Hydrodynamics and heat transfer in cooled active laser mirrors

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Abstract. The paper describes some features of hydrodynamics and heat exchange in cooling systems of optical units of laser active mirrors. The experimental data on hydraulic resistance and heat transfer are given and summarized for the cooling systems which are the most suitable to such applications – a channel system with an intermittent wall and a wafer cooling system.

Introduction

Basic elements of active (deformable) mirrors are an optical unit and piezoelectric actuator. The processes running in them are connected with heating up due to the effect of radiation or electromechanical loading. A power active mirror operates under laser exposure with a high intensity up to \( (3-10) \times 10^7 \text{ W/m}^2 \) and more. Determining characteristics of the active mirror under its operation are the following: both local and total deflections of the optical unit specified by temperatures distribution in the unit, its average overheat relative to a heat carrier and mirror casing. These characteristics are connected with the cooling system efficiency.

The cooling system is effective if it provides minimal thermal distortions and minimal time to reach steady-state conditions at the limited differential pressure and elevated values of radiation exposure. For the purpose of providing amplitude and dynamic characteristics of the mirror, the thickness of its optical unit should not exceed \( (6-8) \times 10^{-3} \text{ m} \). Development of a cooling system for an active mirror is a nontrivial task.

Before we investigated cooled passive mirrors of power metal optics. The emphasis in this paper has been on heat-hydraulic tasks of active mirrors.

1. Hydrodynamics and heat transfer features of cooling systems of active mirrors

At a tendency to minimal thickness of an optical unit in the systems which can be realized in a technical way, a relation of the cooling system components height to their typical lateral dimension is not significant. Wall effects are important, and so it is a problem to use results on a flow around some components (for example, tube bundles).

Non-uniformity of the coolant distribution around the optical unit square and two-dimensionality of flow rate intensity fields occurred in the circular aperture, can take place in cooling systems. It is caused by both heat carrier (coolant) inlet and outlet systems features (e.g. in channel cooling systems with a peripheral inlet and outlet) and anisotropy of hydraulic resistance of the cooling system.

This paper presents the results of experimental investigations into optical unit fragments of the active mirrors, which geometrical dimensions allowed excluding initial areas influence \( L/d_e \geq 100, \)
where $L$ – channel length, $d_g$ – hydraulic diameter) and wall effects which usually occur due to limited lateral dimensions of the fragment.

Various cooling systems have been investigated, but the paper presents the results only for flow channel and wafer cooling systems. In this case the channel systems are made with cross-sections which allowed making the optical unit more isotropic and excluding longitudinal component of thermal expansion of the cooling system. In the general case the formed fins rows can have different longitudinal $S_1$ and transverse $S_2$ pitches. For mechanical isotropy of the optical mirror unit it is more preferable to have $S_1=S_2$ (wafer cooling systems). In the general case in the figure such fins have a form of the rhombs which height is not significant ($h_k/\delta_r\sim1-3$, where $h_k$ – channel height).

For improvement of the optical unit geometrical stability it is necessary to enhance thermal efficiency of the cooling system: a) to increase reduced heat transfer coefficient $\alpha_{pr}$ (heat emission from a heat-exchanging substrate wall), b) to reduce a thermal insulation coefficient $K_i$ (i.e. a relation of the excessive temperatures of the heat-exchanging base surface to the excessive temperature of the heat-exchanging substrate surface). Efforts on choosing a suitable cooling system should be directed to the system having high heat transfer coefficients in the mild region of Reynolds numbers $Re$ ($Re\leq2\cdot10^4$).

For development of an efficient heat-exchanger of an active mirror with a single-phase forced convection it is required: 1) to use the developed heat-exchanging surfaces at small values of a hydraulic diameter (i.e. to make the cooling system more compact); 2) to increase a heat emission coefficient efficiently by creation of hydrodynamic conditions at a moderate growth of hydraulic resistance.

The investigations [1] showed the limits of heat exchange intensification which were $\sim (2-3)\cdot10^5$ W⋅m$^{-2}$⋅K$^{-1}$ in respect to the laser mirrors for the reduced heat-transfer coefficient, $\alpha_{pr}$.

Numerous engineering and technological limitations can strongly reduce this limit, and the problem on searching for a suitable cooling system is becoming compromise.

For analysis of temperature fields in an optical unit of the active mirror with flow cooling systems we have developed and experimentally substantiated a model of temperature distribution calculation in multilayer systems at one-sided heating [2].

2. **Hydraulic resistance and heat transfer of some cooling systems of active mirror**

2.1. **Channel system with intermittent walls**

At increasing in the active mirror diameter and supporting the pressure drop available, an operating area of the cooling system by the Reynolds number can be in the $5\cdot10^2$-$5\cdot10^3$, where laminar and transient conditions of the coolant are implemented. Heat emission intensification under these conditions can be based on different methods of the flow structure implementation and destruction [3].

