

Chapter 2

Vacuum Chamber Design

Abbreviations

AFM	Atomic force microscope
EB	Electron beam
Hz	Hertz cycles per second
K	Thermal conductivity
L/s	Liters per second
NO/NC	Normally open/normally closed
PVD	Physical vapor deposition
RCF	Rolling contact fatigue
RF	Radio frequency
RGA	Residual gas analysis
RPM	Revolutions per minute
Scm	Standard cubic centimeters per minute
VAC	Volts alternating current

A discussion of mechanical testing in high vacuum is presented in this chapter. Vacuum component selection and test chamber materials are discussed with emphasis on what is needed for testing high speed rotating systems in vacuum similar to a rotating anode inside an x-ray tube device. Low speed testing of rotating systems to detect balance shift is also discussed with emphasis on vibration transmittance through key vacuum components. Two case studies are explored that highlight the balance between cost and complexity related to chamber design and its long-term operation.

2.1 Introduction

The first step in designing a vacuum chamber is to determine how the chamber will be used and to have a clear understanding of the processes that will be carried out inside the chamber. The designer must be aware of the risks for contamination to the vacuum chamber, and how will the inside of the chamber be checked for cleanliness prior to starting a process. Some additional design consideration includes what types of coatings are present inside the chamber during the test, at what temperature, and at what pressure will the test be carried out. Will the process involve high voltage, that is, application of high-voltage bias, during the process? Without a clear understanding of the intended use and fine details of the test, you may inadvertently purchase equipment that is improperly sized for your needs. Worse yet, you may purchase incorrect components which will require costly rework of the equipment.

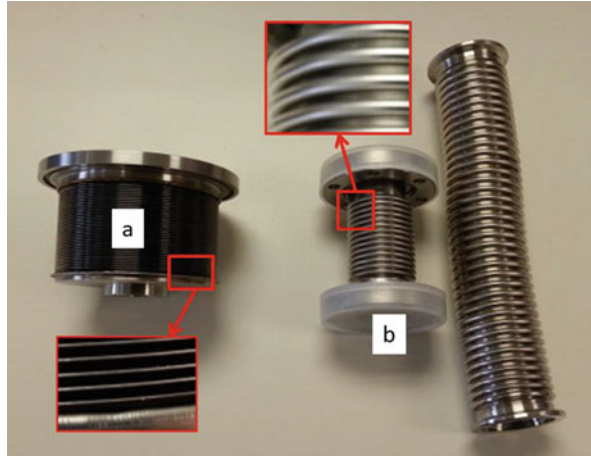
Maintenance costs, along with system performance, need to be considered when selecting vacuum chamber components. For example, a cryogenic vacuum system is very clean and requires little hands-on maintenance. However, it does require uninterrupted facility power to run the helium compressor, usually 480 VAC 3 phase, for example, even when the chamber is vented to the atmosphere. The cold head of the cryogenic system is not at all tolerant of hydrocarbon contamination. Oil and even low vapor pressure vacuum grease can destroy the cold head of a cryogenic pumping system. In comparison, a turbo pumping system can provide the same pumping speed but with a smaller foot print than a cryogenic system. Turbo pumping systems are less affected by hydrocarbon contamination, but are at risk in the event of a rapid venting process. Turbo pumps may be severely damaged by rapid pressure changes within the chamber and may fail catastrophically if the chamber gas load is too high. Each pumping system has its advantages and disadvantages, and the designer needs to decide what level of risk is tolerable concerning their process and who will operate the equipment.

A proper organization and arrangement of components inside the chamber can reduce contamination risk and shorten maintenance time. For example, tight corners and layered structures inside the chamber should be avoided where possible. Layered structure such as shelves or closely spaced plates allows contamination and debris to collect without easy visual detection. All fixtures and components should be easily removable, requiring simple hand tools and little effort and skill for cleaning.

2.2 Vacuum Chamber Design Considerations

The pumps and reciprocating hardware associated with vacuum systems can generate significant mechanical vibrations that are often transmitted to the chamber. These systems require hermetic and structural connection between components

Fig. 2.1 Types of bellows: (a) welded, (b) hydroformed



which can increase vibration transmittance. However, if one separates the hermetic and structural requirements, a design solution may be achieved. For example, the welded-flexible bellows shown in Fig. 2.1a enables a hermetic seal with very little vibration transmittance, but does not support structural load. In comparison, the formed bellows of Fig. 2.1b is less flexible than the welded design and can support light to moderate loads and is not as rigid as a bolted gasket joint. The welded bellows may be used, for example, between the pump and chamber to minimize vibration transmittance from the pump. The pump may then be supported externally from the chamber to prevent vibration transmittance. For example, one novel idea from Iijima (1982) allows atmospheric pressure to offset the weight of the pump and removes fixed bracket attachments to the chamber, thereby minimizing vibration transmittance from the pump.

The stiffness of a bellows assembly may vary depending on how it was manufactured. The two most common manufacturing methods are hermetic welding and hydroforming, both shown in Fig. 2.1. The hydroformed bellows are much stiffer than the welded design and have some load carrying capability. In comparison, the hermetic-weld bellows offer the best vibration isolation but are not capable of supporting a load of any kind. In fact, the welded bellows requires external support fixtures to be effective. The welded bellow consists of a series of thin flat disks, usually at least 20, that are welded together at the inner and outer diameters. The flexibility of each disk provides the vibration isolation.

Chamber material and geometry should be chosen to minimize chamber outgassing time and maintenance. The surface finish of the chamber walls may influence the test inside the chamber. For example, engineers designing a chamber to carryout particle physics experiments similar to Baglin et al. (1998) would be concerned with how the chamber wall surfaces will interact with the electron cloud during their test. If electron-beam or radio-frequency heating is going to be used to heat test components, the surface finish and material of the chamber walls may

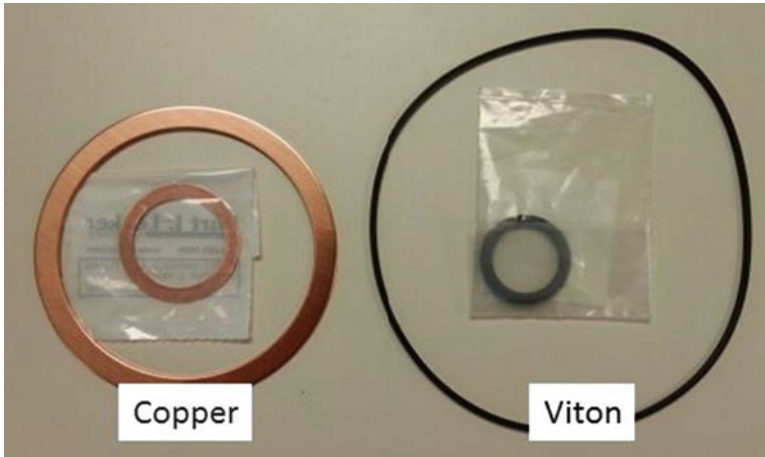


Fig. 2.2 Copper and Viton rubber gaskets

interact with those heating sources, which could result in unintended heating of the test sample inside the chamber or, worse, direct heating of the chamber itself.

Temperature isolation may also influence chamber material selection. For engineers designing a vacuum chamber furnace for a fluxless soldering process as in Lee et al. (2006), minimizing heating time and operation costs was most important. When applying heat inside a vacuum chamber, the primary heat dissipation path will be conduction through the support fixtures and connections to the chamber structure. A simple conduction calculation assuming heat flow through a flat plate $H = k\Delta T/L$ can give insight about how material choice and geometry will affect the test inside the chamber. For example, the thermal conductivity, (k), for metals is much larger than that for ceramics. If the process involves significant heating of 500 °C or more, then ceramic fixtures or even ceramic liners may be needed to protect the chamber against long-term damage. If heat is generated within the test hardware itself, in case of rotating internal friction, then chamber external connections should be isolated to allow use of less expensive and flexible Viton™ o-ring seals instead of rigid copper gaskets. A comparison of Viton and copper gaskets is shown in Fig. 2.2. One reason for using metal gaskets, copper, or silver-coated copper is to minimize outgassing time when the chamber is under high vacuum. Copper gaskets will outgas more quickly than Viton, and Viton gaskets will outgas much quicker than silicon gaskets, for example. Why is outgassing important? It is important because the gaseous molecules and atoms leaving the gasket material will interact with test sample while under vacuum.

