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An Analytical Study of IR Beampipe Cooling

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Abstract

A set of formulae useful for heat transfer analysis in general is presented and their reliablity and limitations are discussed. A particular emphasis is on the cooling analysis of the Belle IR beampipe. Coolants studied are water, PF200, methanol, He (1 atm and 2 atm) and nitrogen.

1 Modes of coolant flow

For a tube of various cross sectional shapes, the flow is expected to be laminar if the Reynolds number (Re) is less than about 2000∼3000, where the Reynolds number is defined by

$$Re = \frac{vD}{\nu} \tag{1}$$

where v (cm/s) is the bulk velocity of the fluid, D (cm) is the effective diameter (or sometimes called the hydraulic diameter) defined as

$$D \equiv 4\frac{\text{cross sectional area}}{\text{perimeter length}}\,, \tag{2}$$

and ν (cm^2/s) is the kinematic viscosity which is related to the dynamic viscosity μ (gr/cm·s ≡ poise) by

$$\nu = \frac{\mu}{\rho}\,, \tag{3}$$

where ρ (gr/cm^3) is the density of the fluid. The bulk velocity v is defined as the total volume flow divided by the cross sectional area of the tube. Sometimes in the literature, the Reynolds number is defined as half of the expression (1); namely, using the radius instead of the diameter D. For that definition, the transition from laminar to turbulent is in the region of 1000 to 1500.

The effective diameter for a tube with a rectangular cross section (a by b) is

$$D = \frac{2ab}{a+b} \tag{4}$$

which in the limit of $a \gg b$ becomes

$$D \sim 2b \quad (a \gg b)\,. \tag{5}$$

Namely, the effective diameter of a channel between two plates separated by a gap b is twice the gap size. For a tube with a circular cross section, the effective diameter is

$$D = 4\frac{\pi r^2}{2\pi r} = 2r \tag{6}$$

as expected.

The transition from laminar to turbulence depends on variety of parameters such as roughness of inner wall, smoothness of inlet, etc. and it could be abrupt or gradual. When the flow is in the transition region, the mode could oscillate between laminar and turbulent ('spouting'). This occurs because the pressure needed for turbulent flow is in general greater than that for laminar flow in the transition region. As one increases flow velocity by increasing pressure, the Reynolds number will increase and the mode will change from laminar to turbulent, then for the given pressure, the flow velocity will drop, the Reynolds number will drop accordingly and the mode will revert back to laminar which will then increase the flow velocity for the given pressure. Such spouting could stress the cooling structure and may also cause shifts of alignment. Operation in the transition region should thus be avoided. In any case, in order for the results (temperature rises, pressure drops, etc.) to be reliable in the transition region, the predictions for laminar case and those for turbulent case should be reasonably similar in the transition region.

When the coefficient of thermal expansion is large, the temperature gradient is steep, and the flow is laminar and placed in the gravitational field, the effect of convection could become important. When the flow is turbulent, the convection can be ignored. The parameter of interest is the Rayleigh number ($\mathcal{R}$) given by [1]

$$\mathcal{R} = \frac{\beta g h^3 \Delta T}{\nu \chi} \tag{7}$$

where ΔT (K) is the characteristic temperature difference, h (cm) is the characteristic height, β (1/K) is the thermal expansion coefficient, g (cm/s^2) is the gravitational acceleration, and χ (cm^2/s) is the thermometric conductance given by

$$\chi = \frac{k}{c_p \rho} \tag{8}$$

where k (W/cm·s) is the thermal conductivity and c_p (J/gr·K) is the equal-pressure specific heat. When the Rayleigh number is of order 1500 or greater, the convection becomes important. As the chracteristic temperature difference ΔT, we will take the bulk to wall temperature difference, and the gap size as the characteristic height h. For $\Delta T = 10$ K, $h = 0.1$ cm, and for water at 20° C, we obtain $\mathcal{R} \sim 300$. The effect of convection is in general not important for our application.

