



## Research paper

# Experimental study of the influence of cold heat exchanger geometry on the performance of a co-axial pulse tube cooler



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## ARTICLE INFO

## Article history:

Received 25 January 2016

Received in revised form 7 June 2016

Accepted 7 July 2016

Available online 7 July 2016

## Keywords:

Cold heat exchanger

Co-axial

Pulse tube cooler

Slot number

Slot width

## ABSTRACT

Improving the performance of the pulse tube cooler is one of the important objectives of the current studies. Besides the phase shifters and regenerators, heat exchangers also play an important role in determining the system efficiency and cooling capacity. A series of experiments on a 10 W @ 77 K class co-axial type pulse tube cooler with different cold heat exchanger geometries are presented in this paper. The cold heat exchangers are made from a copper block with radial slots, cut through using electrical discharge machining. Different slot widths varying from 0.12 mm to 0.4 mm and different slot numbers varying from around 20–60 are investigated, while the length of cold heat exchangers are kept the same. The cold heat exchanger geometry is classified into three groups, namely, constant heat transfer area, constant porosity and constant slot width. The study reveals that a large channel width of 0.4 mm (about ten times the thermal penetration depth of helium gas at 77 K, 100 Hz and 3.5 MPa) shows poor performance, the other results show complicated interaction effects between slot width and slot number. These systematic comparison experiments provide a useful reference for selecting a cold heat exchanger geometry in a practical cooler.

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## 1. Introduction

In comparison with the traditional regenerative coolers such as G-M and Stirling coolers, the pulse tube cooler with no moving parts in low temperature region has the advantages of simple structure, high reliability, low mechanical vibration and low cost [1]. In retrospect, the quick development of Stirling type pulse tube coolers in the past two decades mainly benefited from the improvements of phase shifters [2–4] and regenerators [5–7]. On the other hand, the heat exchangers are also important. Insufficient heat transfer inside both ambient heat exchangers and cold end heat exchangers will influence the system performance seriously.

Some studies on heat transfer of oscillating flow in heat exchangers have been reported. In 1988, theoretical analysis from Swift [8] indicated that in order to obtain better heat transfer performance, the length of the heat exchanger should be comparable to the peak to peak gas displacement inside the heat exchanger. In 1993, Hofler [9] carried out an effectiveness analysis, and concluded that heat exchanger whose fin length is shorter than the

peak gas particle displacement can be effective if the fin spacing is sufficiently small. In 2004 Wakeland [10] made experimental study of heat transfer between two identical parallel-plate heat exchangers in oscillating flow. The experiments were conducted in air at ambient atmospheric pressure and the distance between two heat exchangers was 6.35 mm. The results revealed that the effectiveness of the heat transfer can be increased to nearly 1 by decreasing the plate spacing of the heat exchanger. In 2011, Piccolo [11] developed a two-dimensional numerical model for studying the heat transfer characteristics of parallel plate thermoacoustic heat exchangers. The numerical results illustrated that the optimum fin interspacing should fall in the range 2–4 times of the thermal penetration depth. The author also pointed out that the selection of the optimum fin interspacing is case dependent and the predictions fit the experimental data [12–14] within 36% and 49% respectively at moderate and high acoustic Reynolds numbers and a higher deviation found with low Re. In 2014, Tang et al. [15] set up an experimental apparatus to investigate the heat transfer performance of a water-cooled finned heat exchanger. A new correlation of the Nusselt number to maximum Reynolds number and Valensi number was proposed, with which the results have a maximum deviation of 6.3% compared with the experimental values. The test results indicated that the increase in Reynolds number

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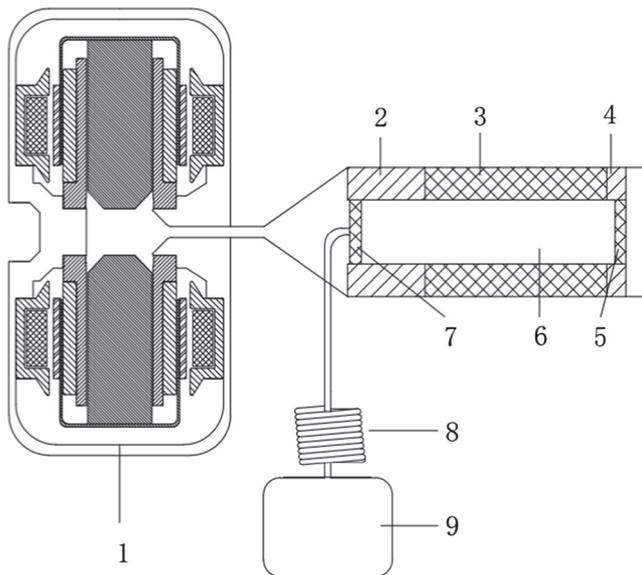
and Valensi number can lead to increases in the Nusselt number, which means heat transfer is enhanced.

