GA-A15336

A STEP-BY-STEP APPROACH TO CONVECTIVE COOLING OF LASER DISC AMPLIFIERS

by

R. F. BOURQUE

NOTICE

This report was prepared as an account of work sponsored by the United States Government. Neither the United States nor the United States Department of Energy, nor any of their employees, nor any of their contractors, subcontractors, or their employees, makes any warranty, explicit or implied, or assumes any legal liability or responsibility for the accuracy, completeness or usefulness of any information, apparatus, product or process disclosed, or represents that its use would not infringe privately owned rights.

**Work supported by
Lawrence Livermore Laboratory
Purchase Order 4406309**

GENERAL ATOMIC PROJECT: 3072

APRIL 1979

GENERAL ATOMIC COMPANY

A STEP-BY-STEP APPROACH TO CONVECTIVE COOLING
OF LASER DISC AMPLIFIERS

Robert F. Bourque

General Atomic Company
San Diego, California 92138

ABSTRACT

A step by step approach is presented to the problem of gas cooling of laser glass amplifiers. The basic equations are given for glass conduction, thermal stress, gas convection, and heat exchanger and blower design. An example calculation is then carried out for helium gas at one atmosphere with the gas flow in the direction orthogonal to the optical path. It is found that pumping powers and temperatures are acceptable for this case. Results are also presented for helium in the slant direction and for nitrogen in both directions. Included also are the effects on pumping power of gas temperature rise, gas pressure, flashtube rep rate, and flow channel width. It is found that, based on temperature rise and pumping power, nitrogen is as viable a coolant as helium.

Work performed under Lawrence Livermore Laboratory Purchase Order 4406309. General Atomic project 3072.

1. INTRODUCTION

1.1 PROBLEM DEFINITION

One of the questions concerning the use of glass lasers in inertial confinement fusion is the requirement to remove the heat deposited in the laser disc amplifiers by the flashtubes. This report addresses the problem of convectively cooling the glass surfaces by means of an inert gas. The objective is to find the proper flow and geometric parameters that allow the glass to be cooled with a minimum power required to pump the gas.

Figure 1 shows the laser glass layout chosen for analysis. It consists of three full width and two half width glass plates 1.0 m long in the slant direction and 0.5 m wide. There are four 0.01 m wide flow channels for the gas. Fig. 2 shows a schematic of the entire coolant circuit, which is made up of the laser glass, a water-cooled heat exchanger, a centrifugal blower, and associated ducting.

1.2 PURPOSE AND SCOPE

This report presents a simplified, step-by-step approach to solving problems of gas cooling of laser amplifier glass. Included also are heat exchanger and blower sizing calculations. The basic equations are first presented without derivation. The use of the equations is then demonstrated by means of an example problem. Results for two gases, helium and nitrogen at one atmosphere, and two flow directions are summarized. Scaling relations are given so that these results can be used to roughly predict performance at different pressures, flashtube repetition rates, and coolant temperature rises without having to go through the detailed calculations.

Fig. 1. Laser glass amplifier layout. Optical path is from left to right. Gas coolant flow is shown in slant (long) direction. Short direction is orthogonal to paper.

Fig. 2. Schematic of coolant flow circuit.

2. BASIC EQUATIONS

2.1 GLASS CONDUCTION AND STRESS

For the purpose of the study, each discharge of the flashtube is presumed to uniformly deposit e Joules/m³ into the glass*. The temperature rise per pulse is then

$$\Delta T_p = \frac{e}{\rho_g C_p} \quad (1)$$

where ρ_g is the glass density and C_p is its specific heat per unit mass. It will be shown later that ΔT_p is small compared to the steady state temperature rise and therefore this pulsed energy input can be regarded as a heat rate per unit volume

$$q = er \quad , \quad \text{watts/m}^3 \quad , \quad (2)$$

where r is the flashtube repetition rate in cycles per second (Hz). If V_g is the glass volume in cubic meters, then the total heat input to the glass is

$$Q = qV_g \quad , \quad \text{watts} \quad . \quad (3)$$

It can be shown (Ref. 1) that the temperature distribution in a slab with uniform internal heating and edge cooling is parabolic of the form

$$T = T_0(1 - z^2/t^2) \quad , \quad (4)$$

where t is the halfwidth of the slab and z is measured normal to the slab surface with $z = 0$ at the center. The peak temperature at the center of the slab T_0 relative to the edge, is given by

$$T_0 = \frac{Qt^2}{2kV_g} \quad , \quad (5)$$

where k is the thermal conductivity. It can be shown (Ref. 2) that the

* Worst case: actual deposition is closer to the surface.