2 Pressure drops

It takes some pressure to accelerate a volume of fluid to a certain bulk velocity. Such pressure is called the dynamic head (ΔP_d) which is given by

$$\Delta P_d = \frac{1}{2}\rho v^2 \quad (\mathrm{dyne/cm^2})\,. \tag{9}$$

The velocity v (cm/s^2) is the bulk velocity as before. Since the actual flow velocity is not uniform across the cross section, this expression is clearly an approximation. For a laminar flow, the velocity in a tube has a parabolic form, and the expression above theoretically underestimates by factor of two. Also, if the flow is supplied to the cooling region with some velocity to begin with, the pressure needed to accelerate the fluid would be less than otherwise. If the flow is turbulent, the pressure needed to accelerate the fluid from a static reservoir would be well approximated by the above expression. The pressure may be converted to psi (pound per square inch) by

$$1 \text{ ps}i = 68950 \text{ dyne/cm}^2 = 6895 \text{ Pascal} \, (\equiv \text{Newton/m}^2\,) \,. \tag{10}$$

We will assume that the dynamic head is given by (9) for both laminar and turbulent flows keeping in mind that the actual number highly depends on each circumpstance.

The pressure drop due to viscosity (ΔP_v) depends on the type of flow. For a laminar flow between two plates, it is given by

$$\Delta P_v = v \frac{12\mu L}{b^2} \quad \text{(laminar, two plates)} \,, \tag{11}$$

where b is the gap size between the two plates, and for a tube of inner radius r it is estimated to be [2]

$$\Delta P_v = v \frac{8\mu L}{r^2} \quad \text{(laminar, round tube)} \,. \tag{12}$$

For both cross sectional shapes, L is the length of the flow. When the fluid enters the channel, it takes a while before the laminar flow is fully developed. The transition length is approximately given by [2]

$$L_{\text{trans}} \sim 0.13 D Re \,. \tag{13}$$

For $D = 1$ mm and $Re = 1000$ this gives $L_{\text{trans}} \sim 13$ cm. While the laminar flow is still developing, the velocity gradient near the wall is steeper than that for a fully developed laminar flow. This results in a greater drag force. Thus, the pressure drops calculated by the laminar formulae above will be smaller than correct values.

For a turbulent flow, the pressure drop of a channel due to viscosity is given by

$$\Delta P_v = \lambda \frac{L}{D} \frac{1}{2} \rho v^2 \,, \tag{14}$$

where the 'friction coefficient' λ is given by [4]

$$\lambda = 0.2 \, Re^{-0.2} \,. \tag{15}$$

The classical form for λ due to Blausius is [2, 3]

$$\lambda = 0.316 \, Re^{-0.25} \,. \tag{16}$$

According to Ref [2], a better expression that works up to $Re \sim 230000$ reads

$$\lambda = 0.00714 + 0.6104 \, Re^{-0.35} \,. \tag{17}$$

All three forms for λ give an almost identical value at $Re \sim 10000$, and agree within 20% up to $Re \sim 200000$. They work reasonably well regardless of the shape of the tube cross section. We will use (17) in our analysis. Total pressure drop is then given by

$$\Delta P = \Delta P_d + \Delta P_v \,. \tag{18}$$

3 Heat Transfer

The heat flow between a bulk fluid and a solid wall can be expressed in terms of the heat transfer coefficient h (W/cm^2·K) defined by

$$Q = hA\Delta T\,, \tag{19}$$

where Q (W) is the heat flow, A (cm^2) is the area where the fluid is in contact with the solid wall, and ΔT is the difference between the wall temperature and the bulk temperature of the fluid (namely, the temperature of the fluid averaged over its entire flow volume corresponding to the area under consideration). Often, the heat transfer coefficient is written in terms of a dimensionless quantity (the Nusselt number) as

$$Nu = \frac{hD}{k}\,, \tag{20}$$

where D is the effective diameter of the cooling tube as before.

For a fully developed laminar flow with uniform viscosity and thermal conductance, the Nusselt number becomes a function only of the cross sectional shape of the cooling tube. For example,

$$Nu(\text{ideal}) = \begin{cases} 4.26 & \text{(circle)} \\ 3.6 & \text{(square)} \\ 8.2 & \text{(parallel plates)} \end{cases} \qquad \text{(ideal laminar case)}\,. \tag{21}$$

In our application, the cooling channel is essentially double parallel plates, but the heat is applied from only one side. The above Nusselt number for parallel plates is for the case where two plates are equally heated, and thus cannot in principle be used for our case. The heat flux is determined by the local temperature gradient of the fluid next to the wall. Then, the Nusselt number above needs to be multiplied by the ratio of bulk temperature (with respect to the wall temperature) for the both-side heating case to that for the one-side heating case where the temperature gradient next to the heated wall is kept the same. This results in a slightly smaller heat transfer coefficient than calculated by using $Nu = 8.2$. However, the difference is within a few 10's of % and the error in the entrance effect which is discussed below is likely to dominate the overall uncertainty in the heat transfer coefficient, so we will adopt $Nu = 8.2$ for this analysis.