Although many researchers have done studies on the heat transfer characteristic of heat exchangers in oscillating flow, the theoretical analyses are mainly performed with some ideal assumptions and most experimental setups also deviate from practical systems. Due to coupling with the stack/regenerator, abrupt cross-sectional area changes and entrance effects of heat exchangers in practice, heat transfer inside the heat exchangers are actually very complicated and experiments in practical systems are still needed. Ki et al. made some studies about the slit-type heat exchanger in practical cryocoolers. In 2010, they designed tapered slit-type heat exchangers and did some experiments. Two air-cooled aftercoolers with different slit configurations had been compared with regard to its cooling capacity [16]. The results indicated that the number and length of slits and the taper angle in axial direction all affected the performance. However there were only two group experiments, and how these factors affected were not shown. In 2011, they then did experiments with slit-type heat exchangers and make comparison with three-dimensional CFD simulation, but most of their studies focused on aftercoolers. [17]. In 2012, Yu et al. [18] used the parallel-plate structure cold end heat exchanger on a 300 Hz pulse tube cooler, and showed that the cooling power at 80 K increased to more than 1 W, compared with about 0.2 W obtained with a simple copper housing.

In this paper, most recent experimental research on the influence of geometrical dimensions of cold end heat exchangers in a 10 W @77 K class co-axial type pulse tube cooler is introduced. The following firstly briefs on the experimental setup and operation parameters. Secondly, the experimental results are presented. Finally, some discussions and conclusions are made.

**2. Experimental setup and operation parameters**

As illustrated in Fig. 1, the system includes a moving-magnet type linear compressor and a co-axial pulse tube cooler. The compressor uses dual-piston configuration to cancel vibration. The



1. compressor 2. main ambient heat exchanger 3. regenerator 4. cold end heat exchanger 5. flow straightener 6. pulse tube 7. secondary ambient heat exchanger and flow straightener 8. inertance tube 9. reservoir

Fig. 1. Schematic of a 100 Hz co-axial pulse tube cooler.

**Table 1**  
Main parameters of the pulse tube cooler.

Components	Diameter (mm)	Length (mm)	Structure
Main ambient heat exchanger	i.d.20	30	Radial slots
Regenerator	i.d.20	34.5	Filled with 600 stainless steel mesh
Cold end heat exchanger		9.5	Radial slots
Flow straightener	10	2	Filled with copper mesh
Pulse tube	10	54	
Secondary ambient heat exchanger	10	5	Filled with copper mesh
Inertance tube	3	1500	Copper tube
Reservoir	300 cm <sup>3</sup>		

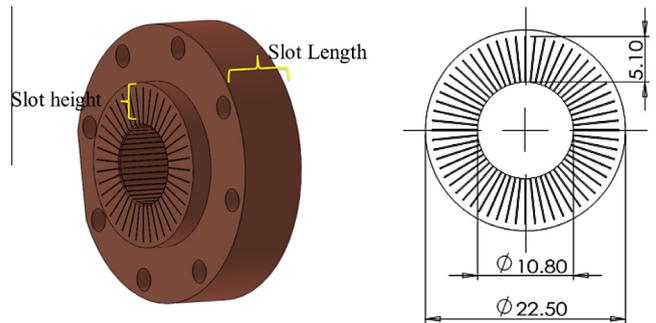


Fig. 2. The structure and dimensions of the cold end heat exchanger.

pulse tube cooler consists of a main ambient heat exchanger, regenerator, cold end heat exchanger, pulse tube and inertance tube plus reservoir. Flow straighteners made of copper wire mesh are installed at the cold end of the pulse tube with a thickness of 2 mm. The dimensions of the main components are listed in Table 1.

