Large diameter cryogenic seals

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Cryogenic vacuum seals are discussed with emphasis on large diameter reusable connections. Both theoretical considerations and experimental results are presented. The main subject is large diameter cryogenic vacuum seals in a horizontal vessel, where thermal distortions are encountered in particular during the cool down stage.
We show a large diameter stainless steel flange design that can take large repeatable thermal shocks and maintain a good vacuum seal at liquid helium temperatures.

Introduction

This work is the summary of the design and initial test stage of the cryogenic system of a gravitational radiation detector to be built at the Nuclear Physics Department of the Weizmann Institute of Science, Rehovot.

The cryogenic system of this detector, now under construction at Ricor Ltd., En Harod Ihud, includes a horizontal vessel, approximately 600 mm I.D., designed to store liquid helium at temperatures between 1.8–4.2 K. The system also includes additional vessels at smaller diameters, and about 20 cryogenic connections such as pipes, electric feed-thrus, liquid level detectors etc. with diameters between 20–50 mm.

The system design emphasizes easy access and disassembly of all parts on one hand, and long range reliability on the other. These requirements, and the operating temperature led to the choice of indium wire for the seal material, and 304N, 316N stainless steel for the vessel and flanges. The considerations involved in these choices will be presented here.

The general approach is based on the practical experience gathered in the operation of liquid helium cryostats of similar dimensions and operating temperatures over the last seven years at the superconducting electron linac project of Stanford University, California.[1]

On this design we introduce a few modifications, which are intended to improve the performance, and which are based on the observation of the sealing properties of Stanford University's cryostats.

The case of a horizontal cylindrical vessel, sealed by end caps which are connected through flanges to the vessel, is illustrated in Figure 1. This design seems common enough in chemical, food and cryogenic industries, however the analysis of stresses and deformations under thermal shock conditions cannot be found in the engineering literature.

This case is characterized by the large gradient of the circumferential temperature distribution, described in Figure 2, achieved during the initial fill-up period, when the cold liquid wets the bottom of the vessel. We can expect large asymmetric thermal stresses and deformations mainly at the early stage of the cool-down, when temperature differentials and linear expansion coefficients are large.

For the theoretical case of a cylindric vessel having the temperature distribution given in Figure 3, we get maximal bending stress given by

$$\sigma_{b\,\max} = \frac{E\alpha\Delta T_{\max}}{2(1-\nu)} \tag{1}$$

and longitudinal compressive stress on the inner side of

$$\sigma_{l\,\max} = \frac{3E\alpha\Delta T_{\max}}{4} \tag{2}$$

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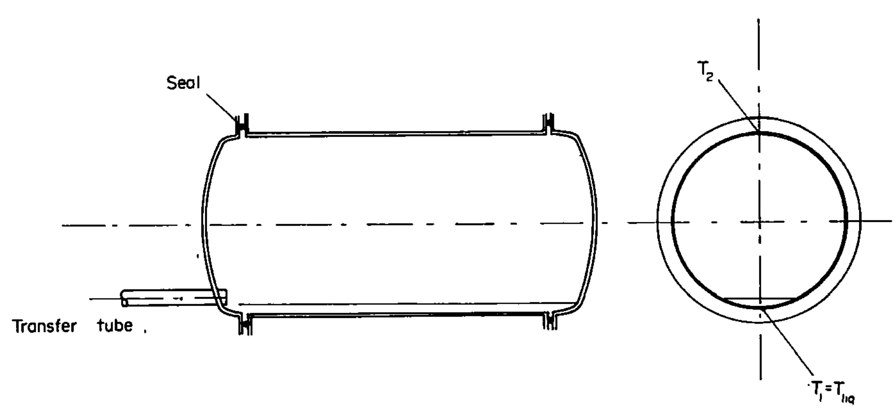

Figure 1. A schematic horizontal vessel for cryogenic fluid storage.

