Development and experimental study of the high efficient flow turbulators for heat transfer enhancement in heat exchangers

V O Kindra¹, A N Rogalev², S K Osipov¹ and I V Shevchenko³

¹ Innovation Department, National Research University “Moscow Power Engineering Institute”, Krasnokazarmennaya 14, Moscow, 111250, Russia
² Department of Innovative Technologies of High-Tech Industries, National Research University “Moscow Power Engineering Institute”, Krasnokazarmennaya 14, Moscow, 111250, Russia

E-mail: kindra.vladimir@yandex.ru

Abstract. The paper discloses investigation of flow turbulator thermal and hydraulic performance. The turbulator intensifies heat transfer in heat exchangers. Two modifications of pin turbulators allow decay of the pin “shadow” zone with low heat exchange coefficient downstream the pin. The intensifier main structural parameters influence thermal and hydraulic channel performance. The results show that the developed intensifiers increase the Nusselt number for 11–36% in the staggered pin-fin arrays. The obtained criteria equations allow the analysis of cooling air heat transfer with ±8% error.

1. Introduction

Traditional methods for heat transfer improvement are the applications of various vortex generators, ribs, dimples or pin fins that increase the heat transfer coefficient by 1.5–2 times against smooth walls through the boundary layer decay and flow mixing [1–5]. The main directions of the convective heat exchange improvement are the development and optimization of heat transfer intensifiers. Scientific reports present numerous works devoted to the investigation of combined pin fin and dimple flow turbulators intended for higher thermal and hydraulic performance of cooling channels. Analysis and test results [6–7] of these structures show possibilities of 8% Nusselt number increase and 18% hydraulic resistance reduction in Reynolds number range of 8200–50500 against the staggered pin fin arrays.

The analysis of heat transfer intensity in channels with pin fins determines the prospects of pin fin-dimple flow turbulators [8–9]. It shows that a single pin fin-dimple intensifier has 7–13% higher Nusselt number than a single pin fin intensifier in Reynolds number range of 8000–70000. This investigation is devoted to the thermal and hydraulic performance of cooling channels with staggered pin fin-dimple intensifiers and array of pin fins placed in grooves normal to flow direction.

2. Computer analysis of thermal and hydraulic performance in channels with flow turbulators

2.1. Analysis object

The analysis objects are the rectangular section blade cooling channels. Their dimensions are length $L=55$ mm, width $B=15$ mm, channel wall thickness $b=2.5$ mm, channel height $h=2$ mm, and channel
hydraulic diameter $d_{\text{hydr}}=3.5 \text{ mm}$ (figure 1). The channel with a staggered pin fins array is shown in figure 1a and the channel with staggered pin fin-dimple intensifiers array is shown in figure 1b. The channel specific section is assumed to be its exit. Calculations were carried out for the following dimple dimension range: depth $h_{\text{dimp}}$ from 0.5 to 1 mm, diameter $D_{\text{dimp}}$ from 3 to 4 mm. The minimal dimple depth of 0.5 mm and diameter of 3 mm are taken in terms of production of stable intensive swirls around the pin fin that intensify heat exchange in this zone. The 1 mm dimple maximal depth is limited by the minimally acceptable blade wall thickness in its trailing edge zone. The maximal diameter $D_{\text{dimp}}=4 \text{ mm}$ is limited by the pin fins location pitch in the channel. The pin fins diameter and height are kept constant $D_{\text{pin}}=H_{\text{pin}}=2 \text{ mm}$, transverse spacing-to-diameter ratio $S_2/D_{\text{pin}}=2.5$, streamwise spacing-to-diameter ratio $S_1/D_{\text{pin}}=2.5$. This produces the dimple to pin fin diameters ratio (dimple relative diameter) $d_{\text{dimp}, \text{rel}}=D_{\text{dimp}}/D_{\text{pin}}=1.5–2.0$.

2.2. Method of thermal and hydraulic processes computer simulation

The cooling channel computer simulation consists of the following four stages:

- 3D model construction.
- FEM mesh buildup.
- Boundary conditions and solver tuning.
- Analysis and parametric studies.