Thermal isolation reduces total energy consumption for a test and prevents damage to heat-sensitive components that may be mounted on the chamber exterior. The researchers in Lee et al. (2006) were able to drive their sample temperature to 400 °C while maintaining chamber base exterior temperature of only 55 °C.

Another consideration in chamber design is determining what level of vacuum is needed for the test or process. It is significantly more work and a lot more expense to reach high-vacuum condition $<10^{-5}$ Torr than it is to test in the rough-vacuum 10^{-4} – 10^{-2} Torr range. Vacuum processes that occur in the 10^{-4} – 10^{-2} Torr range are commonly referred to as “milli-Torr” range vacuum. For example, if the process involves any type of voltage biasing or high-voltage application, then at least high-vacuum conditions will be required. At pressures in the milli-Torr range, there are sufficient number of gas atoms and molecules present inside the chamber to be ionized when high voltage is applied. The ionization is a result of leakage current through the gas during the test. Such leakage current paths can alter your test results and may cause significant damage to the equipment. On the other hand, if the process or test requires only an inert and evacuated chamber environment, that is, no high voltage and no heat, then rough vacuum in the 10^{-2} – 10^{-4} Torr range may be sufficient.

The focus of this chapter is vacuum chamber design for testing components used in high-voltage and high-vacuum devices. One such example is a rotating anode x-ray tube device. The chamber environment must be sufficiently outgassed to prevent ionization and leakage current during operation. For example, the chamber could be in the 10^{-7} Torr range, but may not be sufficiently outgassed for 120 kV operation. The chambers used for component testing should have the same operational characteristics as the end-use application.

2.3 Chamber Material Selection

The most common and recommended material for high-vacuum chamber applications is 304 stainless steel. Quick surface outgassing and resistance to surface oxidation are among the most beneficial aspects of using 304 stainless steel (304SS). However, this material can be difficult to machine and may require frequent tool and cutting bit replacement. Stainless steel chambers also transmit vibration to all attached components. Vibration from pumping systems and those transmitted from the floor can corrupt sensitive vacuum instruments as well as confound vibration measurements of the hardware under test inside the chamber.

2.3.1 *Material Outgassing*

Material surface outgassing characteristics should be considered when choosing vacuum chamber material. For example, 6061 aluminum is hermetic and light weight and will provide necessary structural support for a vacuum chamber application. However, due to its chemical composition, an aluminum surface will collect large amounts of surface moisture each time the chamber is opened to the atmosphere. If the process inside the chamber is sensitive to moisture, nitrogen, and

oxygen, the process should not be started until the chamber is sufficiently outgassed. The surface moisture will need to be removed each time the chamber is opened, resulting in long pump-down times and delays in starting the process. In contrast, 304SS collects very little moisture, thus allowing faster outgassing and improved process cycle time in a production environment. The outgassing characteristics for candidate chamber materials are tabulated in Mattox (1998).

2.3.2 Vacuum Welds, Gaskets, and Attachments

All welded joints that seal the chamber from atmosphere need to pass a leak rate test to $10^{-9} \frac{\text{mBar} \cdot \text{Liter}}{\text{Sec}}$ and should be constructed in such a way as to prevent trapped gas or accumulated debris. The weld must provide sufficient strength to support measurement instruments and also have flexibility to prevent fatigue damage after repeated evacuation and venting. Atmospheric pressure on the chamber during vacuum is 14.6 psi at sea level; therefore, large flat sections within the chamber should be avoided due to repeated bending and flexure at welded joints each time the chamber is pumped down. Corners and cavities must be properly vented or eliminated whenever possible. For example, a blind hole inside the chamber can trap gas if it is not properly vented. With the threaded fastener sealing off the opening of the hole, trapped gas will bleed into the chamber, resulting in a virtual leak and long pump-down time. Vented fasteners should be used on all bolted joint assemblies within the vacuum space of the chamber. Press-fit assemblies inside the chamber can cause virtual leaks as well if not properly vented. In general, make sure all components, fixtures, fasteners, and gaskets are properly vented before installing them on the chamber.

2.3.3 Vacuum Chamber Isolation

Vibration isolation from components outside the chamber may be achieved using a flexible bellows similar to the one shown in Fig. 2.1. For example, if vibration measurements are needed while the test hardware is under vacuum, the chamber will need to be isolated from vibration input from vacuum system components. The hardware required to maintain vacuum can be stiff and heavy, resulting in significant vibration transmittance to the test element(s) inside the chamber. If chamber components heat up during the test, then the vibrational characteristics of the chamber will change and could confound test measurement results. For example, if one wishes to track the peak vibration amplitude of a rotating system as it is heated under vacuum, then any vibration artifacts from the chamber must be accounted for in order to have confidence in the measurement. If temperature

isolation and control is required, the chamber system may require active cooling by forced convection or possibly temperature-controlled process fluid through the chamber structure.

2.4 Vacuum Components Selection

This section is intended as an overview of key components that are needed to support a high-vacuum test chamber. More detail on these components as well as many others may be found in vacuum technology textbooks such as by Mattox (1998) and O'Hanlon (1989). When selecting vacuum components, it is best to work with a brand-name equipment supplier who can help you to make the best choice.

2.4.1 High-Vacuum Pumping System

Cryogenic vacuum-pumping systems are among the cleanest in the industry for use for high-vacuum chambers. A cryogenic vacuum pump is hermetically sealed from outside atmosphere once the vacuum chamber has been evacuated to less than approximately 10^{-3} Torr. Another advantage of the cryogenic system is that there are no moving parts inside the vacuum space so that in the event of a catastrophic leak, or an accidental rapid venting of the chamber, the pumping system will not be damaged. In contrast, a turbo pump is a momentum transfer pumping system. An array of compressor blades attached to a rotor drum and bearing assembly is rotated to about 75 kRPM to force or evacuate gases out of the chamber. Both turbo and cryogenic pumping systems are capable of reaching high vacuum, 10^{-7} Torr. In general, turbo pumping systems are more economical and require less space than cryogenic systems.

2.4.2 Vacuum Measurement System Selection

Vacuum measurement systems are selected based on process sensitivity to pressure and on the method used to control chamber pressure. For example, a thin film PVD system may require chamber pressure control during deposition. Usually two gauge systems are required: one for high-vacuum 10^{-5} – 10^{-7} Torr and one for rough-vacuum 10^{-3} – 10^{-4} Torr. The gauge controllers from each system provide dry contact relay circuits to enable safety interlocks and process protection. The relay circuits are either normally open (NO) or normally closed (NC). These circuits are different from digital communications in that these relays interface well with a large variety of industrial equipment and do not require specialty connector hardware such

as RJ45 or RS-232, 485. Processes in the milli-Torr range, 10^{-3} – 10^{-2} Torr, require a self-heating manometer pressure sensor to give the most accurate pressure measurement during the process. For processes less sensitive to pressure, and occurring in the ultrahigh-vacuum regime such as 10^{-5} Torr or less, either a hot filament or cold cathode magnetron gauge would be sufficient.

2.4.3 *Vacuum System Safety Interlocks*

All vacuum subsystem components have internal safety interlock relays. Safety interlocks apply to three aspects of the chamber system operation: (i) safety of the operator, (ii) protection of the equipment, and (iii) protection of the test or part inside the chamber. Operator safety is the most important aspect of system interlock configuration. Vacuum system processes often bring together voltage and current in the ranges of 10–2,000 V and current as high as 15 amps. Turbo pumps rotating at 75 kRPM, as well as cryogenic pumps that cool to 17 K, may be very dangerous to an operator if they malfunction. Consideration of the rotating or friction-generating devices inside the test must be included as well when designing your chamber. What if the device is rotating very fast and fails, what is the containment capability of your chamber? What if the operator tries to open the chamber before the process is complete? Use of safety interlocks will enable safe operation of the chamber system in the event of operator error or impatience.

Protection of system equipment and the test parts inside the chamber is also very important. What if there is a power or other utility outage during the process? The objective is to bring the system to zero energy as fast as possible without damaging test components inside the chamber. If the chamber is vented during a high-heat situation of the process, it could damage the parts and components inside, rendering some instruments a complete loss if not properly interlocked with a control system. Figure 2.3 contains a safety interlock circuit that protects both operator and test rig. For example, if the RCF chamber is vented while the test is still rotating, the pressure interlock will open, which will shut down all systems immediately. If the system is running unattended and there the process water cooling flow stops, the system will shut down to prevent overheating. Referring to Fig. 2.3, the high-voltage bias heating to the RCF test rod is enabled only with the pressure threshold is below milli-Torr range, to protect the operator from injury while the chamber is open.

