Heat transfer in laser passive and deformable mirrors

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Abstract. In work, the possibilities of using uncooled and cooled optical elements (including laser passive and deformable mirrors) with an increase in the power of laser facilities are analyzed. To increase the permissible light loads acting on the optical elements, the use of highly efficient cooling systems with minichannels (coplanar and multi-tiered), providing a high compactness of the heat exchange surface and the intensification of heat transfer, is considered. The advantages and efficiency of the proposed cooling systems for reducing the displacement of the optical surface of the mirror due to bending are estimated.

1. Introduction

In connection with the increasing power of laser installations, the negative effect on the elements of the optical path (lenses, mirrors, etc.) of the absorbed heat, leading to a distortion of the optical surface of the element and, consequently, the wavefront of radiation, increases. Distinguish between uncooled and cooled optical elements (OE), passive and active (deformable) laser mirrors. The use of interference quarter-wave dielectric coatings has achieved tremendous success, providing practically maximum achievable specular reflection coefficients at the level of (99.99-99.995)% in uncooled OEs. The possibilities of using uncooled optics are not endless, since technological difficulties with further efforts to improve coatings increase manifold. The need to switch to cooled optics [1] dictates the requirement to preserve the quality of the optical surface - its movement due to bending should not exceed (5-10)% of the wavelength of laser radiation in the near infrared range. Passive cooled laser mirrors consist of a thin substrate, a cooling system and a thick base. The task of a laser mirror is to reflect the incident radiation as much as possible and to minimize distortions caused by heating the mirror. The task of the passive mirror cooling system (CS) is to reduce the level of excess temperature relative to the coolant and to protect the mirror base from heat penetration into it. The use of cooling takes the design and technological performance of cooled optics to a qualitatively different level. Designers need to solve a number of contradictory mutually exclusive issues when developing cooling systems for laser mirrors. The tasks become much more complicated when using cooling systems in adaptive (deformable) mirrors (DM) [2]. Here it is necessary to solve the problem of creating a highly efficient compact cooling system in a rather thin optical block (3-9 mm thick), while maintaining the high quality of the optical surface and the ability to control its shape using drives. In this case, the task of optimal cooling is to maximally reduce the temperature of the OE and ensure its uniformity over the thickness of the optical block of the mirror.

In this work, with some simplifying assumptions, the results of solving several heat exchange problems are analyzed for both uncooled and cooled OE [3, 4]. In this case, to obtain qualitative results and upper estimates, only one-dimensional temperature fields in the OE were considered under the conditions of a uniform heat flux acting on a circular optical surface. The paper considers the use
of highly efficient multi-tier CSs with minichannels in the DM, providing a high compactness of the heat exchange surface and the intensification of heat transfer due to the use of the initial thermal sections of the coolant flow. To reduce bending displacements of the optical surface, the use of thermal compensation is considered. Estimated calculations were carried out using programs written in the mathematical editor Mathcad.

2. When to switch to cooled optical elements
For an uncooled OE, the absorption of the heat flux leads to an uneven temperature distribution over the thickness of the element and gradual heating, which lead to a curvature of the optical surface of the element. For a flat round plate loaded from the optical surface with a uniform heat flux and thermally insulated from the other side, the dependence for the maximum value of the optical surface deflection arrow at Fourier numbers \( \mathrm{Fo} > 0.3 \) has the form:

\[
W^\text{max}_\theta = 0.0625 \beta q D^2 / \lambda \approx 0.08 \beta Q_t / \lambda ,
\]

where \( q \) is heat flux density; \( Q_t \) - thermal power; \( \beta \) - coefficient of linear expansion of OE material; \( \mathrm{Fo} = \alpha t / H^2 \) - Fourier number; \( H \) and \( D \) - OE thickness and diameter; \( \alpha = \lambda / (c_p \rho) \) - coefficient of temperature conductivity; \( t \) - time.