The channel 3D models are constructed by SOLIDWORKS code. The FEM mesh was built in ANSYS ICEM consisted of two main areas: the main flow of tetrahedral elements and wall boundary mesh with prismatic layers. The heat transfer analysis involved the conjugate approach, the mesh was built for both the flow and metal. The metal mesh is tetrahedral non-structured with maximal element dimension of 0.5 mm. The total metal analysis mesh was 3.6–3.8 million. The mesh parameters including the flow element size and prismatic element parameters may be iteratively corrected. The final mesh parameters for all channels are summarized in table 1.

Table 1. Flow model mesh parameters.

| Parameter                          | Value          |
|------------------------------------|----------------|
| Global element seed size, mm       | 0.75           |
| Number of prism layers, pcs        | 13             |
| First prismatic layer height, mm   | 0.0015         |
| Height ratio of prismatic layers   | 1.410          |
| Total height of prismatic layers, mm | 0.6         |
| Growth law                         | wb-exponential |
The goal of this computer simulation of the air-cooled channel is to approximate the conditions of the molten metal bath test [10]. The channel inlet conditions were the cooling air mass flow $m_{\text{air}}$ that determined the flow mode and total air temperature $T_0=293$ K. The channel exit condition was constant static pressure $P_{\text{out}}=3$ bar. The channel outer wall temperature is assumed $T_{\text{wall}}=692$ K, which is equal to the zinc crystallization temperature on the model outer surface [10]. The SST turbulence model encloses the Reynolds averaged Navier-Stokes equations. Details of the simulation results analysis are described in [8].

2.3. Results of cooling channel computer simulation

Improvement of the channel heat exchange intensity is evaluated by the $\frac{Nu_{\text{pfd}}}{Nu_{\text{pf}}}$ ratio (Nu$_{\text{pfd}}$ and Nu$_{\text{pf}}$ represent Nusselt number along the pin fin-dimple array and pin fin array channels, respectively). The Nusselt and Reynolds numbers were calculated according to methods described in [11]. The analysis shows that the maximal intensity values $\frac{Nu_{\text{pfd}}}{Nu_{\text{pf}}}=1.07$–$1.13$ are obtained at the dimple depth of 1 mm and dimple relative diameter of 2 (figure 2). Thus, these dimensions are optimal for the heat exchange increase.

One more result is the heat flux distribution along the channel outer surface. For example, figure 3 shows the heat flux density distributions in channels with pin fin and pin fin-dimple intensifiers at Reynolds number of 20000. In the pin fin intensifier case, the heat flux maximum is located in the frontal pin fin zone and the “shadow” zone has lower heat transfer (stagnation zone). Application of pin fin-dimple intensifiers eliminates the stagnation zones and increases heat transfer over the whole intensifier surface.

![Figure 2](image_url)

**Figure 2.** Distribution of $\frac{Nu_{\text{pfd}}}{Nu_{\text{pf}}}$ ratio for constant area channel with different pin fin-dimple intensifiers.

![Figure 3](image_url)

**Figure 3.** Distributions of heat flux in channels with different intensifier types, $Re=20000$. 
3. Experimental investigation of thermal and hydraulic performance in cooling channel models.

3.1. Test model and technology
Two test models manufactured by selective laser melting were the pin fin array B1 and pin fin-dimple array B2. The model dimensions obtained through the computer simulation were \( d_{\text{dimp}, \text{rel}} = 2 \) and \( h_{\text{dimp}} = 1 \) mm. The intensifier arrangement for the B2 model is as in figure 1.a and for the B3 model as in figure 1.b.

The pin fin-dimple intensifier may cause manufacturing problems, specifically concerned with investment casting. In terms of simpler manufacturing, the B3 model has pin fin arrays placed in transversal grooves (figure 4).