If the laminar flow is not fully developed, the narrower laminar layer results in a greater heat transfer coefficient. The correction factor for the entrance effect can be parametrized as [4]

$$Nu = f(q)\, Nu(\text{ideal}) \quad \text{(laminar)}\,, \tag{22}$$

where

$$f(q) = 0.0544\, q^{-0534} + 0.93 \quad \text{(laminar)}\,, \tag{23}$$

with

$$q = \frac{x}{D} \frac{1}{Pr\, Re}, \tag{24}$$

where x is the distance from the entrance and Pr is the Prandtl number defined by

$$Pr \equiv \frac{c_p \mu}{k}. \tag{25}$$

As x, we will take the full length of the cooling channel since what we are interested in is the maximum temperature of the wall: toward the end of the channel, the bulk fluid temperature will be the highest and the bulk to wall temperature difference will also be the highest due to the smaller heat transfer coefficient.

For a turbulent flow, the Nusselt number is well approximated by [3, 4]

$$Nu = 0.023\, Pr^{0.4} Re^{0.8} \quad \text{(turbulent)}. \tag{26}$$

This form is known to work quite well for variety of cross sectional shapes as long as the effective diameter is used everywhere D appears. The entrance effect is not important for a turbulent flow as long as L/D is greater than about 20 for which the error in h is typically less than $\sim$20%.

Thus, the procedure to estimate the wall-to-bulk temperature difference ΔT_{w-b} is to first obtain the Nusselt number by (22) for a laminar flow or by (26) for a turbulent flow, convert it to the heat transfer coefficient h by (20), and then use the total area in contact with the heat as A in (19) to obtain ΔT_{w-b}.

4 Gas coolants

The formulas presented above can be used for gases when the bulk velocity is much smaller than the speed of sound c given by

$$c = \sqrt{\frac{c_p}{c_v} \frac{P}{\rho}}, \tag{27}$$

where P is the pressure of the gas and c_p/c_v is the adiabatic constant. The value of c_p/c_v is about 1.4 for air and 1.66 for He at room temperature.

Sometimes gas coolants are used at higher pressures than the atmospheric pressure. The dynamic viscosity μ does not depend on the pressure to the first order, while the kinematic viscosity ν is inversely proportional to the pressure (for a given temperature). The thermal conductivity is also independent of pressure to a good approximation. Since P/ρ is constant for a given temperature, the speed of sound also is independent of pressure; it does, however, depend on temperature as $\propto \sqrt{T}$.

	water	methanol	PF200	Helium	Nitrogen
ρ (gr/cm^3)	0.998	0.791	0.780	1.78×10^{-4}	1.25×10^{-3}
μ (gr/cm·s)	0.010	0.0054	0.019	1.94×10^{-4}	1.79×10^{-4}
ν (cm^2/s)	0.010	0.0061	0.024	1.09	0.143
k (W/cm·K)	0.0062	0.0021	0.0016	0.00148	0.00026
c_p (J/gr·K)	4.200	2.570	2.300	5.230	1.041
Pr	6.77	5.90	27.20	0.686	0.717
X_0 (cm)	36.1	49.7	52.0	5.28×10^5	3.04×10^4

Table 1: Properties of various coolants. Parameters listed are: ρ (density), μ (dynamic viscosity), ν (kinematic viscosity), k (thermal conductivity), c_p (specific heat), Pr (Prandtl number), and X_0 (radiation length).

Gas coolants often require a large volume flow rate. As a result, the energy dissipated in the cooling channel could become substantial. If the fluid is incompressible, the energy dissipated $E_{\rm diss}$ (W) is given by

$$E_{\rm diss} = \Delta P_v \cdot dV/dt \tag{28}$$

where the pressure drop due to viscosity ΔP_v is in Pascal (Newton/m^2) and the volume flow rate dV/dt is in m^3/s. Some of this energy will be transmitted to the wall and some will be used to heat the coolant. For a compressible fluid, the above expression for the dissipated energy does not apply, but probably it will give an order of magnitude estimate.

