Comparative study of performance of heat exchangers used in hypersonic wind tunnels

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Abstract. During the modernization of the TsAGI hypersonic wind tunnel, which implied an increase in the limiting values of stagnation parameters, and, consequently, the heat load, the operation of the main air cooler structures was analyzed. Two qualitatively different options are considered: Option-1, in which the flow moves in a large number of longitudinal tubes placed in a common compartment with cooling water circulating in the intertube area, and Option-2, that is a bundle of tubes (tubes of circular or elliptical section were considered) containing a moving cooler (water), and the air flow transversely moves in the intertube gaps. In the study, a previously worked out and experimentally verified approach to numerical simulation was implemented, according to which a thorough calculation of the flow and heat transfer was carried out in all areas of the duct with cooled walls (prechamber, nozzle, test section, diffuser) and a cooling compartment for the mode $p_0=12$ atm; $T_0=5000$ K; $M\approx7$. The ANSYS FLUENT software package solved axisymmetric Navier-Stokes equations for viscous and heat-conducting gas using the Spalart-Allmaras turbulence model. For the flow in the area of the cooling compartment, the simplified Darcy’s model and the model of the “schematized axisymmetric tube bundle” were considered.

1. Introduction
Improvement of hypersonic aircraft depends on capabilities of experimental facilities. TsAGI has a hypersonic wind tunnel designed to conduct research on heat resistance of thermal protection for rockets and spaceships; the flow parameters of its test section are Mach number $5<M<8$, stagnation temperature and stagnation pressure $T_0=4000$ K, $p_0\leq10$ atm. Expanding the range of test modes to $T_0=5000$ K, $p_0=12$ atm requires gas dynamic studies of the test section duct of the hypersonic wind tunnel taking into account heat transfer processes in the cooling compartment and on the walls of its circuit. In addition, an increase in heat load entails a change in the design of the air cooler.

It should be noted that at the entrance to the cooling compartment of the hypersonic wind tunnel, the flow is very heterogeneous. The flow core can remain supersonic even after stagnation in the diffuser and have the temperature approximately equal to the stagnation temperature in the prechamber of the nozzle. Therefore, under the influence of high-temperature central jet, deformation or damage to the surfaces of the air cooler can occur.

This paper presents the results of computational studies of the flow and heat transfer in the nozzle, in the test section, in the diffuser and in the air cooler of the hypersonic wind tunnel at $p_0=12$ atm; $T_0=5000$ K; $M\approx7$. Two qualitatively different versions of the air cooler are considered: Option-1, in which the air flow moves in a large number of longitudinal tubes placed in a common compartment...
with cooling water circulating in the intertube, and Option-2 (prototype of the existing air cooler) in
the form of a bundle of serpentine tubes (round or elliptical section), in which moves the cooler -
water, and the cooled air crossflows in the intertube gaps. The advantages and disadvantages of using
the considered options are discussed. The problems of redistributing the heat load of a high-
temperature jet over the surface of a heat exchanger are considered.

2. Calculation of the flow and heat transfer in the duct of hypersonic wind tunnel

Figure 1 shows a direct-flow hypersonic facility which includes: an electric arc heater with an electric
power of \( N = 0.8 \times 10^6 \) W, which is necessary to create the required stagnation parameters in the
prechamber \( (p_0 = 3 \times 10 \text{ atm}; T_0 = 1600 \pm 4600 \text{ K}) \); the nozzle whose circuit provides the specified Mach
numbers; the test section – to study the aerodynamic heating of the model; the diffuser – to stagnate
the flow and the cooling compartment (the air cooler and the heat exchanger) – to cool the operating
gas before being fed into the ejection system [1–3].

![Figure 1. Scheme of straight-through hypersonic wind tunnel.](image)

In the existing air cooler, which is a bunch of tubes, inside which the cooler – water – moves; it
was observed that the long start-up time leads to the formation of steam plugs, leading to tube rupture
and water entering the duct. In addition, the air temperature at the inlet to the ejector system was
unacceptably high.

