CRYOPUMPS APPLIED TO HIGH TEMPERATURE VACUUM PROCESSES

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ABSTRACT

Cryopumps create vacuum by capturing gases on very cold surfaces. Vacuum pump users are often uncertain whether cryopumps are suitable for use in high temperature processes where the high thermal loads could warm the pumping surfaces and release the captured gases. This paper discusses methods for estimating the thermal loads on a cryopump, and makes suggestions for minimizing their effects.

The basics of radiant heat exchange are presented to explain the behavior of high temperature vacuum processes. This understanding can help the system designer make some qualitative predictions of the thermal performance of a cryopumped system.

For a quantitative study, a simple model is presented, which can be easily implemented on a computer spreadsheet, for the transfer of radiant heat to the cryopump. This model can then be used to determine the refrigeration capacity of the cryopump expander that is needed for a particular application, or to predict the thermal shielding that is required for any given cryopump. An example is given, using a new Marathon® cryopump designed by IGC-APD Cryogenics Inc. (APD) for small optical coaters, to demonstrate the effectiveness of the model.

INTRODUCTION

Cryopumps are capture type vacuum pumps that evacuate a chamber by freezing and retaining the gases from the chamber. For this reason, they are ideal for creating clean, high vacuum, and have the added benefit of being able to pump water vapor very effectively.

In order to retain the pumped gases, the cryogenic surfaces in the pump must always remain below a critical temperature or the frozen material will vaporize and vacuum will be lost. The temperature will increase if parasitic heat loads increase to the point that the cryogenic refrigerator can not keep up. These parasitic heat loads can come from poor vacuum or radiant heat exchange.

Conductive heat leak through the vacuum space is negligible for most cryopumps using closed cycle, Gifford-McMahon expanders, as long as the pressure is about 0.3 Pa (2 mil-

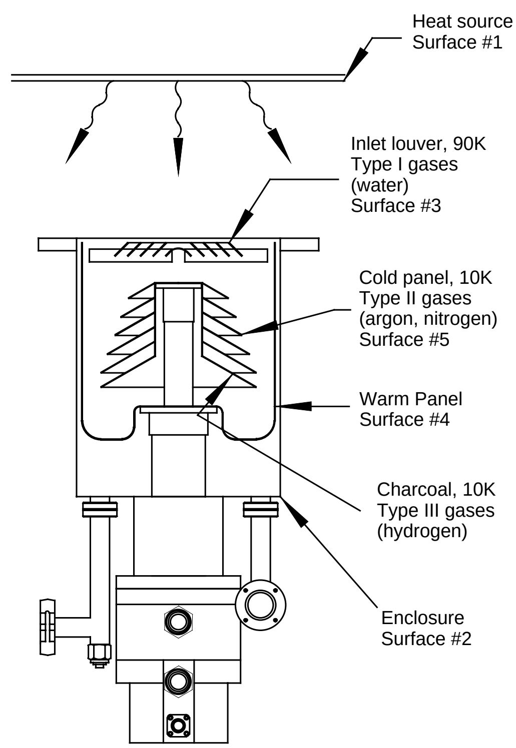



Figure 1 - Cross sectional view of a cryopump, illustrating its operation.

litorr) or less. On the other hand, radiant heat loads from sources hotter than room temperature can be quite significant, especially considering that the radiant heat increases with the fourth power of the temperature of the radiating surfaces. It is this parasitic radiant heat load on which we will concentrate.

## CRYOPUMP OPERATION

Figure 1 illustrates how a cryopump operates. Gases, like water vapor, that condense at relatively high temperatures are captured (or pumped) immediately on the inlet louver of the cryopump. These gases, frequently known as Type I gases, have a very low partial pressure (less than $10^{-8}$ Pa) when cooled to the temperature of the inlet louver, which is typically 90K or colder. Type II gases, such as nitrogen, oxygen, or argon, will pass through the inlet louver until they reach the surfaces of the second stage cryopanel, or cold panel, which is cooled to 20K or less. Hydrogen, and helium are examples of Type III gases which can not be condensed on any of the cold surfaces in the pump. These gases must be captured by cryo-sorption into activated charcoal granules cooled by the second stage cryopanel.

Since cryopumps are capture pumps, it is essential that the frozen Type I and Type II gases remain cold enough to prevent them from outgassing to the vacuum space. The behavior of gases under these conditions is shown in the equilibrium vapor pressure curve of Figure 2 [1]. This chart shows the partial pressure in a vacuum chamber versus temperature for a gas captured and cooled by the cryopump. For example, if nitrogen is captured on a surface at 20K, its partial pressure in the chamber will be about $1x10^{-9}$ Pa. Conversely, to keep the partial pressure of water in the UHV range, requires that all the water in the chamber be cooled to about 120K or less.