Presented next is a case study dealing with design and optimization of a vacuum chamber with heated and rotating components. Three chambers were designed, tested, and optimized for this application. Optimizing an existing chamber design enables one to focus on design aspects that will give the most appropriate solution for the problem under consideration. For example, if the structure and geometry of a tool or piece of equipment is considered expensive to build and to operate, one can use design optimization techniques to construct an objective function to track the cost of the tool or equipment over its projected operating life. In contrast, optimization on a design concept before it has been built is more challenging because all

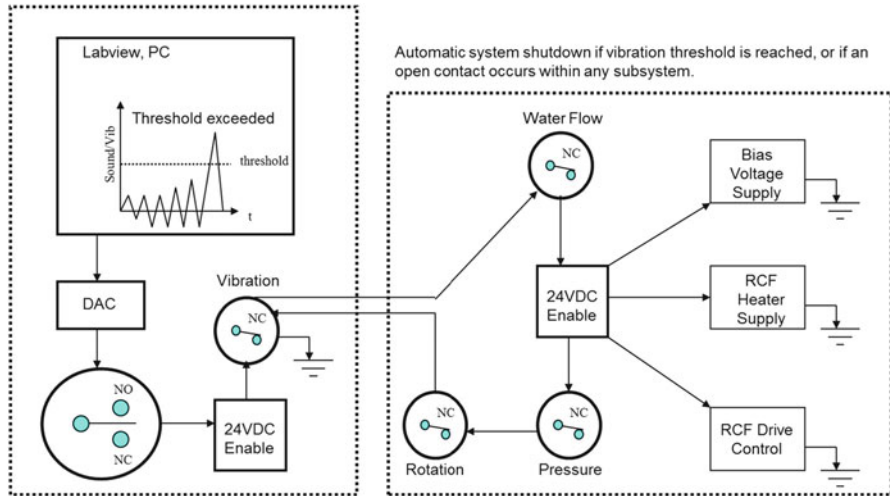


Fig. 2.3 Safety interlock system for a rolling contact fatigue test chamber

of the possible trade-offs are not yet established. The ideal situation is to apply optimization techniques to a prototype design and verify conclusions and assumptions with sub-element testing; then return to update and complete the optimization task using the lessons learned from the first tier-optimized prototype design.

2.5 Case Study: Chamber Vibration Isolation and Measurement

This section and the next present three vacuum chamber test rigs that were designed to measure vibration of rotating components in high vacuum and at high temperature. The designs were optimized using MATLAB Optimization Toolbox, for vibration isolation, cost of operation, and protection of the sensitive measurement tools attached to the chamber. Where ever possible, the systems were constructed using off-the-shelf components from vacuum hardware suppliers. As a requirement, the chambers must be assembled by engineers and technicians using common hand tools with basic mechanical laboratory skills. No specialized training is required, a key factor in keeping maintenance costs low.

2.5.1 Vibration Transmissibility Design

Testing in vacuum presents unique challenges not encountered when testing in air. Vibration isolation in vacuum can be particularly challenging due to the stiffness

Fig. 2.4 Test rig A: configuration of the drive, rotating, and table assemblies in air (Reproduced with permission from Rev. Sci. Instrum. 82, 105113 Copyright 2011, AIP Publishing LLC)

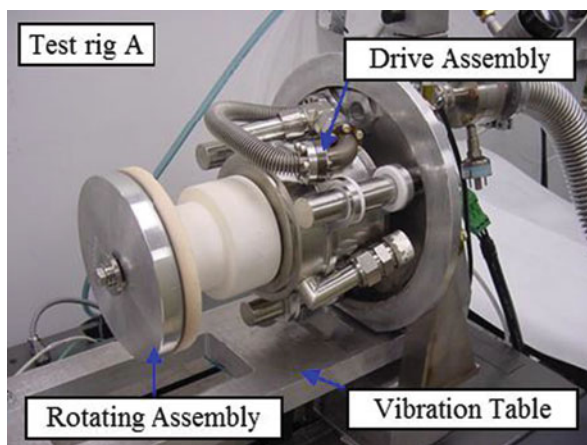
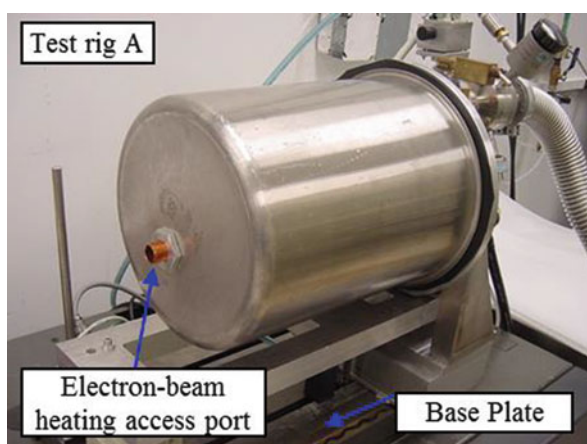


Fig. 2.5 Test rig A under vacuum with electron-beam heater port shown (Reproduced with permission from Rev. Sci. Instrum. 82, 105113 Copyright 2011, AIP Publishing LLC)



properties of materials such as 304 stainless steel. Three vacuum chamber systems are compared in this section. The first test rig, test rig A, is presented in Fig. 2.4 and includes a rotating aluminum–ceramic braze assembly. For this test, the brazed assembly is rotated at 15 Hz in vacuum while being heated to 200 °C.

Vibration data as a function of time and temperature is collected and used to assess the structural integrity of the brazed assembly. Heat is applied using a coiled tungsten filament at 5.5 amps and biased to 1 kV using a dc power supply. The biased coiled filament is an electron-beam heating system, with connection to the power supply through the copper tube as shown in Fig. 2.5.

Test rig A was also used to characterize a welded-flexible bellows assembly for application in the RCF high-vacuum test rig. The goal of this test was to measure vibration transmissibility through the bellows assembly and to compare it with a fixed-flange design. From Fig. 2.4, it can be seen that the drive and vacuum vessel assemblies are mounted independent of the rotating assembly and vibration table.

Fig. 2.6 Close-up of test rig A to point out the weld-flexible bellows assembly (Reproduced with permission from Rev. Sci. Instrum. 82, 105113 Copyright 2011, AIP Publishing LLC)

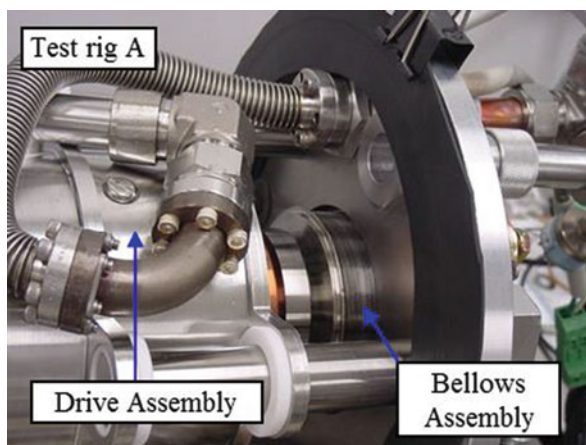
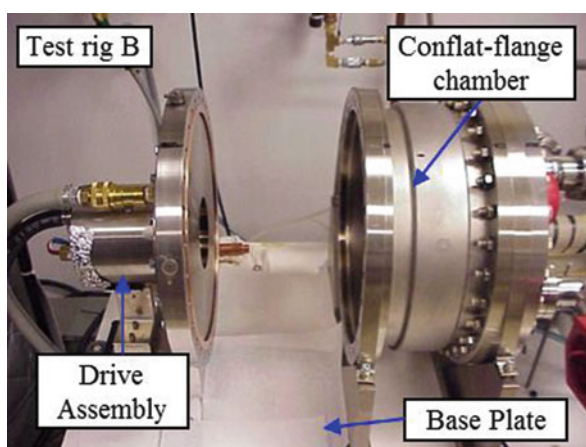


Fig. 2.7 Test rig B uses an off-the-shelf conflat flange joint vacuum chamber (Reproduced with permission from Rev. Sci. Instrum. 82, 105113 Copyright 2011, AIP Publishing LLC)



Specifically, the bearing on the rotating assembly is rigidly attached to the vibration table. The table is supported on two displacement sensing probes. The probes record displacement during rotation of the aluminum–ceramic component spinning in vacuum. The welded-flexible bellows is used to seal the vacuum system between vibration table and drive assembly. The bellows assembly is presented in Fig. 2.6. One end of the bellows is attached to the drive assembly, while the other end is connected to the mounting interface on the bearing of the rotating aluminum–ceramic part.