Wider test range of the considered facility including the following stagnation parameters:
\( p_0 = 12 \text{ atm}; T_0 = 5000 \text{ K} \) with increased electric power of the heater \( N = 1.3 \times 10^6 \) W, requires studying the
features of gas dynamics and heat transfer in the circuit, calculating the required heat capacities and
heat losses, obtaining data for designing a new air cooler.

The study implements a streamlined and verified approach to modeling the flow and heat transfer
in the wind tunnel duct, according to which the axisymmetric Navier-Stokes equations are solved in all
areas (prechamber, nozzle, test section, diffuser, air cooler) with ANSYS FLUENT software package
(TsAGI license No. 501024) for viscous and heat-conducting gas using the Spalart-Allmaras
turbulence model [4–6]. The air cooler was modeled as a semi-permeable region, where the law of
resistance of a porous medium (Darcy’s Law) was added to the Navier-Stokes equations:

\[
\Delta p = \left( C_1 u + 0.5 C_2 \rho u^2 \right) h, \tag{1}
\]

where \( \Delta p \) – differential pressure before and after the heat exchanger;

\[
C_1 = \frac{\mu}{\alpha} \quad \text{– coefficient of linear resistance;}
\]

\[
C_2 \quad \text{– coefficient of nonlinear resistance;}
\]

\[
\mu \quad \text{– dynamic viscosity;}
\]

\[
\alpha \quad \text{– permeability of the partition (of the heat exchanger);}
\]

\[
u \quad \text{– normal component of the area-average velocity at the inlet to the heat exchanger;}
\]

\[
\rho \quad \text{– gas density;}
\]

\[
h \quad \text{– thickness of the partition.}
\]
Coefficients \((C_1=0; C_2=194.7 \text{ m}^{-1})\) of dependence (1) were obtained as a result of series of gas dynamics calculations and Option-1 heat transfer (see below).

The gas-dynamic and thermophysical properties of the flow were modeled on the basis of the molecular-kinetic theory of gases, on the assumption that a mixture moves in the duct: \(\text{N}_2; \text{O}_2; \text{Ar}; \text{CO}_2\) in proportions corresponding to air. Chemical reactions, which take place mainly in the nozzle prechamber, were not considered at this stage of the study, but the change in the heat capacity of the gas mixture because of temperature was taken into account. The temperature on the duct walls was \(T_w=350\) K (the walls are cooled with water). At the exit of the computational domain, “soft” boundary conditions were set with a pressure level selected taking into account the capacity of ejectors and the stationary flow regime in the test section and in the diffuser \((p=350\) Pa).

After checking the mesh independence, the results showed that, in order to obtain a qualitative resolution of the viscous sublayer of the turbulent boundary layer in the wall region, an acceptable resolution of the shock waves and flow separation zones, for the axisymmetric problem, a computational grid containing \(~0.8\) million cells is sufficient. A grid with so many cells was used as the base. However, to obtain the best quality results, all final calculations were performed on a more detailed grid containing \(~1.1\) million cells.

The results of the calculation of the flow and heat transfer in the duct of the hypersonic wind tunnel are presented in Figures 2 and 3.

In the nozzle and in the test section, the flow velocity increased, reaching \(M\approx 7\), while the static pressure and temperature decreased to \(p\approx 300\) Pa and \(T\approx 500\) K. The air flow rate was \(G=0.075\) kgs\(^{-1}\). At the entrance to the diffuser, the Mach number continued to increase, reaching the value of \(M\approx 9\). Then oblique shock waves could be considered in the diffuser, due to which the velocity and stagnation pressure sharply decreased. Accordingly, the value of static pressure along the length of the diffuser increased. In the expanding part of the diffuser, a separation of the flow from the walls was observed, while near the walls of the cooling compartment the formation of recirculation zones occurred. The recirculation zones were also present in the test section and affected the jet flowing from the nozzle.

The distributions of the values of the flow parameters at the inlet and outlet of the air cooler are presented in Figure 4.

