By Luke Hinkle, *Contributing Editor*

VT&C Guides to Vacuum Technology

Heat Transfer in Vacuum

PART I: How Pressure Affects Your System's Thermal Conduction

Many of us from time to time still envision heat as a fluid. In fact, a few hundred years ago it was the accepted model -- heat *was* a fluid -- and the lexicon has persisted in expressions like "heat flow" and "heat reservoir" that we still use today. But the fluid model, while intuitively satisfying for some of us, falls short even as an analogy when considering the details of the various heat transfer mechanisms occurring in a vacuum system. As we investigate how each of the basic transfer mechanisms is affected by pressure and gas type it will be clear that understanding the principles of heat transfer is necessary for solving thermal challenges in a vacuum environment.

Heat transfer mechanisms are critical design factors in a vacuum processing system. Careful consideration for gas pressure effects must be exercised when designing for heating and cooling of system components and the process product. Common situations include the delivery of thermal energy to a substrate (for example, to raise the temperature of a wafer in a CVD reactor or annealing chamber) or the removal of excess heat from a substrate (for example, backside helium cooling in a wafer chuck). Executing an effective design involves both understanding the various mechanisms and exploiting the benefits of each. The three basic mechanisms for heat transfer are:

1) Conduction – the transfer of heat through motionless matter
2) Convection – the transfer of heat carried by moving matter
3) Radiation – the transfer of heat by electromagnetic waves

In this multipart series, a brief overview of each mechanism including how pressure and gas type influence its characteristics will be followed by practical examples demonstrating how thermal design for a vacuum system is often very different from that for ambient conditions. In Part I of this series, thermal conduction will be the focus.

Basics of Thermal Conduction

Conduction is the transfer of heat within motionless matter by the direct communication between molecules. We often associate heat conduction with solids, as when you solder a fitting on one end of a copper pipe and the rest of the pipe is soon warm to the touch. Perhaps the association of conduction and solids is because solids (opaque ones) can only transfer heat using conduction. However, conduction is not exclusive to solids; liquids and gases also conduct heat and conduction cannot be overlooked as a significant mechanism in many situations involving those states of matter too.

The model for conduction relates the amount of heat transferred through a material to the temperature gradient (temperature difference per length) in the material. Heat transfer is also proportional to a property of the material, the thermal conductivity. A simple diagram to illustrate the principles of conduction is shown in **Figure 1.** In this arrangement, two parallel surfaces each having area A are separated by a distance d and held at specified temperatures T_1 and T_2. The space between them is filled with the test material and in this ideal scenario no heat is transferred through any other path. The heat transferred from plate 1 to plate 2 in a steady state condition is given by,

$$\dot{Q}_h = \frac{kA(T_1 - T_2)}{d} \qquad \textbf{Equation 1}$$

where k is the thermal conductivity for the test material.

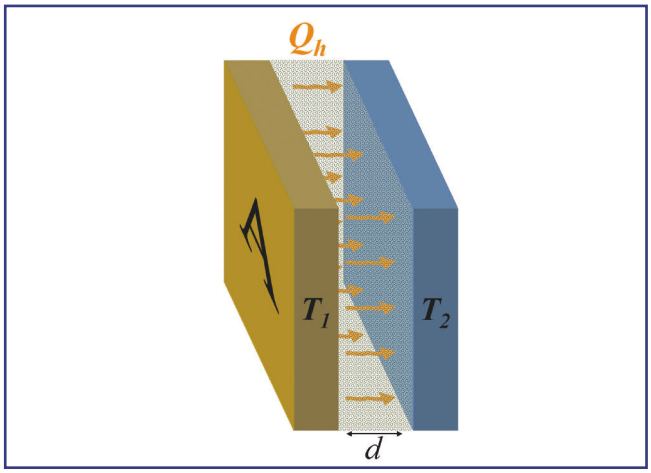

Figure 1. The parallel plate model for thermal conduction.

Thermal conductivity values for a selection of materials is listed in **Table 1.** It's important to note that the "from plate 1 to plate 2"directionality and the order of the temperature difference in the equation embody the concept that heat is transferred from higher temperature to lower temperature. And while the diagram implies that plate 1 is hotter than plate 2, Equation 1 also handles the reverse case by producing a negative value for heat transfer, indicating that heat is transferred from plate 2 to plate 1.

Pressure Dependence of Thermal Conduction

So it seems like conduction is a relatively simple concept. But before we get too comfortable with this heat transfer mechanism, we have to deal with the fact that the material's thermal conductivity is not necessarily a constant. The values listed in **Table 1** are for room temperature and atmospheric pressure conditions, but conductivity depends on temperature for all materials and for gases it additionally has significant pressure dependence. For many engineering situations the temperature dependence can be ignored without consequence. And for most applications at atmospheric (or higher) pressure the conductivity of gases is essentially constant, meaning that the pressure dependence can be overlooked. But we vacuum technologists cannot safely assume this as we often deal with low pressure conditions. So let's dig a little deeper into gas conductivity.

To be precise, the thermal conductivity of gases is not directly pressure dependent; it is Knudsen number dependent. The Knudsen number (the molecular mean free path divided by the characteristic dimension) certainly depends on pressure, but it also depends on the length scale of the problem and the gas type. A simple formula for the Knudsen number at room temperature is:

$$Kn = \frac{\beta}{Pd} \qquad \textbf{Equation 2}$$

Where β is a gas type dependent factor (listed for several gases in **Table 2**), P is the pressure and d is a characteristic dimension (intentionally chosen to be the same symbol as the distance between the parallel plates). How does gas conductivity depend on Knudsen number? Its behavior is best characterized within the gas regimes commonly used in vacuum technology: viscous, transition, and molecular. In the viscous regime where $Kn < 0.01$, gas conductivity is nearly constant; in the molecular regime where $Kn > 1$, it is linear with pressure; and in the transition regime where Kn is between 0.01 and 1, well, it gradually transitions from one to the other. This is shown in **Figure 2**.

Now we'll pull this information together and use a practical method to estimate the thermal conductivity for a particular case of gas type and characteristic dimension. It involves simply rescaling the graph in **Figure 2** using the following procedure:

1) Set the vertical scale of **Figure 2** by labeling the point where the horizontal dashed line meets the axis with the value for thermal conductivity at atmospheric pressure (from **Table 1**).

Table 1.