From the analysis (1) it follows that an increase in the threshold for the use of uncooled optics is achieved due to the use of material with the smallest \( \beta \) and largest \( \lambda \), and a decrease in the power absorbed by OE. The latter is achieved by increasing the specular reflection coefficient (SPC) by using multilayer interference coatings. To perform its functions of transmitting laser radiation without large distortions, the bending movements of the optical surface of the OE are usually limited \( W^\text{max}_\theta < (\eta/20-\eta/10) \) (where \( \eta \) is the wavelength of the laser radiation). Then the maximum absorbed power is estimated for the case in which the bending movements do not exceed \( \eta/10 \), according to the formula:

\[
Q^\text{lim}_t = 1.25 \lambda \cdot \eta / \beta .
\]

Estimates of the limiting thermal power under the action of laser radiation with a wavelength \( \eta = 1 \mu \text{m} \) on various optical materials are given in Table. 1. With a known SPC, using (2), one can estimate the maximum power of laser radiation from the point of view of ensuring the geometric stability of the optical surface, or, with a known laser power, the required SPC.

| Material      | \( \beta \cdot 10^6 \), 1/K | \( \lambda \), W/(m⋅K) | \( Q^\text{lim}_t \), W |
|---------------|-----------------------------|------------------------|------------------------|
| ULE           | 0.03                        | 1.31                   | 54.6                   |
| Zerodur       | 0.05                        | 1.65                   | 41.3                   |
| Silicon       | 2.60                        | 130.00                 | 62.5                   |
| Silicon carbide | 2.80                       | 190.00                 | 84.8                   |
| Leucosapphire | 5.60                        | 27.20                  | 6.1                    |
| Quartz KU-1   | 0.55                        | 1.38                   | 3.1                    |
| K8 glass      | 7.10                        | 1.11                   | 0.2                    |
| Glassceramic SO115M | 0.05                  | 1.18                   | 29.5                   |

When the capabilities of uncooled optics to maintain the permissible curvature of the optical surface are exhausted, forced cooling of the OE is used either by gas or liquid. For an OE in the form of a round plate with freely supported edges and when its rear side is cooled, the maximum steady-state flexural thermal displacement of the optical surface when exposed to it with a constant uniform heat flux \( q \) is equal to:

\[
W^\text{max}_\theta = \beta q D^2 / (8\lambda).
\]
and does not depend on the thickness of the plate and the heat transfer coefficient on the cooled surface, which determines only the level of the average temperature of the OE. Thus, the deformation of the EO (mirror) cooled from the back side is two times higher than that of the uncooled one (see formula (1) for the case \( F_0 > 0.3 \)), due to the linear temperature distribution over the OE thickness.

For a mirror made of a homogeneous material, made in the form of a package of a thin reflecting mirror plate-substrate (thickness \( h \)), a compact CS (height \( h_k \)) and a thicker base (total thickness of the mirror - \( H \)), with some simplifications, thermal displacement of the optical surface of the mirror due to bending can be estimated by the formula:

\[
W_{\text{fl}}^\text{max} = \frac{1}{8} \beta h \left( \frac{D}{H} \right)^2 \left[ 6 - \frac{q h}{\alpha_{\text{th}}} \left( 1 - \frac{h}{H} \right) + \frac{q h}{\lambda} \left( 3 - 2 \frac{h}{H} \right) \right],
\]

where \( \alpha_{\text{th}} \) is the reduced heat transmission coefficient to the heat exchange surface of the substrate. For \( h/\lambda << (1/\alpha_{\text{th}}) \) and \( h/H<<1 \), Eq. (4) will be simplified

\[
W_{\text{fl}}^\text{max} = \frac{3}{4} \beta h \left( \frac{D}{H} \right)^2 \frac{q}{\alpha_{\text{th}}} = \frac{3}{4} \beta \frac{h}{H^2} \frac{Q_t}{\alpha_{\text{th}}}. \]

From the analysis of (5) it follows that to reduce the bending it is necessary to decrease the values of \( \beta, h, Q_t \), and to increase the values of \( H \) and \( \alpha_{\text{th}} \). Structurally, it is possible to select a material and change the thickness of the substrate and the base of the mirror. The most effective effect on the thickness of the base, which is usually realized in cooled passive metal optics. It is also necessary to strive to increase the SPC (while \( Q_t \) decreases). The design of the mirror CS should ensure the maximum \( \alpha_{\text{th}} \) at a given pressure drop of the coolant across the mirror. From equation (5), one can obtain an estimate of the \( Q_t^\text{lim} \) for cooled optics, similar to (2):

\[
Q_t^\text{lim} \approx 0.1 \eta H^2 \alpha_{\text{th}}/(\beta h). \]