It was established that the thermal power required to implement the regime with the maximum stagnation parameters differed significantly from the electric power and amounted to \(Q=0.462\cdot 10^6\) W. In the duct of the hypersonic wind tunnel, the thermal power \(Q_w=0.116\cdot 10^6\) W was removed through the walls to the air cooler inlet, and the total temperature decreased from \(T_{01}=5000\) K to \(T_{02}=3410\) K.

Thus, the value of the maximum thermal power absorbed in the cooling compartment had to be \(Q_{\text{max}}=0.346\cdot 10^6\) W, and to ensure an acceptable output temperature for the operation of the ejector \((T\approx 600\) K) \(Q_{\text{cooler}}=0.3\cdot 10^6\) W.

Figure 2. Flux line.
Mach number

Total pressure

Static pressure

Total temperature

Static temperature

**Figure 3.** Flow calculation in the duct of hypersonic wind tunnel.
3. Calculation of the main design options for the air cooler

As mentioned earlier, two qualitatively different design options for the heat exchanger were considered in the study (Figure 5).

The coefficients of the porous wall resistance which was used to simulate the flow in the duct of the hypersonic wind tunnel correspond to the actual parameters of Option-1 heat exchanger. The heat exchanger is a horizontal cylindrical tank, inside of which there are \( n \approx 900 \) steel tubes with inner diameter \( d_{\text{тр}} = 0.01 \) m. At the first stage Steel 20 was taken (\( \rho = 7845 \text{ kgm}^{-3}; \ c = 461 \text{ Jkg}^{-1}\text{K}^{-1}; \ \lambda = 58 \text{ Wm}^{-1}\text{K}^{-1} \)). The tubes were located along the length of the tank and parallel to the direction of the flow core. Air moved inside the tubes. Cooling was carried out due to the movement and heat transfer of water in the intertube space.

In accordance with the data presented in Figure 4, the following options for the implementation of air flow in tubes were considered (Table 1).

At the first stage of calculations, it was assumed that the temperature of the outer surface of the tubes and mounting discs was \( T_{\text{w}} = 350 \) K. It should be noted that in the peak mode (Mode 1) only the central tube of the heat exchanger operated, while in Mode 7 – up to 40% of the tubes.

**Figure 4.** Distribution of parameters along the radius of the air cooler.
Figure 5. Options of air cooler design.

Figure 6 shows some results of the calculation of the flow and heat transfer in the air cooler tubes.

The subsonic flow regime was realized along the length of the tubes. The air outflow from the central tubes was supersonic, the outflow from the tubes at the periphery (Mode 7) was subsonic. It should be noted that at a relatively small distance from the outlet cross section of the air cooler tubes, the flow became completely subsonic.

The average total temperature of the flow at the outlet of the air cooler was \( T_0 \leq 600 \text{ K} \), and the average static temperature was \( T = 570 \text{ K} = 300 \text{ °C} \). The air cooler provided the removal of \( Q_{\text{cooler}} \approx 0.3 \times 10^6 \text{ W} \) of thermal energy and a quite acceptable flow temperature at the inlet to the ejector.

Increasing the value of thermal power which the air cooler discharges is possible by increasing the length of the tubes or reducing their diameter. However, it is advisable to reduce the diameter only for tubes operating in modes No. 1 and No. 2 (central tubes) so as not to increase the overall resistance of the heat exchanger.

Option-2 of the air cooler is a staggered bundle of tubes, inside of which the cooler (water) moves. Structures with round tubes (external diameter 0.012 m) and elliptical tubes (external dimensions: major axis \( a = 0.0089 \text{ m} \); minor axis \( b = 0.006 \text{ m} \)) were considered.