5 Analysis of various coolants

As a coolant, we consider water, methanol, PF200 [5], Helium and Nitrogen. Relevant parameters of the coolants are listed in Table 7. The geometry of the beampipe is taken to be radius 2 cm and length 10 cm. The coolant gap is centered around $r = 2$ cm and is 0.5 mm for liquids and 2 mm for gases. The heat deposit is assumed to be 200 W uniformly distributed over the inner surface of the beampipe. For now, we will assume that there is no additional cooling such as the conduction through the ends of Be beampipe. We will come back to this issue later.

The results are given in Tables 2-7. In each table, volume flow rate (dV/dt), Reynolds number (Re), flow mode (laminar 'L' or turbulent 'T'), temperature rise of bulk fluid (ΔT_b), dynamic head (ΔP_d), pressure drop due to viscosity (ΔP_v), heat transfer coefficient (h: W/cm^2·K), temperature difference between the wall and bulk fluid (ΔT_{w-b}), and dissipated energy ($E_{\rm diss}$) are given for different values of flow velocity v. The bulk temperature rise ΔT_b is calculated simply by assuming that the total heat input (200 W) is absorbed by the coolant. We assume that the transition from laminar to turbulent occurs at $Re = 2500$. For heat deposits different from 200 W, we simply note that ΔT_b and ΔT_{w-b} scales linearly with the heat deposit.

Comparing the tables for the three liquid coolants, we see that water provides an excellent cooling capacity. Since the radiation length of water is about the same as that of Be (36.1 cm for water, 35.3 cm for Be), with two layers of Be 0.5 mm, 0.5 mm of water contributes 1/3 of the total radiation length assuming that there is no additional beampipe material such as heavy metal coatings. With total pressure $\Delta P_d + \Delta P_v$ of 0.77 psi, one obtains an wall-to-bulk temperature difference of 2.1 K and a bulk temperature rise of 0.7 K. The Reynolds number is about 1000 indicating that it is a stable laminar flow. A drawback of water is that it can easily corrode the Be beampipe if care is not taken. For the CLEO 2.5 beampipe, a double coating of BR127 (a single component epoxy) was applied to the inside of the cooling channel. No problem was noticed during its lifetime of operation.

With PF200, one can achieve overall temperature rise of 6-7 K for the inner Be layer with about 2 psi of pressure. Higher values of pressure and temperature rise are mainly due to its higher viscosity and smaller thermal conductance compared to those of water. The temperature rise of the outer Be layer is about the same as the bulk temperature rise which is about 1-1.5 K. The flow is well within laminar region due to its high viscosity. PF200 have been used to cool the drift chamber, the IR masks, etc. of CLEO detector, and scheduled to be used for the CLEO III beampipe [6]. Its compatibility with Be has been tested by immersing a Be coupon in PF200 for about two years: so far no corrosion problem has been noted. PF200 is similar to kerosene and is flammable: Its auto-ignition point is 210°C and its flash point is 93°C.

An advantage of methanol is its low viscosity as reflected in the low values of pressure needed to push the fluid though the channel. Its specific heat and thermal conductivity are quite good and with 1.25 psi of total pressure, one can obtain $\Delta T_b = 0.75$ K, and $\Delta T_{w-b} \sim 2.5$ K. Disadvantages are that it is flammable and its boiling point is low (64° C). Also, it has not been tested for Be corrosion as far as I know. One could operate it at even higher flow rate than shown in the table. It would also put it well within the turbulent range which helps to avoid spouting.

For Helium at 1 atm flowing at 300 m/s, we see that the bulk temperature rise is 2.7 K, the wall-to-bulk is 12 K with the total pressure of about 2 psi. Even though this may be acceptable, one should keep in mind that 300 m/s is a substantial fraction of the speed of sound (about Mach 0.3), and thus the calculation may not be reliable. Also, the dissipative heating at the operating point is about 400 W which is twice as large as the heat it is designed to remove. Even though the outer Be temperature is quite manageable, we would like to keep the inner Be temperature reasonably low so that it will not cause alignment problems and thermal fatigue. When the operating pressure is raised to 2 atm, the situation is better but not by much. Similar comments applie to Nitrogen where the wall to bulk temperature difference is even larger due to the small thermal conductivity. It appears that gas cooling is not suited for a heat deposit that is as high as 200 W or more.