Taking \( \eta=1 \, \mu\text{m} \) and using the sizes \( h = 1 \, \text{mm} \) and \( H = 8 \, \text{mm} \) most suitable for cooled optical units of DM, in table 2 shows an estimate of the \( Q_t^\text{lim} \) for various structural materials and the reduced heat transmission coefficients \( \alpha_{\text{th}} \).

| Material          | \( \beta \times 10^6 \) | \( \lambda \), W/(m-K) | \( Q_t^\text{lim} \), W |
|-------------------|-------------------------|------------------------|-------------------------|
|                   |                         | \( \alpha_{\text{th}}=5 \times 10^4 \), W/(m^2-K) | \( \alpha_{\text{th}}=1 \times 10^5 \), W/(m^2-K) | \( \alpha_{\text{th}}=1.5 \times 10^5 \), W/(m^2-K) |
| Copper            | 16.7                    | 385                    | 19.1                    | 38.2          | 57.3          |
| Molybdenum        | 5.1                     | 130                    | 62.7                    | 125.4         | 188.1         |
| Bronze BrH-0.8    | 16.2                    | 314                    | 19.8                    | 39.6          | 59.4          |
| Silicon carbide   | 2.8                     | 190                    | 114.3                   | 228.6         | 342.9         |
| Silicon           | 2.6                     | 140                    | 123.1                   | 246.2         | 369.3         |
| Invar             | 1.0                     | 11                     | 320.0                   | 640.0         | 960.0         |

Note that for highly heat-conducting structural materials, it is possible to increase \( \alpha_{\text{th}} \) by improving the design, for example, by reducing the hydraulic diameter and increasing the compactness of the CS. For low heat-conducting materials (Invar) \( \alpha_{\text{th}} \) will always be at the level of the surface heat transfer coefficient.

Depending on the \( Q_t \), duration and frequency of loading, it is possible to use both cooled and uncooled OE. For example, if the OE are made of the same material and have the same SPC, then the
same bending displacement in the steady state (for an uncooled OE at \( F_o > 0.3 \), for a cooled OE at a
stationary mode) under loading with a constant and uniform heat flux, they will approximately have, subject to observance conditions

\[
12 \alpha f h / (\alpha_{th} H^2) = 1. \tag{7}
\]

If the material and dimensions \( h \) and \( H \) are chosen, then from (7) follows the requirement for the
value of the minimum \( \alpha_{th} \) when passing from an uncooled to a cooled laser mirror. It is advisable to
increase \( \alpha_{th} \) up to the values \( \alpha_{th} = 2 \lambda / h \). For cooled mirrors made of highly heat-conducting materials
(copper, molybdenum, silicon carbide, etc.), the values of \( 2 \lambda / h \) exceed \( 2 \cdot 10^5 \) W/(m²·K).

Given the applicability of the simplified equation (5), we estimate the minimum \( \alpha_{th} \), at the second
it is necessary to proceed to the cooling of the DM optical unit. In this case, we take the same design
dimensions \( h \) and \( H \), as in the calculations for table. 2. We obtain \( \alpha_{th} = 2 \cdot 10^3 \) W/(m²·K) for invar and
\( \alpha_{th} = 7.2 \cdot 10^4 \) W/(m²·K) for copper. Let us estimate the error introduced by the assumption \( h/H \ll 1 \),
which does not hold for the remote sensing (where \( h/H > 0.1 \)). For low heat-conducting metals (Invar),
the error reaches 66 % at \( \alpha_{th} = 5 \cdot 10^4 \) W/(m²·K) and increases with increasing \( \alpha_{th} \). For copper, it ranges
from −7 % to 5 % with a change in \( \alpha_{th} \) from \( 5 \cdot 10^4 \) W/(m²·K) to \( 1.5 \cdot 10^5 \) W/(m²·K).

3. Cooled deformable mirrors
Earlier in [4], experimental data on hydraulic resistance and heat transfer for the two most suitable CS
for DM were presented and summarized - a channel system with an intermittent wall (undercuts) and a
wafer cooling system. Here, calculations of its thermally deformed state are given for the wafer
channel CS of the DM, implemented in practice. A multi-tiered CO with minichannels is considered as
a promising system.