Test rig B is shown in Fig. 2.7 and does not include the welded-flexible bellows assembly. The rotating assembly is rigidly attached to the drive assembly, and therefore the vibration measurement is confounded with external inputs from the vacuum pumps. Accelerometers are used to measure the vibration transmission from the drive assembly of rigs A and B to the measurement table.

Fig. 2.8 Accelerometer locations on test rig A
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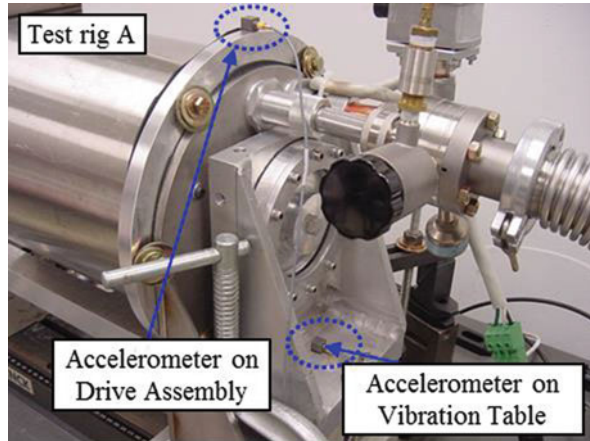
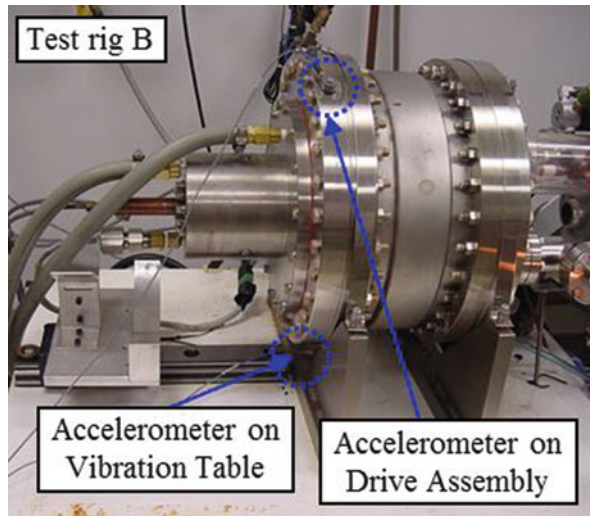


Fig. 2.9 Accelerometer locations on test rig B
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Figures 2.8 and 2.9 illustrate the accelerometer measurement locations for each test rig A and B. The location of each accelerometer on the test rigs A and B is the same with respect to the drive assembly and vibration table. The point of comparison between the two test rigs is the vibration transmittance between accelerometers for each rig.

The vibration transmittance may be modeled as a collection of springs and masses, one for each component of the test rigs. For example, Fig. 2.10 illustrates the connectedness of the rotating assembly, the drive assembly, and the vibration measurement table. An excitation force $F_2(t)$ originating from an imbalance of the rotating system is applied to the vibration table. The subsequent displacements of the vibration table and drive assembly, $x_2(t)$ and $x_3(t)$, are measured using the

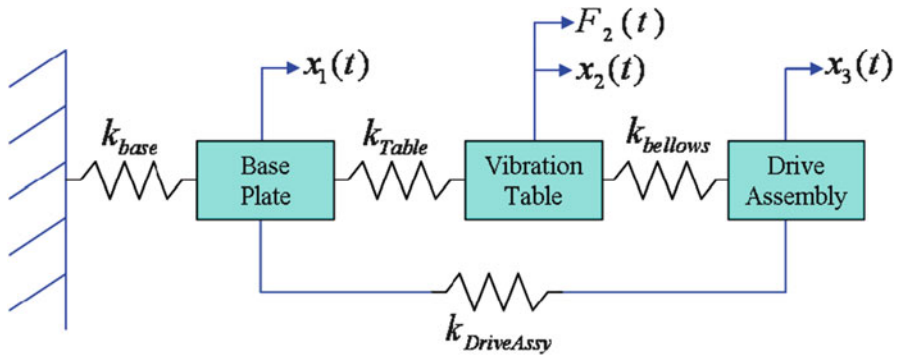


Fig. 2.10 A spring-mass system model illustrating the force transmissibility test for test rig A (Reproduced with permission from Rev. Sci. Instrum. 82, 105113 Copyright 2011, AIP Publishing LLC)

accelerometers presented in Fig. 2.8 and compared with the vibration response of Fig. 2.9.

For the spring-mass model shown in Fig. 2.10, two test conditions for vibration transmittance are considered: (i) in high vacuum and (ii) in air with the bellows detached from the vibration table. The stiffnesses, k_{Table} and $k_{DriveAssy}$, are nonzero for both test conditions since both the vibration table and drive assembly are present for each test condition (i) and (ii). The bellows stiffness, $k_{bellows}$, is nonzero only when vacuum is applied; otherwise it is not connected to the vibration table during non-vacuum test conditions. An analysis of the model in Fig. 2.10 suggests that if $k_{bellows}$ is very small during high-vacuum testing, then the responses $x_2(t)$ and $x_3(t)$ should not change or will change very little between conditions (i) and (ii). This is achievable using the welded-bellows design of test rig A shown in Fig. 2.6. In contrast, applying the model of Fig. 2.10 to test rig B of Fig. 2.7 results in a rigid connection between the vibration table and drive assembly. The stiffness between the vibration table and drive assembly is much larger than that of test rig A due to the presence of the thick and heavy conflat flange joints used in rig B.

2.5.2 Accelerometer Response Comparison

A comparison between accelerometer responses for test rigs A and B is presented in Fig. 2.11 through Fig. 2.14. Figure 2.11 contains a baseline vibration signature of test rig A presented under high vacuum with the rotating assembly installed but not rotating. No attempt was made to further isolate the vacuum system from the drive assembly and base plate. Instead, the external inputs from the vacuum pumps, 60 Hz from the roughing pump and 1,250 Hz from the turbo pump, provided a traceable and known input to the system in addition to the input of highest interest, a 15 Hz excitation from the rotating assembly. The goal of the accelerometer tests

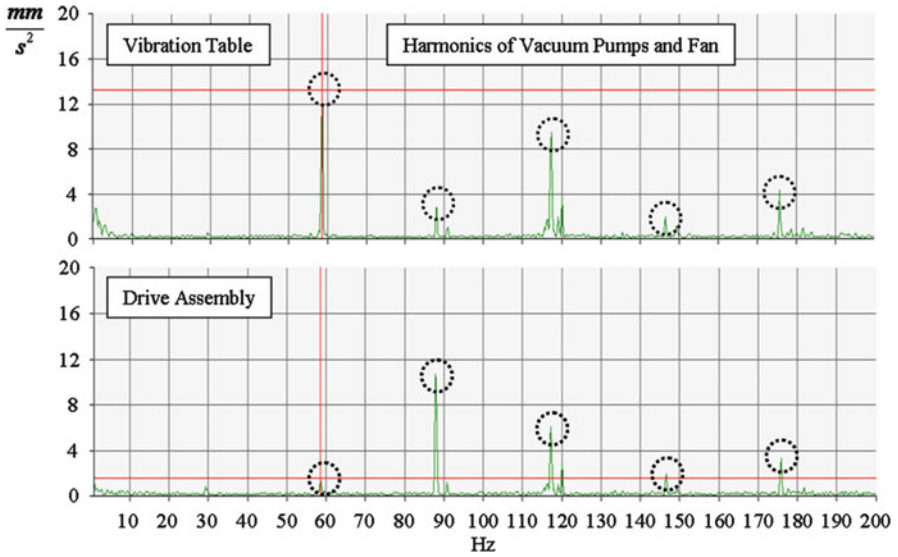


Fig. 2.11 Baseline response of the vibration table and drive assembly of test rig A under vacuum with the rotating assembly installed but not rotating (Reproduced with permission from Rev. Sci. Instrum. 82, 105113 Copyright 2011, AIP Publishing LLC)