Three Dynamic pressure transducers (113B21, PCB piezotronics Inc.) are placed at the inlet of the pulse tube cooler, the backside of the compressor piston and the inlet of the inertance tube, respectively, to measure local pressure wave amplitude. Platinum resistance thermometer (PT 100) with an accuracy of ±0.1 K is attached to the cold end heat exchanger to measure the temperature. Cooling power is measured through a heating wire mounted on cold end heat exchanger which is powered by a DC voltage source.

The influence of cold end heat exchanger on the system is studied here. The components of the pulse tube cooler are kept the same except the cold end heat exchanger, which is made from a copper block with radial slots cut through electrical discharge machining as shown in Fig. 2.

In all the experiments here, the working gas is helium. The mean pressure is kept at 3.5 MPa. The ambient heat exchanger is cooled by water maintained at a temperature of 293 K. The system operates at 100 Hz. The cooling power is measured at the temperature of 77 K unless otherwise stated.

**3. Experimental results**

The geometry dimensions of the cold end heat exchanger such as slot width, length and slot number are critical parameters, because they determine the available heat exchange area and gas velocity, etc. In this paper, we focus on studying the influence of slot width and slot number, and the length of all cold end heat exchangers studied here are kept constant at 9.5 mm.

**Table 2**  
Dimensions of the cold end heat exchangers.

	Slot width (mm)	Slot number	Heat exchanger area(mm <sup>2</sup> )	Porosity
First Group (same heat exchange area)	0.12	20	1961	0.04
	0.2	20	1976	0.067
	0.3	20	1995	0.1
	0.4	20	2014	0.133
Second Group (same porosity)	0.12	60	5882	0.12
	0.2	36	3557	0.12
	0.3	24	2394	0.12
	0.4	18	1813	0.12
Third Group (same fin-spacing)	0.12	20	1960	0.04
	0.12	40	3921	0.08
	0.12	60	5882	0.12

Three groups of experiments have been done. Due to the restrictions of the process, the narrowest slot width is 0.12 mm. The first group has different slot width varying between 0.12 mm and 0.4 mm but with the same slot number of 20. The second group has different slots width but with different slots number to keep the porosity constant. The last group has the same slot width of 0.12 mm but with different slot number. The geometry dimensions of various cold end heat exchangers are shown in Table 2.

Although the geometrical changes may bring about an apparent change of local viscous loss, the overall pressure drop across the regenerator and heat exchangers does not change much. This is evidenced by nearly same pressure ratio at the inlet of the inertance tube given the same pressure ratio at the compression space, which means the volume flow rate across the cold end heat exchanger does not change much since there is no change on the pulse tube, inertance tube and reservoir. Meanwhile, the numeric calculations based on Sage software indicate that the mass flow at the exit of cold end heat exchanger is roughly the same (the volume change do have some effect on the mass flow inside, but with an estimated maximum variation below 5%). The peak to peak gas displacement inside the cold end heat exchanger in all the following experiments roughly falls within the range from 100% to 500% of the heat exchanger length, corresponding to an average velocity of 3.0–15.0 m/s inside the cold end heat exchanger.

3.1. Influence of the slot width with the same slot number of 20

In this section, four cold end heat exchangers with different slot width of 0.12 mm, 0.2 mm, 0.3 mm, 0.4 mm are tested. The slot number is 20 for all cases. Compared with the slot height of 5.1 mm, the heat transfer area change due to the change of slot width is negligible and the area is around 2000 mm<sup>2</sup>.

Fig. 3 shows that the no-load temperature decreases with the increasing pressure ratio. Among the four cases, slots width of 0.3 mm and 0.2 mm give the best results and both lead to almost the same no-load temperatures. The case of 0.4 mm is the worst with roughly 3 K higher than the case of 0.2 mm and 0.3 mm.

Fig. 4 shows that the cooling capacity increases with the increasing of pressure ratio. The case of 0.3 mm has the maximum cooling power given the same pressure ratio. The performance of 0.2 mm slot is slightly worse than that of 0.3 mm slot. Smaller (0.1 mm) and larger (0.4 mm) slots have negative effect on the cooling capacity.