thermal stress in a freely supported slab caused by the above temperature distribution is given by

$$\sigma = \frac{\alpha E T_0}{1-\nu} (z^2/t^2 - 1/3) \quad , \quad \text{N/m}^2 \quad , \quad (6)$$

where α is the coefficient of thermal expansion, E the modulus of elasticity, and ν is Poisson's ratio. The peak tensile stress occurs at the edge ($z/t = 1$) and is

$$\sigma_t = \frac{2}{3} \frac{\alpha E T_0}{1-\nu} \quad , \quad \text{N/m}^2 \quad . \quad (7)$$

Note again that T_0 is relative to the temperature at the edge of the glass. The peak compression occurs at the center of the glass and has a magnitude equal to half the peak tension.

2.2 GAS CONVECTION

The mass flow of the gas m_g depends on the allowable gas temperature rise ΔT_r and is given by

$$m_g = \frac{Q}{C_p \Delta T_r} \quad , \quad \text{kg/sec} \quad , \quad (8)$$

where C_p from here on refers to the gas. The gas temperature rise should in general be small because the glass temperature will track that of the gas creating an optical wedge.

A useful term in heat transfer calculations is the mass flow per unit area, G . If A_c is the cross sectional flow area, then

$$G = \frac{m_g}{A_c} \quad , \quad \text{kg/sec-m}^2 \quad , \quad (9)$$

and the flow velocity, u , is given by

$$u = \frac{G}{\rho} \quad , \quad \text{m/sec} \quad , \quad (10)$$

where ρ is the gas density in kg/m^3 . Other equations involving A_c are

$$A_c = \frac{m_g}{G} \quad , \quad \text{m}^2 \quad , \quad (11)$$

and the thickness of a heat exchanger with a core volume V_c is given by

$$t_c = \frac{V_c}{A_c} \quad , \quad m \quad , \quad (12)$$

The Reynolds number N_r is a dimensionless variable that relates dynamic forces to viscous forces. Large values of N_r indicate turbulent, inviscid flow while small values indicate laminar, viscous flow. The Reynolds number is defined by

$$N_r = \frac{4r_h G}{\mu} \quad , \quad (13)$$

where $4r_h$ is the hydraulic diameter and is defined as the cross sectional flow area divided by the wetted perimeter. For a circular tube, $4r_h$ is the tube diameter. Use of hydraulic diameter allows one to use experimental flow data for tubes for analysis of a channel of any cross sectional shape. μ is the gas viscosity.

The equation for heat flow by convection across a surface with area A is

$$Q = hA\Delta T_{bl} \quad , \quad \text{watts} \quad , \quad (14)$$

where h is the heat transfer coefficient across the gas boundary layer film and ΔT_{bl} is the temperature difference across the boundary layer. Experimental data for film coefficients are usually plotted against Reynolds number as a factor J , defined as

$$J = \frac{h}{GC_p} N_{pr}^{2/3} \quad , \quad (15)$$

where N_{pr} is the dimensionless Prandtl number and is equal to $\mu C_p / k$. Solving Eq. 15 for the film coefficient gives

$$h = \frac{JGC_p}{N_{pr}^{2/3}} \quad . \quad (16)$$

The film coefficient h can be found using Eq. 16 and experimental data for J . Then the boundary layer temperature rise can be determined from Eq. 14. The maximum glass temperature is then given by

$$T_{max} = T_{in} + \Delta T_r + \Delta T_{bl} + T_o \quad , \quad (17)$$

where T_{in} is the gas inlet temperature and T_0 is the temperature rise from the edge of the glass to the center.