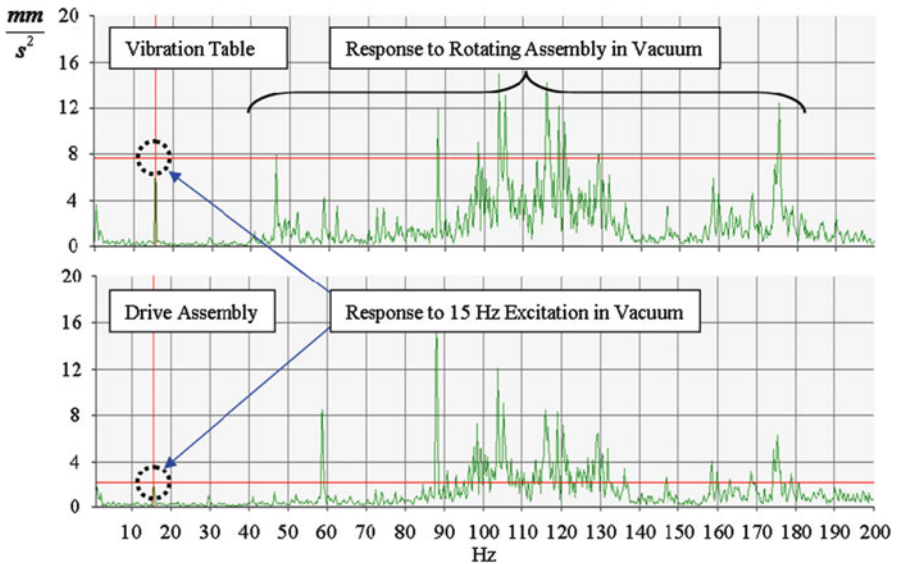


Fig. 2.12 Accelerometer response of the vibration table and drive assembly from test rig A. Response to a 15 Hz excitation from the rotating assembly at $8.6\text{E-}7$ Torr (Reproduced with permission from Rev. Sci. Instrum. 82, 105113 Copyright 2011, AIP Publishing LLC)

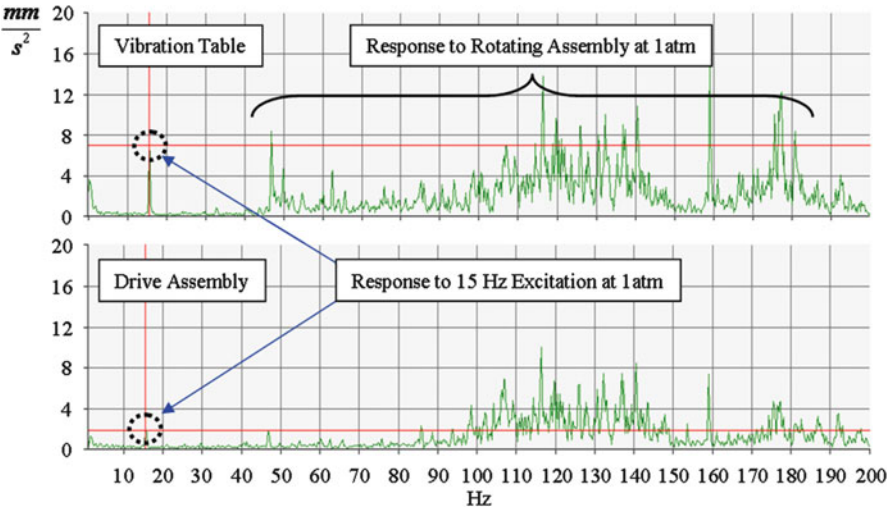


Fig. 2.13 Accelerometer response of the vibration table and drive assembly from test rig A. Response to a 15 Hz excitation from the rotating assembly at 760 Torr (Reproduced with permission from Rev. Sci. Instrum. 82, 105113 Copyright 2011, AIP Publishing LLC)

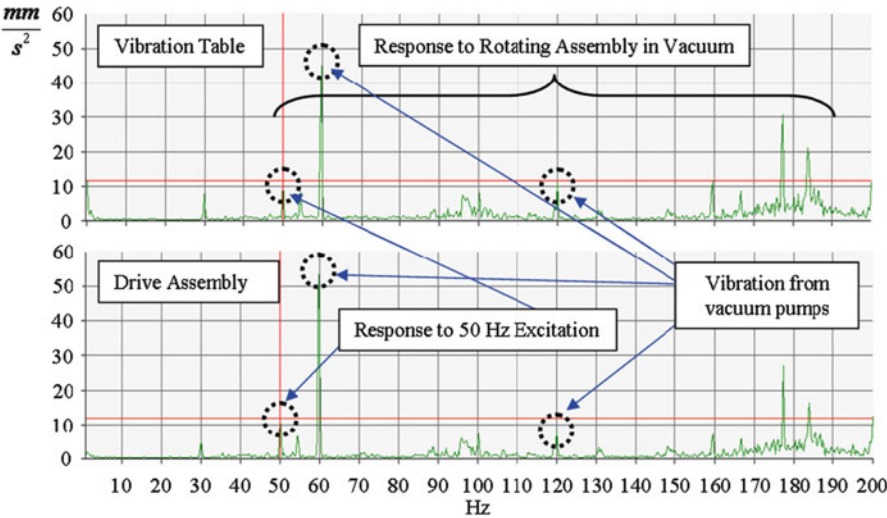


Fig. 2.14 Accelerometer response of the vibration table and drive assembly from test rig B. Response to a 50 Hz excitation from the rotating assembly at 3.6E-7 Torr (Reproduced with permission from Rev. Sci. Instrum. 82, 105113 Copyright 2011, AIP Publishing LLC)

was to check if the welded bellows would in any way alter the vibration response to the vibration table while the system is under vacuum. A cyclic force input is generated by applying an unbalance to the rotating assembly and then rotating it at 15 Hz. The nature of this force input is representative of the type of vibration characteristic to be detected under heat and vacuum conditions using these test rigs.

The accelerometer response of the vibration table and drive assembly under vacuum are presented in Fig. 2.12. Similarly, Fig. 2.13 has the accelerometer response of the vibration table and drive assembly at atmospheric pressure. Concerning Figs. 2.12 and 2.13, the values to compare are the acceleration magnitudes of the vibration table at 15 Hz under both vacuum and non-vacuum conditions. The acceleration responses of the vibration table and drive assembly for the vacuum and non-vacuum tests are nearly the same at 15 Hz. These results indicate that use of the welded bellows does not significantly influence the response of the vibration table while operating under vacuum. Specifically, the presence of the bellows with vacuum applied does not enable cross talk between the vibration table and drive assembly. For comparison, this result is equivalent to a very small spring constant, k_{bellows} , in Fig. 2.10 such that unwanted excitation from the drive assembly and its mass does not influence the response of the vibration table, $x_2(t)$. For comparison, the same rotating assembly was tested in test rig B and the results are presented in Fig. 2.14. Comparing the acceleration magnitude of Figs. 2.12 and 2.13 with that of Fig. 2.14 suggests difficulty in resolving low amplitude vibration. The 15 Hz input unbalance was not detectable in test rig B. Instead, the rotation speed of the rotating assembly was increased until the vibration magnitude response matched with that of test rig A. Specifically, the speed of the rotating assembly was increased to 50 Hz before reaching the magnitude response of test rig A. These results suggest that the mass and stiffness of the conflat flanges used in test rig B preclude sensitive and low amplitude vibration detection. Concerning test rig B in Fig. 2.7, it is impossible to isolate the vibration table from the drive assembly using thick-walled conflat-flange-type attachments. These results suggest that a welded-bellows assembly should be considered in the design optimization of the next test rig.

2.6 Case Study Continued: Optimum Chamber Design

Test rig C was built to accommodate larger diameter rotating assemblies and to enable operation at higher temperatures. This design incorporates conflat flange joints with the isolation aspects of the welded-bellows assembly used in test rig A. The new chamber, test rig C, is an extension of the chamber design of test rig B with inclusion of the welded bellows from test rig A. The singular assembly that was optimized for test rig C was the main chamber shown in Fig. 2.15. The chamber design has four aspects: two welded bellows, tubular interface attachments, tubing wall thickness, and chamber wall thickness.