### Table 1. Modes of operation for the heat exchanger tubes

| Number | \( r/R \) (radius of the tube from the axis of symmetry, referred to the radius of the heat exchanger) | Inlet parameters | Outlet parameters |
|--------|-------------------------------------------------|-----------------|------------------|
|        |                                                 | \( p_0, \text{ Pa} \) | \( T_0, \text{ K} \) | \( \mu_0/\mu \) | \( p, \text{ Pa} \) |
| 1      | 0                                               | 20760           | 4470             | 400             | 644             |
| 2      | 0.05                                            | 16000           | 4250             | 350             | 471             |
| 3      | 0.15                                            | 9000            | 3920             | 240             | 224             |
| 4      | 0.30                                            | 5000            | 3520             | 142             | 102             |
| 5      | 0.50                                            | 3300            | 3240             | 113             | 60              |
| 6      | 0.65                                            | 2380            | 3080             | 110             | 290             |
| 7      | 0.65-1.00                                       | 1830            | 2800             | 10              | 640             |
The purpose of calculating the flow and heat transfer in the air cooler was mainly to study the possibilities of using the existing structure and its modifications with increasing test parameters to maximum values: $p_0=12$ atm; $T_0=5000$ K; $M\simeq 7$.

When modeling at the inlet (the average cross section of the diffuser), the flow field obtained by calculating the duct of the hypersonic wind tunnel was set. The boundary conditions at the exit were selected taking into account the implementation of the stationary flow regime in the duct ($p \approx 150–350$ Pa). The temperature of the walls of the diffuser and the cooling compartment of the air cooler, as well as the surface of the tubes was $T_w=350$ K.

The calculation results are presented in Figure 7.

At the diffuser exit one could see the separation of the supersonic flow and the formation of recirculation zones near the walls of the air cooler inlet. In front of the tube bundle, a shock wave was observed that is close to a direct one, beyond which the flow changed from supersonic to subsonic. However, local supersonic zones were formed in the intertube area, especially in the central part of the tube bundle. The transition to a fully subsonic flow occurred at the outlet of the air cooler.

Despite the complexity of the flow, the spreading of the high-temperature central jet at an angle of $\sim 60^\circ$ between the tubes was clearly visible. A part of the tubes of the first and second row located at the periphery (near the wall of the cooling compartment) interacted with the recirculation zone formed in front of the tube bundle and did not actually participate in heat exchange with the main stream.

Due to the nature of the flow of the high-temperature jet in the cooling compartment, a significant number of tubes worked under conditions of high thermal loads. The heat flux density to the surface of the tubes could locally reach $q_{w,\text{max}} \approx (2–3) \times 10^6$ Wm$^{-2}$ (central tubes of the first row).

Such high heat loads made it impossible to use serpentine tubes, since boiling water could lead to their rupture. It would be possible to use short tubes, the flow of the cooler in which is organized in one direction from the inlet to the outlet manifold. This requires high flow rates of water in the tubes, and, consequently, high costs of the cooler. According to preliminary estimates, the flow rate in the air coolers of Option-2 should be more than 10 times higher than the flow rate for Option-1.
Figure 7. Flux lines, Mach number fields and stagnation temperature values in the air cooler.
In addition, the use of a single tube bundle (both round and elliptical) does not provide the required temperature at the outlet of the air cooler ($T_0 \approx 600$ K). The use of a double beam reduces the temperature at the outlet of the air cooler to $T_0 \approx 900$ K.

Thus, from the point of view of the possibilities of absorbing thermal energy, the cost of cooling water and the margin of the possibility of absorbing heat, the best air cooler is Option-1. However, both options do not provide uniform thermal loading of heat exchange surfaces.

4. Calculation of the modified design of Option-1 air cooler

The studies have shown that to ensure a more uniform thermal loading of the heat exchange surfaces (Option-1 of the air cooler), the static pressure behind the cooling compartment should be increased (with the help of the ejectors) and the most heat-loaded central tubes should be blocked. In this case, the high-temperature jet would interact with the surface of the tube plate and spread along this surface, loading the peripheral tubes. Using the method of successive approximations, it was found that by blocking the central tube and increasing the outlet pressure to $p=900$ Pa, the previous level of hydraulic resistance of the entire cooling module can be ensured.

A comparison of the pressure and temperature distributions at the inlet of the air cooler types Option-1.1 (in the presence of a central tube) and Option-1.2 (in the absence of a central tube and an increased value of the outlet pressure) are presented in Figure 8.