Especially for cooled optical units, previously we proposed the cooling system design that combined short channels with creation of pressure non-uniformity (figure 1) [4]. Structurally it was designed by making some additional channels arranged with a pitch $i$ (T) under angles $\pm \gamma$ to the basic channels in the channel cooling system.

The results on measurements of resistance, heat emission, and temperature fields for such cooling systems (Table 1) at the Reynolds numbers $Re=70\cdot3\cdot10^4$ and Prandtl numbers $Pr=5.5-8$ are given in details in [4]. The results on the resistance, heat emission are approximated by power functions $\xi=\zeta_i Re^{n_i}, \quad \alpha_{pr}=C_2 Re^{n_2}$ and $\alpha_0=C_3 Re^{n_3}$ (where $C_1, C_2, C_3$ are constant values) and given in figure 2. The fields of $Re$ numbers were revealed in which such intensification was advantageous from the point of view of $\alpha_{pr}$ value (figure 3).

The proposed method of the heat exchange intensification is advanced since at increase in the design complexity by 8-10 %, it allows attaining the intensification that is advantageous in the wide range of coolant flow rates.
Figure 1. Cooling systems: a) channel system; b) channel system with transverse grooves; c) channel system with grooves (“herringbone”); d) channel system with grooves (“zigzag”) (refer to designations in Table 1).

Table 1. Characteristics of studied system with intermittent wall.

| Mockup No. | Cooling system type                          | $\delta_a$ (mm) | $h_k$ (mm) | $d_y$ (mm) | $\delta_t$ (mm) | $\varepsilon^b$ | Note                  |
|------------|----------------------------------------------|-----------------|------------|------------|-----------------|-----------------|-----------------------|
| 1          | Channel system with transverse grooves       | 1.0             | 1.457      | 1.0        | 0.5             | $\delta_{v}=1.1$ mm; T=19 mm |                      |
|            | (figure 1b)                                  | 2.68            |            |            |                 |                 |                       |
| 2          | Channel system with inclined grooves         | 1.0             | 1.462      | 1.0        | 0.5             | $\delta_{v}=2.0$ mm; T=36.5 mm; $\gamma=38^\circ$ |                      |
|            | (figure 1c)                                  | 2.69            |            |            |                 |                 |                       |
| 3          | Channel system (figure 1a)                   | 1.0             | 1.456      | 1.0        | 0.5             | Reference specimen |                      |
| 4          | Channel system with inclined grooves         | 1.0             | 1.444      | 1.0        | 0.49            | $\delta_{v}=1.1$ mm; T=26 mm; $\gamma=66^\circ$ |                      |
|            | (“zigzag”, figure 1d)                       | 2.60            |            |            |                 |                 |                       |
| 5          | Channel system with inclined grooves         | 1.0             | 1.457      | 1.0        | 0.5             | $\delta_{v}=1.0$ mm; T=36 mm; $\gamma=40^\circ$ |                      |
|            | (“herringbone”, figure 1c)                  | 2.68            |            |            |                 |                 |                       |
| 6          | Channel system with inclined grooves         | 2.0             | 2.4        | 2.0        | 0.5             | $\delta_{v}=2.0$ mm; T=43 mm; $\gamma=55^\circ$ |                      |
|            | (“zigzag”, figure 1d)                       | 3.0             |            |            |                 |                 |                       |

$^a$ Cooling channel width.

$^b$ Porosity.

Figure 2. Dependence of heat-hydraulic characteristics on a Reynolds number (the numbers of curves 1-5 correspond the mockup numbers in Table 1): a) $\xi$, curve 10 - $\xi=66.6/\text{Re}$ – hydraulic resistance of a prismatic channel in the laminar flow region, curve 20 - $\xi=0.316/\text{Re}^{0.25}$ – hydraulic resistance of a prismatic channel in the turbulent flow region; b) $\alpha_{pr}$, $\alpha_0$, W·m$^{-2}$·K$^{-1}$. Curve 30 – surface heat emission in the prismatic channel at the turbulent flow, curve 40 – calculated $\alpha_{pr}$ of the prismatic channel, obtained from curve 30.

2.2. Wafer cooling systems

The wafer structures are obtained by milling or electro-erosion machining in material of the rows
mutually crossing under angle $\varphi$ of the channels having the same height $h_k$ (Refer to figure 4).

Resistances and heat exchange were previously investigated in such cooling systems at $h_k/\delta_k=1.3$ for 24 mockups [5-7, 8]. The parameter changes ranges were: Reynolds numbers $Re=100-2.5\times10^4$, Prandtl numbers $Pr=5.5-8.5$, porosity - $\varepsilon=0.53-0.8$, channel crossing angles $\beta=60^\circ, 90^\circ$, angles of incidence $\gamma=0-120^\circ$ and a coefficient of thermal conductivity of the fins material $\lambda=100-400$ W·m$^{-2}$·K$^{-1}$. The results have been summarized as power dependences.

