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Thermoelectric Cooling of High Power Extremely Localized Heat Sources: System Aspects

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Abstract

A generalized approach to the cooling of localized high power heat sources is given. A system is considered, which consists of a heat source and a cooling unit including a cascade thermoelectric cooler (TEC) and a finned heat exchanger with a fan. The method of optimal integration of a TEC into the cooling system, which provides minimal input power is described. The cooling unit destined to maintain a concentrated (0.8x2.5mm) 8 W heat source at 25°C, the ambient temperature being of 70°C, is fabricated and tested.

Introduction

Many modern electronic devices such as power amplifiers, semiconductor lasers and some other electronic and electro-optic components act as high density exclusively localized heat sources. Their heat dissipation can achieve 10W and more while base area is of the order of 1mm^2. Thus, the heat flux density at their interface with a TEC can reach 1kW/cm^2. No one standard TEC can meet such demands. Even with TE leg as short as 0.1mm, TE cooler has cold flux density not more than 100W/cm^2 [1,2]. Thus, the TEC cold side dimensions must be much greater than those of the object to be cooled. This leads to a considerable thermal resistance of the thermal path from the heat source to the TEC cold junctions. Even with such high thermal conductive substrates as beryllium or aluminum nitride ceramic, temperature drop at a TEC cold side can exceed 10K. This unfavorable factor reduces greatly TEC efficiency and leads to further increase in its dimensions. Thus, the minimization of a TEC cold side thermal resistance is one of the main aspects when cooling localized heat sources. Another problem is connected with thermal resistance at a TEC hot side. It is not practical to use a cumbersome heat sink in the device intended to cool a tiny object. Thus, in applications discussed, a heat sink must meet definite space restrictions, what can lead to a severe thermal problem at a TEC hot side.

A generalized approach to the cooling of localized high power density heat sources is presented. A system is considered, which consists of a heat source and a cooling unit including a multi-stage TEC and a finned heat exchanger with a fan. To minimize total input power, the overall aspects of all these components are involved into optimization. An advanced system for temperature control of a laser diode with 8W of heat dissipation is described.

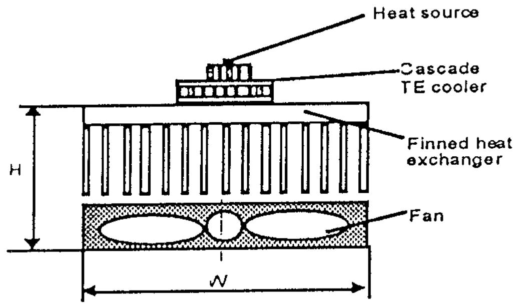

Figure 1. Scheme of thermoelectric cooling unit.

1. Thermal model of a cooling system

Let us consider a thermal model of a N-stage TEC with a concentrated heat source attached to its cold substrate (Fig.1). Specified are heat load from cooled object Q_c, its obligatory temperature T_c and ambient temperature T_a. Given are also the thermoelectric properties of TE materials in the range from T_c to T_a. The thermal resistance of the cold substrate R_s and that of a heat sink R_h are taken into account, while intercascade thermal resistances are neglected.

Let us consider stationary operation of this cooler. When electrical current is supplied to the TEC, heat fluxes Q_{0k}, Q_{1k}, k=1,...,N, arrive at cascades cold and hot sides. The corresponding equilibrium temperatures at their boundaries are T_k, k=0,...,N. The heat fluxes obey the continuity condition

$$\begin{aligned} Q_{01} &= Q_c \\ Q_{0k+1} &= Q_{1k}, \quad k = 1,\dots,N-1 \end{aligned} \tag{1}$$

and the boundary temperatures are defined as follows:

$$T_0 = T_c - R_s Q_c \tag{2}$$

$$T_N = T_a + R_h Q_h \tag{3}$$

where Q_h=Q_N is the total heat transferred to the heat sink.