While on the subject of temperatures, another useful temperature difference is the log-mean temperature difference, ΔT_{lm} , which is used when calculating the heat flow in a heat exchanger from one convecting fluid to another. Eq. 14 can be used for this purpose with ΔT_{lm} replacing ΔT_{b1} . If ΔT_1 is the bulk temperature difference between the two fluids at one end of a heat exchanger and ΔT_2 is that at the other, then

$$\Delta T_{lm} = \frac{\Delta T_2 - \Delta T_1}{\ln(\Delta T_2 / \Delta T_1)} \quad (18)$$

The purpose of this exercise is to determine if the laser glass can be adequately cooled by a convecting gas with acceptably low pumping powers. Pumping power depends on the pressure drop of the gas, which is given by

$$\Delta P = \frac{G^2 f A}{2 \rho A_c} \quad (19)$$

where f is a friction factor determined from experimental data and is usually plotted against Reynolds number along with the J factor. The total pressure drop is the sum of that across the glass, entrance and exit, ducts, and the heat exchanger that removes the heat from the gas. That is

$$\Delta P_{tot} = \Delta P_{glass} + \Delta P_{hx} + \Delta P_e + \Delta P_{duct} \quad (20)$$

where ΔP_{hx} and ΔP_e refer to the heat exchanger and entrance plus exit pressure drop, respectively. The former is calculated explicitly while the latter can be taken to be a fraction of the dynamic head, that is,

$$\Delta P_e = C \rho u_d^2 / 2 \quad (21)$$

where u_d is the duct flow velocity and is taken here to be half the flow velocity in the glass ($u_d = u/2$) and C is a constant taken here to be

0.4. Eq. 21 is also used for ΔP_{duct} with the same value of u_d but with $C = 0.2$ (0.6 total). This is a very conservative estimate for C . Sound engineering practices, such as fairings and smooth surfaces, normally yield a much lower number.

The ideal pumping power, not accounting for blower losses, is

$$W_1 = \frac{m_g \Delta P_{\text{tot}}}{\rho} \quad (22)$$

If the blower has an adiabatic efficiency η (taken here to be 0.5), then the power delivered to the blower shaft must be

$$W = W_1 / \eta \quad (23)$$

Note that this power ultimately goes to the gas and must be included in the heat flow to be removed by the heat exchanger.

2.3 BLOWER SIZING

The air blower that drives the gas is taken to be a simple squirrel-cage type centrifugal blower. In this section, equations are presented for the blower tip speed, inner and outer diameter, and thickness. The ratio of inner to outer diameter is defined as

$$\alpha = d_1 / d_0 \quad (24)$$

where d_1 and d_0 are the blower inner and outer diameters, respectively. The outlet tip speed depends on the pressure rise and is

$$u_0 = \left(\frac{2\Delta P}{\rho \eta (1 - \alpha^2)} \right)^{1/2} \quad (25)$$

and the inlet tip speed is

$$u_1 = \alpha u_0 \quad (26)$$

Blower stress is dependent on tip speed u_0 . In general, it is prudent to keep u_0 below about 250 m/sec. If too high, it can be lowered by staging more than one blower in series. Once the blower rpm is specified, the inlet and outlet diameters can be determined from

$$d_0 = \frac{60u_0}{\pi r \omega} \quad (27)$$

$$d_1 = \alpha d_0 \quad (28)$$

The radial gas velocity u_r is set low so as not to interfere with the pressure rise. In this case, we set

$$u_r = u_1/5 \quad (29)$$

The gas volume flow rate is

$$V = m_g/\rho \quad (30)$$

and so the axial thickness of the blower is given by

$$t = \frac{V}{\pi d_1 u_r} \quad (31)$$

2.4 SCALING RELATIONS

Calculated results are given in Section 3 for a specified flashtube rep rate, gas pressure, and coolant temperature rise. Scaling laws are presented in this section to facilitate estimates of performance at other values of these three variables. The approximation used is that the J and friction factors are constant, which is exact for varying gas pressure and is reasonably accurate over fairly wide Reynolds number ranges for varying rep rate and coolant temperature rise.

At a fixed pressure and coolant temperature rise, the pumping power fraction W/Q varies with heat deposition rate, Q , or repetition rate, r (for a given energy per pulse), according to

$$W/Q \sim Q^2 \sim r^2, \quad (32)$$

where the " $\sim$ " indicates proportionality.