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At the results summarizing, a hydraulic channel diameter $d_g$ was taken as a typical dimension. Filtration rate $W_\varepsilon$, average velocity $W_2=W_\varepsilon \varepsilon^{-1}$, maximal velocity $W_1=W_\varepsilon \varepsilon_m^{-1}$, where $\varepsilon=\varepsilon_m(2-\varepsilon_m)$ were applied as a typical velocity. Adequate Reynolds numbers $Re_i=W_i d_g/\nu_l$ hydraulic resistance coefficients $\xi_i=\Delta P/\rho W_2^2/(d_g/l)$ (where $\Delta P$ - pressure drop on length $l$; $\nu_l$, $\rho$ - a kinematic viscosity coefficient and density of coolant) were calculated based on these velocities.

A version with a corridor wafer structure was selected as a basic version for revealing a dependence of resistance $\xi_2(\gamma)$ and heat emission $Nu(\gamma)$ (where $Nu=\alpha d_g/\lambda_l$ - a Nusselt number, $\lambda_l$ – liquid thermal conductivity) on the incidence angle structure. The following dependences were obtained by summarizing the results:

$$\xi_2=0.75Re_2^{-0.036}, \quad K_1=\text{NuPr}^{-1/3}=0.75Re_1^{0.74} \quad (1)$$

for $\varphi=60^\circ, \gamma=0^\circ, 60^\circ$ and $3\times10^2<Re_2<8\times10^3$;

$$\xi_2=0.72Re_2^{-0.12}, \quad K_1=\text{NuPr}^{-1/3}=0.115Re_1^{0.73} \quad (2)$$

for $\varphi=90^\circ, \gamma=0^\circ$ and $8\times10^2<Re_2<1.5\times10^4$.

Figure 5a shows the diagrams of relative resistance $\bar{\xi}=\xi_2(\gamma)/\xi_2(\gamma=0^\circ)$ and relative heat emission $\bar{Nu}=Nu(\gamma)/Nu(\gamma=0^\circ)$ dependences on the incidence angle $\gamma$, which were obtained by normalization for the baseline characteristics (Formulas (1) and (2)) at the constant Reynolds number. A common feature for all structures is increase in resistance and heat emission at increasing in $\gamma$. When changing a corridor structure with a chessboard structure, hydraulic resistance growth is 14-16 times, and heat emission – 1.6-2.2 times.

Figure 3. Dependence of characteristics of channel systems with intermittent wall on number Re: a) relative number $\tau_{pr}=\alpha_{pr}/\alpha_{pr}^0$; b) energy efficiency $\eta=\alpha_{pr}/\xi_2$. Normalization has been carried out for the reference specimen data (mockup 3 in Table 1).

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Figure 4. Wafer structures view in the drawing: a) corridor, b) chess-board order.

The paper [8] presents the formulas for approximation of experimental data by polynomials of $n^{th}$ power ($n=4$) by hydraulic resistance and heat emission in dependence on an angle.

Relation $\eta = \overline{\text{Nu}}/\overline{\xi}$ (figure 5b) shows how many times the heat emission is more than resistance at increasing in $\gamma$. At the specified values of heat power influencing the laser mirror and heat exchange surface, the heat emission intensification is specified by power consumption for the coolant circulation. Having characteristics for the basic version as power functions $\text{Nu}=C_1\text{Re}^n$ and $\xi_2=C_2\text{Re}^m$ (where $C_1$, $C_2$ – constants) and dependence $\overline{\xi}(\overline{\gamma})$, $\overline{\text{Nu}}(\overline{\gamma})$, a relation of powers for the coolant circulation is defined by the complex:

$$N = \frac{\text{Nu}^{(m+3)/n}}{\overline{\xi}},$$

where $(m+3)/n$ is equal to 4 (for $\phi = 60^\circ$) and 3.93 (for $\phi=90^\circ$).

This complex is suitable for comparison of the heat power of heat-exchangers at the same coolant rates and hydraulic resistances, and in dependence on angle $\overline{\gamma}$ it is given in figure 5b. Its behavior differs from behavior $\eta = \overline{\text{Nu}}/\overline{\xi}$ in that for some cases a minimum point by $N$ is available with its following increase. Directly value $N$ does not differ much in comparison with $\eta$. For $\phi = 60^\circ$ at $\gamma > 96^\circ$ we have $N \geq 1$.

