0.5$ m.

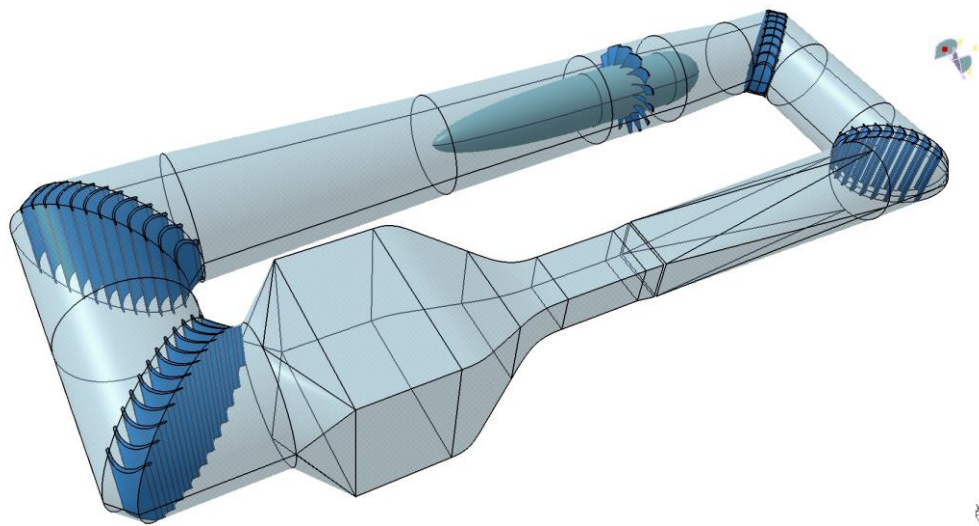


Figure 1. Subsonic wind tunnel (closed test section).

The initial design suggested that each module contains 60 vertical tubes of elliptical cross section. The length of each tube is 8.25 m. The cross section major axis is 0.06 m, the minor axis is 0.02 m, and the wall thickness is 0.0016 m. The ellipse lateral (major) axis coincides with the air flow direction. Rectangular fins are installed normally to the tubes on their external side. Each module has 1016 yellow cooper fins; the dimensions of one fin is $1.5 \times 0.5 \times 0.0003$ m. The distance between fins is 7.8 mm.

3. Calculation method of thermal and hydraulic characteristics

Air mass flow G , and average flow rate on the air cooler clear opening w can be found with the flow rate in the test section:

$$G = w\rho F = w_{\max}\rho_{TS}F_{TS}, \quad (1)$$

where F – area of air cooler clear opening with account for its shading;

F_{TS} – area of nozzle exit ($F_{TS}=40.75$ m²).

Air density in the test section is expressed with the density in the stagnation point:

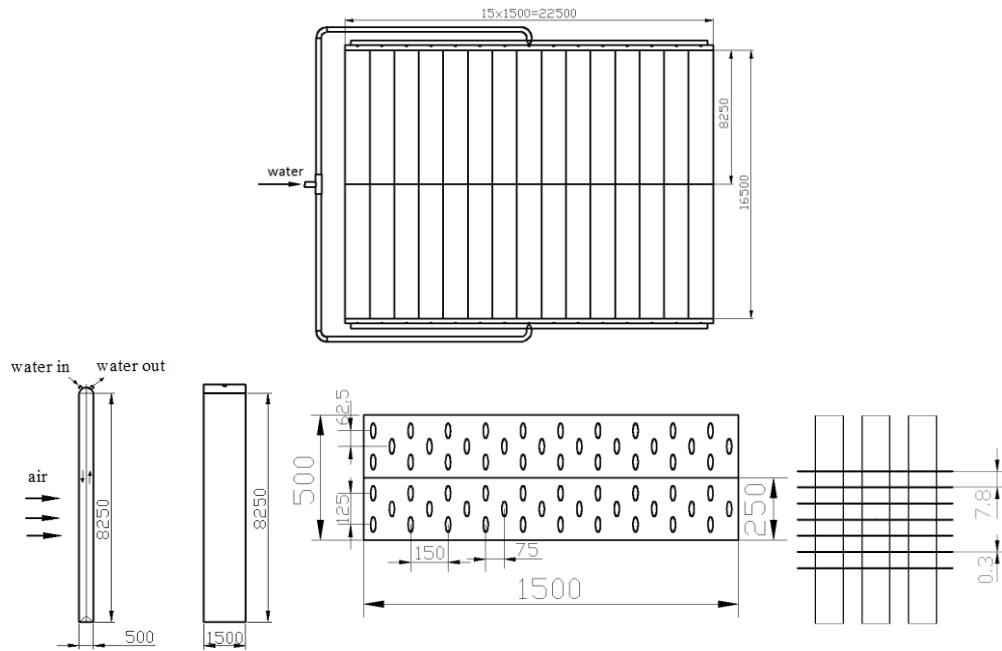


Figure 2. Air cooler drawing.