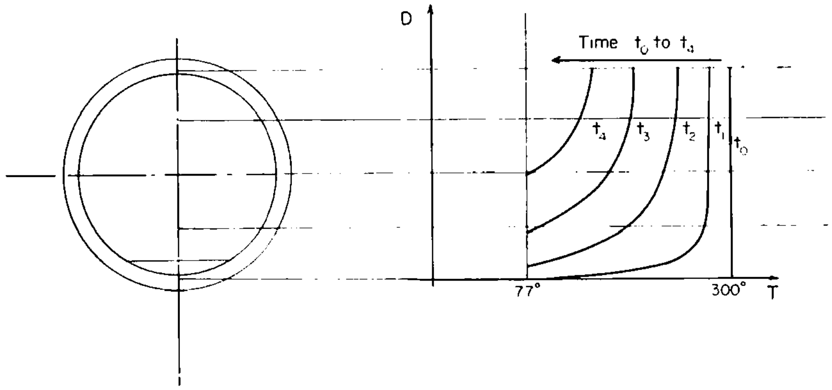

Figure 2. Circumferential temperature profiles showing progressive stages of LN_2 transfer.

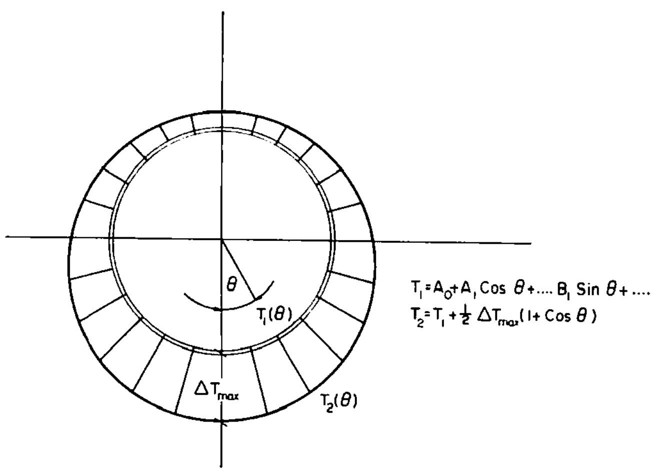

Figure 3. Circumferential temperature profiles according to Ref. 2.

and longitudinal tensile stress on the outer wall

$$\sigma_{l\,max} = \frac{E\alpha\Delta T_{max}}{4} \qquad (3)$$

In these equations E is Young's modulus, α is the linear expansion coefficient, ν is Poisson's number, and ΔT is the local temperature difference.

These stresses will produce bending deformations in two planes and asymmetric twisting in the connecting flanges.

On top of that we have to take into account other deformations which endanger the seal as well. The unequal thermal inertia of the flange-cover assembly and flange-body assembly, which may produce shearing motion between them due to unequal rate of cooling. Residual stresses stored in the flanges after machining and welding can produce deformations which appear later, either at room temperature or during cool down.

The absolute value of all these deformations is in direct proportion to the dimensions of the vessel. Usually it is not practical to keep the ratios between the various dimensions of the vessel parts constant while we increase the overall size. Therefore we have to find methods to insure dependable seals as we increase the size from small vessels to larger ones, where dimension ratios become worse as far as the deformations are concerned.

Design and Manufacturing considerations

Here we would like to present some of the considerations used in the design of the flange assemblies and seals.

(a) The visco-elastic approach. To solve the two main problems, deformations and low operating temperature, we selected indium wire as the sealing medium. This is the viscous member in the approach. Indium has a relatively low sealing pressure, good rheological properties down to cryogenic temperatures,[3] good cohesion to the seal surfaces (in particular when the indium and seal surfaces are free of oxides), and low price when it is reused (in house wire extrusion is quite simple). Limitations of flatness and roundness in the manufacture of large flanges should be the guide in the choice of the wire diameter. After compression the indium must spread over a large enough strip to cover surface defects, and compensate for mismatches at the same time. In our case the strip is designed to cover 8 mm.

The elastic member of the seal system is a conic disc spring—Bellevile spring—which is machined on the flange, and can be seen in Figure 4 which describes the flange structure.