![Figure 8](image-url)
Table 2 shows main modes of airflow in the tubes of the air cooler (Option-1.2).

### Table 2. Modified heat exchanger tube modes

| Number | \( r/R \) (radius of the tube from the axis of symmetry, referred to the radius of the heat exchanger) | Inlet parameters | Outlet parameters |
|--------|-----------------------------------------------|------------------|------------------|
|        |                                               | \( p_0, \text{Pa} \) | \( T_0, \text{K} \) | \( \mu/\mu' \) | \( p, \text{Pa} \) |
| 1      | 0                                             | -                | -                | -                | -                |
| 2      | 0.05                                          | 10265            | 4210             | 67               | 1786             |
| 3      | 0.15                                          | 7920             | 3946             | 169              | 1635             |
| 4      | 0.30                                          | 5133             | 3736             | 159              | 1646             |
| 5      | 0.50                                          | 3840             | 3605             | 153              | 1657             |
| 6      | 0.65                                          | 3200             | 3525             | 148              | 1738             |
| 7      | 0.65-1.00                                     | 2900             | 3500             | 95               | 2000             |

Some results of the calculation of the flow and heat transfer in the tubes of the air cooler (Option-1.2) are shown in Figure 9.

**Figure 9.** Fields of Mach numbers and values of stagnation temperature for heat exchanger tubes (Option-1.2)

Compared to the initial version, most of the tubes of the modified heat exchanger operate in a subsonic mode, while also providing the removal \( Q_{\text{cooler}} \approx 0.3 \times 10^6 \text{ W} \) of thermal energy. A comparative analysis of the operation of all considered air cooler designs is given in Table 3.
Table 3. Comparison of the parameters of the considered options for heat exchangers

| Number | Number | $F$, m$^2$ | $Q_{\text{cooler}}$, W | $T_0$, K | $q_{\text{hcp}} / q_{\text{max}}$, Wm$^{-2}$ | $G$, kgs$^{-1}$ |
|--------|--------|-----------|---------------------|--------|---------------------------------|--------------|
| 1.1    | 1.1    | 14.34     | $0.3 \cdot 10^6$    | 600    | $21 \cdot 10^3 / 1042 \cdot 10^3$ | 2            |
| 1.2    | 1.2    | 14.32     | $0.3 \cdot 10^6$    | 600    | $30 \cdot 10^3 / 683 \cdot 10^3$ | 2            |
| 2.1    | 2.1    | 2.66      | $0.18 \cdot 10^6$   | 1600   | $84 \cdot 10^3 / 2332 \cdot 10^3$ | 7.3 / 26.3   |
| 2.2    | 2.2    | 3.33      | $0.21 \cdot 10^6$   | 1600   | $75 \cdot 10^3 / 3053 \cdot 10^3$ | 9.4 / 54     |
| 2.3    | 2.3    | 4.88      | $0.25 \cdot 10^6$   | 900    | $62 \cdot 10^3 / 2405 \cdot 10^3$ | 10.4 / 55    |

Thus, from the point of view of the level of thermal loading in the tubes, the temperature at the outlet, the cost savings of cooling water and the margin of the heat absorption, the best air cooler is Option-1.2.

5. Conclusions

Numerical studies of the flow and heat fields in the duct of the hypersonic wind tunnel made it possible to obtain the distribution of Mach number, pressure, and temperature in the nozzle, test section, diffuser, and air cooler under accelerated mode. It is shown that $\approx 25\%$ of the total thermal power of the installation is discharged through the cooled walls of the duct. Providing an acceptable temperature at the inlet to the ejector $T \approx 300 \, ^\circ C$, the air cooler would divert the heat output $Q_{\text{cooler}} \approx 0.3 \cdot 10^6$ W.

Based on these results, several air cooler options were proposed and investigated, for which the flow and heat transfer parameters were determined. It is shown that the most acceptable design that excludes damage to heat exchange surfaces under the action of the central high-temperature jet is a design made in the form of a compartment with longitudinal tubes inside of which the air moves, and in the intertube gaps the movement of the cooling water is organized.

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