With q_{0k} and q_{1k} as heat flux densities and F_k as junctions surface, the heat fluxes at cascade boundaries are defined by the equalities:

$$Q_{0k} = F_k q_{0k}, \qquad Q_{1k} = F_k q_{1k} \tag{4}$$

To find exact q_{0k} and q_{1k} values, the system of differential equations for heat transport and heat balance within a TE leg have to be solved. With x axis directed from the cold junction to the hot one, the system has the form:

$$\left.\begin{aligned} q &= -\lambda \frac{dT}{dx} + \alpha i T \\ \frac{dq}{dx} &= \alpha i \frac{dT}{dx} + i^2 \rho \end{aligned}\right\}_{n,p} \tag{5}$$

where $T(x)$ is absolute temperature, $q(x)$ is a generalized heat flux density, i is electrical current density, $\alpha(T)$, $\rho(T)$, $\lambda(T)$ are Seebeck coefficient, electrical resistivity and thermal conductivity of TE material, all the kinetic coefficients being dependent on temperature.

With α and i taken as their absolute values, the system (5) is valid both for n-type and p-type TE legs. So the "n" and "p" indices in (5) mean that these equations are to be written for both legs of a thermocouple, which for a N-stage TEC gives a system of 2N equations. A numerical solution of the system with boundary conditions $T(0)=T_{k-1}$, $T(l_k)=T_k$, gives desired heat flux densities, the formulas being as follows:

$$q_{0k} = \frac{1}{2}(q(0)_n + q(0)_p) - i_k^2 r_c$$
$$q_{1k} = \frac{1}{2}(q(l_k)_n + q(l_k)_p) + i_k^2 r_c \qquad (6)$$

where l_k is TE leg length, r_c is electrical contact resistance.

For relatively small temperature range when single stage configuration is available, the simplified model with constant kinetic parameters can be accepted with satisfactory accuracy. For this case, the system (5) has the well known solution:

$$q_{0k} = \alpha_k i_k T_{k-1} - \frac{1}{2} i_k^2 (\rho_k l_k + 2r_c) - \frac{\lambda_k}{l_k}(T_k - T_{k-1})$$
$$q_{1k} = \alpha_k i_k T_k + \frac{1}{2} i_k^2 (\rho_k l_k + 2r_c) - \frac{\lambda_k}{l_k}(T_k - T_{k-1}) \qquad (7)$$

where α_k, ρ_k, λ_k are averaged (for n-type and p-type TE materials) kinetic coefficients related to the cascade averaged temperature $T_{mk}=(T_{k-1}+T_k)/2$.

The problem is to find such a cooling unit configuration, which satisfies specified restrictions with minimal energy consumption. As a first step, the method of minimization of thermal resistance R_h and R_s under specified constraints has to be developed to provide minimal losses at a TEC boundaries. Then the optimal integration of a TEC into the cooling system must be done.

2. Heat Sink Optimization

We define heat sink thermal resistance R_h as a ratio

$$R_h = (T_h - T_a)/Q_h = \Delta T_h / Q_h \qquad (8)$$

with $T_h=T_N$ as a TEC hot side temperature. The maximum dimensions of the heat sink base $F_b=LW$ and its height H are regarded as specified. The goal is to optimize heat sink internal configuration to a minimal R_h. The problem was considered earlier [3] for specified fan and heat sink combinations. A more general approach is given here.

Heat sink thermal resistance. Let us express R_h as a function of heat exchanger dimensions and air flow rate. Fig. 2 shows schematically the temperature distribution in the air flow along a heat exchanger. The temperature of the heat sink base T_h is regarded as uniform. It is greater than the temperature of the ambient air T_a. Two unfavorable factors contribute to this overheating. The first is air flow heating ΔT_a, when moving along heated fins. In addition, the heat exchanger's own temperature always exceeds that of the exiting air (ΔT_2). These quantities define the mean integral temperature difference ΔT_m between air flow and heat sink as follows:

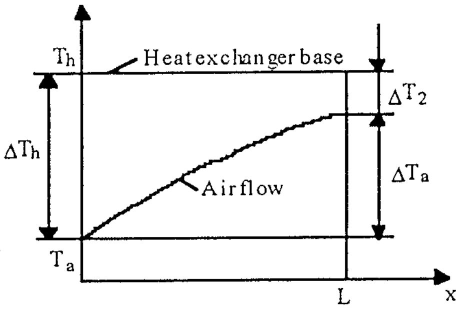

Figure 2. Temperature distribution along a heat sink.