6 Other miscellaneous heat transfers

Conduction though the end of Be beampipe can be roughly estimated as follows: The heat flow when there is an temperature gradient of $\Delta T/\Delta x$ is given by

$$Q = A\,k\,\frac{\Delta T}{\Delta x}\,, \tag{29}$$

where A is the cross sectional area. Assuming that the thickness of Be is 0.5 mm for each layer (namely, $A = 0.628$ cm^2) and that the heat flows over about $\Delta x = 3$ cm length of beampipe (with $k = 2.0$ W/cm·K for Be), we have

$$Q(W) \sim 0.4\Delta T(K) \tag{30}$$

for each layer of Be, where the number $\Delta x = 3$ cm is a pure dead-reckoning. Thus, if $\Delta T_b = 2$ K and $\Delta T_{w-b} = 10$ K, then the inner Be would be 12 K, and the outer would be 2 K, and thus total heat escaping though the ends is $0.4 \times (12 + 2) \sim 5.6$ W which should be compared to the input heat of 200 W. The amount of heat escaping from the ends becomes important when the cooling is weak. It is usually small enough to be negligible for a liquid cooling but important for a gas cooling at low flow rates.

Heat loss due to radiation is given by

$$\frac{Q}{A} = \epsilon\,\sigma\,T^4 \tag{31}$$

where Q/A (W/cm^2) is the heat flux emitted from the surface, ϵ is the emissivity of the surface, T is in Kelvin, and

$$\sigma = 5.67 \times 10^{-12} \ \ \mathrm{W/cm^2{\cdot}K^4} \tag{32}$$

is the Stephan-Boltzman constant. The unit emissivity corresponds to the ideal black body, and the actual emissivity is highly dependent on the condition of the surface, changing from a few % for polished metal surfaces to ~90% for black oxidized steel surfaces. For $T = 300$ K (27° C), the radiation heat flux is

$$\frac{Q}{A} = \epsilon \times 0.046\ (\mathrm{W/cm^2})\,. \tag{33}$$

Taking the emissivity of the outer Be surface to be 0.25 (probably good to factor of two), and using $A = 126$ cm^2, the radiation flux amounts to 1.5 W. We have to also include the radiation absorbed by the surface in order to make a better estimation of net radiation loss. If we assume that the outer Be surface and the environment was in equilibrium at T_0 and the Be surface temperature is then raised to T, the net radiation loss is (assuming that there is no extra reflection back from the environment)

$$\frac{Q}{A}(\text{net}) = \epsilon\,\sigma\,(T^4 - T_0^4) \tag{34}$$

which gives 0.45 W for $T_0 = 300$ K and $T = 320$ K. The heat loss due to radiation is usually negligible for our application.

References

[1] L. D. Landau and E. M. Lifshitz, 'Fluid Mechanics' 2nd edition, Pergamon press (1987).

[2] L. Prandtl and O. G. Tietjens, , 'Applied Hydro- and Aerodynamics', Dover (1934).

[3] J. P. Holman, 'Heat Transfer' 7th edition, McGraw Hill (1990).

[4] A. P. Fraas, 'Heat Exchanger Design' 2nd edition, John Wiley and sons (1989).

[5] PF^{TM}-200 IG is manufactured by P-T Technologies inc. Its properties are similar to those of kerosene.

[6] D. Cinabro and S. McGee, CLEO internal memo CBX 98-8 (1998).

v (cm/s)	dV/dt (cc/s)	Re		$f(q)$	ΔT_b (K)	ΔP_d (psi)	ΔP_v (psi)	h	ΔT_{w-b} (K)	E_{diss} (W)
10.	7.	100.	L	1.08	7.23	0.00	0.07	0.550	2.76	0.
20.	13.	200.	L	1.15	3.62	0.00	0.14	0.584	2.60	0.
30.	20.	299.	L	1.20	2.41	0.01	0.21	0.611	2.48	0.
40.	26.	399.	L	1.25	1.81	0.01	0.28	0.634	2.39	0.
50.	33.	499.	L	1.29	1.45	0.02	0.35	0.654	2.32	0.
60.	40.	599.	L	1.32	1.21	0.03	0.42	0.673	2.25	0.
70.	46.	699.	L	1.36	1.03	0.04	0.49	0.690	2.20	0.
80.	53.	798.	L	1.39	0.90	0.05	0.56	0.706	2.15	0.
90.	59.	898.	L	1.42	0.80	0.06	0.63	0.721	2.10	0.
100.	66.	998.	L	1.45	0.72	0.07	0.70	0.735	2.06	0.
110.	73.	1098.	L	1.47	0.66	0.09	0.77	0.749	2.02	0.
120.	79.	1198.	L	1.50	0.60	0.10	0.84	0.762	1.99	0.
130.	86.	1297.	L	1.52	0.56	0.12	0.91	0.775	1.96	1.
140.	92.	1397.	L	1.55	0.52	0.14	0.97	0.787	1.93	1.
150.	99.	1497.	L	1.57	0.48	0.16	1.04	0.799	1.90	1.
160.	106.	1597.	L	1.59	0.45	0.19	1.11	0.810	1.87	1.
170.	112.	1697.	L	1.62	0.43	0.21	1.18	0.821	1.85	1.
180.	119.	1796.	L	1.64	0.40	0.23	1.25	0.832	1.82	1.
190.	125.	1896.	L	1.66	0.38	0.26	1.32	0.843	1.80	1.
200.	132.	1996.	L	1.68	0.36	0.29	1.39	0.853	1.78	1.