As presented in Fig. 2.15, a conflat flange joint is used at one end of the chamber to enable removal and installation of the rotating assembly. During a test, the rotating assembly may be heated to 600 °C while the vibration response of the vibration table is measured over time. With the rotating assembly at 600 °C, the exterior of the chamber may reach 200 °C due to radiative heat transfer inside the chamber. Hard copper gaskets are required for all conflat joints due to

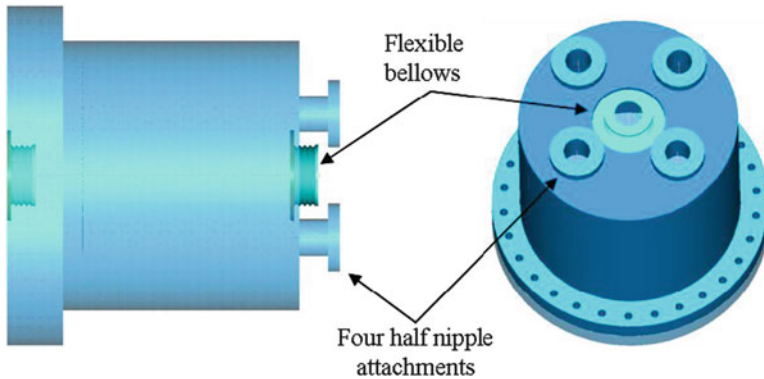


Fig. 2.15 Design optimization of chamber section of test rig C. The design couples the welded-bellows aspect with conflat flanges to enable vibration isolation at high temperature under vacuum (Reproduced with permission from Rev. Sci. Instrum. 82, 105113 Copyright 2011, AIP Publishing LLC)

the heating conditions of this test rig. The rotating assembly is driven by a hermetically sealed drive assembly similar to the one used on test rig A in Fig. 2.4. Two flexible bellows are used in the new design to enable more test configuration flexibility.

2.6.1 Design Optimization Analysis

The new chamber is made from 304 stainless steel (304SS), and all permanent seams are welded with no long-term gasket joints. Fabrication with stainless steel is expensive, and therefore, the amount of 304SS used should be minimized. The cost of cooling the chamber will be minimized through optimization of geometry and heat conduction through the half nipple attachments in Fig. 2.15. The interface attachment tube walls conduct heat from the chamber to the interface attachments. The design requirement concerning the half nipple attachments is that they cannot exceed room temperature, approximately 20 °C, to protect and isolate temperature-sensitive measurement equipment from heat damage and leakage. The conductive heat transfer through the attachment tubes will be minimized within the constraints that the tube section of the attachments be able to support a 25 N cantilever load with an end deflection less than 0.1 mm. This requirement applies equally to all four tubes so that any instrument or pump can attach at any interface flange.

The front surface of the chamber in Fig. 2.15 must not deflect more than 3 mm, and the chamber walls around the circumference must not deflect more than 1 mm. It is desirable that the entire chamber system has a first natural frequency greater than 600 Hz or three times the anticipated rotational speed of the rotating assembly. The cost of materials, fabrication, and operation to be minimized is given as

$$f(X) = \text{Cost}_{\text{metal}} + \text{Cost}_{\text{welding}} + \text{Cost}_{\text{operation}}. \quad (2.1)$$

The cost of the stainless steel for the chamber and tube interfaces is assumed constant. The cost of the stainless steel weld is assumed to be \$5/cm due to weld prepping of the surfaces and the required leak check after the weld is complete. The heat transferred to the chamber during the test emanates from the rotating assembly inside. Cooling costs per unit surface area of attachment tubes are based on use of one 120 VAC fan. Labor cost for assembling the test rig is not included in the optimization; rather, it is assumed that the test rig will be assembled by a staff engineer or laboratory technician with basic knowledge of hand and shop tools. The labor costs associated with assembling the rig are included in the overhead costs of the facility.

The design constraints for the problem will first be stated and then translated into mathematical form using established equations and expressions found in Young (1989), O'Hanlon (1989), and Incropera and DeWitt (1996). The design constraints are:

1. Chamber diameter is fixed at 36 cm.
2. Length of the chamber is at least 30 cm.
3. Chamber wall thickness strong enough to maintain pressure of 10^{-8} Torr.
4. The chamber first natural frequency must be greater than 600 Hz.
5. Force and displacement transmissibility of rotating part attachments less than 0.1.
6. Deflection of the front plate less than 0.1 mm but should be as small as possible.
7. Diameter and length of attachments on front plate must satisfy minimum molecular flow conductance requirement for a 60 L/s pump. Pumps and instrumentation are allowed to mount on any of the four attachments.
8. Wall thickness and length of the front plate attachments must not conduct heat to half nipple flange interfaces.
9. The attachments on the front plate must be capable of supporting up to a 25 N load, the nominal weight of instrumentation such as pyrometers, analyzers, and laser probes.
10. The first natural frequency of the attachments must be greater than or equal to that of the chamber, 600 Hz.

Constraints 1 and 2 are classified as size constraints and 3 through 10 may be classified as material and behavioral constraints. Expressions and references for material and behavioral constraints 3 through 10 are established in the fields of mechanical engineering and applied physics and are presented below. The mathematical form of the optimization problem is stated as,

$$\min f(X) = \rho(A_1 L_1 + A_3 L_3 + A_2 t_2) \text{Cost}_{\text{material}} + \pi(2d_1 + 8d_3) \text{Cost}_{\text{weld}} + h \times \text{Cost}_{\text{cooling}} \quad (2.2)$$

subject to the constraint expressions in Eqs. 2.3, 2.4, 2.5, 2.6, 2.7, 2.8, 2.9, 2.10, 2.11, 2.12, 2.13, 2.14, 2.15, and 2.16 for the chamber shown in Fig. 2.15. Constraints g_1 through g_3 relate to radial and hoop stress and deflection of the chamber:

$$g_1 = p \frac{d_1}{t_1} - \sigma_y \leq 0 \quad (2.3)$$

$$g_2 = \frac{p \left(\frac{d_1}{2}\right)^2}{Et_1} \left(1 - \frac{\nu}{2}\right) - \text{cdif} \leq 0 \quad (2.4)$$

$$g_3 = p \frac{d_1}{2t_1} - \sigma_y \leq 0. \quad (2.5)$$

Constraints g_4 and g_9 account for the first bending mode natural frequency of the chamber section and for each attachment tube, respectively. Since the rotating device inside the chamber may have an unbalance, or could develop an unbalance during testing, it is good practice to design for at least $2\times$ the frequency response of the chamber system to internal excitation:

$$g_4 = \text{freq} - \frac{6.93}{2\pi} \sqrt{\frac{EI_1 g}{wL_1^3}} \leq 0 \quad (2.6)$$

$$g_9 = \text{freq} - \frac{1.73}{2\pi} \sqrt{\frac{EI_3 g}{wL_3^3}} \leq 0. \quad (2.7)$$

Constraints g_5 and g_6 relate to front plate stress and bending deflection. With the chamber evacuated, under vacuum, the external pressure on the chamber is 14.6 psi. The load on the chamber is opposite that of a pressure vessel. For the present case study, the chamber must not deflect during heating:

$$g_5 = \frac{p \left(\frac{d_1}{2}\right)^4}{64D} \frac{(5 + \nu)}{(1 + \nu)} - \text{pdif} \leq 0 \quad (2.8)$$

$$g_6 = 6 \frac{\left(\frac{d_1}{2}\right)^2}{16} \frac{(3 + \nu)}{t_2^2} - \sigma_y \leq 0. \quad (2.9)$$

Limits on bending and deflection of instrumentation attachments is accounted for using constraints g_7 and g_8 . The attachments need to support the weight of instruments as well as maintain vacuum integrity for the life of the test:

$$g_7 = \frac{2wL_3^2}{6EI_3} - \text{adif} \leq 0 \quad (2.10)$$

$$g_8 = \frac{wL_3}{I_3} \left(\frac{d_3}{2} - t_3\right) - \sigma_y \leq 0. \quad (2.11)$$