$$\Delta T_m = \frac{\Delta T_h - \Delta T_2}{\ln(\Delta T_h / \Delta T_2)} \qquad (9)$$

Solution of this equation as to ΔT_h and its substitution into (8) gives R_h in the following form:

$$R_h = \frac{\Delta T_a}{Q_h}\frac{\exp(\vartheta)}{\exp(\vartheta)-1} = \frac{1}{C_p \gamma V}\frac{\exp(\vartheta)}{\exp(\vartheta)-1} \qquad (10)$$

where

$$\vartheta = \frac{\Delta T_a}{\Delta T_m}; \qquad \Delta T_a = \frac{Q_h}{C_p \gamma V}; \qquad \Delta T_m = \frac{Q_h}{h F_e} \qquad (11)$$

C_p is air specific heat, γ is air density, h is heat exchange coefficient, F_e is the total effective (with consideration of fin efficiency) heat sink surface defined as follows:

$$F_e = F_b\left(1 + K_f \tanh(mH_f)/(mH_f)\right) \qquad (12)$$

where

$$m = \sqrt{\frac{2h}{\delta \lambda_f}} \qquad (13)$$

$K_f=2H_f/t$ is coefficient of finning, H_f is fin height, λ_f is fin thermal conductivity, t and δ are fin pitch and fin thickness.

Using (11) and (12), one obtains the following expression for ϑ value

$$\vartheta = \frac{hF_b}{C_p \gamma V}\left(1 + \frac{2H_f}{t}\frac{\tanh(mH_f)}{mH_f}\right) \qquad (14)$$

Thus, the heat transfer coefficient h remains the only unknown variable. To calculate this quantity, the criterial equations for forced convection in plain rectangular channels must be used. These well known equations [4] define h value as a function of Reynolds number, which is specified itself by the air flow rate. Thus, the relations (10) and (14) together with above mentioned criterial equations define R_h as a function of heat sink dimensions and air flow rate

$$R_h = f_1(L, W, H; t, \delta, V) \qquad (15)$$

Heat sink hydraulic characteristics. The next question to be answered is as follows: which will be air flow rate through the given heat exchanger when connected to the given fan? To solve the question, the fan's measured dynamic characteristic

$$V = f_2(\Delta P) \qquad (16)$$

must be considered together with the equation for pressure head in heat exchanger. This last one can be expressed as:

$$\Delta P = f_3(L, W, H; t, \delta, V) \qquad (17)$$

To find a practical form of Eq.(17), one must express the ΔP as a sum of separate components of hydraulic resistance in a heat exchanger. All these quantities are represented in data books on heat transfer [4], so it would be superfluous to include such details into this paper. It is important for us that

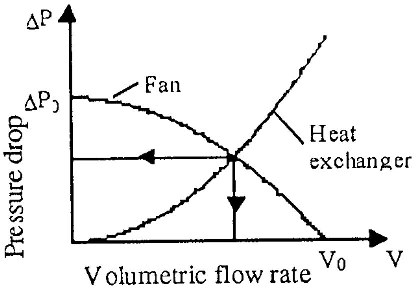

Figure 3. Typical hydraulic characteristics of a fan and heat exchanger.

they are all direct functions of air flow rate and heat exchanger dimensions, which identifies uniquely the form of equation (17). At Fig.3 a typical view of dependencies (16) and (17) is shown. Their intersection gives resultant V and ΔP values for given heat sink configuration. An iteration method can also be used to receive the solution.

Heat sink optimization. Parameters L, W, H in (15) are fixed, while t and δ are considered as independent variables. The goal is to find such t and δ values, which provide minim heat sink thermal resistance compatible with restrictions (16), (17). The following considerations prove the existence of such optimum. When reducing fin pitch, the number of fins increases and heat transfer increases simultaneously due to reduction of channels width. Thus, if a fan can supply enough air flow, the t value must be chosen as small as possible. However, in real case the pressure head in a heat exchanger increases greatly with t reduction and air flow rate decreases accordingly to the fan $V(\Delta P)$ characteristic. So, the optimum t value must exist as a reasonable compromise of heat sink thermal and hydraulic characteristics. To find this optimum, the general dependences (15) to (17) must be determined. Then one of standard procedures of nonlinear programming can be used to solve the problem.