Figure 5. Dependences for wafer cooling systems: a) strength ratio $\overline{\xi}$ and relative heat emission $\overline{\text{Nu}}$ on the flow (incidence) angle $\gamma$: 1 - $\phi=90^\circ$, $\text{Re}_2=2.5\cdot10^3$; 2 - $\phi = 60^\circ$. b) Dependence of values $\eta$ and $N$ on (incidence) angle $\overline{\gamma}$: 1 - $\phi=60^\circ$, $0^\circ \leq \gamma \leq 30^\circ$, $\overline{\gamma}=\gamma/30$, $\text{Re}_2=2\cdot10^3$; 2 - $\phi=60^\circ$, $60^\circ \leq \gamma \leq 120^\circ$, $\overline{\gamma}=(\gamma-60)/60$, $\text{Re}_2=2\cdot10^3$; 3 - $\phi=90^\circ$, $0^\circ \leq \gamma \leq 45^\circ$, $\overline{\gamma}=\gamma/45$, $\text{Re}_2=2.5\cdot10^3$. Solid lines - $N$, dashed lines - $\eta$.

3. Comparative analysis of cooling systems

A choice of a cooling system for an optical mirror unit is a multifactorial task. In this paper the cooling systems are analyzed taking into account their thermohydraulic and thermal characteristics only.

A thermal displacement of the optical mirror surface is specified by three components: thermal expansion, thermal shear deformation, and bending thermal deformations of the optical unit which are proportional to the integrals of the same type: $\int_0^h \beta \vartheta(y)dy$, $\int_{-h/2}^{h/2} E\beta \vartheta(y)dy$, $\int_{-h/2}^{h/2} E\beta \vartheta(y)dy$ (where $\beta$ – linear expansion coefficient, $E$ – Young’s modulus, $y$ – coordinate by mirror thickness, $h$ - mirror thickness). A temperature profile $\vartheta(y)$ depends on a particular model of a cooling system, and its
maximum on the mirror surface can be expressed as \( \vartheta_p = q [h/(\alpha_{pr} + h/\lambda + S_h/(Gc_p))] \) (where \( q \) – heat flow, \( h \) – mirror faceplate thickness, \( S_h \) – irradiated surface, \( G \) – flow rate, \( c_p \) – liquid flow thermal capacity), and all three components contribute comparably. For increasing in the surface temperature \( G \) and \( \alpha_{pr} \) rate should be reduced, and \( h/\lambda \) component is regulated by material selection \( \lambda \) and substrate thickness \( h \). Usually an available pressure drop of liquid (\( \Delta P/l \)) is specified or set, and the first estimation of the cooling systems can be made based on dependences \( \alpha_{pr}(\Delta P/l) \). The summarized experimental data show that maximal values \( \alpha_{pr} \) are reached in the wafer systems, then in the channel systems with notches, and then in the channel ones. The obtained dependences for the temperature distribution [9] allow them to be integrated by the optical unit thickness. Taking into account that main contribution is given by the component determining a shear deformation, we can obtain the following:

- for channel cooling systems

\[
E \int_0^L \varphi(y) dy = \frac{E\beta q}{\sqrt{a\lambda P_f/S_c}} \left\{ \text{ch} \varphi \left[ h_c \text{ch}(mh + \varphi) + h_c \text{ch} \varphi \right] + \frac{2}{m} \text{sh} \left( \frac{mh}{2} \right) \text{ch} \left( \frac{mh + 2\varphi}{2} \right) + \frac{h_c^2 m}{2} \right\} \Rightarrow \min \quad (4)
\]

- for wafer cooling systems and channel cooling systems with notches

\[
E \int_0^L \varphi(y) dy = \frac{E\beta q}{\sqrt{a\lambda P_f/S_c}} \left\{ \text{ch} \varphi \left[ h_c \text{ch}(mh + \varphi) + h_c \text{ch} \varphi \right] + \frac{2}{m} \text{sh} \left( \frac{mh + 2\varphi}{2} \right) + \frac{h_c^2 m}{2} \right\} \Rightarrow \min , \quad (5)
\]

where \( P_f, S_c \) – fin perimeter and square; \( m = \sqrt{a\lambda P_f/(S_c \lambda)} \); \( \varphi = \varepsilon \alpha/(1 - \varepsilon) m \lambda \) \leq 1.

Comparative analysis of these expressions gives: 1) taking into account the materials properties by complex \( E\beta/\sqrt{\lambda} \), they are arranged as follows: copper, silicon carbide, molybdenum, and tungsten; 2) the bracketed expression (5) is less the bracketed expression (4), i.e. the wafer cooling systems and systems with notches are preferable; 3) surface heat emission in the wafer and notched channel systems is higher, and at the expense of this fact the expression (5) will be less than (4).

Thus from a heat-hydraulic point of view it is recommended for an optical unit of an active mirror to be made of copper with wafer or notched channel cooling systems which should be formed in the substrate.

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