At a given heat deposition rate and coolant temperature rise, pumping power fraction and gas flow velocity vary with gas pressure according to

$$W/Q \sim \frac{1}{P^2} \quad , \quad u \sim \frac{1}{P} \quad . \quad (33)$$

At a given pressure and heat deposition rate, pumping power fraction varies with coolant temperature rise according to

$$W/Q \sim \frac{1}{\Delta T_r^3} \quad . \quad (34)$$

Note the extreme sensitivity of pumping power to coolant temperature rise.

The blower tip speed varies with pumping power according to

$$u_0 \sim W^{1/2} \quad . \quad (35)$$

This completes the specification of equations. An example calculation using the above are given in the next section.

3. EXAMPLE CALCULATION

In this section, the reader is led through an example numerical calculation using the equations presented in Section 2. The example chosen is helium at one atmosphere with the gas flow in the direction transverse to the optical path.

The glass physical properties are given in Table 1 and the geometry of the glass is given in Table 2. Physical properties of helium and nitrogen are given in Table 3. Fig. 3 gives experimental heat transfer and friction data for circular tubes, which can be employed for the rectangular glass cross section by means of the hydraulic diameter. Fig. 4 shows a schematic of the heat exchanger and Fig. 5 shows experimental heat transfer and friction data for this heat exchanger. Figures 3 and 5 are taken from Ref. 3.

3.1 GLASS CONDUCTION AND STRESS

The energy deposited in the glass per unit volume per pulse is taken as a worst case to be

$$e = 0.4 \text{ J/cm}^3 = 400,000 \text{ J/m}^3 .$$

From Eq. 1 and Table 1

$$\Delta T_p = \frac{400,000}{3000(840)} = 0.16 \text{ C} .$$

This is low enough compared to the steady state glass temperature rise so that the heating can be considered steady state. From Eqs. 2 and 3 and Table 2

$$q = 400,000(1.0 \text{ Hz}) = 400,000 \text{ watts/m}^3 ,$$

$$Q = 400,000(0.04) = 16000 \text{ watts} .$$

TABLE 1
LASER GLASS PROPERTIES

Thermal Conductivity, k , watt/m-C	0.86
Density, ρ_g , kg/m ³	3000
Specific Heat, C_p , Joules/kg-C,	840
Thermal expansivity, α , C ⁻¹	10 ⁻⁵
Modulus of elasticity, E , N/m ²	$8 \cdot 10^{10}$
psi	8 $\cdot 10^6$
Poisson's ratio, ν ,	0.238

TABLE 2
LASER GLASS GEOMETRY

Thickness, $2t$, meters	0.02
Heat transfer area, A , m ²	4.0
Coolant flow areas, A_c , m ² :	
long direction	0.02
short direction	0.04
Glass Volume, V_g , m ³	0.04
Hydraulic Diameter, $4r_h$, m (both ways)	0.02

TABLE 3
GAS PROPERTIES

	HELIUM	NITROGEN
Density, ρ , kg/m ³ 1.0 atm 100 C	0.12	1.30
Specific heat, C_p , J/kg-C	5193	1000
Thermal conductivity, k , w/m-C	0.18	0.029
Viscosity, μ , kg/m-sec	$0.25 \cdot 10^{-4}$	$0.20 \cdot 10^{-4}$
Prandtl number, N_{pr}	0.67	0.69
$N_{pr}^{2/3}$	0.766	0.781

TUBE INSIDE DIAMETER = 0.231 IN.

HYDRAULIC DIAMETER = 0.231 IN., 0.01925 FT

FLOW LENGTH/HYDRAULIC DIAMETER, $L/4r_h = 79.2$

FREE-FLOW AREA PER TUBE = 0.0002508 FT²

Fig. 3. Experimental heat transfer and friction data for flow inside circular tubes. Bottom curve is J factor; top is friction factor. From Ref. 3 (used with permission of McGraw-Hill Book Company)

Fig. 4. (a) Kays and London heat exchanger surface 1/6-12.18(D). From Ref. 3 (used with permission of McGraw-Hill Book Company). (b) flow geometry.

FIN PITCH = 12.18 PER IN.

PLATE SPACING, $b = 0.353$ IN.

SPLITTER SYMMETRICALLY LOCATED

FIN LENGTH FLOW DIRECTION = 0.178 IN.