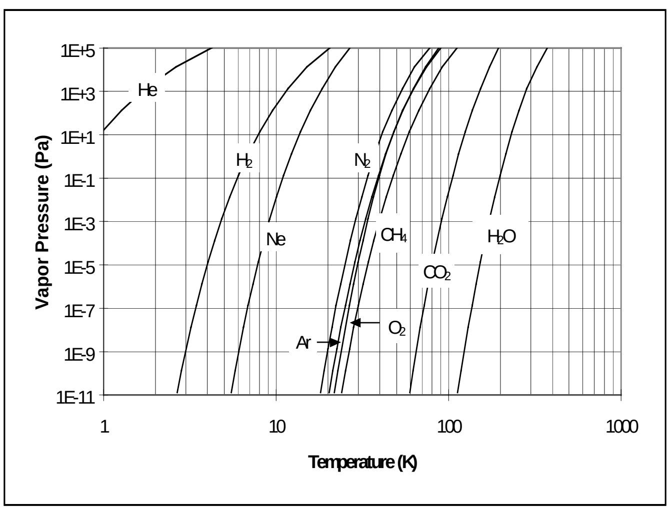



Figure 2 - Equilibrium vapor pressure curves for common gases.

The behavior of the Type III gases adsorbed by the charcoal is more complicated. For these gases, it is generally accepted that the charcoal must be kept below 20K to maintain high vacuum.

If the inlet louver is operating at a high enough temperature, water vapor will pass through the inlet louver and be captured on the second stage cryopanel with the Type II gases. Also, if the inlet louver is warmed after it has already captured water vapor, the water will be released and will condense onto the 20K second stage panel. This process will cause the pressure to linger at the vapor pressure of the water, corresponding to the temperature of the inlet louver, until all of the water has transferred to the second stage cryopanel. In general, it is not necessary for the water vapor to be pumped by the inlet louver, although the pumping speed for water will be much higher if it is captured by the inlet louver instead of the cold panel.

## RADIANT HEAT TRANSFER

In high temperature vacuum processes, the dominant mode of heat transfer is by radiation. When the pressure falls below about 0.3 Pa, heat transfer via conduction and convection is negligible. To appreciate the difference, if two, black, parallel plates, 1000 $cm^2$, separated by three centimeters were in a vacuum of 0.3 Pa, and one plate was at room temperature while the other was at 10K, the radiant heat transfer between the two would be about 40 watts compared to 50 milliwatts for the conductive heat transfer. Clearly, the radiant heat transfer will be of primary concern for cryopump applications.

Of course, the example above is for two black bodies: a perfect thermal emitter and perfect thermal absorber. For this situation the heat transfer is proportional to the difference of the surface temperatures to the fourth power. The equation is:

$$q = \sigma A(T_1^4 - T_2^4) \qquad \text{Eq (1)}$$

where
$q$ = heat transfer rate (watts)
$\sigma$ = Stefan-Boltzmann constant ($5.677x10^{-12}$ watts-$cm^{-2}$-$K^{-4}$)
$A$ = area of emitting surface ($cm^2$)
$T_1$ = hot surface temperature (K)
$T_2$ = cold surface temperature (K)

In the real world case, where the interacting surfaces might be shiny, rather than black, the heat transfer will be reduced, probably quite dramatically. A shiny absorbing surface will reflect most of the incident radiation, thereby lessening the amount of energy absorbed. Likewise, a shiny emitting surface will exhibit far less emissive power than a black surface at the same temperature. Frequently, to simplify calculations, the emissivity and absorptivity of a real surface are taken to have the same value[2]. While this may not be totally accurate, the error is small for most engineering applications.

The radiant heat exchange between two surfaces is also reduced from the ideal by the spatial orientation of the surfaces. The ideal case in equation one is for flat parallel surfaces extending infinitely in both directions. Here, radiation emitted in any direction from any point on the hot surface will fall on the cold surface. For other cases where finite surfaces are oriented in such a way that emitted energy may "miss" the absorbing surface, the radiant exchange must be modified by a shape factor which is always less than unity.