Table 2: Water, gap = 0.5 mm, beampipe radius = 2 cm, beampipe length 10 cm. Heat is 200 W uniform over the inside of the beampipe. From the left, v is the bulk flow velocity, dV/dt is the volume flow rate, Re is the Reynolds number, 'L' for laminar and 'T' for turbulent, $f(q)$ is the correction factor for the entrance effect (used only if the flow is laminar), ΔT_b is the bulk temperature rise, ΔP_d is the dynamic head, ΔP_v is the pressure drop due to viscosity, h (W/cm^2·K) is the heat transfer coefficient, ΔT_{w-b} is the maximum wall to bulk temperature difference, and E_{diss} is the dissipated energy.

v (cm/s)	dV/dt (cc/s)	Re		$f(q)$	ΔT_b (K)	ΔP_d (psi)	ΔP_v (psi)	h	ΔT_{b-w} (K)	E_{diss} (W)
10.	7.	41.	L	1.13	16.90	0.00	0.13	0.148	10.24	0.
20.	13.	82.	L	1.22	8.45	0.00	0.26	0.160	9.50	0.
30.	20.	123.	L	1.29	5.63	0.01	0.40	0.169	8.99	0.
40.	26.	164.	L	1.34	4.22	0.01	0.53	0.176	8.59	0.
50.	33.	205.	L	1.40	3.38	0.01	0.66	0.183	8.27	0.
60.	40.	246.	L	1.44	2.82	0.02	0.79	0.190	8.00	0.
70.	46.	287.	L	1.49	2.41	0.03	0.93	0.195	7.76	0.
80.	53.	328.	L	1.53	2.11	0.04	1.06	0.201	7.55	0.
90.	59.	369.	L	1.57	1.88	0.05	1.19	0.206	7.36	0.
100.	66.	411.	L	1.61	1.69	0.06	1.32	0.211	7.19	1.
110.	73.	452.	L	1.64	1.54	0.07	1.45	0.215	7.04	1.
120.	79.	493.	L	1.68	1.41	0.08	1.59	0.220	6.90	1.
130.	86.	534.	L	1.71	1.30	0.10	1.72	0.224	6.76	1.
140.	92.	575.	L	1.74	1.21	0.11	1.85	0.228	6.64	1.
150.	99.	616.	L	1.77	1.13	0.13	1.98	0.232	6.53	1.
160.	106.	657.	L	1.80	1.06	0.14	2.12	0.236	6.42	2.
170.	112.	698.	L	1.83	0.99	0.16	2.25	0.240	6.32	2.
180.	119.	739.	L	1.86	0.94	0.18	2.38	0.243	6.23	2.
190.	125.	780.	L	1.88	0.89	0.20	2.51	0.247	6.14	2.
200.	132.	821.	L	1.91	0.84	0.23	2.65	0.250	6.05	2.

Table 3: PF200, gap = 0.5 mm, 200 W. See Table 2 for explanations.