Material	Thermal Conductivity Room Temperature, Atmospheric Pressure Condition W/(m·°C)	Reference
Aluminum	250	1
Stainless Steel	16	1
Glass	1.0	1
Mercury	8	1
Water	0.58	1
Alcohol	0.17	1
Helium	0.14	2
Hydrogen	0.17	2
Argon	0.017	2
Air or Nitrogen	0.024	2
Carbon Dioxide (CO_2)	0.015	2
Ethane (C_2H_6)	0.017	2

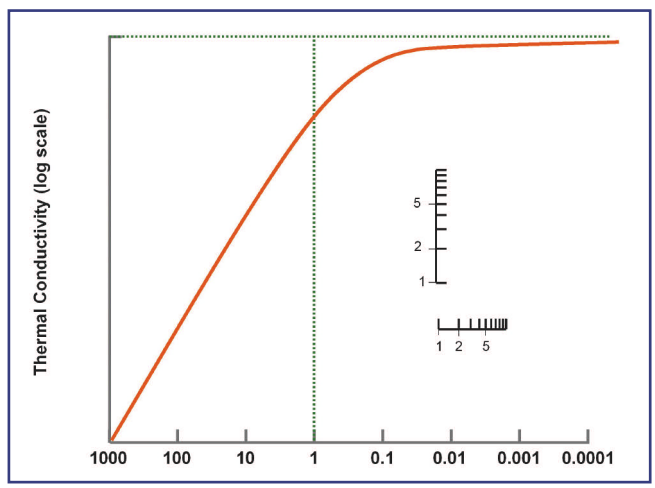

Figure 2. The thermal conductivity for a gas as a function of Knudsen number.

Table 2. (adapted from Ref 2)

Gas Type	Helium	Hydrogen	Argon	Air or Nitrogen	CO_2	C_2H_6
β (Torr·m)	1.5×10^{-5}	9.3×10^{-6}	5.3×10^{-6}	5.1×10^{-6}	3.4×10^{-6}	2.5×10^{-6}

2) Next convert the horizontal scale from Knudsen number to pressure. To do this, first divide β by *d* and use the result to label the vertical dashed line (the former *Kn* = 1 mark) on the horizontal axis.

3) Finally, sketch grid lines and tic marks on both axes using the "floating" decade scales and your new labels as anchor points.

Following this procedure produces a graph customized to meet your particular gas type and dimension. As examples, **Figure 3a** shows this for the case of nitrogen in a 2 cm gap; **Figure 3b** is for helium in a 30 μm gap. Now we can use this type of graph and Equation 1 to estimate the heat transfer between two plates in a vacuum system.

Example 1: Cold Plate Standby

In a degassing chamber system, a cold plate condenser is suspended in a foreline to trap water vapor (this protects the mechanical pump from high vapor loads). The front face of the cold plate is held at -30˚C during operation. It is 10 cm by 15 cm and is located 2 cm away from an opposing wall at room temperature (20˚C). To make it more interesting, let's pretend an argument has erupted during a managers' meeting over the best way to put the system in standby mode. Some say the foreline should be evacuated to 1 Torr and isolated to maintain the cold plate temperature; others say it's no different from simply venting with dry nitrogen and isolating it. We have to decide what the thermal load on the cold plate is at 1 Torr of nitrogen versus 760 Torr of nitrogen. (We'll focus on the conduction through the gas for now. In later parts of this series, we'll revisit this example to calculate how the other heat transfer mechanisms contribute.)

First, sketch the graph of conductivity versus pressure for a 2 cm gap with nitrogen gas. Oh wait, we already did that in **Figure 3a**. Then use that graph to estimate the conductivity values for the two pressures: about 0.02 W/(m·˚C) at 1 Torr and 0.024 W/(m·˚C) at 760 Torr. Finally, use the numbers for this problem in Equation 1 for each pressure to estimate the heat transferred to the plate – a negative answer – from the room temperature wall. (Note: make sure to convert all dimensions to meters.) What are the results? The cold plate receives about 0.9 Watts at atmospheric pressure and 0.75 Watts at 1 Torr. In other words, pumping to 1 Torr isn't going to make much difference. Just one look at the conductivity graph and we know that to reduce thermal conduction significantly, the pressure must be well below 10 mTorr.

Example 2: Cooling an Etched Wafer

An electrostatic wafer chuck holds a 300 mm wafer against a 30μm egg-crate-textured aluminum oxide surface. Since the atomically smooth wafer and rough aluminum surfaces have no compliance, their contact points are exceedingly small; perhaps only 0.01% of the wafer's surface is in contact. Thus, not much area is available through which to conduct heat. And although aluminum oxide is fairly conductive at 30 W/(m·˚C) there is only 0.000007 m² of the total 0.07m² surface available for conduction. Let's estimate how much thermal transfer the contact points provide. If the wafer temperature is processed at 50˚C and the chuck

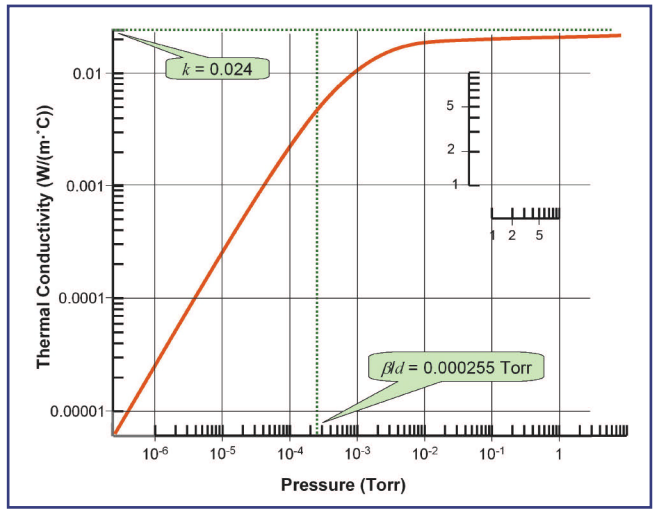

Figure 3a. Graph of thermal conductivity versus pressure for nitrogen in a space with characteristic dimension 2 cm.

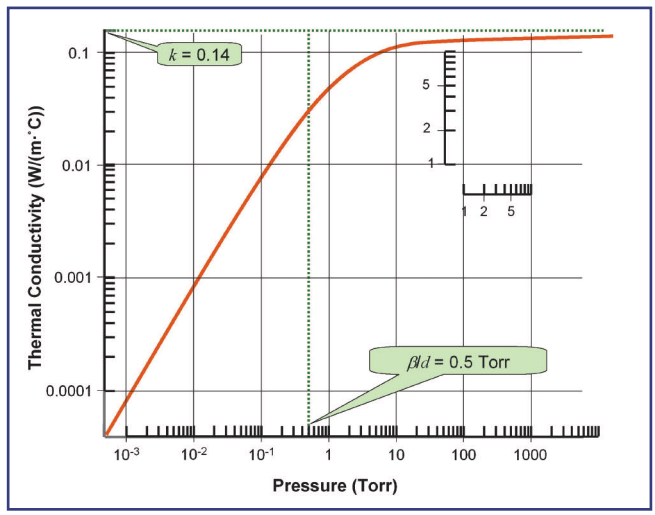

Figure 3b. Graph of thermal conductivity versus pressure for helium in a space with characteristic dimension 30μm.

is water-cooled to 15˚C, Equation 1 predicts that the contact conduction only transfers about 250 Watts across the 30μm textured surface. To provide more heat transfer from the wafer to the chuck and to ensure uniform wafer cooling (since the contact points are not distributed evenly), a common practice is to introduce gas into the texture gap to function as a "thermal paste." Often helium is used since it is inert and highly conductive. To fully understand this function, we must create another plot of conductivity versus pressure for helium and a 30μm dimension. You guessed it. It's already done as **Figure 3b.** As we test some numbers in Equation 1, remembering to use the entire wafer area for *A*, we find that at atmospheric pressure the helium conducts almost 12,000 Watts. This dominates the heat transfer relative to that resulting from the direct contact. What makes this even more interesting and useful is that because of helium's large β and the relatively small gap, the point where conductivity begins changing is at about 50 Torr. And by adjusting the helium pressure in the gap, nearly any heat transfer rate from 250 Watts to 12,000 Watts can be achieved.