The purpose of this spring is to provide a uniform and constant pressure on the indium through all the assembly, cool-down and operation stages. Its dimensions were calculated

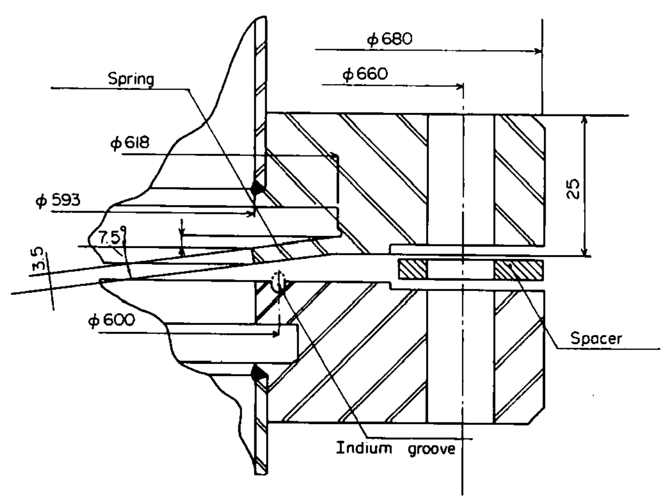

Figure 4. Sealing flanges detail.

to provide a force of 50 kg per centimeter of contour, independent of the flanges coupling method and force. When we have the ratio

$$\frac{\text{settlement}}{\text{initial elevation}} \approx 1$$

(as it is in our case after the flanges are tightened), this spring has the nice property of a nearly constant clamping force, which is not sensitive to small deformations.[4]

In addition we checked the maximal stresses at the edge and root of the Bellevile spring under the conditions of the planned deformation and found them to lie deep in the elastic range of the flange material.

(b) Flange material and production method. The experience gathered at Stanford showed clearly that the sealing surfaces must be of the highest quality, free of local defects or manufacturing deformations. Stanford's cryostats were made of 5083 aluminium, and this made them somewhat sensitive to scratches and hits during assembly and disassembly. Another problem was deformation of the machined flange during welding to the vessel or cover. With these considerations in mind, we decided to produce the vessel and flanges from stainless steel types 304LN, 316LN. The purpose of the nitrogen stabilized stainless steel is to keep the magnetic permeability properties over the designed temperature range.[5] The manufacturing will be done in the following order: initial turning of the flanges, welding to a sleeve, stress relief at 350° for 8 h, final turning down to 32 μm surface quality, and finally shaping of the Bellevile spring. The flange plus sleeve will then be welded to the vessel, and the length of the sleeve will protect the flange from the heat generated by the welding process. This method protects the finished flange from deformations since it is impossible to turn the whole vessel on a lathe to take a final cut off the flange.

To reduce the danger of local damage at the seal surfaces due to handling of the heavy vessel and end caps, these surfaces were located near the inner edge of the flanges.

(c) Flange coupling. At Stanford the flanges were tightened with a V clamp (Marman V clamp MVL64146-2600). We prefer the use of stainless steel bolts, in order to improve the uniformity of the clamping force distribution. The number of the bolts has been determined according to the approach given in reference 6. The force should be 50 kg cm^{-1} at the bolt position, and no less than 35 kg cm^{-1} circumference at midpoint between two bolts. The total number of bolts designed for each flange pair is 24, bolt size 3/8 in., torque 12 ft. lb., and the total clamping force is 20,600 kg. This force designed to provide 2800 kg for pressure differences on the cover, 9400 kg for the nominal 50 kg per cm sealing force, and a reserve force of 8400 kg to prevent relative motions during the cool-down process.

(d) Indium wire problems. Since we are dealing with relatively heavy equipment of large dimensions, we considered a number of details of the equipment operation whose significance was shown by the past experience. One of these is the problem of positioning the indium wire in its groove on the cover, and moving the cover to the vessel without moving the wire from its place, and without producing a relative lateral motion between the flanges. To solve this problem, we take the following steps: the covers have handles and a suspension point for a hoist. The vessel flanges have centering pins to guide the cover flange. The dimensions of the wire groove are designed to prevent the wire from getting loose when the cover is tilted from the hoirizontal to the vertical position (the first position is for fitting the wire into the groove, the other for assembly). The indium wire, 2 m long and with a cross section shown in Figure 5 we produce in house using a hand operated hydraulic press.[7] This method has three advantages—recycling of the indium, producing wires of unavailable diameters, and oxide free 'fresh' wire, which adheres best to the flanges.