3.1. Implemented CS with water cooling
Estimation calculations were carried out for the realized optical block of a water-cooled deformable
mirror made entirely of Mob copper with an optical surface diameter of 410 mm. The thickness of the
mirror substrate is \( h = 1.5 \) mm and the thickness of the optical block is \( H = 12 \) mm. Cooling system
checkerboard waflle. Channels \( \delta_c = 1.3 \) mm wide and \( h_c = 3.5 \) mm deep are cut with a step \( t_c = 2.55 \) mm
(rib thickness \( \delta_c = 1.25 \) mm) at an angle of 60°. The entire optical surface was affected by a uniform
heat flux with a power of \( Q_c = 1 \) kW. The calculation used experimentally obtained formulae [4] for the
hydraulic resistance coefficient \( \xi \) and the Nusselt number \( Nu \), which are true in the range of the
Reynolds number change from \( Re = 6 \cdot 10^2 \) to \( Re = 6 \cdot 10^4 \).

\[
\xi_2 = 3.14 \text{Re}_2^{0.12},
\]

\[
K_1 = \text{NuPr}^{-1/3} = 0.41 \text{Re}_1^{0.646} \tag{8}
\]

A compensatory uniform heat flux with a power \( Q_c = (0.1-0.2)Q_t \) was applied on the reverse side
of the optical unit. The reduced heat transfer coefficient varied in the range \( \alpha_{th} = (4.2-13.0) \cdot 10^4
\)
W/(m²·K). In this case, the maximum deflection of the optical surface of the mirror was for: 1) for
calculations according to (8) - (5-2.73) \( \mu m \) at \( Q_c = 0 \), (3.74-1.85) \( \mu m \) at \( Q_c = 0.1Q_t \), (2.9-1.23) \( \mu m \) at
\( Q_c = 0.2Q_t \), 2) for calculations by the formula (4) - (4.15-1.56) \( \mu m \) at \( Q_c = 0, (2.9-0.85) \mu m \) at
\( Q_c = 0.1Q_t , (1.6-0.14) \mu m \) at \( Q_c = 0.2Q_t \).

3.2. Promising multi-tiered minichannel CSs
Reducing the hydraulic diameter of the channel and thereby increasing the compactness of the heat-
exchange surface seems to be the most promising way of forcing the heat-exchange characteristics of
CS for DM. A multi-tiered minichannel CS with short channels is able to significantly increase \( \alpha_{th} \) and
reduce bending displacements. Here are the results of computational estimates of the thermally
stressed state of the cooled DM unit with a minichannel CS made in copper. Three sizes of a channel
with a square cross-section with a hydraulic diameter $d_h=0.2$, 0.15, 0.1 mm are considered. The channels are made in thin plates with a thickness of $h_p=0.3-0.15$ mm by means of photolithography. The rib thickness is equal to the channel width (i.e. porosity $\varepsilon=0.5$). The relative length of the channels is constant $l_c/d_h=100$. The compactness of CS was estimated according to the formula $K_{cs}=2/h_p$ and varied from $6.67 \times 10^3$ to $13.3 \times 10^3$ 1/m with a change in $d_h$ from 0.2 mm to 0.1 mm. The plates are packaged and connected by thermosetting soldering. Particular attention in such CSs should be paid to the creation of a network of distributing and collecting channels to ensure uniform supply and removal of liquid without distortions of the temperature field at the base of the mirror.

Calculations were carried out sequentially with an increase in the number of tiers of the system from one to six. At the same time, a laminar water flow regime was maintained in the channels, the Reynolds number did not exceed the critical value for a laminar flow $Re_{cr} \leq 2.3 \times 10^3$. The calculations were performed for a round deformable mirror 200 mm in diameter, substrate and entire optical block thicknesses of 1.5 and 10 mm, respectively. The entire aperture of the mirror was affected by a uniform heat load with a power $Q_t=1$ kW. A compensatory uniform heat flux with a power $Q_{tc}=(0.1-0.2)Q_t$ was applied on the reverse side of the optical unit.

Calculations of the temperature distribution over the thickness of the optical block were carried out according to the models proposed in [5, 6]. In this case, the assumption of an approximately uniform temperature distribution along the perimeter of the minichannel made in [6] is in satisfactory agreement with the solution proposed in [5] only when the number of tiers does not exceed three.