FLOW PASSAGE HYDRAULIC DIAMETER, $4r_h = 0.008648$ FT

FIN METAL THICKNESS = 0.004 IN., ALUMINUM

SPLITTER METAL THICKNESS = 0.006 IN.

TOTAL HEAT TRANSFER AREA/VOLUME BETWEEN PLATES, $\beta = 422.4$ FT²/FT³

FIN AREA (INCLUDING SPLITTER)/TOTAL AREA = 0.847

Fig. 5. Heat transfer and friction data for heat exchanger configuration shown in Fig. 4. Bottom curve is J factor; top is friction factor. From Ref. 3 (used with permission of McGraw-Hill Book Company).

The temperature at the center of the glass relative to the edge is, from Eq. 5 and Tables 1 and 2

$$T_o = \frac{16000(0.01)^2}{2(0.86)(0.04)} = 23 \text{ C} ,$$

and peak tensile stress is, from Eq. 7 and Table 1

$$\begin{aligned} \sigma_t &= \frac{2}{3} \frac{10^{-5}(5.54 \cdot 10^{10})(23)}{1-0.238} = 1.11 \cdot 10^7 \text{ N/m}^2 \\ &= 1620 \text{ psi} . \end{aligned}$$

This stress is probably quite reasonable even for glass which has not been preloaded.

3.2 GAS CONVECTION

A ten degree temperature rise, ΔT_r , is specified for the coolant gas. Then the mass flow is, from Eq. 8 and Table 3

$$\dot{m}_g = \frac{16000}{5193(10)} = 0.308 \text{ kg/sec} .$$

The mass flow per unit area is, from Eq. 9 and Table 2

$$G = \frac{0.308}{0.04} = 7.7 \text{ kg/sec-m}^2 .$$

The flow velocity is, from Eq. 10 and Table 3

$$u = \frac{7.7}{0.12} = 64 \text{ m/sec} .$$

The Reynolds number, from Eq. 13 and Tables 2 and 3, is

$$N_r = \frac{0.02(7.7)}{0.25 \cdot 10^{-4}} = 6160 .$$

The bump in the curves in Fig. 3 indicate the transition from laminar flow on the left side of the bump to turbulent flow on the right. With the Reynolds number calculated above, flow is just turbulent. From Fig. 3

$$J = 0.003 , \quad f = 0.0077 .$$

The value of f is for surfaces of average roughness. In view of the smoothness of the glass, it is clearly an overestimate. From Eq. 16 and Table 3

$$h = \frac{0.003(7.7)(5193)}{0.766} = 157 \text{ watts/m}^2\text{-C} .$$

The boundary layer temperature rise, from Eq. 14 and Table 2, is then

$$\Delta T_{bl} = \frac{16000}{157(4.0)} = 25 \text{ C} ,$$

and the maximum central glass temperature is, from Eq. 17

$$T_{max} = 50+10+25+23 = 108 \text{ C} ,$$

which is probably acceptable. If not, then there is margin for lowering the gas inlet temperature.

The gas pressure drop across the glass, from Eq. 19 and Tables 2 and 3, is

$$\Delta P = \frac{7.7^2}{2(0.12)} 0.0077 \frac{4.0}{0.04} = 190 \text{ N/m}^2 .$$

To this must be added the drops from entrance and exit, ducting, and the heat exchanger, which are calculated below.

3.3 HEAT EXCHANGER SIZING

To begin the heat exchanger sizing, a value of G is chosen and the resulting heat exchanger configuration is examined afterwards to see if it appears reasonable. The value selected is

$$G = 2 \text{ kg/m}^2\text{-sec} .$$

From Fig. 5, the hydraulic diameter is

$$4r_h = 0.008648 \text{ feet}/3.28 \text{ feet/meter} = 0.00264 \text{ m} .$$

From Eq. 13 and Table 3,

$$N_r = \frac{0.00264(2)}{0.25 \cdot 10^{-4}} = 211 .$$

From Fig. 5,

$$J = 0.035, f = 0.14,$$

and the film coefficient, from Eq. 16 and Table 3, is

$$h = \frac{0.035(2)(5193)}{0.766} = 475 \text{ watts/m}^2\text{-C}.$$

This film coefficient is for the gas side only. Conduction through the fins and thermal resistance of the water on the other side should also be considered. However, they can be expected to offer little thermal resistance compared to the gas. To a first approximation, the overall film coefficient is taken to be 80 percent of the gas side, or