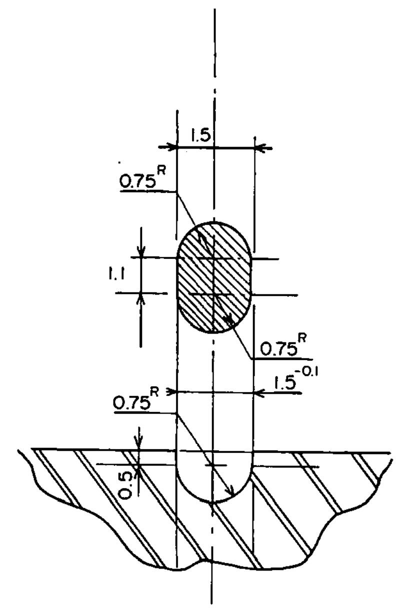

Figure 5. Indium wire cross section and positioning groove.

(e) The circumferential heat conduction: The poor heat conduction of stainless steel increases the temperature differences at the initial cooling stage. Although we believe that even so stainless steel is a better choice than aluminum, and the experiments described in the next section justify this, at very large diameters we may have to improve the conduction. We are considering two alternatives—soldering copper braids in a groove around the flange circumference, and a dynamic method, injecting the cryogen tangentially into the groove at a velocity which will provide centrifugal coupling of the liquid along the whole perimeter.

Sealing experiment

To test some of the considerations mentioned in this communication, we installed a pair of flanges in a top loading cryostat. The diameter of the helium vessel of this cryostat (Figure 6) is just 385 mm, therefore the flange size, the spring size and the number of bolts were scaled down appropriately, to conserve the dimensionality. Four copper-constantan thermocouples were connected to the flanges as shown in Figure 6, to provide data on the cooling rate and temperature profiles in the flanges during cool down. For a continuous sealing quality inspection, the vacuum jacket of the cryostat was connected to a Veeco MS-9 helium mass spectrometer, and the exhaust of a helium gas source placed at the bottom of the helium vessel.

The test was made at four stages.

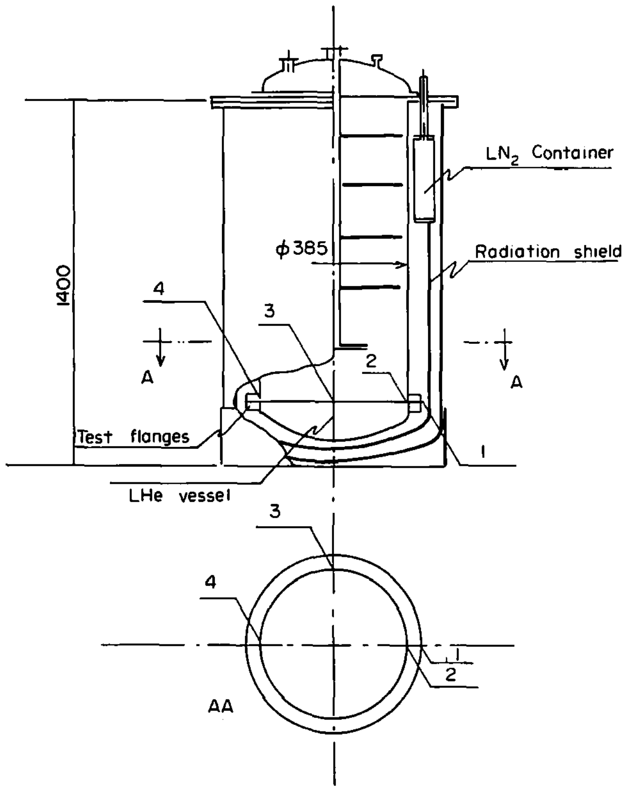

Figure 6. Experimental cryostat for seal test, and the positions of the thermocouples.

(a) Vertical cryostat (uniform circumferential temperature distribution), and cooling rate 2–3 K min^{-1}. Cooling was done by pouring liquid nitrogen on the bottom, and controlling the cooling rate by control over the liquid level. Following the cool-down the cryostat was warmed up, taken apart for inspection, and then reassembled.