When we modify Equation 1 to take into account real surfaces having emissivities, abosorptivities, and shape factors that are not unity, the radiant heat exchange is calculated by Equation 2 below[2].

$$q = A_1 F_{1\text{-}2} \sigma \frac{(T_1^4 - T_2^4)}{(1/\varepsilon_1)+(1/\varepsilon_2)-1} \qquad \text{Eq. (2)}$$

where

- $q$ = heat transfer rate (watts)
- $A_1$ = area of the emitting surface ($cm^2$)
- $F_{1\text{-}2}$ = shape factor
- $\sigma$ = Stefan-Boltzmann constant ($5.677 \times 10^{-12}$ watts-$cm^{-2}$-$K^{-4}$)
- $T_1$ = hot surface temperature (K)
- $T_2$ = cold surface temperature (K)
- $\varepsilon_1$ = emissivity of hot surface
- $\varepsilon_2$ = emissivity of cold surface

We can take this information and do another comparison with the 1000 $cm^2$ parallel plates separated by a 3 cm gap. The shape factor for this geometry is nearly unity. The emissivity of a surface painted flat black is about 0.96, while that of a polished stainless steel surface is about 0.1. If the emitting surface is painted flat black at room temperature, and if the absorbing surface is polished stainless at 10K, then the heat transfer between them is about 4.3 watts. If both surfaces are polished stainless then the heat transfer is about 2.3 watts. Compare these values to the cooling capacity of a typical Marathon® cryopump refrigerator which can sink about 8 to 10 watts of power on the second stage without warming above 20K.

QUALITATIVE ANALYSIS

The theoretical discussion above may lead the engineer using a cryopump in a high radiant heat environment to decide that polishing all the surfaces inside the chamber will have the greatest effect on minimizing the heat load on the cryopump. In reality, this is almost never the case. Lets look at a specific example to see why this is so.

Suppose we have a thermal process in which aluminum is being vaporized for deposition. This will require that the aluminum be heated to at least 1500K to achieve a vapor pressure of $1 \times 10^{-2}$ torr. Our 1000 $cm^2$, black, parallel plates would exchange about 27 kilowatts with the hot plate at 1500K! It's easy to see that we have to dissipate most of this heat before it reaches the cryopump.

One way to intercept the heat is to place a barrier between the source and the inlet of the cryopump. The side facing the heat source should be high emissivity (black) so that the energy can be absorbed and dissipated rather than reflected to some other place where it could cause problems. The side facing the pump inlet should be low emissivity (shiny) so that the re-radiated energy will be minimized. Since this baffle will be in a vacuum, the heat must be carried away by some method that has the capacity to handle the potentially large quantities of energy. Cooling the baffles by circulating water is much more effective than by conducting the heat through appendages in contact with the shell of the vacuum vessel.

If this water-cooled baffle is to be placed between the process and the pump inlet, it must be made in such a way that it does not restrict the passage of gas. A common way is to create "louver" or "chevron" type baffles. These two types of baffles are illustrated in Figure 3.

The louver is a series of angled slats (or concentric conical surfaces) that are spaced in such a way as to create a blind to incoming radiant heat, yet allow the process gas to pass by. It is quite possible for thermal photons to get past the louver

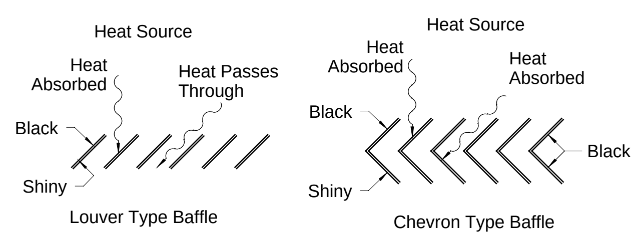
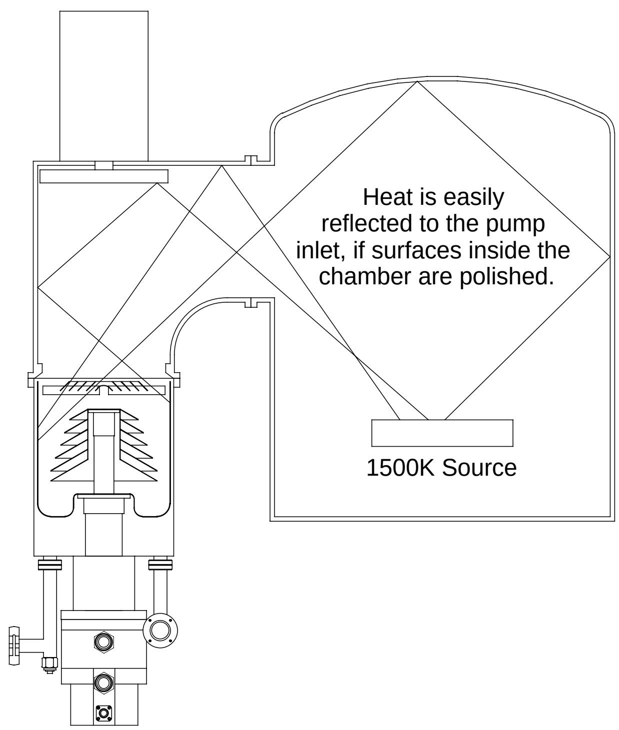