$$h_o = 0.8(475) = 380 \text{ watts/m}^2\text{-C}.$$

The log mean temperature difference, from Eq. 18 and the bottom of Fig. 4 is given by

$$\Delta T_{lm} = \frac{30-15}{\ln(30/15)} = 21.6 \text{ C}.$$

The heat transfer area is, from Eq. 14 with ΔT_{lm} replacing ΔT_{bl}

$$A_{hx} = \frac{2(16000)}{380(21.6)} = 3.90 \text{ m}^2,$$

where the heat to be removed by the heat exchanger is taken to be twice that delivered to the gas by the laser glass. The heat input from the blowers, which the heat exchanger must also remove, is therefore conservatively accounted for.

The heat exchanger core volume, from Fig. 5, is

$$V_{hx} = \frac{3.90}{422.2(3.28)} = 0.00282 \text{ m}^3,$$

and the gas flow area is, from Eq. 11

$$A_{chx} = 0.308/2 = 0.154 \text{ m}^2.$$

The heat exchanger, if square, would be 0.39 m on a side, not counting water jackets or the small solid fraction due to the fins.

The heat exchanger thickness, from Eq. 12, is

$$t_{hx} = 0.00282/0.154 = 0.018 \text{ m} .$$

The core pressure drop, from Eq. 19 and Table 3, is

$$\Delta P_{hx} = \frac{2^2}{2(0.12)} 0.14 \frac{3.90}{0.154} = 59 \text{ N/m}^2 .$$

3.4 DUCTS, ENTRANCE, AND EXIT LOSSES

The duct flow velocity is taken to be half that in the glass or $64/2 = 32 \text{ m/sec}$. The duct diameter required to pass 0.308 kg/sec at this velocity and one atmosphere is 0.32 m and scales with $1/\sqrt{P}$. From Eq. 21, with $C = 0.6$, the duct, entrance, and exit combined loss is

$$\Delta P_e + \Delta P_{duct} = 0.6(0.12) \frac{32^2}{2} = 37 \text{ N/m}^2 .$$

3.5 PUMPING POWER

From Eq. 20, the total circuit pressure drop is

$$\Delta P_{tot} = 190 + 59 + 37 = 286 \text{ N/m}^2 \quad (0.041 \text{ psi}) .$$

The ideal pumping power, from Eq. 22 and Table 3, is

$$W_1 = \frac{0.308(286)}{0.12} = 734 \text{ watts} ,$$

and, for a blower efficiency of 0.5, the power input is, from Eq. 23

$$W = 734/0.5 = 1468 \text{ watts} .$$

The pumping power, as a fraction of the heat transferred, is then

$$W/Q = 1468/16000 = 0.092 .$$

3.6 BLOWER SIZING

The blower is a simple squirrel cage type that must provide a gas pressure rise of 286 N/m^2 . We first choose a ratio of inlet to outlet diameter Eq. 24 of $\alpha = 0.6$. If α is too close to unity, the tip speed would be too high; if too close to zero, the inlet would be too small. From Eq. 25, the outlet tip speed is

$$u_0 = \left(\frac{2(286)}{0.12(0.5)(1-(0.6)^2)} \right)^{1/2} = 122 \text{ m/sec} ,$$

which is well under the stress limited speed of 250 m/sec.

The inlet tip speed is, from Eq. 26

$$u_1 = 0.6(122) = 73 \text{ m/sec} .$$

From Eq. 27, the blower outlet diameter is, for 3600 rpm

$$d_0 = \frac{60(122)}{\pi(3600)} = 0.65 \text{ m} ,$$

and the inlet diameter is, from Eq. 28,

$$d_1 = 0.6(0.65) = 0.39 \text{ m} .$$

The radial gas velocity at the inlet is, from Eq. 29,

$$u_r = 73/5 = 14.6 \text{ m/sec} ,$$

and the volume flow rate is, from Eq. 30,

$$V = 0.308/0.12 = 2.57 \text{ m}^3/\text{sec} .$$

The blower axial thickness is therefore, from Eq. 31

$$t = \frac{2.57}{\pi(0.39)(14.6)} = 0.14 \text{ m} .$$

3.7 SCALING RELATIONS

The scaling relations below, based on Eqs. 32 to 35 provide estimates for different operating conditions. They are based on the assumption that the J and f factors are constant. This is exact for pressure variations and nearly true for others. The scaling laws for heat flow and repetition rate are