(b) Vertical cryostat, cooling rate 5–7 K min^{-1} down to 80 K, and fast warm up by hot air jet.

(c) Cryostat tilted at 45°, and fast cool-down, with the liquid nitrogen running down on the vessel wall and wetting the flanges initially at one point. This situation is very close to the case of a horizontal cryostat. The cooling rate at the sensor points is given in Figure 8.

(d) Test at liquid helium temperature. One does not expect any problems at the cool-down from liquid nitrogen to liquid helium temperature, (small temperature difference, small linear expansion coefficients and slower cooling rate). However for completeness sake, we went through two complete cycles down to 4.2 K.

In parallel to the large seal test, we made tightness tests for two types of seals which are planned for the small diameter cryogenic vacuum connections. The purpose of these tests was to find a quick connecting reusable seal. The first type is a telescopic vacuum seal, VAT telescopic sealing ring NW25,[8] which is based on an indium silver alloy, and is specified as reusable at room temperature. The other type is a seal based on a Virgin Teflon wire, 1.6 mm in diameter. The test set-up of these seals is given in Figure 9. Each of these seals have undergone three cycles of immersion in liquid nitrogen and warm-up to room temperature while continuously leak tested. The telescopic seal was taken apart and reassembled before each cycle, to check its reusability. Finally both seals were tested for leaks at liquid helium temperature.

Results

(a) Helium vessel flanges and indium seal—no leak was detected at a sensitivity of 2×10^{-10} atm cc/s^{-1} through all cooling cycles, including inclined dewar test and liquid helium temperature test.

(b) A check of the uniformity of the Bellevile spring, made between the first two cool down cycles, has shown a maximal deviation of 0.5 mm. This result is particularly promising since the flanges were not stress relieved thermally during the manufacturing stages.

(c) The telescopic seal proved completely reliable through all cool-down cycles, including thermal shock and operation at liquid helium temperature. This seal was proved to be reusable under these conditions as well.

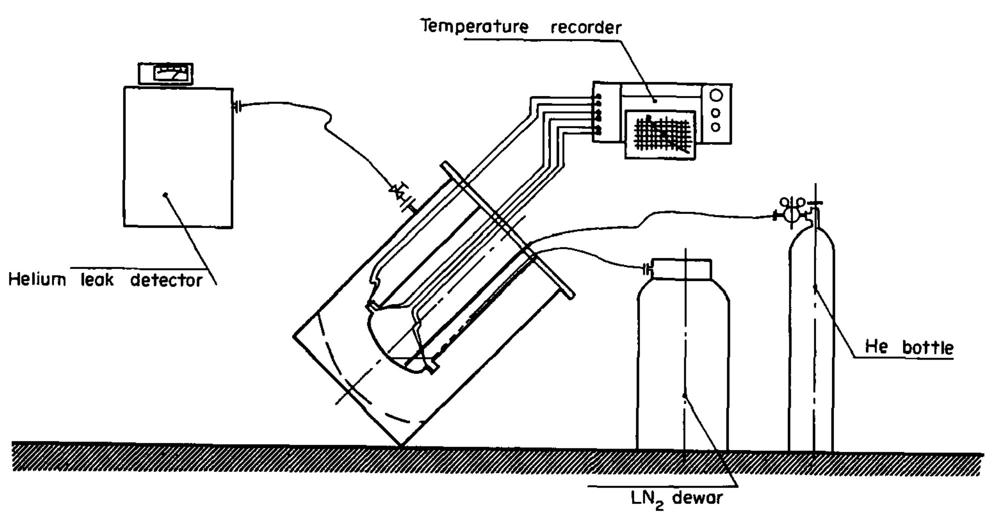

Figure 7. Set-up for the seal and cooling rate test.