For a better insight of the influence of slot width on system performance, Fig. 5 plots the cooling capacity with the pressure ratio of 1.24 as a function of  $r_h/\delta_k$ .  $r_h$  is the hydraulic radius which is half of the slot width here.  $\delta_k$  is the thermal penetration depth, 0.04 mm for helium at 77 K, 3.5 MPa and 100 Hz. It can be seen that the cooling performance reaches optimum with the  $r_h/\delta_k$

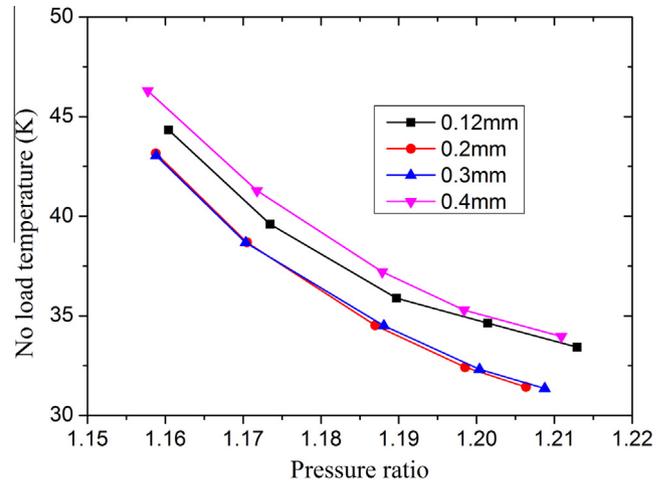


Fig. 3. No-load temperature as a function of pressure ratio with the same slot number of 20.

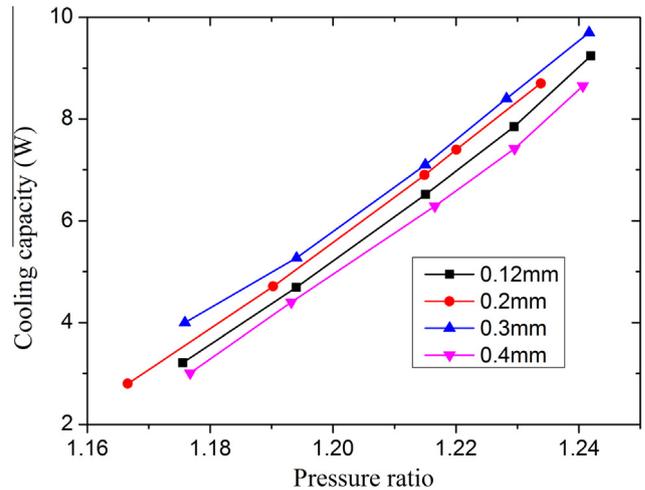


Fig. 4. Cooling capacity as a function of pressure ratio with the same slot number of 20.

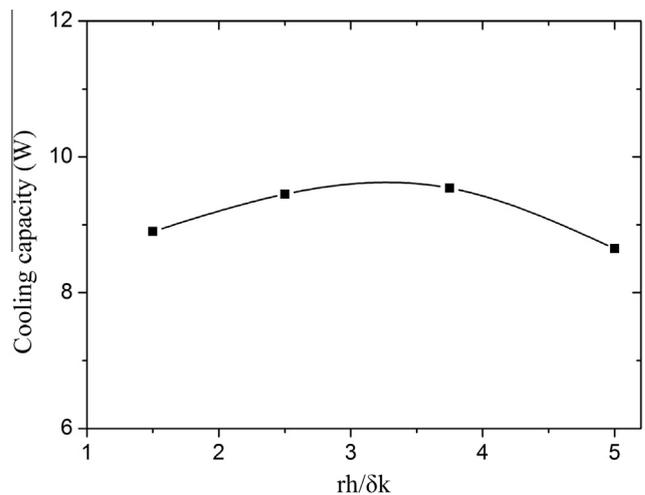


Fig. 5. Cooling capacity as a function of  $r_h/\delta_k$  with the same pressure ratio of 1.24 and slot number of 20.

between 2.5 and 3.75, corresponding to the slot width of 0.2 mm and 0.3 mm, respectively. While it is easy to understand that too big slot width means that the center part of the flow may take no part in the heat exchange process, a much smaller slot width may cause much local viscous dissipation which also leads to a lower efficiency.

3.2. Influence of different slot width and number combinations with the same porosity

In the section above, as the slot width changes, the porosity of the heat exchanger changes from 0.04 to 0.13. The porosity of the cold end heat exchanger determines the void volume in cold end and local velocities which is related to viscous dissipation and minor losses [10]. Thus, to study the influence of slot width with the same porosity, another group of experiments are carried out with four different slot widths but with the same porosity of 0.12. The slot widths are 0.12 mm, 0.2 mm, 0.3 mm and 0.4 mm, respectively. The corresponding slot number is 60, 36, 24 and 18, respectively.