$$\frac{W}{Q} = 0.092 \left(\frac{Q}{16000} \right)^2 = 0.092 r^2 .$$

The scaling laws for gas pressure are

$$\frac{W}{Q} = \frac{0.092}{P^2} , \quad u = \frac{64}{P} \text{ m/sec} , \quad P \text{ in atmospheres} .$$

The scaling law for coolant temperature rise is

$$\frac{W}{Q} = 0.092 \left(\frac{10}{\Delta T_r} \right)^3 .$$

The scaling law for blower tip speed is

$$u_0 = 122 \left(\frac{W/Q}{0.092} \right)^{1/2} .$$

The scaling law for glass central temperature is

$$T_{\max} = 50 + \Delta T_r + 25r \frac{\Delta T_r}{10} + 23r .$$

And the scaling law for peak tensile stress in the glass is

$$\sigma_t/Q = 0.1 \text{ psi/watt} .$$

4. RESULTS

4.1 TABULATED SUMMARY

The above results are summarized in Table 4. Also summarized in the table are results for helium in the slant direction and nitrogen in both directions. These results are for a heat input of 16000 watts, one atmosphere gas pressure, and 10 degree coolant temperature rise from inlet to outlet.

As seen in the table, nitrogen is almost as good a coolant as helium in terms of pumping power. Also, because of its higher density, it is easier to design a blower to pump nitrogen. Peak glass temperature, however, tends to be about 10 degrees higher with nitrogen. This is because nitrogen is less effective in removing heat; therefore, boundary layer temperature differentials must be higher.

Regardless of coolant, pumping power is much less when the gas flow is in the short direction (normal to the optical path) than in the long direction (slant direction).

4.2 SCALING OF OTHER RESULTS

The four terms A,B,C,D at the bottom of Table 4 can be used to scale the results to other heat flows or repetition rates, gas pressures, flow channel width w, and coolant temperature rises by means of the following equations.

Effect of heat flow or repetition rate:

$$\frac{W}{Q} = A \left(\frac{Q}{16000} \right)^2 = Ar^2$$

Effect of gas pressure:

$$\frac{W}{Q} = \frac{A}{P^2}, \quad u = \frac{B}{P}, \quad P \text{ in atmospheres}$$

TABLE 4

SUMMARY OF RESULTS FOR $P = 1.0 \text{ ATM}$ AND $\Delta T_r = 10 \text{ C}$

	SHORT PATH		LONG PATH	
	Re	N ₂	Re	N ₂
Gas flow, kg/sec	0.308	1.60	0.308	1.60
m ³ /sec	2.57	1.60	2.57	1.60
Velocity over glass, m/sec	64	40	128	80
Reynolds number	6160	40000	12320	80000
Film coefficient, w/m ² -C	157	113	313	174
Boundary layer ΔT , C	25	35	13	23
Max glass temp, C	108	118	96	106
Gas pressure drop, N/m ²	190	368	1383	2688
Heat exchanger Reynolds No.	211	660	211	660
Overall HX film coeff, w/m ² -C	380	82	380	82
HX flow area, m ²	0.154	0.32	0.154	0.32
HX thickness, m	0.018	0.041	0.018	0.041
Gas pressure drop, N/m ²	59	39	59	39
Nominal gas vel in ducts, m/sec	32	20	64	40
Duct+entrance+exit ΔP , N/m ²	37	120	148	480
Total flow circuit ΔP , N/m ²	286	527	1590	3207
Pumping power ($\eta = 0.5$), watts	1468	1686	8156	10262
Pumping power fraction W/Q	0.092	0.105	0.510	0.641
No. blower stages	1	1	2	1
Blower tip speed, m/sec	122	57	203	142
Blower rpm	3600	1200	3600	2400
Outer diameter, m	0.65	0.91	1.08	1.13
Inner diameter, m	0.39	0.54	0.65	0.68
Thickness, m	0.14	0.13	0.05	0.05
Scale Factors:	A	0.092	0.105	0.510
(see text)	B	64	40	128
	C	122	57	203
	D	25	35	13

