Optimization of micro gas-turbine-recuperator heat transfer surface

Yu A Gavrilova¹⁴, V N Beschatnykh³, Yu A Borisov¹⁴, D A Achkasov³⁴ and A S Kosoy¹³

¹Joint Institute for High Temperatures, Russian Academy of Sciences, Moscow
²Russian University of Peoples Friendship, Moscow
³AO “Scientific and Production Enterprise “LEMZ”, Moscow
⁴Bauman Moscow State Technical University, Moscow

e-mail: borisovyu@gmail.com

Abstract. Making highly efficient recuperator, that will have minimal dimensions, high recovery rate, and low-pressure loss is one of the key tasks in a micro gas turbine design process. This paper presents recuperator with Frenkel-type heat-exchange matrix. Design and dimension choice of envelopes, which comprise recuperator heat-exchange surface is validated through gas-dynamic calculations, which are described in this paper.

Introduction

One of the main prerequisites in successfully developing highly efficient micro gas turbine installation is the presence of an efficient recuperator. Such a recuperator must meet certain requirements, which are as follows: a high degree of recuperation (90% and more), minimal hydraulic resistance (total pressure losses in the recuperator must not be higher than 5-6% of the compressor head), reliable design, and a possibility to withstand for a great amount of thermal cycles due to numerous start-ups, shutdowns and changes in the operation regimes. Minimal mass and size indexes must be also provided.

While designing any heat exchanger, the maximal attention is paid for providing the efficiencies of heat transfer surfaces. Frenkel-type surfaces have been taken as the basic ones in the designing process because they are thoroughly studied [1] and their advantages are confirmed both experimentally [2] and by calculations [3].

The developed recuperator is designated for utilizing heat of waste gas-turbine gases for regenerative heating of air downstream of the compressor in a micro gas-turbine electricity-generation module of 30 kW. This module comprises of electric generator, compressor, gas turbine, and combustion chamber, which are arranged within a single cylindrical casing. To minimize unit size, this module is placed inside of circular recuperator.

Inside the recuperator casing heat-transfer matrix is formed. It consists of many envelopes (segments) formed by two metal plates connected with each other and having mutually coplanar corrugations. To provide compactness of the installation, the metal plates are fabricated as curvilinear parts having involute-like generatrix. Envelope plate surfaces have a stamped relief formed by corrugations and internal partitions, which are connected in such a manner that provide multi-pass...
cross flow of a cold coolant with a general counterflow of hot and cold fluids. A heat-exchanger size directly depends on a channel equivalent diameter. Therefore, its value was chosen as minimal one with an account for technological potentialities of fabrication process. The choice of the multi-pass scheme allows also considerably reducing heat transfer area. However, with an increase in number of passes this effect gradually decreases. As a result, 6 passes were taken. The cold-coolant channel matrix formed in envelopes is divided into 6 sections. This makes it possible to reduce a diameter and increase a length of the recuperator. The sections are interconnected by channels.

The envelopes have rims, which protrude out of their peripheral edges. These rims constitute inlet and outlet collector windows on the internal cylindrical surface of the matrix. Such a design reduces the impact of thermal stresses on recuperator parts, and the accepted arrangement of the distributing collectors simplifies coolants input and removal. This allows excluding excessive hydraulic losses in inlet pipes. Figure 1 shows the envelope design.

![Figure 1. Envelope with channels between the sections: a) Top view, b) Side view](image)

If we take an assumption that a coolants distribution is uniform, thermohydraulic calculations can be conducted with sufficient accuracy employing commonly used criterion correlations and the known experimental data. These calculations have been used for determining dimensions and configuration of the corrugated plates as the first approximation. However, the requirement of minimizing hydraulic losses leads to such a situation where it becomes difficult, in terms of technical sense, to uniformly distribute the coolant between the channels. To solve this problem, the heating surfaces were optimized for providing minimal nonuniformity in the coolant distribution with the allowable values of the total pressure losses being retained.