Constraints g_{10} , g_{11} , and g_{12} relate to hoop stress and radial wall deflection in the half nipple attachment tubes. For some test applications, the wall thickness of an attachment tube may need to be minimized to allow optimal passing of magnetic

fields through the wall. For example, if electron-beam heating through one of the half nipple attachments is needed, then external magnets are used to guide the electrons through nipple. The magnets are typically attached to the exterior of the tube so the magnetic fields must pass through the tube wall:

$$g_{10} = p \frac{d_3}{t_3} - \sigma_y \leq 0 \quad (2.12)$$

$$g_{11} = p \frac{d_3}{2t_3} - \sigma_y \leq 0 \quad (2.13)$$

$$g_{12} = \frac{p \left(\frac{d_3}{2}\right)^2}{Et_3} \left(1 - \frac{\nu}{2}\right) - adif \leq 0. \quad (2.14)$$

The constraints g_{13} and g_{14} track molecular flow and heat transfer in the attachment tubes. Molecular flow through the attachment tubes will determine system pump-down time and may limit recovery during chamber outgassing. In fact, there may be a decade difference in pressure between the pump and the chamber if the attachment tube is too small. In addition, the larger diameter tube attachment will increase convective heat flow to the ambient air and will assist with chamber cooling:

$$g_{13} = \text{pump} - 7.66 \times 10^5 \frac{\pi d_3^2}{4(1-pr)} pr^{0.712} \sqrt{1-pr^{0.288}} \leq 0 \quad (2.15)$$

$$g_{14} = T_2 + (T_1 - T_2) \times \exp\left(-L_3 \sqrt{\frac{h \times pc}{k \times ac}}\right) - T_2 \leq 0. \quad (2.16)$$

Equations 2.3, 2.4, 2.5, 2.6, 2.7, 2.8, 2.9, 2.10, 2.11, 2.12, 2.13, 2.14, 2.15, and 2.16 may be found in Young (1989), O'Hanlon (1989), and Incropera and DeWitt (1996). While optimizing the chamber parameters, be sure that you can justify your optimization analysis with equations from a good reference to take advantage of previous work of others to improve the design of your chamber.

The calculated variables in Eqs. 2.3, 2.4, 2.5, 2.6, 2.7, 2.8, 2.9, 2.10, 2.11, 2.12, 2.13, and 2.14 are defined as

$$I_1 = \pi \left(\frac{d_1}{2} - d_1\right)^3 t_1, I_3 = \pi \left(\frac{d_3}{2} - d_3\right)^3 t_3, D = \frac{Et_2^3}{12(1-\nu^2)}, \quad (2.17)$$

$$ac = \pi \left(\frac{d_3}{2} + t_3\right)^2 - \pi \left(\frac{d_3}{2} - t_3\right)^2, pc = \pi d_3, A_1 = \frac{\pi}{4} [(d_1 + 2t_1)^2 - d_1^2], \quad (2.18)$$

$$A_3 = \frac{\pi}{4} [(d_3 + 2t_3)^2 - d_3^2], A_2 = \frac{\pi}{4} d_1^2 - 4\pi d_3^2. \quad (2.19)$$

The associated design vector, $X = \{L_1, t_1, t_2, L_3, t_3, d_3, h\}$ for Eq. 2.2, contains seven independent design variables. The design variables are defined as L_1 and t_1 , the

length and wall thickness of the chamber, and t_2 , the thickness of front plate section. The variables L_3 , t_3 , and d_3 are the length, wall thickness, and diameter of the attachment tubes, respectively. The heat transfer coefficient h applies to the half nipple tube attachments only. The bounds on the design variables were chosen with observation of readily available flanges, nipples, and tubes. Based on a survey of off-the-shelf components from leading vacuum hardware retailers, the bounds on the design variables were chosen as

$$\begin{aligned} 300 \leq L_1 \leq 390, \quad 1.5 \leq t_1 \leq 5, \quad 1.5 \leq t_2 \leq 5, \quad 50 \leq L_3 \leq 150, \\ 1.5 \leq t_3 \leq 5, \quad 30 \leq d_3 \leq 70, \quad 80 \leq h \leq 150. \end{aligned} \quad (2.20)$$

All lengths in Eq. 2.20 have units of mm, and h has units of $\frac{\text{W}}{\text{m}^2 \cdot \text{K}}$. The bounds on the thicknesses t_1 , t_2 , and t_3 were chosen with respect to weldability and manufacturability range of material thicknesses for 304 stainless steel tubing and flat stock. For this analysis, the following fixed material and system parameters were used:

$$\begin{aligned} d_1 = 350 \text{ mm}, \quad g = 9.81 \frac{\text{m}}{\text{s}^2}, \quad k = 14 \frac{\text{W}}{\text{m} \cdot \text{K}}, \quad w = 25 \text{ N}, \quad E = 210 \text{ GPa}, \quad \nu = 0.29 \\ \sigma_y = 280 \text{ MPa}, \quad p = 100.6 \text{ kPa}, \quad \text{freq} = 600 \text{ Hz}, \quad \text{pump} = 60 \frac{\text{L}}{\text{s}}, \quad pr = 0.52, \\ T_1 = 200^\circ \text{C}, \quad T_2 = 20^\circ \text{C}, \quad cdif = 1 \text{ mm}, \quad pdif = 3 \text{ mm}, \quad adif = 0.1 \text{ mm}. \end{aligned} \quad (2.21)$$

The parameter pr is the ratio of the vacuum pump pressure to chamber pressure. The parameters $cdif$, $pdif$, and $adif$ are the chamber radial deflection, end plate deflection, and attachment deflection limits, respectively. The parameter p in Eqs. 2.3, 2.4, 2.5, 2.6, 2.7, 2.8, 2.9, 2.10, 2.11, 2.12, 2.13, 2.14, 2.15, 2.16, 2.17, 2.18, 2.19, 2.20, and 2.21 accounts for atmospheric pressure at sea level in kilopascals applied to the external surfaces of the chamber and nipple sections. The cost parameters in the objective function of Eq. 2.2 are defined as

$$\text{Cost}_{\text{material}} = \$2.39/\text{kg}, \quad \text{Cost}_{\text{weld}} = \$500/\text{m}, \quad \text{and} \quad \text{Cost}_{\text{cooling}} = \$7.52/\text{h}. \quad (2.22)$$

The cooling costs for this rig are a function of the electricity used to operate the cooling fan. Defining h in Eq. 2.20 as a design variable allows flexibility within the optimization problem to reach the lowest cost cooling solution using fans. The program does not consider the details of the heat transfer coefficient, such as surface area, heat flow, and temperature differential, etc. Rather, the program considers the amount of h that is needed to meet the design constraints. The cost per unit h was calculated based on 8,000 h of testing at an electricity cost rate of \$0.877/kWh.

The optimization problem presented in Eqs. 2.2, 2.3, 2.4, 2.5, 2.6, 2.7, 2.8, 2.9, 2.10, 2.11, 2.12, 2.13, 2.14, 2.15, 2.16, 2.17, 2.18, 2.19, 2.20, and 2.21 contains nonlinear constraints and is therefore classified as a nonlinear programming problem. This problem is solved using a sequential quadratic programming algorithm found in Rao (2009). The basis of quadratic programming algorithms is to solve the

Table 2.1 Results for three optimization options for new system, test rig C

	1. fval = \$1049.20		2. fval = \$837.81		3. fval = \$983.74	
	Upper bound	<i>Medium fan</i>	Upper bound	<i>Small fan</i>	Upper bound	<i>Medium fan</i>
		Freq. minimum		Freq. minimum		Freq. minimum
		$3 \times \textit{run speed}$		$3 \times \textit{run speed}$		$2 \times \textit{run speed}$
L_1	390 mm	300 mm	Unchanged	300 mm	Unchanged	300 mm
t_1	5 mm	1.5 mm	Unchanged	1.5 mm	Unchanged	1.5 mm
t_2	5 mm	4.72 mm	Unchanged	4.72 mm	Unchanged	4.72 mm
L_3	150 mm	145.52 mm	170 mm	170 mm	Unchanged	150 mm
t_3	5 mm	1.5 mm	Unchanged	1.5 mm	Unchanged	1.5 mm
d_3	70 mm	70 mm	90 mm	81.28 mm	Unchanged	55.71 mm
h	150 W/ m ² K	111.43 W/ m ² K	Unchanged	81.65 W/m ² K	Unchanged	104.87 W/ m ² K

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Lagrange multiplier problem directly with application of the Kuhn-Tucker conditions for optimality. The optimization calculations were carried out in MATLAB using the *fmincon* function in the Optimization Toolbox. The *fmincon* function requires that the system of constraint equations be continuous and differentiable within the upper and lower bounds of the constraint variables. At each iteration, the Kuhn-Tucker conditions are solved through calculation of the second order Hessian matrix. If all equations are quadratic, a solution will be reached within two iterations. Due to the non-quadratic nature of the constraint equations, numerical Hessian updating is needed, and the optimum solution was located after 12 iterations.