![Figure 4. Test model with pin fin array placed in transversal grooves.](image)

It is worth mentioning that the pin fin-dimple and pin fin in grooves intensifiers have equal dimensions: the dimple depth is equal to the groove depth and the dimple diameter is equal to the groove width. The optimal intensifier dimensions of the pin fin-dimple channel were retained in the “pin fin in groove” model, the groove depth and width of 1 mm and 4 mm, respectively. The B1, B2 and B3 models have the same pin fin diameter of 2 mm. The model material was chromium-nickel alloy with heat transfer coefficient of 16 W/mK. The models were tested by the calorimetry method in the liquid metal bath [10].

The method is used for measurement of local thermal performances in cooling channel models. It is based on the effect of phase transition in chemically pure metals. The testing technology is the following. The test model is fitted with cooling air supply and discharge tube lines. The model is submerged into the melted zinc heated above its melting temperature. Then, the model and zink are cooled together down to the zinc melting temperature. The model internal channel is blown with the cooling air flow for given time \( \tau \) and then taken out from the melted zinc. While the model is cooled with air, some heat is transferred to the air and a shell of crystalized metal grows on the model surface.

An important feature of this method is that all tests are carried out at the constant heat load determined by constant outer surface temperature equal to the zinc crystallization temperature \( T_{\text{cr}} = 692.4 \) K [10].

3.2. Test results analysis
The blow test results are shown in figure 5. Model B2 has the larger flow path area than B1 but its flow capacity is smaller. The models B1, B2 and B3 flow path areas \( F \) in the pin fin row section are 36 mm\(^2\), 41.14 mm\(^2\) and 54 mm\(^2\), respectively. The B3 mass flow is 50% larger than the B1 one at \( P/P_0 = 1.1 \) and 33% larger at \( P/P_0 = 1.6 \), temperature factor \( T_{\text{wall}}/T_{\text{mean}} = 2.3 - 2.5 \). Here \( P \) and \( P_0 \) are the
model inlet and exit pressures; $T_{\text{wall}}$ and $T_{\text{mean}}$ are channel wall and channel mean air temperature. The experimental investigations were conducted at the fixed exit pressure $P_0$ of 3 bar. The inlet air temperature is equal to 293 K.

The pin diameter $d_{\text{pin}}=2$ mm is assumed as the specific dimension in Nusselt number calculation, in Reynolds number calculation, $F=h\cdot(S-d\cdot n)$ is assumed the minimal channel area, where $h$ is channel slot height, $S$ is channel width, $n$ is a number of pin fins in the channel cross-section.

The model heat transfer performance at Reynolds number of 4000–14000 range show 11% Nu increase for the pin fin-dimple intensifiers and 36% increase for the pin-in-grooves ones. The number equations for mean Nusselt values along the intensifier zone (figure 6) are calculated by the RMS method. In model B1 with pin fin intensifiers the equation is $\text{Nu}_{\text{mean}}=0.245\cdot\text{Re}^{0.6}$, in model B2 with dimple-pin intensifiers, it is $\text{Nu}_{\text{mean}}=0.271\cdot\text{Re}^{0.6}$ and in model B3 with pin-in-grooves intensifiers, it is $\text{Nu}_{\text{mean}}=0.332\cdot\text{Re}^{0.6}$. The pin fin in grooves version shows the maximal heat exchange intensity. The number equations allow calculation error of ±8% for heat transfer to cooling air coefficients.
4. Conclusion
1) The analysis shows optimal in terms of heat transfer dimensions of pin fin-dimple intensifier: dimple relative diameter $D_{\text{dimp.rel}}=2$, and dimple depth $h_{\text{dimp}}=1$ mm. Computer simulation results for the Reynolds number range $Re=20000–85000$ represent the 7–13% increase in Nusselt number against the common pin fin intensifier.

2) The flow capacity test shows the smallest flow capacity in the pin fin array intensifier model and the largest capacity in the “pin fin in grooves” model.

3) The pin fin-dimple model thermal test at Reynolds number range of 4000–14000 shows the 11% higher mean Nusselt number over the channel path than the pin fin intensifiers value. This increase for the “pin fin in groove” version is 36%.

Acknowledgments
The results were obtained during implementation of the project of the state task of the Ministry of Education and Science of the Russian Federation in the field of scientific activity No. 13.7616.2017/8.9.

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