The curves of cooling capacity are given in Fig. 6. The curves of 0.12 mm, 0.2 mm and 0.3 mm slot width almost overlap each other. Performance with 0.4 mm slot width is the worst with only 8.7 W cooling power at a pressure ratio of 1.24. As compared with the experimental group above, the increased slot number from 20 to 60 with 0.12 mm slot width leads to a better cooling performance. This trend will be further investigated in the following section.

3.3. Influence of slot number with the same slot width of 0.12 mm

This section mainly compares how the slot number affects system performance with the same slot width of 0.12 mm. Due to the limitation of the heat exchanger dimension, only 20, 40 and 60 slots are machined. Although the experimental results on 20 and 60 slots number have been introduced in the sections above, they are also presented here for a better comparison.

Fig. 7 shows how the slot number affects the dependence of no-load temperature on the pressure ratio. The cases with 40 slots and 60 slots reach 31.1 K lowest no-load temperature. And the case with 20 slots can only reach 34 K lowest no-load temperature.

Fig. 8 further shows that how the slot number influences the cooling capacity. The cooling power of 40 slots at 77 K reaches 9.5 W under the pressure ratio of 1.24, which is almost as same

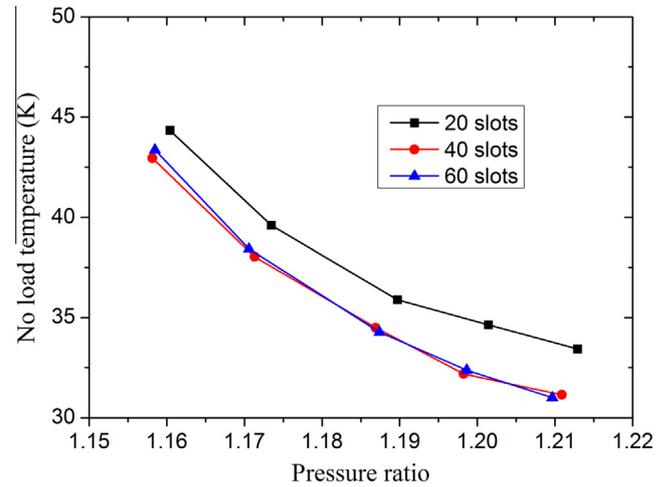


Fig. 7. No-load temperature as a function of pressure ratio with the 0.12 mm slot width.

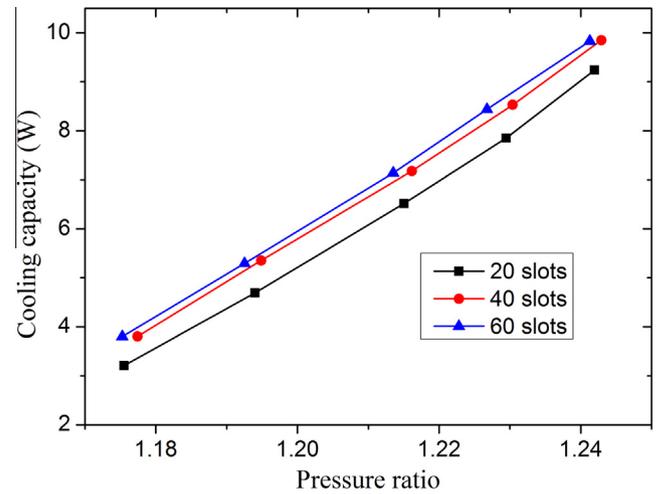


Fig. 8. Cooling capacity as a function of pressure ratio with the 0.12 mm slot width.

as the cooling capacity with 60 slots. The case of 20 slots is worse than the other two cases. Better performance with 40 and 60 slots may be attributed to the following two factors, improvement of the available heat exchange area and reduced flow resistance due to increases of the slot number.

4. Discussions and conclusions

Experiments here demonstrate how the slot number and width affect the cooling performance. The results above show that the maximum cooling capacity difference between different configurations is about ~13% at high pressure ratio and ~34% at low pressure ratio. As said in the introduction section, lots of factors may affect the performance of cold end heat exchanger in a practical cooler system. And it is difficult to change only one geometric parameter without affecting some other parameters. However, through three representative groups of experiments presented here, some tendencies can be found. For a more clear review, Fig. 9 generalizes the results on 0.12 mm and 0.2 mm slot width series.