3. Cold side thermal resistance

A model of a TEC cold substrate is shown at Fig. 4. Heat flux Q_c from heat source enters into the substrate uniformly through rectangular area $F_c = 2a\text{x}2b$ at the center of its top surface and leaves through the whole surface of its bottom F_s=2Ax2B. The rest of the surface is adiabatically insulated. With these assumptions, one has three-dimensional temperature distribution inside the substrate. We define the thermal resistance R_s as a ratio $R_s=(T_c-T_0)/Q_c$, where T_c and T_o are mean integral temperatures inside F_c and F_s areas correspondingly. Earlier [5] a three-dimensional thermal model of intermediate substrate for a two stage cooler was developed and its thermal resistance was found as a function of its configuration. This solution is also valid for a cold substrate of a single stage TEC. With the nomenclature given here, the relation for R_s is as follows:

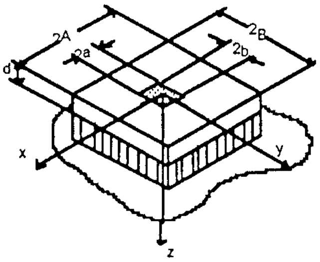

Figure 4. Scheme of a top substrate with a heat source

$$R_s = \frac{d}{\lambda_s F_s}\left(1 + 2S_1 + 2S_2 + 4S_3\right) \qquad (18)$$

where

$$S_1 = \sum_{n=1}^{\infty}\left(\frac{\sin(\nu a)}{\nu a}\right)^2 \frac{R(\nu d)}{\nu d}$$

$$S_2 = \sum_{m=1}^{\infty}\left(\frac{\sin(\mu b)}{\mu b}\right)^2 \frac{R(\mu d)}{\mu d}$$

$$S_3 = \sum_{n=1}^{\infty}\left(\frac{\sin(\nu a)}{\nu a}\right)^2 \sum_{m=1}^{\infty}\left(\frac{\sin(\mu b)}{\mu b}\right)^2 \frac{R(\alpha d)}{\alpha d}$$

$$R(\eta d) = \frac{g\cosh(\eta d) + \eta d\sinh(\eta d)}{g\sinh(\eta d) + \eta d\cosh(\eta d)}$$

$$\nu = \frac{n\pi}{A}, \qquad \mu = \frac{m\pi}{B}, \qquad \alpha^2 = \nu^2 + \mu^2$$

$$g = \beta\frac{d}{\lambda_s}\frac{\lambda}{l}\left(1 + \frac{\alpha i l}{\lambda}\right) \qquad (19)$$

$\beta=F_l/F_s$ is TE leg packing density, λ_s is substrate thermal conductivity.

Variable g reflects influence of a TEC operation mode on the temperature distribution in substrate through the complex $\varphi=\alpha il/\lambda$. This complex can vary in the range $0 < \varphi < zT_0$, z being TE material figure of merit. Our estimations show that with reasonable choice of ceramic thickness, the φ value does not affect practically substrate thermal resistance. So in our calculations the value $\varphi=0.5zT_a$ was used as giving negligible error. With this assumption, the above mentioned thermal model can be as well applied to the case of arbitrary cascades number.