$$\rho_{TS} = \rho^* \frac{1}{\left[1 + \frac{\kappa - 1}{2} \left(\frac{w_{TS}}{a} \right)^2 \right]^{2.5}}, \quad (2)$$

where κ – adiabatic exponent;

$a = 20.1\sqrt{T^*}$ – local velocity of sound.

Let us suppose that $T^* = 298$ K; $\kappa = 1.4$. Then $G = 6963.4$ kg/s.

With neglect of wind tunnel structure heating and component heating the air flow during the time $\Delta\tau = 1$ sec. will be heated for $\Delta t_{\max} \approx 5$ °C.

The factor of air flow hydraulic resistance is calculated with the formula [1]:

$$\xi = 2Eu_{f_2} z, \quad (3)$$

where Eu_{f_2} – Euler number;

z – number of rows in a matrix.

For the staggered arrangement of finned tubes (if $Re_{f_2} = 10^3 \dots 10^5$):

$$Eu_{f_2} = 3.2\varepsilon^{0.5} Re_{f_2}^{-0.25} a^{-0.55} b^{-0.5}, \quad (4)$$

where ε – finned surface factor;

Re_{f_2} – Reynolds number (for average air temperature in the air cooler);

a ; b – geometrical constants of staggered arrangement of tubes [1-3].

The calculation of hydraulic resistance of tubes on the water side is made with the formulae presented in [4].

The mean thermal conductivity coefficient (as the ratio between the area and its external surface) is:

$$K = \frac{1}{\frac{1}{\alpha_1} \cdot \frac{F_{\text{ex}}}{F_{\text{in}}} + \frac{F_{\text{ex}}}{2\pi\lambda nl} \ln \frac{d_2}{d_1} + \frac{1}{\alpha_2}}, \quad (5)$$

where α_1 – heat-exchange coefficient on the water side;

α_2 – heat-exchange coefficient on the air side;

λ – heat-exchange coefficient for tubes;

n – number of tubes;

F_{in} – area of tube surface on the water side;

F_{ex} – area of tube external surface.

The heat-exchange coefficient on the water side is found with M. Mikheev's similarity solutions.

The heat-exchange coefficient of finned surface on the air side is:

$$\alpha_2 = \alpha \left[1 - (1 - \psi E) \frac{F_{ed}}{F_{ex}} \right], \quad (6)$$

where α – heat-exchange coefficient of external surface of tubes;

ψ – correction for fin effectiveness because of the irregularity in the distribution of the heat-exchange coefficient on the fin surface;

E – fin effectiveness coefficient;

F_{ed} – fin area.

Coefficient α is defined with the similarity solution presented in [1] (where $Re_{f_2} = 2 \cdot 10^4 \dots 2 \cdot 10^5$):

$$Nu_{f_2} = 0,0507 \left(\frac{a}{b} \right)^{0,2} \left(\frac{s}{d} \right)^{0,18} \left(\frac{h}{d} \right)^{-0,14} Re_{f_2}^{0,8} Pr_{f_2}^{0,4} \left(\frac{Pr_{f_2}}{Pr_{w_2}} \right)^{0,25}, \quad (7)$$

where h – fin height;

s – geometric parameter equal to the amount of distances between fins and fin thickness;

Pr_{f_2} ; Pr_{w_2} – Prandtl number (with average air temperature in the air cooler and temperature of external surface of tube) $Pr_{f_2} \approx Pr_{w_2}$;

The coefficient of fin effectiveness E shows quite a difference between the heat transfer on the fin and the heat transfer on the tube with a coolant inside. Since the fin temperature on the distance h from the tube is higher than the temperature of tube external surface, the fins absorb heat worse than the coolant tubes, therefore their effectiveness is always less than 1.