**Calculation of flow inside the envelope**

The necessity of reducing hydraulic losses is especially urgent for the cold side of the recuperator. Since the cold coolant flows inside the envelopes, a mathematical model of this region has been developed and corresponding calculations of the gas flow were conducted. This was done taking distributed parameters assumption and using numerical calculation methods. The model is based on Navier-Stokes and continuity equations, and SST turbulence model (Menter’s model [4]) is applied for closing the equation system.

Because it is impossible to analytically solve the problem for the entire recuperator, and the corresponding numerical solution requires great computing resources, the calculations were conducted for a single envelope. In so doing, the following assumptions were accepted to simplify the model: the gas flow at the inlet to the calculation region of the envelope is uniform, the working gas is considered as a perfect one and it does not contain admixtures. The first assumption is correct because an annular
chamber of a large volume is located between the compressor diffuser and the collector windows and this makes it possible to maximally level-off the flow. The second assumption is valid because the changes in pressure and flow velocity in the recuperator flow path are small and, hence, a change in the gas density at a presence of pressure drop is also small as compared to the mode with zero pressure drop.

The total pressure and gas temperature at the inlet to the calculation region and the flow rate at the region outlet were used as boundary conditions. These parameters were determined in the course of designing the compressor of the electrogenerating micro gas turbine module.

Method of control volumes was applied for discretizing the system of differential equations.

In consequence of the calculations of the gas flow inside the envelope, a nonuniformity of the velocity distribution over the connecting channels was recognized. It was determined that the flow rate through the central channels is more than 2.5 times larger than that through the peripheral ones. Moreover, in the corner areas of the sections and in some diagonal channels adjacent to them, the gas flow is almost completely absent. Figure 2 represents streamlines in the first two envelope sections. Maximal velocity in central channels is more than 25 m/s and this fact leads to an increase in the pressure losses in the recuperator envelope. When this happens, the gas velocities in the crossing channels, which outgo from the central connecting channels, also exceed 20 m/s. Such a nonuniformity of the gas flow considerably reduces recuperator efficiency.

On the basis of the obtained data analysis, it was proposed to change envelope configuration. Seven connecting channels between the sections were substituted by a single combined V-shaped collector, with the collector cross section area being increased in transversal direction to keep constant gas velocity as new channels were added.

![Figure 2. Gas flow streamlines in the sections of recuperator with connecting channels.](image)

Figure 3 shows the envelope design with V-shaped collectors. This alternative also has 6 sections.
The calculation results showed that the gas flow velocity practically became constant in all envelope channels and equal to 5-7 m/s. This considerably reduces pressure losses. The maximal velocity value reaches 15 m/s and it takes place only in small areas near the V-shaped collectors. Figure 4 shows streamlines in two leading envelope sections. We should point out that collector surfaces are not involved in heat transfer process between the coolants. Therefore, the total dimensions of the new envelope are somewhat higher than those in the previous one. However, the uniform distribution of the coolant flow between the channels significantly increases the envelope efficiency.

**Conclusion**

A recuperative heat-exchange unit with heat-exchange matrix comprised of Frenkel-type involute surfaces was described in this work. For dimensions minimization, a multipass crossing heat-exchange fluid flow chart was chosen. A number of crossing channel nets variants were analyzed. It was discovered, that envelope with combined V-shaped collectors, a cross-section of which is increased as new channels are connected to it, has least pressure loss and most uniform channel gas flow.
References
[1] Tikhonov A M 1977 *Heat regeneration in air-craft gas-turbine engines* (Moscow: Mashinostroenie) p 108
[2] Savostin A F and Tikhonov A M 1970 *Teploenergetika* 9 75-78
[3] Stasick J, Collins M V, Ciofalo M and Chew P E 1996 *Int. J. of Heat and Mass. Transfer*. 39(1) 149-164
[4] Menter F R 1993 *Tech. Rep AIAA93-2906* p 33