Figure 3 - Types of water cooled baffles used in vacuum systems.

if they are on an oblique path, or if they are reflected from the back side of the louver. Figure 3 shows the sides of the louver that face the heat source are made to have a high absorptivity and the sides that would face the pump inlet are shiny. With this arrangement, the radiated heat will be more effectively absorbed by the louver so it can be removed by the cooling water. The side which faces the pump is radiating heat at the cooled temperature of the louver, so it is more effective to make this side shiny to reduce its emissivity.

Chevron type baffles can be made opaque to thermal radiation by incorporating another set of slats angled in the opposite direction of the first to cut off any heat that is trying to pass by the first set. While the chevron baffle is very effective in intercepting radiant heat, it has a much lower conductance than the louver type. The conductance probability for the louver illustrated in Figure 3 is 0.4, but the chevron is almost half that, with a probability of 0.25 [3]. Again, most sides of the chevron are black to absorb the radiant heat and sink it to the cooling water, and the side facing the pump is shiny to reduce its emissivity.

Another way to protect the pump from radiant heat is to place it so that there is not a "line of sight" path from the




heat source to the pump inlet. A common way to do this is to connect the pump to the process chamber by way of a right angle poppet valve as illustrated in Figure4. The figure

Figure 4 - The cryopump is positioned out of the direct "line of sight" of heat radiating from the process.

also shows that, if the inside surfaces of the chamber and elbow are polished stainless steel, much of the heat will be reflected to the pump inlet anyway. Even if it is reflected from a cool surface, the radiation will still have the characteristics of the 1500K source. In order to attenuate the energy, the inside surfaces need to be treated to give them a high absorptivity. Then the heat that is absorbed can be carried away by cooling the outside surface of the elbow. The remaining energy will be re-radiated at the temperature of the surface. Because of the fourth power relationship of radiant heat to the temperature, the energy getting to the pump will be reduced by two or three orders of magnitude.

## QUANTATIVE ANALYSIS

IGC-APD Cryogenics Inc. has developed a simple, but accurate method to predict the operating temperatures of the pumping surfaces of a cryopump. The method employs a personal computer spreadsheet program to calculate the operating temperatures from input data about the source of heat and the features of the cryopump. Thermal performance information for the cryogenic refrigerator powering the pump is also contained in the spreadsheet and is used to calculate the operating temperatures.

Refer back to Figure 1, the cross section of the cryopump. The figure shows an emitting surface, surface #1, opposite the pump inlet, which represents the source of radiant heat in the process chamber. Another source of heat that affects the operation of the cryopump, comes from the vacuum enclosure of the pump itself which is identified in the schematic as surface #2. The various emitting and absorbing surfaces are considered in the spreadsheet, in all significant combinations, to arrive at the total load on the refrigerator.

The process heat source radiates heat directly to the inlet of the pump, which is broken into three separate components. One component is the heat radiated to the louver, surface #3, and usually has a high shape factor for the exchange. The next element is heat radiated past the inlet louver to the inside surface of the first stage cryopanel (or warm panel), surface #4 in Figure 1. The final component is heat which is absorbed by the second stage cryopanel, surface #5.

The inlet louver is plated with a highly reflective nickel finish that can be approximated as having an absorptivity of about 0.1. However, if a significant amount of moisture, about a gram or more of water, has been pumped, then the inlet louver will behave more like a black body with an absorptivity of about 0.8. The area of the inlet louver is calculated from its area projected onto the plane of the pump inlet. This is usually the area of a circle whose diameter is the outside diameter of the outer ring of the louver.

The first stage cryopanel, or warm panel, is a sheet of copper, cooled by the first stage of the refrigerator, which lines the vacuum enclosure of the cryopump, and shields the pumping surfaces from thermal radiation. Its surfaces are usually painted flat black to give them an absorptivity of about 0.9. With these highly absorbing surfaces, incident radiation is captured and sunk to the first stage of the refrigerator rather than be reflected to the pumping surfaces as would happen with a shiny finish. The shape factor for the exchange between the heat source and the warm panel is usually moderately low since the surfaces of the panel are oblique to the incoming radiation.