Working this type of example from the other direction, if we know the power being delivered to the wafer by the process, we can tune the helium pressure to control the wafer temperature. For instance, let's take as design input that the process delivers 6000 Watts to the wafer and we want the wafer temperature to be 60˚C. First we can estimate how much heat is transferred through contact. With the same assumptions as before the contact should transfer 315 Watts. So the helium must transfer the remaining 5685 Watts. Solving Equation 1 for k and using the numbers for this case gives,

$$k = \frac{d\dot{Q}_h}{A(T_1 - T_2)} = \frac{(3 \times 10^{-5} m)(5685 W)}{(0.07 m^2)(45°C)}$$
$$= 0.054 W/(m \cdot °C)$$

Finally, use the graph (**Figure 3a**) to determine the helium pressure corresponding to this conductivity. About 1 Torr should do the job.

Summary

Part I of this Series on heat transfer in vacuum environments has focused on thermal conduction. Conduction appears to be a straightforward and fundamental heat transfer mechanism; that is until a gas at low pressure is considered. However, once understood, the complicated interaction between gas type, dimensions, and pressure allow the clever technologist an opportunity to optimize and invent new strategies for moving and controlling heat in a vacuum system.

Next month, the second heat transfer mechanism, convection, will be explored. It also has opportunities waiting for us discover.

References

1. http://www.engineeringtoolbox.com/thermal-conductivity-d_429. html
2. John F. O'Hanlon, *A User's Guide to Vacuum Technology*, 3rd Ed. (Wiley, Hoboken, NJ) 2003, p 467.

By Luke Hinkle, *Contributing Editor*

VT&C Guides to Vacuum Technology

Heat Transfer in Vacuum

PART II: Convection in Your System

In Part I of this Series, the three traditional mechanisms for heat transfer – conduction, convection, and radiation – were introduced. Conduction was then examined more closely in the context of transferring heat through a partial vacuum environment. A simple graphical technique was used to estimate the thermal conductivity for a particular situation and a couple practical examples demonstrated the way gas type, pressure, and characteristic dimension all interact to affect the amount of heat conducted through a gas. In Part II, the mechanism of convection will be considered. While convection can be a major factor at atmospheric pressure, as the pressure drops beyond the rough vacuum range it often becomes negligible relative to the other mechanisms. But this "often becomes negligible" presents a practical concern. For a given circumstance, how do you know whether convection effects are negligible? In this part of the Series, heat transfer by convection will be analyzed with an emphasis on how you can quickly estimate its importance and, if significant, how it affects your vacuum processing system.

Basics of Thermal Convection

Convection is the transfer of heat as carried by a moving liquid or gas. This mechanism is often further categorized as either forced convection or free convection. These are distinguished by the cause driving the motion: external in the case of forced, internal (by temperature-induced buoyancy) in the case of free. If you happen to be in a lab or production facility and you'd like to see some examples of each, go look at the back of a typical instrument rack. Electronics enclosures with vertically oriented, finned heat sinks rely on free convection between and around the fins to carry heat away. Enclosures with "muffin fans" actively drawing air across internal components use forced convection.

A full understanding of engineering and design for convective heat transfer requires far more detail and effort than can be discussed here.[See classic texts such as Refs 1 and 2.] And in fact, most serious design work is executed by heat transfer experts aided by computational fluid dynamics (CFD) analysis. But there's still an opportunity for us novices to dabble a bit, if for no other reason, to know when and by how much a design may be affected by convection. So let's explore convection just a bit further, enough to develop a keen intuition and a suite of estimating techniques. After all, for a given situation, you may need to determine whether convection is a contributing factor before you bother with more precision.

Forced Convection

In the March 2008 edition of the Guides to Vacuum Technology, a hot purge gas technique was discussed as a way of enhancing surface desorption.[3] In that context, its function was to accelerate the process of surface moisture outgassing from the chamber. The heat delivered to (or distributed within) the chamber was analyzed using some gross assumptions; but nonetheless, this exercise allowed us to gain some insights regarding the flow rate and gas temperature required. This estimating method will be repeated in a more general sense here.

First, assume there are two regions between which the gas flows. This arrangement can be thought of as two sections of a system separated by an insulating pipe as shown in **Figure 1.**

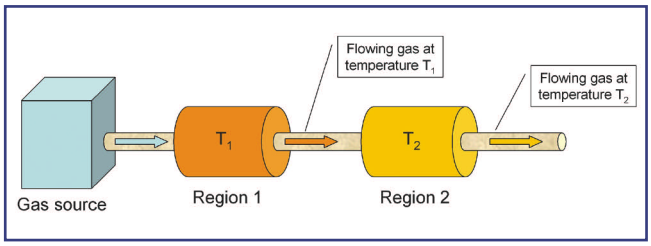

Figure 1. A simple model for forced convection heat transfer

Next assume that as the gas resides within Region 1 or Region 2, it thermally equilibrates to each. In other words, the gas leaving Region 1 is at T_1 and the gas leaving Region 2 is at T_2. Note that these assumptions lead to a maximum heat transfer situation, thus, the resulting value from this estimate should be interpreted as an upper bound.

Now we can estimate the amount of heat carried by the gas into Region 2 and not carried out. This depends on how much the gas temperature changes, the mass of the gas, and the heat capacity of the gas. In equation form it's represented by,

$$Q_h = cm(T_1 - T_2),$$

where Q_h is the heat transferred to Region 2, c and m are the gas's heat capacity (at constant pressure) and mass, respectively. This expression suffices if all we want to do is send a slug of gas from one region to the other and estimate how many Joules were transferred. But we want to estimate the rate of heat transfer for a *continuous flow* of gas: we want a "how many Watts are transferred with this much flow" kind of answer. To get this, we divide both sides of the equation by time, giving a heat transfer rate on the left (indicated by adding a dot over the Q_h) and a mass flow rate Q_m on the right replacing the mass. The resulting equation becomes,

$$\dot{Q}_h = cQ_m(T_1 - T_2). \qquad \textbf{Equation 1}$$

OK, here is where our choice of symbols can cause confusion. The traditional symbols for both heat and mass flow are the letter Q, so we have to be vigilant in our use of the subscripts: h is for heat, m is for mass.