With a change in the specific water consumption $G_m$, the following were calculated: 1) the surface heat transfer coefficients on the channel walls $\alpha$; 2) the temperature distribution over the thickness of the optical block; 3) the reduced heat transfer coefficients for each layer and for the entire system $\alpha_{ht}$; 4) the excess temperature of the heat exchange surface of the substrate $\Delta \theta_{hs}$; 5 ) the maximum bending displacements of the optical surface $W_{fl}(G_m)$ both by integrating the temperature distribution over the thickness and by applying formula (4); 6) heating the liquid $\Delta t(G_m)$ and the pressure drop due to friction in the CS, 7) reducing thermal displacements under the influence of thermal compensation (including cases of determining the required flow rate and the thermal power of compensation $Q_{hc}$, leading to the absence of bending), 8) determination of the specific flow rate of the liquid, upon reaching which the option of a cooled mirror is preferable to an uncooled one. Selectively, the results of the calculations are shown in Table 3.

### Table 3. Calculation results for $d_h=0.1$ mm at Reynolds number $Re_{cr}=2.3 \times 10^3$ (i.e. at maximum specific flow rate $G_m$)

| Calculated value |          | Number of tiers |
|------------------|----------|----------------|
|                  | 1        | 2              | 3              | 6              |
| Speed $v$ (m/s), $G_m$ (kg/(m$^2$·s)), consumption $G_m$ (kg/s) | 20/100/2 | 20/200/4       | 20/300/6       | 20/600/12      |
| $\alpha \cdot 10^{-5}$, $\alpha_{ht} \cdot 10^{-5}$ (W/(m$^2$·K)) | 0.53/1.04 | 0.53/1.97      | 0.53/2.65      | 0.53/3.44      |
| $\Delta \theta_{hs}(G_m)$, °C | 0.460 | 0.290 | 0.254 | 0.218 |
| Bend $W_{fl}(G_m)$ (µm) at $Q_{tc}=0$ | 0.706 | 0.81 | 1.0 | 1.32 |
| Bend $W_{fl}(G_m)$ (µm) at $Q_{tc}=0.1Q_t$ | 0.285 | µ/p | µ/p | µ/p |
| $\Delta t(G_m)$, °C | 0.13 | 0.065 | 0.044 | 0.022 |
| Transition to cooling at $G_m$ (kg/(m$^2$·s)) | $>34.4$ | $>12.6$ | $>7.3$ | $>14.3$ |

The predicted tendencies of channel refinement in a single-tier system were: 1) an increase in both $\alpha$ and $\alpha_{ht}$; 2) an increase in $W_{fl}$ when using thicker heat transfer plates; 3) a decrease in $W_{fl}$ with an
increase in $Q_{tc}$. The same tendencies were observed with an increase in the number of tiers. The maximum coefficient of reduced heat transfer for the dimensions of the cooling system specified above is determined from the formula:

$$
Nu_{rht} = \varepsilon Nu + (1 - \varepsilon)\sqrt{2\lambda Nu/\lambda_w},
$$

where $Nu_{rht}$, $\alpha_{rht}$, $\lambda$, $\lambda_w$ are the thermal conductivity coefficients of the material and water. Calculations carried out according to (9) and the law of heat transfer in a laminar flow regime are shown in Figure 1 for CS from various materials.

![Figure 1](image)

**Figure 1.** Maximum possible values of the reduced heat transfer coefficient depending on the Reynolds number for laminar flow in the minichannels of a multi-tiered CS. Indices in $\alpha_{rht}$: 1 - invar, 2 - molybdenum, 3 - silicon, 4 - silicon carbide, 5 - bronze, 6 - copper

### 4. Conclusion

The presented formulas (1), (4) for estimating the maximum bending displacements of the optical surface are suitable for both uncooled and cooled optics. The condition for the transition (7) from uncooled to cooled optical elements makes it possible to determine their design dimensions and choose a suitable CS. For short operating times of laser installations (on the order of several minutes), one should strive to use uncooled optics by selecting a material and using highly retractive (or transmissive) coatings. The restrictions imposed on the maximum absorbed thermal power by the optical element (2), (6), at which the geometric stability of the optical surface is ensured, will be useful when choosing a material, forming requirements for SPC, and $\alpha_{rht}$.

The evaluations carried out for the implemented and future systems of water cooling of DM showed that systems with wafer channels are technologically simple to implement and allow one to obtain acceptable thermohydraulic characteristics. For a significant increase in the heat exchange characteristics of CSs, it is necessary to switch to minichannel ($d_h=0.1-0.2$ mm), multi-tiered systems with short channels operating in the laminar flow regime and allowing $\alpha_{rht}$ to be increased several times compared to channel, wafer and coplanar systems.

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