If we know a fin height, a fin thickness and a coefficient of heat exchange of fin material we can evaluate the fin effectiveness according to the data [1]:

$$E = \frac{\text{th}(\beta h)}{\beta h}, \quad (8)$$

$$\beta = \sqrt{\frac{2\alpha}{\lambda \delta}}. \quad (9)$$

The correction for the wall effectiveness with the consideration of irregularity in the distribution of heat exchange,

$$\psi = 0,97 - 0,056\beta h. \quad (10)$$

In the initial structure the fin effectiveness $E=0,42$ was low enough. So there are two solutions: to double the number of fins in the initial structure or to increase the fin effectiveness. Besides, the calculations show that the elliptical section of tubes is rather efficient as for the decrease of air flow hydraulic resistance but it leads to the deformation of tubes during the operation (they tend to take round shape). Thus, further considered structures have round section tubes.

4. Thermal effectiveness of fin

To avoid excessive hydraulic resistance (of air) and metal waste on producing fins whose surfaces are not used in the process of heat exchange it is necessary to evaluate their effectiveness.

The design performance of initial air cooler is studied in terms of dependence between the fin effectiveness and their height, thickness and thermophysical properties of material.

Figure 3 shows the graph where the effectiveness of fins made of material whose thermal conductivity coefficient is higher is better, other things being equal. Effectiveness $E=0.85$ can be obtained by using the fins $\delta \approx 1.5$ mm thickness and $h \leq 0.035$ m height.

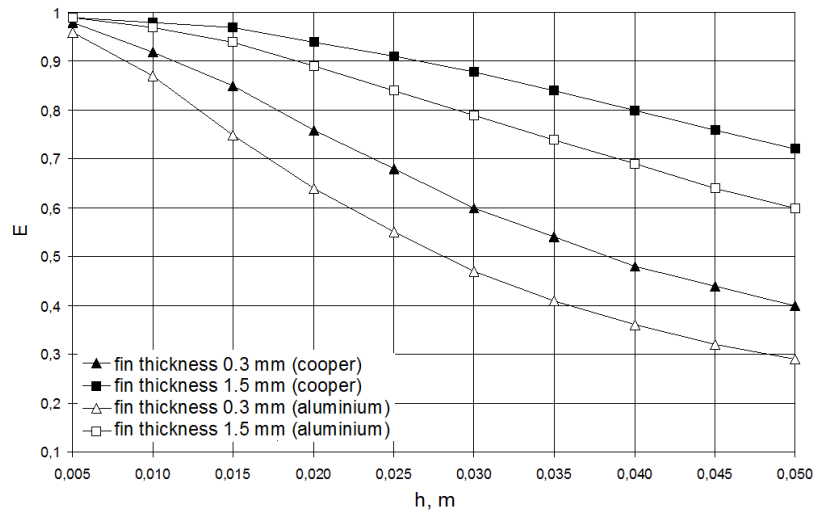


Figure 3. Dependence between the fin effectiveness and its height.

5. Final structure of air cooler

The analysis of manufacturing technology for finned surfaces shows that the thickness of plates is often not more than 0.25 mm. The largest producer of heat-exchange equipment Icarus-hex-group can change their standard forming and can manufacture 0.4 mm thickness plates. Further evaluation of air cooler design for the subsonic wind tunnel shows that the dimensions of each module can be $8.2 \times 1.4 \times 0.36$ if separate modules are installed in the stilling chamber. The modules are disposed according to the scheme shown on figure 2. If thinner plates are used tubes of less diameters are to be installed more closely (the chosen tube cross section is 21.3 mm). The basic configuration is shown on figure 4.

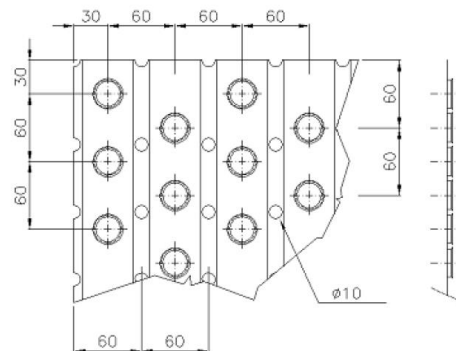


Figure 4. Disposition of tubes in the air cooler.

The stainless tubes are 8.2 m long each. The tube thickness is 1.5mm. Each module has 138 tubes and 975 aluminium fins, the dimensions of each fin are 1.4×0.36×0.0004 m. The distance between fins is 8.4 mm.

The calculations use the following fixed parameters: temperature of incoming water $t'_1=20\text{ }^{\circ}\text{C}$ (in summer), $t'_1=5\text{ }^{\circ}\text{C}$ (in winter); temperature of outgoing water $t''_1=27\text{ }^{\circ}\text{C}$; temperature of outgoing air $t''_2=35\text{ }^{\circ}\text{C}$; temperature of incoming air $t'_2=40\text{ }^{\circ}\text{C}$.