One last issue remains to be resolved before we can conveniently use this analysis: the units. Mass flow instruments don't actually measure mass flow, they measure molar flow and report in units of standard volumetric flow such as standard liters per minute (slm).[4] One way to make the estimation process easier is to make the values for heat capacity (which we have to consult a table for anyway) in units ready-made for our purpose. **Table 1** shows *our version* of heat capacity for various gases in the second column. To arrive at these values, the original J/(gram·˚C) numbers have been converted using the standard density for each gas and the fact that there are 60 seconds in a minute. The result is an uncommon unit, but convenient for our purpose, W/(slm·˚C). With **Equation 1** and **Table 1**, the forced convection heat transfer can quickly be estimated for many typical cases.

For example, if a flow of 20 slm of argon passes from a hot process chamber at 160˚C to a house vacuum line at 30˚C, the heat transfer rate to the line can be predicted as follows:

In nearly every vacuum technology application, forced convection calculations can begin and end with this estimating technique because either: (**1**) the rate of heat transfer is so minor that it does not affect the system, or (**2**) the value is insignificant relative to those of other heat transfer mechanisms. So, estimating forced convection is often simply a checklist item, ensuring that its value can be neglected. But occasionally the exercise reveals that forced convection cannot be neglected. If you find yourself in that situation, you'll be glad you did the simple check. However, then it becomes necessary to analyze the convective heat transfer with more rigor. For most engineers and technologists today, that means we call for help. Thermal modeling and analysis packages and services have become almost a commodity in the recent decade. Further, the standard fluid dynamics and heat transfer models are ideally suited for this task since the circumstances where forced convection becomes significant only arise when the system is in the viscous flow regime (as opposed to molecular flow). Even when we resort to more sophisticated models and techniques, the simple estimating exercise outlined above still serves a role. It establishes an upper bound for results obtained from the model, guarding against input mistakes and faulty interpretation.

$$Q_h = \left(0.015\frac{W}{slm\cdot{}^\circ C}\right)(20slm)(130^\circ C) = 39W.$$

Free Convection

Both forced and free convection transfer heat by transporting gas; the difference lies in what's making the gas move. While an externally-generated pressure source drives the gas in the case of forced convection, the temperature difference within the gas itself drives the free convection gas motion. To describe the effect

Table 1.

Gas Type	Volumetric Heat Capacity Room Temperature, Atmospheric Pressure Condition W/(slm ·˚C)	Density Room Temperature, Atmospheric Pressure Condition kg/m³
Helium	0.016	0.17
Hydrogen	.021	0.083
Argon	0.015	1.7
Air or Nitrogen	.022	1.2
Carbon Dioxide (CO_2)	0.028	1.8
Ethane (C_2H_6)	.036	1.3

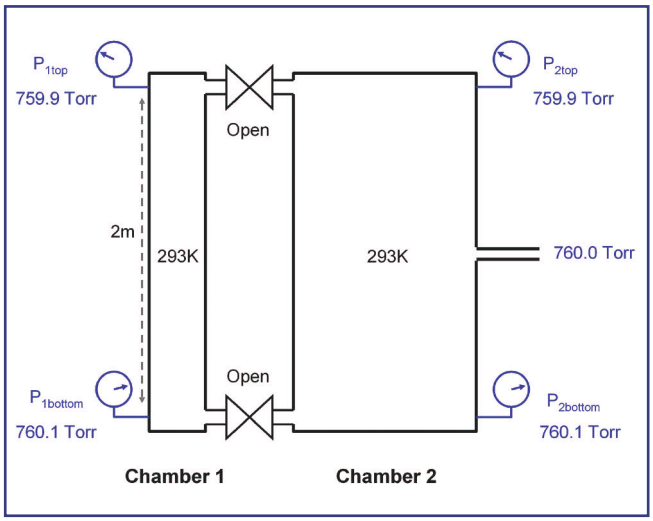

Figure 2a. Free convection thought-experiment apparatus; initial state

in more detail requires a careful, but short, thought process. (By the way, the "hot air rises" cliché will not be used here.) The free convection phenomenon can be illustrated by a simple thought-experiment involving two gas-filled chambers connected by two pipes, each with a valve, as shown in **Figure 2a.** The experiment begins with the entire apparatus at room temperature and both valves open. The pressures at the top of each chamber are the same and the pressures at the bottom of each chamber are the same.

$$P_{1top} = P_{2top} \text{ and } P_{1bottom} = P_{2bottom}$$

However, it's important to note that the top and bottom pressures within each chamber are not equal. The weight of the gas itself generates a pressure gradient from the top to the bottom. This is often overlooked in vacuum technology because the pressure difference between the top and bottom of a chamber is so minor that it is, with rare exception, negligible. But sit up straight now! This is one of the rare exceptions. How much higher is the pressure at the bottom than at the top of a chamber? The pressure-height relationship for fluids (gases or liquids) is given by:

$$P_{bottom} - P_{top} = \rho g\,(h_{top} - h_{bottom}) \qquad \textbf{Equation 2}$$

Where ρ is the fluid density (found in **Table 1** for various gases at room temperature and atmospheric pressure), g is the gravitational acceleration, and h is the height. Many of us are aware of this relationship's historical significance when using columns of liquid (such as mercury or water) to measure vacuum or barometric pressure. Here it's being used to predict the vertical pressure gradient within the gas. In this example, the chambers each contain room temperature air at one atmosphere (760 Torr). What is the pressure difference between the top and bottom? Using **Equation 2**:

$$P_{bottom} - P_{top} = (1.2 kg/m^3)\,(9.8 m/s^2)(2m) = 24 Pa$$

That's about 0.2 Torr. So there would be 759.9 Torr at the top and 760.1 Torr at the bottom.

The key to understanding free convection is buried under the density symbol ρ. How would this vertical pressure gradient be affected if the system were at a different pressure or at a different temperature? Higher pressure means higher density (linear within limits) so the pressure gradient effect is proportional to absolute pressure. So if the system were filled to 76 Torr it would only have a 0.02 Torr difference, 75.99 Torr at the top and 76.01 Torr at the bottom. Higher temperature means lower density (inversely proportional to absolute temperature) so a higher temperature gas will develop a lower pressure difference from top to bottom. With these dependencies in mind, let's return to the thought-experiment.

Recall that the chambers are each filled with air at room temperature and atmospheric pressure. The next step is to isolate them by closing both valves, then heat Chamber 1 to twice the absolute temperature. As the temperature is increased, we can expect the pressure in Chamber 1 to also double. Next we open the upper valve. Immediately a burst of gas rushes into Chamber 2. (This carries a little heat with it of course, but this transfer is not what we're focused on now.) When the top pressures have equalized the flow stops, leaving only half the original amount of gas in Chamber 1. For a moment consider how the pressures at the bottom of the chambers compare? They are not the same. As shown in **Figure 2b**, the gas in Chamber 1 has half the density and therefore half the gradient to the bottom when compared to Chamber 2. So now if we open the lower valve, gas will begin flowing from the bottom of Chamber 2 at 760.1 Torr to the bottom of Chamber 1 at 760.0 Torr. That flow tends to raise the average pressure of Chamber 1 a little, which causes the top pressures to become unbalanced and gas begins to flow from Chamber 1 to Chamber 2 through the upper valve. When these events eventually settle – the steady-state condition – the upper and lower flows are equal and opposite and the pressure differences have equilibrated as shown in **Figure 2c.** So the gas motion is driven through a series of causes beginning with the temperature difference itself. Finally, the flow passing from Chamber 1 to Chamber 2 delivers heat, much the same way as in the forced convection case. How much heat is transferred? That depends on how much flow is being generated, which depends on the restriction presented by the connecting pipes and valves. But before we get too far into the numbers for this example, let's think about how the critical parameters affect the free convection heat transfer.