Effect of coolant temperature rise:

$$\frac{W}{Q} = A \left(\frac{10}{\Delta T_r} \right)^3$$

Effect of flow passage width:

$$\frac{W}{Q} = \frac{A}{w^3}$$

Effect on blower tip speed:

$$u_0 = C \left(\frac{W/Q}{A} \right)^{1/2}$$

Effect of rep rate r , coolant temperature rise ΔT_r ,
and flow channel width w on glass central temperature:

$$T_{\max} = 50 + \Delta T_r + Drw \frac{\Delta T_r}{10} + 23r, \quad w \text{ in cm}$$

Calculated results using the above scaling relations are plotted in Figures 6 through 11 for pumping power fraction and in Figure 12 for glass central temperature. A plot is not presented for blower tip speed because most overspeed conditions can be corrected by adding stages.

As seen in Figs. 6 to 11, decreasing pumping powers occur when one increases gas temperature rise, gas pressure, and flow channel width. The greatest gains occur when gas temperature rise is increased. Because boundary layer temperature rise follows the gas temperature rise, there is a limit to the latter that is dictated by the maximum allowable glass temperature, which is shown as a function of gas temperature rise and rep rate for a 1.0 cm flow channel. It may be possible to cancel out optical wedges by clever coolant path arrangements.

The short flow direction gives pumping powers about 6 times smaller than the long direction for the same gas temperature rise. As seen in Fig. 12, however, glass temperature are higher for a given gas temperature rise for the short flow direction. The limiting gas temperature rise is therefore more constrained in this case. There is a similar tradeoff with flow channel size. For the same gas temperature

Fig. 6. Effect of gas temperature rise and pressure, repetition rate, and flow direction on pumping power fraction W/Q . Helium gas, 1.0 cm flow channel

Fig. 7. Nitrogen gas. 1.0 cm flow channel.

Fig. 8. Helium gas. 2.0 cm flow channel.

Fig. 9. Nitrogen gas. 2.0 cm flow channel.

Fig. 10. Helium gas. 0.5 cm flow channel.

Fig. 11. Nitrogen gas. 0.5 cm flow channel.

Fig. 12. Glass central temperature as a function of coolant temperature rise and flashtube repetition rates. 1 and 2 cm flow channels.

rise, peak glass temperature increases with increasing channel width, as seen in Fig. 12, even though pumping power falls off.

Despite these tradeoffs, a detailed examination of the results in Figs. 6 to 12 reveal a number of parametric arrangements which will yield acceptable pumping powers. Presuming thermal stresses pose no problem, rep rates as high as 4 Hz can be adequately cooled at least in the short flow direction. An example favorable selection of parameters is shown in Table 5.

TABLE 5

EXAMPLE DESIGN PARAMETERS

Helium at one atmosphere
Short flow direction
2 cm flow channel width

	<u>1 Hz</u>	<u>2 Hz</u>	<u>4 Hz</u>
Gas temperature rise, C	5	8	8
Pumping power fraction	0.1	0.1	0.5
Glass central temperature, C	125	230	300
Glass thermal stress, psi	1600	3200	6400

5. CONCLUSIONS

Either helium or nitrogen can be used to remove the heat deposited by the flashtubes on the laser glass. Both can give acceptable pumping powers although, with nitrogen, glass temperatures are higher. Cooling can be accomplished at one atmosphere gas pressure. The short flow direction is by far preferable, as is a 2.0 cm flow channel width between glass plates. Gas temperature rises around 10 C are acceptable; higher values tend to give glass temperatures that may be too high. Provided glass temperatures near 300 C and thermal stresses around 6000 psi are acceptable, then flashtubes repetition rates as high as 4.0 Hz can be used.

REFERENCES

1. H.S. Carslaw and J.C. Jaeger, Conduction of Heat in Solids, second edition, Oxford:Clarendon Press, 1959.
2. B.A. Boley and J.H. Weiner, Theory of Thermal Stresses, John Wiley, 1967. See Eq. (9.5.4).
3. W.M. Kays and A.L. London, Compact Heat Exchangers, second edition, McGraw-Hill, 1964.