A simple physical analysis shows that some optimal relationship of substrate dimensions exists, which gives minimum to its R_s value. When substrate thickness is small, its central part mostly carries heat, the heat spread to the periphery being negligible. If substrate thickness increases, the conditions for heat spread improve but the thermal resistance of its central part increases simultaneously. Thus, the optimum substrate thickness d_{opt} exists, which provides its minimal thermal resistance. This thesis is confirmed by

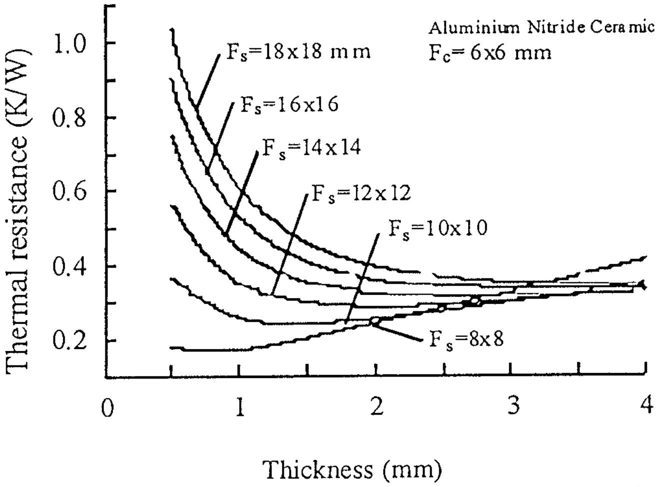

Figure 5. Dependence of substrate thermal resistance on its dimensions (λ_s=1.7W/(cm·K), λ=0.015 W/(cm·K), l=0.5mm, β=0.6).

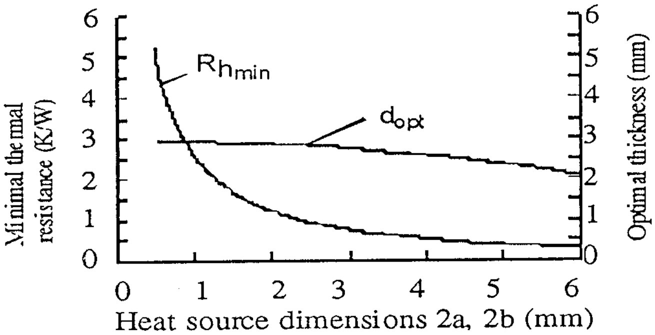

Figure 6. Dependence of minimal substrate thermal resistance and its optimal thickness on heat source dimensions (F_s=12x12mm, the rest of input data are the same as in Fig. 5).

calculations (Fig.5). For square heat source with F_c=6x6mm d_{opt} value lays in the range from 1 to 3.5 mm, increasing with its area. It is seen also that $d_{opt}\to 0$ when $F_s\to F_c$. On the contrary, for great F_s/F_c ratios d_{opt} increases significantly, moreover the deviation to the left from d_{opt} becomes extremely undesirable as leading to high thermal losses.

Fig.6 shows how critical is the influence of heat source dimensions on cold side thermal resistance even for such a highly thermal conductive substrate as aluminum nitride ceramic. When F_c value reduces from 6x6 to 1x1mm, the temperature drop at the TEC cold side increases by factor of ten achieving 25K at Q_c=8W. Thus, to provide high TEC efficiency, the use of effective intermediate heat spreaders at the interface of a heat source with a TEC is desirable.

4. System approach to the TEC optimization

Let us consider first of all the case with zero thermal resistance at a TEC boundaries. The temperatures at the TEC top and bottom sides become fixed: $T_0=T_c$, $T_N=T_a$. For this simple case the problem of cascade TEC optimization is well studied. It reduces to the minimization of the function [6]:

$$\mu = \prod_{k=1}^{N} \frac{q_{1k}(i_k, T_k, T_{k-1})}{q_{0k}(i_k, T_k, T_{k-1})} \qquad (20)$$

To identify the form of functions q_{0k} and q_{1k}, different methods are used including a simple approach with constant TE properties [6], the method based on temperature averaging of kinetic coefficients [7] and a more complex approach considering their exact temperature dependence [8]. In any case, the solution reduces to determination of the optimal intensive parameters i_k, k=1,...,N, and T_k, k=1,...,N-1, which give the corresponding q_{0k} and q_{1k} values. Then the cascade's areas F_k can be calculated consequently from the TEC top to its bottom using obvious equalities $F_1=Q_c/q_{01}$ and $F_{k+1}=F_k q_{1k}/q_{0k}$.