First, let's consider gas type. Higher molecular weight (denser) gases will produce more of a pressure gradient and therefore more of driving force for the convective flow. But they tend to be a little more viscous (which lessens the volumetric flow). On the other hand, they tend to have a higher heat capacity which delivers more heat for a given flow. The net effect is that the denser gas species are more effective at free convection with a roughly linear relationship between molecular weight and free convection heat transfer. Next, let's consider pressure. Raising the pressure makes the gas denser which actually contributes in two ways: (**1**) it increases the driving force which moves more volumetric flow and (**2**) the volume of moving gas carries more heat with it. These two phenomena work together to produce a dramatic square law dependence on pressure. Going the other way, reducing the pres-

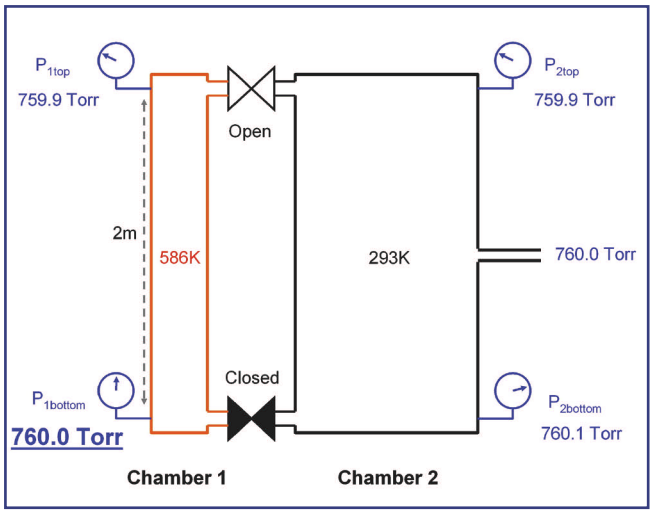

Figure 2b. Lower valve closed; Chamber 1 at twice the absolute temperature, half the density, and half the vertical pressure gradient

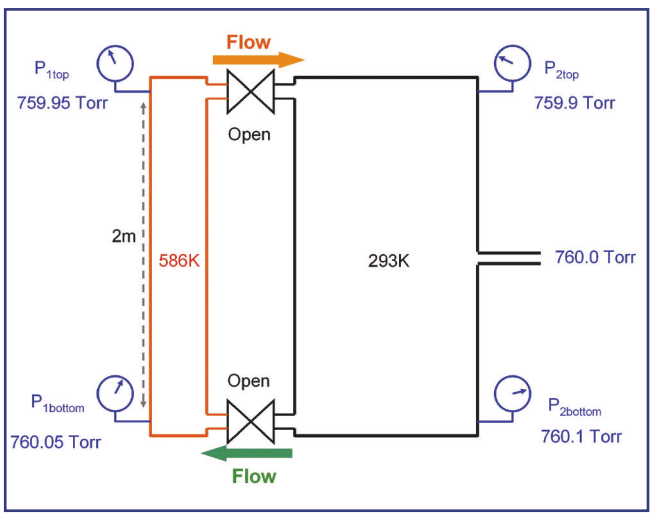

Figure 2c. Both valves open; lower pressure gradient in Chamber 1 produces convective flow and associated heat transfer

sure rapidly diminishes the free convection heat transfer. **So qualitatively speaking, heavy gases at high pressure maximize the free convection effect; light gases at low pressure minimize it.**

Now that we're getting a feel for how free convection works, let's establish an evaluation method which helps us decide whether we have to worry about it in a given setting. This method is based on a dimensionless number called the Rayleigh Number, R_a, which essentially compares the "convectability" to the "conductability" of the particular situation. For a gas the Rayleigh Number formula is:

$$R_a = \frac{g\rho_o^2 P^2 c\,(\Delta T) L^3}{T P_o^2 \eta k} \qquad \textbf{Equation 3}$$

Where ρ_o is the density at standard conditions, η is the viscosity, k is the thermal conductivity, P_o is standard atmospheric pressure, and L is the characteristic dimension. How does this help us? For a vertically oriented situation, if the Rayleigh Number is equal to 1700, then the heat transfer by convection and conduction are equally important. **As the number goes higher than 1700, the convection becomes more important; as the number goes lower, the conduction becomes more important.** Before we get into the details, let's take a quick look at how the parameters affect the formula. We see that the gas density and pressure are in the numerator, meaning that as those are increased the convection effect becomes more important – but we already knew that from our intuitive reasoning above, right? Good. Notice that there is a very strong dependence on the characteristic dimension L. Is this why double-pane windows are spaced so closely, to reduce convection in the gap? Yes. While on the topic of characteristic dimension, note that many important dimensionless numbers used in vacuum technology (e.g. Knudsen, Reynolds, Nusselt, and Péclet) include this parameter in their formulae and the choice of its value is open to interpretation. What happens if you choose the wrong dimension in a particular problem? For precise work, this can be an issue. But the intent here is only to use the Rayleigh Number to establish a "convection alarm" setting – a warning that free convection cannot be ignored. So if any reasonable choice for the characteristic dimension puts you close to the boundary, it's time to sound the alarm.

Before we task ourselves with looking up all those gas properties and punching buttons to find the Rayleigh Number for a particular example, let's convert this approach into another bounding method. Instead of calculating the R_a at a set of conditions to see if it's above or below the critical 1700 value, we can calculate the pressure required to make the $R_a = 1700$. Then if a system is near or above that $R_a = 1700$ pressure we'll know we must account for free convection, otherwise, we can ignore it. To make your task easier, the pressure numbers have been run for a combination of conditions and are shown in **Figure 3.** The parameters have been selected to span a wide range of applications: hydrogen, nitrogen, and sulfur hexafluoride (SF_6) to represent "light, medium, and heavy" gases; a ΔT from 1˚C to 100˚C; and a characteristic dimension of 1cm, 10cm, and 50cm. For many system designs, you can select the graph and find the point that best represents your conditions. Remember, these graphs are interpreted as follows: for pressures above the line, free convection is significant; below the line it rapidly becomes negligible. Observe that for a typical vacuum processing application below 1 Torr, free convection is rarely a factor. However, between 1 Torr and atmospheric pressure, its significance must be assessed – at least by using this quick evaluation method. If you determine that free convection could be a factor relative to conduction, then it may be time to call in the heat transfer expert again.