For the laminar flow in the narrow channel, smaller channel width normally brings about a larger heat transfer coefficient. However, in the first group of experiments with 20 slot number

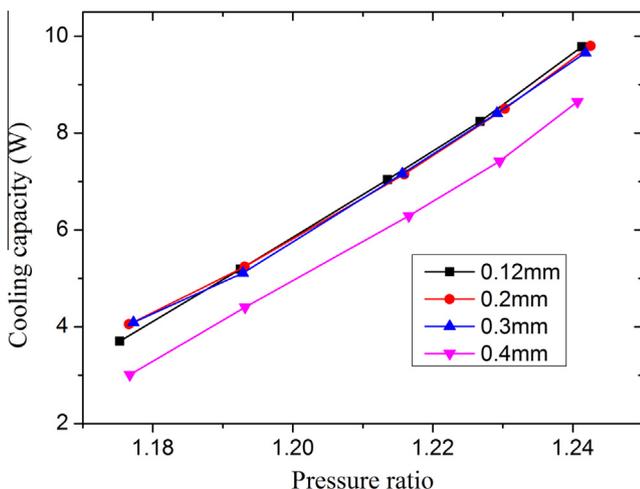


Fig. 6. Cooling capacity as a function of pressure ratio with the same porosity of 0.12.

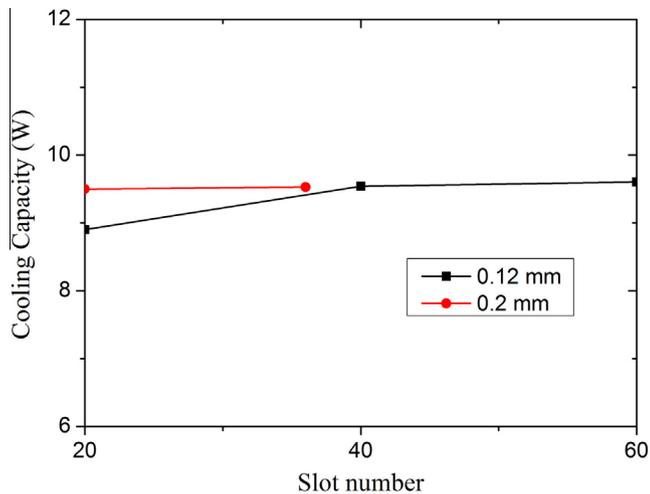


Fig. 9. Cooling capacity as a function of slot number with different slot width, the pressure ratio is kept at 1.24.

and roughly the same heat transfer area, 0.12 mm slot width is not the best. Only when the number increases to 40 and 60 for 0.12 mm wide slot does the performance become comparable with that of 0.2 mm and 0.3 mm slot width. As the local viscous dissipation is roughly related to the cubic of velocity, the slot number changing from 20 to 40 leads to a dissipation reduction by over 80%, which may explain the performance improvement. As the slot number further changes from 40 to 60, the reduction on dissipation is much smaller. Meanwhile, it also implies that heat transfer with 40 slots is good enough.

Fig. 9 also compares the performance of 0.2 mm slot width with slot number of 20 and 36. The cooling capacity at 77 K is almost the same, which implies that heat transfer is good enough. Either increased heat transfer area or decreased fin thickness has no effect on the cooling performance. Compared with slot width 0.12 mm and slot number 40, a much less machining effort is needed with slot width 0.2 mm and slot number 20 to generate similar cooling performance.

In the case study here, the results show that a ratio between hydraulic radius and local thermal penetration depth may better be kept at between 2.5 and 3.5 when given the same heat transfer area. A bigger value will bring adverse effects, while too small values also leads to worse performance due to local viscous dissipation. The latter effect can be compensated by decreasing the velocity through increasing the slot number and porosity, i.e. the flow area, as indicated above in the case of 0.12 mm slot width.

Due to complexity of oscillating flow heat transfer, more study needs to be done in the future. The results here show that, through carefully selecting the combination of slot number and slot width, an optimum cooling performance can be achieved without excessive machining efforts. Furthermore, CFD simulation using some correction factors based on systematical experiments will be done to reveal more details of the interacting mechanism inside.

### Acknowledgement

This work is financially supported by National Natural Science Foundation of China (Contract No. 51376187).

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