v (cm/s)	dV/dt (cc/s)	Re		$f(q)$	ΔT_b (K)	ΔP_d (psi)	ΔP_v (psi)	h	ΔT_{b-w} (K)	E_{diss} (W)
10.	7.	147.	L	1.11	14.90	0.00	0.04	0.192	7.91	0.
20.	13.	293.	L	1.19	7.45	0.00	0.08	0.206	7.37	0.
30.	20.	440.	L	1.26	4.97	0.01	0.11	0.217	6.99	0.
40.	26.	586.	L	1.31	3.73	0.01	0.15	0.226	6.70	0.
50.	33.	733.	L	1.36	2.98	0.01	0.19	0.235	6.46	0.
60.	40.	879.	L	1.41	2.48	0.02	0.23	0.242	6.26	0.
70.	46.	1026.	L	1.45	2.13	0.03	0.26	0.249	6.08	0.
80.	53.	1172.	L	1.49	1.86	0.04	0.30	0.256	5.93	0.
90.	59.	1319.	L	1.52	1.66	0.05	0.34	0.262	5.79	0.
100.	66.	1466.	L	1.56	1.49	0.06	0.38	0.268	5.66	0.
110.	73.	1612.	L	1.59	1.35	0.07	0.41	0.273	5.54	0.
120.	79.	1759.	L	1.62	1.24	0.08	0.45	0.279	5.44	0.
130.	86.	1905.	L	1.65	1.15	0.10	0.49	0.284	5.34	0.
140.	92.	2052.	L	1.68	1.06	0.11	0.53	0.289	5.24	0.
150.	99.	2198.	L	1.71	0.99	0.13	0.56	0.294	5.16	0.
160.	106.	2345.	L	1.73	0.93	0.15	0.60	0.299	5.08	0.
170.	112.	2491.	L	1.76	0.88	0.17	0.64	0.303	5.00	0.
180.	119.	2638.	T	1.79	0.83	0.19	0.85	0.561	2.70	1.
190.	125.	2785.	T	1.81	0.78	0.21	0.94	0.586	2.59	1.
200.	132.	2931.	T	1.84	0.75	0.23	1.02	0.610	2.48	1.

Table 4: Methanol, gap = 0.5 mm, 200 W. See Table 2 for explanations.

v (cm/s)	dV/dt (cc/s)	Re		$f(q)$	ΔT_b (K)	ΔP_d (psi)	ΔP_v (psi)	h	ΔT_{b-w} (K)	E_{diss} (W)
2000.	5278.	736.	L	1.20	40.57	0.01	0.02	0.036	41.61	1.
4000.	10556.	1473.	L	1.32	20.28	0.02	0.03	0.040	37.79	2.
6000.	15834.	2209.	L	1.42	13.52	0.05	0.05	0.043	35.26	6.
8000.	21112.	2946.	T	1.50	10.14	0.08	0.09	0.044	34.75	13.
10000.	26389.	3682.	T	1.57	8.11	0.13	0.13	0.052	29.07	25.
12000.	31667.	4419.	T	1.63	6.76	0.19	0.18	0.060	25.12	40.
14000.	36945.	5155.	T	1.70	5.80	0.25	0.24	0.068	22.21	61.
16000.	42223.	5892.	T	1.75	5.07	0.33	0.30	0.076	19.96	88.
18000.	47501.	6628.	T	1.81	4.51	0.42	0.37	0.083	18.16	121.
20000.	52779.	7365.	T	1.86	4.06	0.52	0.44	0.091	16.69	161.
22000.	58057.	8101.	T	1.90	3.69	0.63	0.52	0.098	15.47	209.
24000.	63335.	8838.	T	1.95	3.38	0.75	0.61	0.105	14.43	265.
26000.	68612.	9574.	T	2.00	3.12	0.88	0.70	0.112	13.53	329.
28000.	73890.	10311.	T	2.04	2.90	1.02	0.79	0.119	12.75	403.
30000.	79168.	11047.	T	2.08	2.70	1.17	0.89	0.126	12.07	487.
32000.	84446.	11784.	T	2.12	2.54	1.33	1.00	0.132	11.46	581.
34000.	89724.	12520.	T	2.16	2.39	1.50	1.11	0.139	10.92	685.
36000.	95002.	13257.	T	2.20	2.25	1.68	1.22	0.145	10.43	801.
38000.	100280.	13993.	T	2.23	2.14	1.87	1.34	0.152	9.99	929.
40000.	105558.	14730.	T	2.27	2.03	2.07	1.47	0.158	9.59	1069.