Summary

Part II of this Series on heat transfer in vacuum environments has focused on thermal convection, both forced and free. Because these mechanisms are so complicated, only estimates and bounding methods have been addressed. However, with these techniques in your "intellectual toolbox," you can quickly decide how much impact convective heat transfer may have on your system. Most times, convection effects can be ignored. That's usually good news for those of us who have bigger design and operational problems

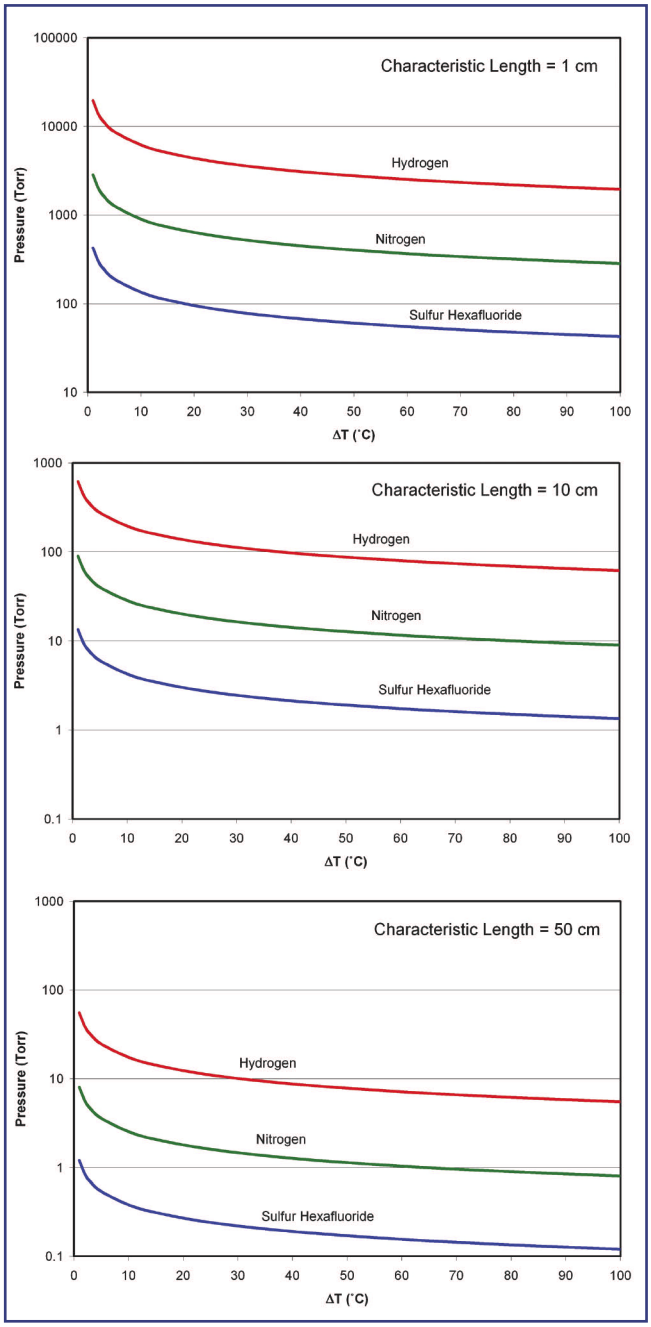

Figure 3. Graphs showing the pressure where free convection and conduction are of equal importance. Above the line, convection becomes progressively more important.

to solve. Unfortunately, the estimating method will occasionally reveal bad news: convection does have a significant effect. Luckily, computational methods have progressed remarkably and there are plenty of experts willing and able to help solve your problem.

Next month, the third heat transfer mechanism, radiation, will be investigated. We'll make quick work of that particular topic since the wavelength range that carries most of the thermal energy is transmitted through air as effectively as through vacuum. Then we'll return to some earlier examples and apply all three heat transfer mechanisms simultaneously.

In the real world, all the mechanisms are at play.

References

1. Sadik Kakac and Yaman Yener, *Convective Heat Transfer*, 2nd Ed. (CRC Press) 1994.
2. Adrian Bejan, *Convection Heat Transfer*, 3rd Ed. (Wiley, Hoboken, NJ) 2004.
3. "Outgassing Part 1: Desorption of Molecules from Surfaces," Luke Hinkle, Vac. Tech. and Coating (March 2008) pp 32-38.
4. "If They Don't Control Mass Flow, Why Do We Call Them 'Mass Flow Controllers?"" Luke Hinkle, Vac. Tech. and Coating (Sept. 2007) GD&D, pp 4-6.

By Luke Hinkle, *Contributing Editor*

VT&C Guides to Vacuum Technology

Heat Transfer in Vacuum

PART III: Radiation and examples with all three mechanisms

In Parts I and II of this Series, two of the three traditional mechanisms for heat transfer — conduction and convection — were the focus.[1,2] Both of these mechanisms have a strong and complicated dependence on pressure and gas type. On the other hand, since air and most other gases are essentially transparent to infrared and visible wavelengths, radiation heat transfer doesn't change as pressure is reduced. Does this mean that radiation heat transfer can be ignored in vacuum technology? No, quite the opposite. By remaining constant with respect to pressure, radiation's *relative* importance increases as pressure is reduced. Further, it becomes the dominant mechanism at high vacuum, where the others are negligible.

In this Part III of the Series on heat transfer, the basics of radiation will be developed and estimating methods for its effect will be demonstrated. Then we'll return to the practical situations raised in Part I; but this time, all three heat transfer mechanisms will be involved. As you gain confidence analyzing these examples, you'll know another arrow has been added to your quiver, another club has been added to your bag, another wrench has been added to your toolbox…

Basics of Radiation Heat Transfer

From our common experience, when the current through a light bulb filament is gradually increased (with a dimmer switch) it appears to go through several stages of glowing: first a dull red, then orange, then yellow, then bright white. With the aid of infrared photography, this color/temperature relation is extended to less extreme temperatures. So for cases where objects are less than "red hot," most heat transfer by radiation is carried by the invisible infrared portion of the electromagnetic spectrum. So how much heat is carried by radiation? The rate of thermal transfer by radiation from a surface is characterized by the Stefan-Boltzmann equation,

$$\dot{Q}_h = \sigma e A (T_1^4 - T_2^4) \qquad \textbf{Equation 1}$$

where $\dot{Q}_h$, is the heat transfer rate, σ is the Stefan-Boltzmann constant with a value of 5.67×10^{-8} W/(m^2·K^4), e is the emissivity of the surface with values from 0 to 1, A is the surface area, T_1 is surface temperature, and T_2 is the surroundings' temperature. Before we think about applying this equation, notice the temperature dependence: the difference of their fourth powers! This is not the same as the fourth power of their difference. This distinction can be demonstrated by a dull-red object in a furnace example where T_1 = 1000K and T_2 = 999K. If we were to do the temperature part of Equation 1 wrong, subtracting first and then doing the fourth power, it would be 1K^4. Whereas the correct way, applying the fourth power to each then subtracting, results in about 4 billion K^4. Being off by a factor of 4 could happen to the best of us; but being off by a factor of 4 billion is another matter altogether. Now back to the other parameters in Equation 1, how do we decide what the emissivity of a surface is? A perfect reflector (no absorption, no emission) is 0; a perfect blackbody is 1. So we could imagine that for every kind of substance, there is an emissivity value listed in a table somewhere, right? Sorry. It seems like every time a relatively simple equation is used to describe some phenomenon, the complicated behavior is hidden in something called a material property! Tables for emissivity values can be found, but the condition of the surface (e.g. pol-