Now let us consider the real case. Initially the R_s value is not known because of lack of preliminary information about top substrate dimensions. The minimal R_h value is defined already but the total heat dissipation Q_h is not specified yet. So neither T_0, nor T_h are known from the very beginning. To obtain an exact solution for such conditions, the following iteration process is recommended:

1. As a first step, the optimal cooler configuration is to be found for R_s=0, $T_0=T_c$. The problem is to find the TEC hot side temperature $T_h=T_N$ corresponding to the specified R_h value. To solve the problem, the "internal" iteration process can be used. To start the iterations, the value $T_h^{(1)}=T_a$ can be accepted as the initial T_h value. Then the TEC is to be optimized with $T_0=T_c$ and $T_h=T_h^{(1)}$ using the above described algorithm. In such a way the first $Q_h^{(1)}$ value will be obtained, which gives a new $T_h^{(2)}$ value from Eq.(3). Then the iterations are to be renewed until the deviation of the next T_h value from previous one is less than the specified small error. This initial step gives the first approach to the cascade's areas $F_k^{(1)}$, k=1,...,N, and to the substrates' dimensions, which can be defined approximately as follows: $A_k^{(1)}=B_k^{(1)}=\sqrt{F_k^{(1)}/\beta}$.

2. Now the second $R_s^{(2)}$ value, minimized accordingly to the method described in item 3, can be calculated with obtained $A_1^{(1)}=B_1^{(1)}$ quantities and a new $T_0^{(2)}$ value will be obtained from the Eq. (2). The step 1 must be repeated with this new $T_0^{(2)}$ value, and so on, until the deviation of a new T_0 value from the previous one becomes acceptably small.

This algorithm was successfully used in this study to develop a cooling unit for a laser diode with 8W dissipation.

5. Advanced cooling system for high power laser diode

System requirements. A laser diode with 8W dissipation to be kept at 25°C, the ambient air temperature being of 70°C. The bottom dimensions of the diode are 0.8x2.5mm. To reduce the diode overheating, it is supplied with a segmented thermal bridge with bottom surface of 6x5.4mm, with a thermal resistance of 1K/W. This 6x5.4mm surface at the center of a TEC top ceramic was accepted as a reference one for the TEC evaluation. To keep the laser diode at 25°C with Q_c of 8W, this surface have to be kept at 17°C.

The heat sink (including a heat exchanger and a fan) must be inside a cube with a base of 60x60mm, this being the size of an available fan. The heat sink total height must be not more than 60mm. With these requirements, the goal is to minimize the overall system electrical power including a TEC and a fan, this to be below 50W. Variables to be optimized are: TE material parameters, TEC configuration, electrical current, substrates' material and dimensions, fin dimensions and spacing, fan model.

Heat sink. Several fan configurations with different heat exchangers were examined so as to obtain minimal heating of the TEC hot side. Small-sized fans from Yen Sun Technology Corporation were tried. For each fan model with its individual characteristic [9], the heat exchanger configuration was adapted to have a minimal R_h value compatible with specified space restrictions. Optimal fin spacing and dimensions were evaluated. Fig. 7 shows how important it is to choose appropriate fan together with optimal fin spacing. The best results are predicted for the fan FD1260257B having advanced hydraulic characteristics. In spite of reduced room for fins, the minimal thermal resistance (0.21K/W) can be obtained with this fan model.

Cooler. Several candidate TECs including 1-stage and 2-stage were examined to determine best efficiency. In all cases TE leg length of 0.5mm was chosen as giving reasonable compromise of high heat flux density with moderate losses at electrical contacts. AlN substrates were used, their configuration being optimized together with other system components. Predicted values of input power for 1-stage TEC and for 2-stage one are 30.7 and 26.8W respectively.