Table 5: Helium at 1 atm, gap = 2 mm, 200 W. See Table 2 for explanations.

v (cm/s)	dV/dt (cc/s)	Re		$f(q)$	ΔT_b (K)	ΔP_d (psi)	ΔP_v (psi)	h	ΔT_{b-w} (K)	E_{diss} (W)
2000.	5278.	1473.	L	1.32	20.28	0.01	0.02	0.040	37.79	1.
4000.	10556.	2946.	T	1.50	10.14	0.04	0.05	0.044	34.75	3.
6000.	15834.	4419.	T	1.63	6.76	0.09	0.09	0.060	25.12	10.
8000.	21112.	5892.	T	1.75	5.07	0.17	0.15	0.076	19.96	22.
10000.	26389.	7365.	T	1.86	4.06	0.26	0.22	0.091	16.69	40.
12000.	31667.	8838.	T	1.95	3.38	0.37	0.30	0.105	14.43	66.
14000.	36945.	10311.	T	2.04	2.90	0.51	0.40	0.119	12.75	101.
16000.	42223.	11784.	T	2.12	2.54	0.66	0.50	0.132	11.46	145.
18000.	47501.	13257.	T	2.20	2.25	0.84	0.61	0.145	10.43	200.
20000.	52779.	14730.	T	2.27	2.03	1.04	0.73	0.158	9.59	267.
22000.	58057.	16203.	T	2.34	1.84	1.25	0.87	0.171	8.88	347.
24000.	63335.	17676.	T	2.41	1.69	1.49	1.01	0.183	8.29	441.
26000.	68612.	19149.	T	2.47	1.56	1.75	1.16	0.195	7.77	549.
28000.	73890.	20622.	T	2.53	1.45	2.03	1.32	0.207	7.33	673.
30000.	79168.	22095.	T	2.59	1.35	2.33	1.49	0.219	6.93	813.
32000.	84446.	23568.	T	2.65	1.27	2.65	1.67	0.230	6.58	971.
34000.	89724.	25041.	T	2.71	1.19	2.99	1.85	0.242	6.27	1147.
36000.	95002.	26514.	T	2.77	1.13	3.36	2.05	0.253	5.99	1342.
38000.	100280.	27987.	T	2.82	1.07	3.74	2.25	0.264	5.74	1558.
40000.	105558.	29460.	T	2.87	1.01	4.14	2.46	0.275	5.51	1794.

Table 6: Helium at 2 atm, gap = 2 mm, 200 W. See Table 2 for explanations.

v (cm/s)	dV/dt (cc/s)	Re		$f(q)$	ΔT_b (K)	ΔP_d (psi)	ΔP_v (psi)	h	ΔT_{b-w} (K)	$E_{\rm diss}$ (W)
2000.	5278.	5587.	T	1.75	29.12	0.04	0.03	0.013	116.45	1.
4000.	10556.	11173.	T	2.11	14.56	0.15	0.11	0.023	66.88	8.
6000.	15834.	16760.	T	2.40	9.71	0.33	0.23	0.031	48.36	25.
8000.	21112.	22346.	T	2.65	7.28	0.58	0.39	0.039	38.41	57.
10000.	26389.	27933.	T	2.86	5.82	0.91	0.58	0.047	32.13	106.
12000.	31667.	33520.	T	3.06	4.85	1.31	0.81	0.055	27.77	177.
14000.	36945.	39106.	T	3.24	4.16	1.78	1.07	0.062	24.55	273.
16000.	42223.	44693.	T	3.41	3.64	2.32	1.36	0.069	22.06	397.
18000.	47501.	50279.	T	3.57	3.24	2.94	1.68	0.075	20.08	552.
20000.	52779.	55866.	T	3.73	2.91	3.63	2.04	0.082	18.46	741.
22000.	58057.	61453.	T	3.87	2.65	4.39	2.42	0.089	17.10	968.
24000.	63335.	67039.	T	4.01	2.43	5.22	2.83	0.095	15.95	1235.
26000.	68612.	72626.	T	4.15	2.24	6.13	3.27	0.101	14.96	1545.
28000.	73890.	78212.	T	4.28	2.08	7.11	3.73	0.107	14.10	1902.
30000.	79168.	83799.	T	4.40	1.94	8.16	4.23	0.114	13.34	2307.
32000.	84446.	89385.	T	4.53	1.82	9.28	4.75	0.120	12.67	2764.
34000.	89724.	94972.	T	4.64	1.71	10.48	5.29	0.126	12.07	3275.
36000.	95002.	100559.	T	4.76	1.62	11.75	5.87	0.131	11.53	3843.
38000.	100280.	106145.	T	4.87	1.53	13.09	6.47	0.137	11.04	4471.
40000.	105558.	111732.	T	4.98	1.46	14.50	7.09	0.143	10.60	5162.

Table 7: Nitrogen at 1 atm, gap = 2 mm, 200 W. See Table 2 for explanations.