ished to what level, oxidized to what extent, cleaned with what procedure, anodized with what recipe, etc.) is a necessary qualifier. Even with the surface condition specified, sample-to-sample variations of 20% are possible and the measured emissivity often increases with temperature (as if a fourth power effect isn't enough!). One more note of caution, there are two different versions of emissivity: spectral and total. Spectral emissivity values, used for temperature measurement and thermal imaging, indicate emissivity at specific wavelengths. For this discussion, we need to use the total emissivity value since heat transfer occurs throughout a broad range of the spectrum. Total emissivity values for materials and surface conditions commonly used in vacuum systems have been compiled from References 3-6 and are presented in **Table 1.**

As a quick example, let's investigate the question that cooking experts have pondered for years: Should I or should I not wrap a potato in aluminum foil before baking? The foil changes the cooking process in many ways including maintaining the moisture level so that the wrapped potato tends to steam cook. Here though, we'll focus the analysis on the differences in radiation heat transfer. Assume a 10 cm diameter (spherical) potato is placed in an oven preheated to 480K (that's about 200˚C or 400˚F). How much heat is transferred to the potato by radiation from the oven's internal surfaces? First, take the case of the bare potato. If we can assume that the emissivity of a washed potato is about 0.9 (a typical number for wood), from Equation 1 we get:

$$\dot{Q}_h=[5.67\times10^{-8}W/(m^2K^4)](0.9)(0.0314m^2)[(293K)^4-(480K)^4]=-73W$$

So the potato initially absorbs (net negative heat transfer) 73 Watts from the oven. How does wrapping it in foil change this result? Only the emissivity is changed, and it is reduced to about 0.09 (taking the upper end of the range from **Table 1** for aluminum foil). This reduces the initial radiation heat transfer by an order of magnitude to about 7 Watts. An interesting result, but this analysis of the initial heat transfer to the potato is far from conclusive regarding the entire baking process. Further, any decision about the best way to bake a potato depends on your preference for the epicurean qualities in the finished product: a crispy or tender skin, a fluffy or moist center, etc. These are important issues, but well beyond the scope of this guide.

Table 1.

Material and Surface Condition	Total Emissivity
Aluminum - Foil/Rolled Sheet	0.04-0.09
Aluminum – Highly Polished	0.039-0.057
Aluminum – Heavily Oxidized	0.20-0.31
Aluminum – Anodized	0.77-0.88
Mild Steel – Polished	0.10
Mild Steel – Machined	0.20-0.32
Steel – Oxidized	0.8
Cast Iron – Oxidized	0.6-0.7
Copper – Polished	0.023-0.052
Copper – Thick Oxide Layer	0.78
Stainless Steel – Polished	0.075-0.28
Stainless Steel – Oxidized	0.66-0.91
Ceramic – White Alumina	0.90
Silicon – Wafer (lightly doped at < 300˚C)	0.1
Silicon – Wafer (heavily doped at < 300˚C)	0.7
Silicon – Wafer (>600˚C)	0.7
Silicon – Wafer with Oxide or Nitride Film	0.65-0.85

For the remainder of this Guide, we will revisit the practical, vacuum processing examples which were introduced in Part I of this Series on heat transfer. Now that we're equipped with techniques for evaluating and estimating the three common mechanisms for heat transfer – conduction, convection, and radiation – we can assess these examples with more confidence.

Example 1: Cold Plate Standby

From Part I of the Series: "In a degassing chamber system, a cold plate condenser is suspended in a foreline to trap water vapor (this protects the mechanical pump from high vapor loads). The front face of the cold plate is held at -30˚C during operation. It is 10 cm by 15 cm and is located 2 cm away from an opposing wall at room temperature (20˚C). To make it more interesting, let's pretend an argument has erupted during a managers' meeting over the best way to put the system in standby mode. Some say the foreline should be evacuated to 1 Torr and isolated to maintain the cold plate temperature; others say it's no different from simply venting with dry nitrogen and isolating it. We have to decide what the thermal load on the cold plate is at 1 Torr of nitrogen versus 760 Torr of nitrogen."

In Part I we found that there was little difference in the conduction through the gas — about 0.9 Watts at atmospheric pressure and 0.75 Watts at 1 Torr. But what about the other transfer mechanisms?

In Part II of the Series, we established methods for evaluating the role of convection. In this example, forced convection is not occurring since the trap is isolated during standby, but is there significant free convection transfer? Let's use the Rayleigh number graphs (**Figure 3** from last month's Part II) to evaluate the importance of free convection. To decide which graph to use and how to interpret it, we must first collect the relevant parameters for our situation. The characteristic dimension is 2 cm (the distance between the plates), the temperature difference is 50˚C, and the gas is nitrogen. So we'll be using the top graph (it's for a 1 cm dimension, the closest to our 2 cm). Following the green (nitrogen) line to where it is over the 50˚C mark, and noting the pressure at that point we find that the critical Rayleigh pressure is about 400 Torr or so. This is shown in **Figure 1.** This tells us that above 400 Torr we need to be concerned that free convection is significant relative to conduction. So our atmospheric case may have free convection whereas the 1 Torr case will not. Now that we are aware of the situation, let's try to be a little more precise about how much convection heat transfer can be expected at atmospheric pressure. The Rayleigh Equation (Equation 3 in Part II) for our exact parameters produces a value of about 50,000. Compared to the critical value of about 1700

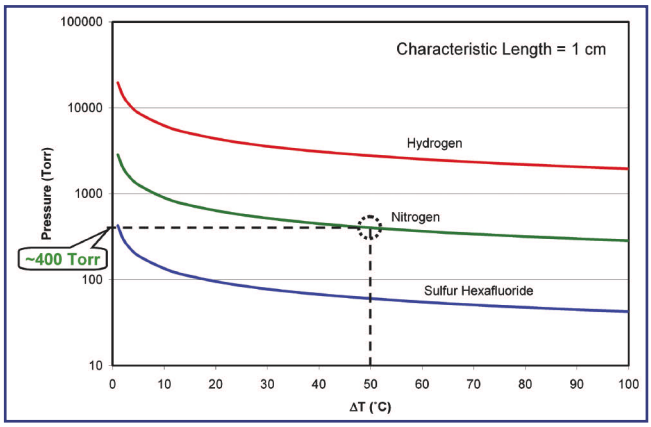

Figure 1. Graph showing the pressure where free convection and conduction are of equal importance. Above the line, convection becomes progressively more important. The dashed lines indicate the critical Rayleigh point for conditions similar to those in Example 1.

this is nearly 30 times higher. So, very roughly speaking, we could expect as much as 30 times more heat transfer by convection than by conduction at atmospheric pressure, or 27 Watts.