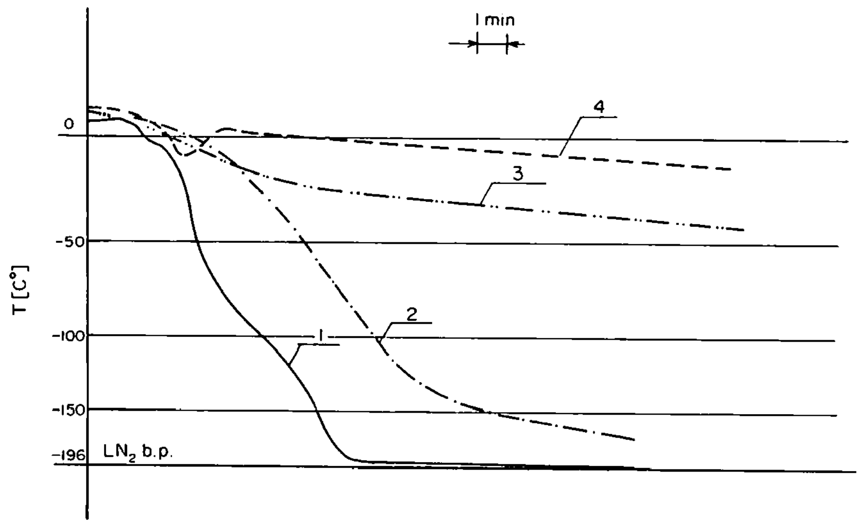

Figure 8. Flange temperature profiles of the tilted cryostat (tilt angle 45°) at initial cool-down.

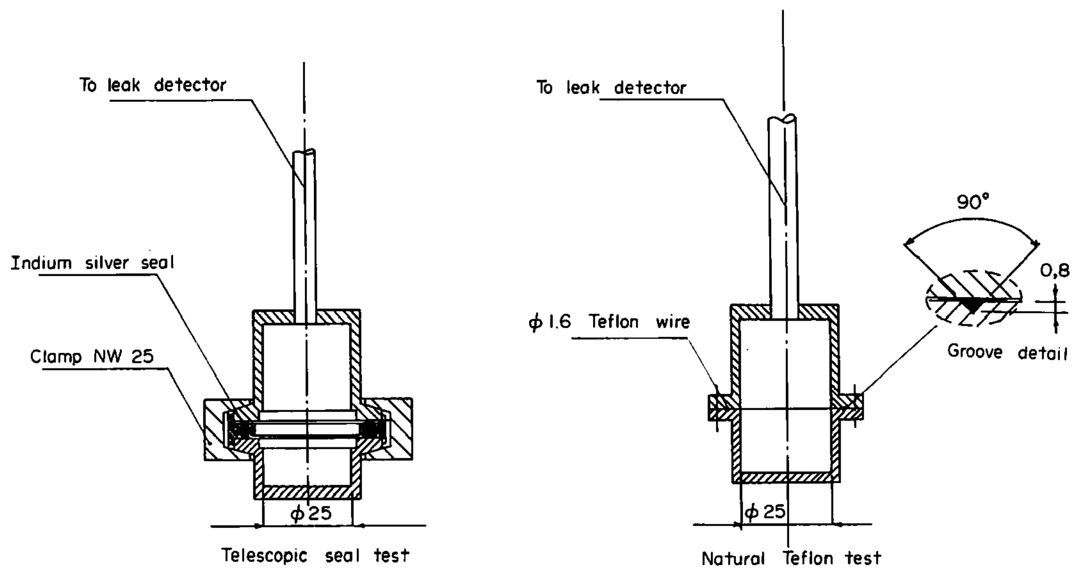

Figure 9. Details of the cryogenic seal tests of a telescopic seal and Teflon wire seal.

(d) the virgin Teflon wire was used similarly to indium wire, (including overlap of the wire ends), and was leak tight at all test stages.

Summary

Although the large diameter seal tests were made at 66% of the required flange diameter, the choice of its relative dimensions and the good results obtained can be considered as justifications for the design of the large vessel.

According to the test there is no need to provide additional circumferential heat conduction, but this point should be checked again at the final dimensions, and one of the methods given in this communication may be used as seen fit.

For the small diameter cryogenic vacuum seals the use of the telescopic seal seems reasonable for diameters below 40 mm, and virgin Teflon or indium wire for the larger diameter.

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