Some thermal radiation is usually able to pass through, or around the inlet louver and be absorbed directly by the second stage cryopanel, or cold panel. Like the inlet louver, the surface of the cold panel is nickel plated to reflect incident radiation. Most of the type II gases, captured by the cold panel, however, do not cause a significant increase in the absorptivity of the of the surface the way water vapor affects the inlet louver. The shape factor for this interaction is usually quite low because of the orientation of the cold panel surfaces, and the effects of the inlet louver.

In addition to the heat coming directly from the thermal process itself, radiation is also emitted from the room temperature vacuum enclosure, and exchanged between internal surfaces of the pump. The internal exchange is between the warm panel and the cold panel, or between the inlet louver and the cold panel. Since the inlet louver and warm panel are at nearly the same temperature, their interaction will be neglected.

The warm panel of the cryopump lines the vacuum enclosure, so it will absorb all of the heat coming from the enclosure. The exchange between these surfaces is characterized by a low emissivity of about 0.1 for the inside surface of the enclosure, and a high absorptivity of about 0.9 for the outside of the warm panel, and a very high shape factor of at least 0.9.

The internal exchange between the warm panel and inlet louver, and the cold panel is the smallest component, particularly after the cryopump has been regenerated and the emissivity of the inlet louver is very low.

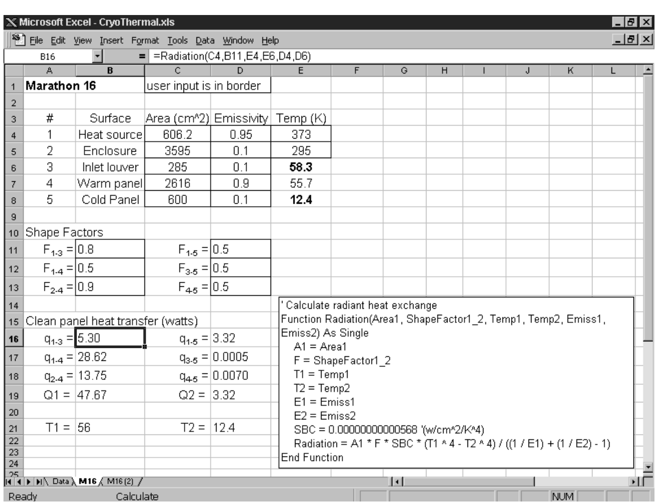



Figure 5 - Spreadsheet used to predict the thermal performance of cryopumps. The example is for the Marathon® 16 cryopump from IGC-APD Cryogenics Inc.

SPREADSHEET SOLUTION

Having broken the cryopump into its thermal components, it is now possible to calculate the effect of the heat source on the pump. APD has created a spreadsheet for this purpose that also includes the performance information for different cryogenic refrigerators used in the cryopumps. The program user enters dimensions of the pump and emissivity information about the interacting surfaces. The operating temperatures of the pumping surfaces are calculated by using Equation 2 from above.

Figure 5 shows the Microsoft® Excel worksheet in which the calculations are made. The user is required to enter data in the cells that are highlighted with a border. The program then calculates the heat transfer between the individual elements identified by the variables $q_{n-m}$ in the lower left side of the spreadsheet. From these values, the heat load on the first stage of the refrigerator, Q1, and the second stage, Q2, are calculated. Then the operating temperatures, T1 and T2, of the first and second stages respectively are calculated by using known relationships between the heat loads and the temperatures of the refrigerator. Finally, the operating temperatures of the pumping surfaces are calculated from the heat load on the surface and the performance of the thermal connection to the refrigerator heat stage. The temperature of the inlet louver is shown in cell E6 in the spreadsheet, and the temperature of the cold panel is calculated in cell E8. Remember that we want the inlet louver to be colder than about 120K and the cold panel no warmer than about 20K for best performance in most applications.

In Figure 5 cell B16 is selected and the formula to calculate the load from the heat source to the inlet louver is shown in the editing toolbar at the top. The formula is written as a user defined macro and the Visual Basic for Applications™ code for the macro is listed in the text box in the lower right corner of the window. The macro takes values for the area, shape factor, temperature, and emissivity as arguments and calculates the heat transfer by Equation 2.

CONCLUSIONS

Cryopumps can be used very effectively in high temperature process applications if the system is properly designed. We know at what temperature the cryopump must operate in order to do its job effectively, and that to be within the operating range, the cryopump cannot be exposed to more than several watts of radiant heat. If hundreds or thousands of watts are radiating through the chamber, then provisions must be made to intercept and sink the heat before it reaches the inlet to the cryopump.

APD has an accurate method to predict the effect a high temperature heat source will have on the operation of a cryopump. If the analysis indicates that the heat will cause the operating temperatures of the pumping surfaces to exceed critical values, then the cryopump user can decide to employ thermal baffles to shield the pump inlet.

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