2.6.2 Compare Optimum Designs

Results of the top three optimization runs are presented in Table 2.1 as Options 1, 2, and 3. Option 1 is the optimum solution using the initial upper and lower bound constraints on the design vector as given by Eq. 2.20. Option 2 is the optimum solution using a slightly modified upper bound vector, as indicated in Table 2.1. Option 3 is the optimum solution using the same upper and lower bounds as Option 1, but with a chamber first natural frequency limit of 400 Hz instead of 600 Hz.

An inspection of the program output for Option 1 indicates that the inequality constraints 5, 9, and 14 are active at the final design. For example, constraint 5 influences the front plate thickness, and constraints 9 and 14 control the attachment tube first frequency and the half nipple attachment convection coefficient, *h*, respectively. Constraints 4 and 13 are very far from influencing the final design as was indicated by their large negative values from the program output compared with other constraints in the program output.

The final cost of fabricating the chamber parts and operating the new test rig is mostly influenced by the attachment tube geometries. The cost of off-the-shelf flanges and other parts common to both test rigs B and C have been excluded from the results in Table 2.1. The cost of common parts for all designs is about \$2,000.00. Two competing constraints on the attachment half nipple structure were stiffness and heat conduction in the tube walls. For example, increasing the wall thickness will increase the first natural frequency, but it will also increase the heat conduction from the chamber and ultimately require more forced convection cooling which will drive up the cost.

The cooling costs associated with test rig B when testing a rotating assembly at 600 °C were compared with the results in Table 2.1 for the new rig, test rig C. The cooling fan costs alone for test rig B were more than double the cooling costs of the options presented in Table 2.1. Test rig B requires two large cooling fans during operation in order to meet the 20 °C temperature constraint on the half nipple attachments. The cooling cost of test rig B is \$15.04/h for comparable ranges of h presented in Eq. 2.20 due to the necessity of the two large fans instead of one medium-sized fan for test rig C. The operating costs of the designs presented in Table 2.1 are about half the operating cost of test rig B based on 8,000 h of testing.

Referring to Option 2 in Table 2.1, the program reached a compromise by increasing the half nipple attachment tube diameter and length and effectively raising the first natural frequency while only slightly raising tube heat conduction and length and improving forced convection efficiency via increased surface area. However, this option is problematic in that larger attachment tube diameters could result in accessibility issues among measurement instrument attachments. Option 2, though least expensive to build and operate, could restrict measurement hardware attachment accessibility and necessitate custom attachment interfaces on the instruments. Option 3 is less expensive than Option 1, but it also has a lower first natural frequency capability, and the diameter of the half nipple attachments is close to the minimum desirable. The reduced frequency capability of Option 3 may render the rig obsolete. In comparing all three options, Option 1 meets all the upper bound requirements as outlined in Eq. 2.20 without any modifications.

2.7 Assembly and Cleaning

The manufacturing history of all vacuum components can have a profound influence on the process results. For example, if the welded joints within the chamber were fabricated using dirty tools, including trace elements of cutting or grinding tool metals, oils, paint, lint from clothing or clothes, etc., then the chamber could outgas for months, thereby corrupting the vacuum space inside. As a best practice, only lint-free towels and high purity alcohol should be used to wipe down the inside of the chamber and components. Oil-free gloves and hairnets should be worn at all times when the chamber is open. One piece of hair, glove, or even finger prints can cause excessive pressure in the chamber, which will delay the process during pump down.

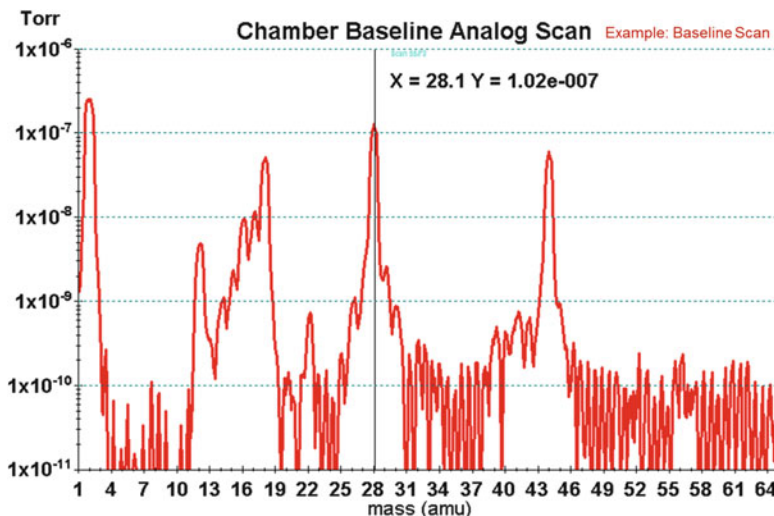


Fig. 2.16 RGA scan of chamber with oxygen (16 amu) and nitrogen (28 amu) present, and hydrocarbon or oil contamination (44 amu)

2.8 Residual Gas Analysis

Residual gas analysis (RGA) is an excellent chamber diagnostic tool for commissioning a new system. RGA should also be used after any major work has been done to the chamber or its components. For example, during a complete system teardown and reassembly, there are lots of opportunities for contamination. Conducting an RGA scan and then comparing it to the baseline scan, when the chamber was new, will put to rest any suspicions about the cleanliness, or it will at least confirm that the chamber is still contaminated.

The SRS100 RGA system is used in Chaps 6 and 7 to track exiting argon gas from the coating layers during the ion-plating process. Using RGA, one can track the concentration of argon present in the system to confirm correct operation of the argon mass-flow controllers. The vacuum chamber for the rolling contact fatigue (RCF) test rig in Chap. 6 uses RGA to assess outgassing during run-in of the silver-coated balls. The RGA tracks the amount of argon that has been squeezed out from the silver coating during run-in. When all of the argon is gone, the silver on the surface of the balls has been sufficiently run-in.

2.8.1 Establishing Chamber Baseline

A residual gas analysis of a chamber similar to Figs. 2.9 and 2.15 is presented in Fig. 2.16. The data is presented as a frequency spectrum of mass elements in atomic

mass units (amu). With this spectrum, one can deduce what gases are present in the chamber under vacuum. For example, there are three peaks of interest in Fig. 2.16, and the 28 amu peak indicates that nitrogen N_2 is present in the chamber. The other two peaks of interest indicate that there is oxygen present as well, $O_2 \rightarrow 16$ amu, and some form of oil contamination showing up as a hydrocarbon molecule at $C_nH_n \rightarrow 44$ amu. If the process is sensitive to nitrogen, oxygen, or oil, the presence of these contaminant molecules and gases in sufficient quantity would corrupt and alter the deposition process. If the hydrocarbon molecules are present during the RCF test, they could influence the fatigue life of the silver coating. The amount of each gas present correlates with the partial pressure of the chamber. Partial pressure is indicated by the height of each peak given in units of Torr. The sum of all peaks is the total pressure inside the chamber.

2.8.2 Helium Leak Check

Prior to operation of the chamber, a leak check using helium gas should be done to confirm that there are no slow leaks in the system. Helium gas is used because it is the second smallest molecule on the periodic table, second to hydrogen. Literally, the small helium molecule and/or atoms will seep through any gaps within the chamber structure. It is important to detect and repair leaks in the system as soon as possible. For example, even if the chamber reaches “good” vacuum, to 10^{-6} Torr range, a very small leak in the chamber will allow contaminant gas to corrupt the process inside, even though the base chamber pressure is sufficiently low. Large pumping capacity, for example, will hide the presence of a leak. This is of particular concern with cryogenic systems which would require more frequent regeneration if a small leak was present, but would maintain “good” vacuum regardless.

2.9 System Commissioning

After assembly, cleaning, leak check, and RGA baseline, the system is ready for commissioning. Vacuum systems are not all the same, and slight differences in control connections, grounding, and cooling can make each system unique with respect to process recipe. Documentation, operating instructions, and data logs should be reviewed before the system can be trusted for use. More detailed best practices may be found in Mattox (1998); the reader is encouraged to consult this text as well when designing and commissioning any vacuum chamber application.

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