Will radiation play a significant role? It's easy to find out. Putting this example's data into the Stefan-Boltzmann Equation, using an emissivity of 0.3 for heavily oxidized aluminum, results in a transfer of about 1 Watt. Even if the plate is covered with ice, increasing its effective emissivity, the radiation transfer will still be limited to about 3 Watts. As a graphical summary, the estimates for each heat transfer mechanism at each pressure condition are illustrated in **Figure 2.**

So it would seem that the best mode for standby, from the thermal perspective, is to pump and isolate the trap at a modest vacuum level to eliminate free convection transfer effects. As long as the pressure remains below about 100 Torr, the thermal transfer is dominated by the combination of conduction through the gas (at about 1 Watt) and radiation (at between 1 and 3 Watts). By the way, conduction paths through the mechanical and coolant connections cannot be overlooked in an actual setting and may easily dominate the heat transfer at vacuum conditions.

Example 2: Cooling and Etched Wafer

From Part I of the Series: "An electrostatic wafer chuck holds a 300 mm wafer against a 30µm egg-crate-textured aluminum oxide surface. Since the atomically smooth wafer and rough aluminum surfaces have no compliance, their contact points are exceedingly small; perhaps only 0.01% of the wafer's surface is in contact. Thus, not much area is available through which to conduct heat. And although aluminum oxide is fairly conductive at 30 W/(m·°C) there is only 0.000007 m² of the total 0.07m² surface available for conduction. Let's estimate how much thermal transfer the contact points provide. If the wafer is processed at 50°C and the chuck is water-cooled to 15°C, Equation 1 predicts that the contact conduction only transfers about 250 Watts across the 30µm textured surface. To provide more heat transfer from the wafer to the chuck and to ensure uniform wafer cooling (since the contact points are not distributed evenly), a common practice is to introduce gas into the texture gap to function as a 'thermal

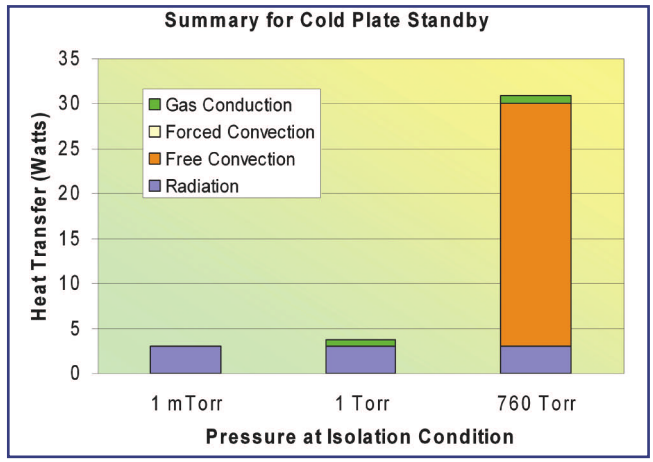

Figure 2. A graphical comparison of the estimated heat transfer, broken-down by mechanism, for Example 1.

paste.' Often helium is used since it is inert and highly conductive."

In Part I we found that by varying the helium pressure between 0.1 and 50 Torr the thermal conduction through the gas can be varied from an insignificant value to 12,000 Watts. This is a remarkably useful technique, but can we safely ignore the convection and radiation effects?

In this case, forced convection can be considered since the helium is continuously flowing between the wafer and textured chuck surfaces. Often the helium gas enters this gap at the center of the chuck and flows at rates typically not exceeding 0.1 slm radially to the edges. To estimate the heat transferred by the forced flow of helium, we can use the analysis summarized as Equation 1 in Part II with the numbers for our example here:

$$\dot{Q}_h = \left(0.016 \frac{W}{slm \cdot °C}\right)(0.1 slm)(35°C) = 0.06W$$

This is indeed negligible compared with both the direct conduction through the contact points and the conduction through the gas. Likewise a glance at the free convection graphs (see **Figure 3** of Part II) tells us that the free convection, even if the gap were much larger, is not a factor.

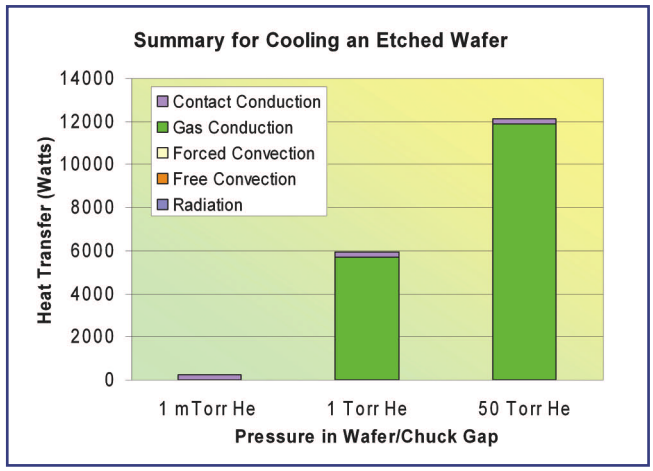

Figure 3. A graphical comparison of the estimated heat transfer, broken-down by mechanism, for Example 2.

Finally, radiation must be considered. Putting the numbers for this example into the Stefan-Boltzmann Equation, assuming the extreme case of a heavily doped wafer, results in a transfer of about 11 Watts — small in comparison to the conduction mechanisms. As a graphical summary, the estimates for each heat transfer mechanism are illustrated in **Figure 3.**

So it appears that the analysis of the helium-cooled wafer can safely ignore both convection and radiation effects. This confirms that helium pressure in the gap between the chuck and wafer is an effective control parameter to regulate the conduction heat transfer and hence the wafer temperature during process.

Summary

In this three-part Series on heat transfer in vacuum, we investigated the three mechanisms for heat transfer and how each is affected by the pressure and gas type. Rather than attempting to address all the nuances and details — a task which might require thousands of pages and many lifetimes of reading — this three-part Guide has provided the basics of each mechanism and a few handy tools for evaluating the relative importance of each for a given situation.

We don't all have to be experts in heat transfer, but we should all know a few things: how to estimate each mechanism, how to recognize a problem, and when to call an expert.

References

1. Luke Hinkle, "Heat Transfer in Vacuum Part I," Vac. Tech. and Coating (July 2008) pp 39-42
2. Luke Hinkle, "Heat Transfer in Vacuum Part II," Vac. Tech. and Coating (August 2008) pp TBD
3. http://www.engineeringtoolbox.com/emissivity-coefficients-d_447.htmlAdrian
4. http://www.omega.com/literature/transactions/volume1/emissivity.html
5. http://snap.fnal.gov/crshield/crs-mech/emissivity-eoi.html
6. Robert Doering, Yoshio Nishi, "Handbook of Semiconductor Manufacturing Technology" (CRC Press) 2000, pp 215 & 223.