Air consumption is $G=6963.4\text{ kg/sec}$.

Average velocity of flow in the most narrow section part of air cooler is $w_2=29.2\text{ m/sec}$. The air-contacting surfaces of air cooler are $F_{ed}=26290.7\text{ m}^2$; $F_{ex}=28424.3\text{ m}^2$, consequently. The heat transfer coefficient of finned surface is $\alpha_2=105.5\text{ W/(m}^2\cdot\text{K)}$. The fin effectiveness is $E=0.57$. The heat transfer coefficient disregarding the contamination of air cooler surfaces is $K=85.9\text{ W/(m}^2\cdot\text{K)}$. The heat transfer coefficient regarding the contamination is approximately $K\approx 81.0\text{ W/(m}^2\cdot\text{K)}$, and the necessary area of heat exchange surface is $F\approx 27034.3\text{ m}^2$.

Thus, the heat exchanging surface margin is 5 %.

The coefficient of hydraulic resistance for air is $\xi=3.31$ ($\Delta p=458\text{ Pa}$).

In the software environment MatLab the calculations of temperature field of finned surface are made. According to the calculations the average air temperature in the air cooler is $t_{p2}=37.5\text{ }^{\circ}\text{C}$, and the temperature of external surface of tube (at the bottom of fins) is $t=27\text{ }^{\circ}\text{C}$. The temperature distribution on the fin is shown on figure 5.

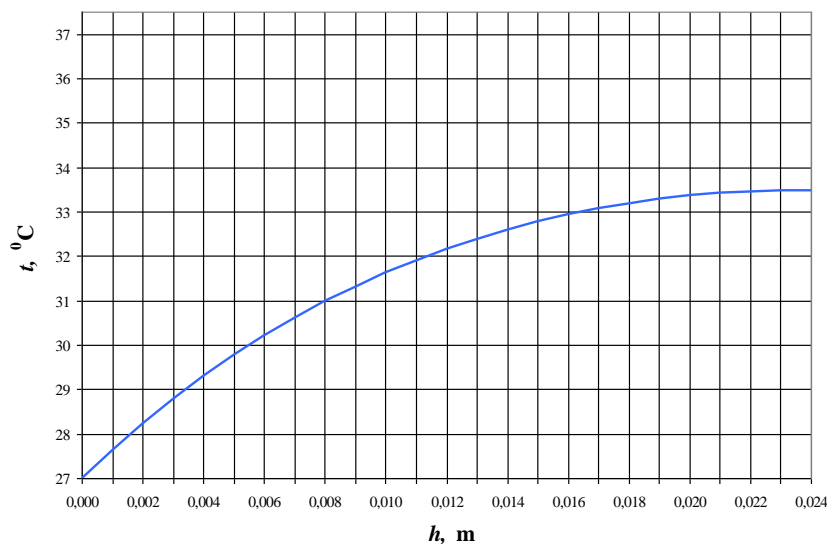


Figure 5. Fin temperature field.

Thus, the entire surface of the fin participates in the heat exchanging. Also the temperature along the fin is augmented by $5.2\text{ }^{\circ}\text{C}$.

6. Conclusion

Basing on the results of calculations some configurations of air cooler are proposed to provide 30 megawatt heat take-off. The configurations represent a recuperative type of cross flow design water-air heat exchanger and consist of six rows of tubes with lateral fins in the form of plates.

During the studies the dependencies are analyzed to calculate hydraulic and thermal parameters of air cooler; this analysis contributes to low hydraulic resistance on the air side ($\Delta p\leq 500\text{ Pa}$) and operating temperatures in the test section below $40\text{ }^{\circ}\text{C}$.

It is shown that the effectiveness of heat exchanging surfaces rises with the fin thickness and depends on the fin material. The fins have higher effectiveness if they are made of material with higher heat exchange coefficient, for example, cooper or aluminium.

The obtained results have become the ground of R&D support for designing the large subsonic wind tunnel and they have been generalized for this class of wind tunnels.

References

- [1] Zhukauskas A.A. Convective transport in heat exchangers. – M. : Nauka, 1982.
- [2] Mukhachev G.A., Shchukin V.K. Thermodynamic and heat exchange. – M. : Vysshaya shkola, 1991.
- [3] Isachenko V.P., Osipova V.A., Sukomel A.S. Heat exchange. – M. : Energiya, 1969.
- [4] Idelchik I.E. Guide on hydraulic resistance. – M. : Mashinostroenie, 1975.