6. Prototypes testing

1-stage and 2-stage prototypes optimized for specified requirements were fabricated and tested. Photograph of the 2-

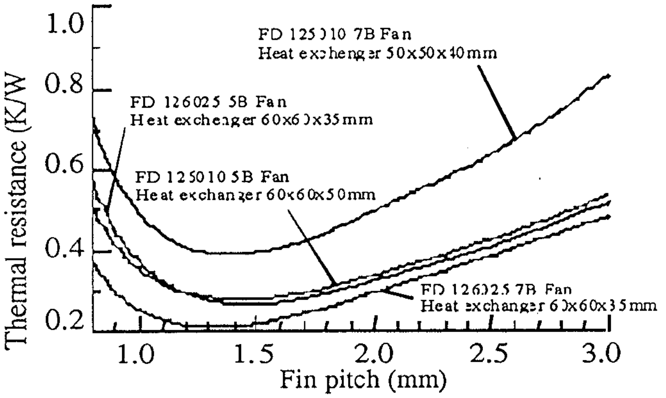

Figure 7. Dependence of heat sink thermal resistance on fin pitch (total heat sink dimensions including fan and heat exchanger fit to the volume 60x60x60mm).

stage prototype is given in the figure 8.The thermoelectric part was designed as a removable block consisting of a TEC soldered to a copper base. The block was mounted onto the heat exchanger with the use of thermal grease. The heat sink was initially tested with a planar electrical heater simulating the heat load from the TEC. The measured R_h value was 25% higher than its predicted available minimum. Also an additional unexpectedly high thermal resistance (of about 0.1K/W) was found at the interface of the cooling block with the heat sink, this was due to insufficient surface flatness.

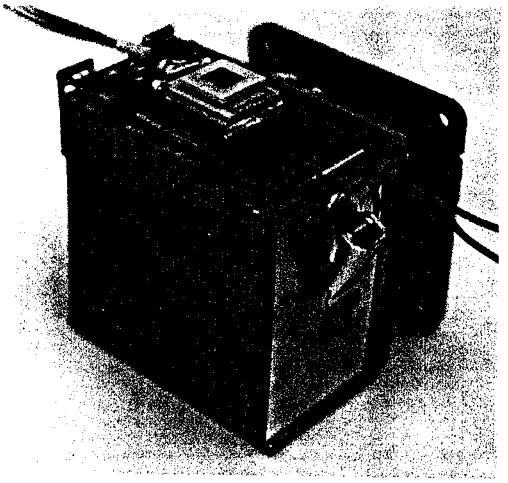

Figure 8. Photograph of the cooling unit for laser diode.

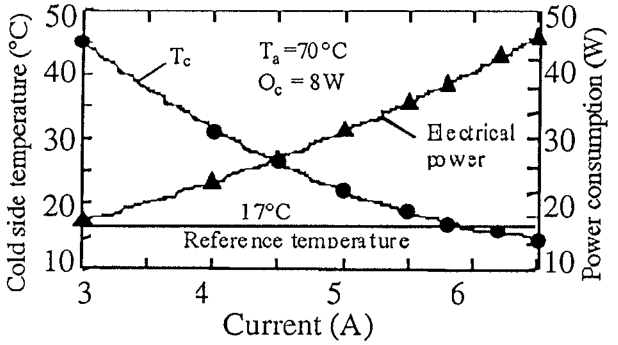

Figure 9. Dependence of cold side temperature and TEC power consumption on electrical current.

The total measured R_h value was of 0.4K/W instead of predicted 0.21K/W. Thus, the thermal problem at the hot side occurred to be more severe than it was expected. This was the reason why a 1-stage prototype failed to meet specified requirements. The required temperature of 17°C was obtained at ambient temperature not higher than 64°C. Test results for the 2-stage prototype (Fig. 9) shows that it meets the specification with a power input of 35.7W.

Conclusion

Increase in power density of modern thermoelectrically cooled electronic components gives rise to a severe thermal problem both at TEC cold and hot sides.

To minimize the total input power, all the components of a device including a heat source, a TEC and a heat sink in their interaction have to be involved into the optimization.

Correlation of heat exchanger internal configuration with a fan hydraulic characteristics proves to be the most important means to reduce TEC hot side thermal resistance.

Substrate thickness is the decisive factor which defines the thermal resistance at the TEC cold side. Its optimization can considerably improve TEC performance.

The feasibility of a compact cooling unit (total volume 60x60x60mm) destined for maintaining laser diode with 8W heat dissipation at 25°C, the ambient temperature being of 70°C, is proved theoretically and practically.

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