REVIEW OF THE OXFORD CRYOCOOLER

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ABSTRACT

The Oxford Cryocooler incorporates a linear drive compressor operating close to resonance. All dynamic seals are non-contacting clearance seals maintained by mounting the piston and displacer on mechanical suspension systems with infinite fatigue life. The displacer is pneumatically driven but controlled by a miniature linear motor. The cooler is therefore non-wearing and performance can be maintained even in adverse environments by servo control of piston and displacer strokes and relative phase. The cooler was originally intended for spaceflight applications but has since been developed for military applications and there are lower cost versions for civil commercial applications. Many different coolers have been developed. Split and integral, single- and two-stage coolers have been produced with operating temperatures between 30K and 200 K, refrigeration powers between 50 mW and several watts and capable of operating in ambient temperatures from -40°C to 70°C. A current project aims to extend the refrigeration power to 500 watts at 80 K. Experimental optimisation techniques have been devised for rapid development of high efficiency coolers. Recently there has been considerable progress in the design, manufacture and assembly methods for linear motors, clearance seals and mechanical suspension systems.

INTRODUCTION

Just over 10 years ago, a research and development programme was started in the Department of Engineering Science, Oxford University, aimed at producing a very long life, high reliability cryocooler. Initially, the programme was dedicated to the development of a cooler for the Improved Stratospheric and Mesospheric Sounder (ISAMS) experiment being prepared in the Department of Atmospheric Physics. Since the emphasis was on very long, maintenance-free operating life allied with high efficiency, the choice of a Stirling cycle machine was inevitable. Since a minimum of two years operating life in space was required the decision to use non-contacting clearance seals throughout the cooler was also taken at an early stage. Various methods of suspending the reciprocating parts were investigated so as to maintain non-contacting dynamic pressure seals but it was necessary to concentrate on just one because of the limited resources which were allocated to the project. The decision to use a

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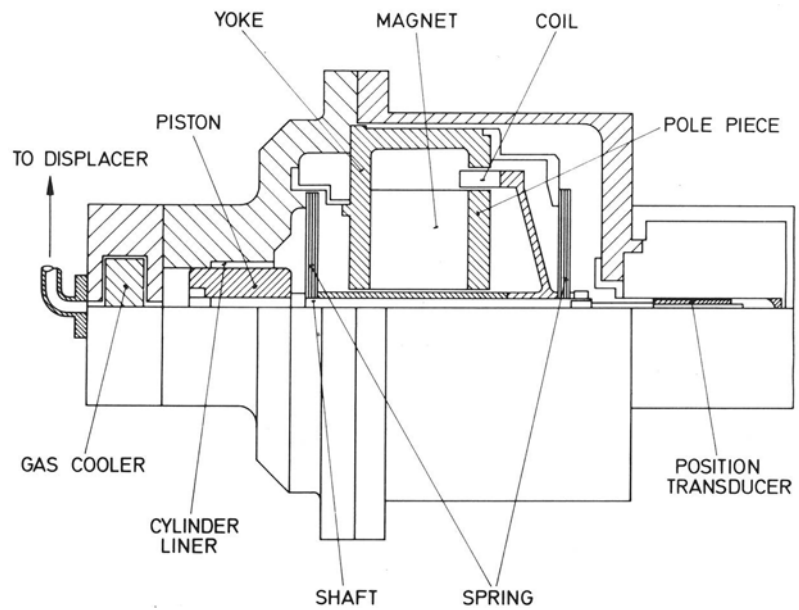

Fig. 1. Schematic diagram of compressor.

mechanical suspension system was due to its inherent compactness and simplicity, but also because the Department of Atmospheric Physics already had experience in the application of such systems

Once a non-wearing prototype had been demonstrated which had the appropriate performance and an acceptable format for the ISAMS experiment, the detailed development of the cooler for the experiment was taken over by the Department of Atmospheric Physics. The Department of Engineering Science then started a much more extensive programme to develop cryocoolers using clearance seal technology for a wide variety of applications. The original technology was suitable for the production of small quantities of high cost units with a refrigeration capacity of around one watt at 80 K. It was not applicable to much smaller or much larger coolers, those which needed to operate in harsh environments or coolers which needed to be produced at low cost or in large quantities. The continuing programme is aimed at making this type of technology widely available in any cryocooler application where very long life operation is desirable.

DESIGN FEATURES

The main features of a compressor for a Stirling cycle cryocooler are shown in Fig. 1. It is a reciprocating compressor driven by a simple linear motor directly coupled to the piston. The motor consists of a permanant magnet and soft iron flux path giving a large magnetic flux density in an air gap occupied by a coil. The coil is rigidly connected via a shaft to the piston and the whole system oscillates when alternating current flows in the coil. The compressor operates close to resonance on the spring provided by the working fluid. The piston is supported on two sets of springs attached to the piston shaft. The springs have considerable radial stiffness which ensures that the piston moves in a precise linear fashion even in the presence of large radial forces. Each spring is of a flat disc, spiral arm design so that sufficient numbers can be stacked to meet the requirements of a given application.

The mechanical suspension system maintains a radial clearance between

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piston and cylinder so that there is no contact and no wear. The magnitude of the clearance is determined by the criterion that the power loss due to gas leakage through the clearance seal is only a few percent of the total power required by the cooler. For coolers with a refrigeration capacity of a few watts at 80 K, the radial clearance is around 0.01 mm. It is also necessary to ensure that the axis along which the piston moves coincides with that of the cylinder axis. Typically, the suspension system design defines the piston motion so that it deviates radially, at full stroke, by less than one quarter of the radial clearance gap.

In early designs, the radial stiffness of the suspension system was such that contact between piston and cylinder was possible intermittently under environmental vibration which was typical in some applications. A hard piston was therefore used, together with a relatively soft cylinder liner chosen for thermal expansion compatibility and good wear characteristics. This arrangement has been retained in current designs because it has proved extremely resistant to damage caused by the presence of very small particles which are difficult to eliminate even if stringent cleaning procedures are used together with a clean room assembly environment.

A typical displacer unit for a small cryocooler is shown in Fig. 2. The displacer with integral regenerator is mounted rigidly on a shaft. The shaft passes through a clearance seal which separates the working volume from a volume containing gas close to mean cycle pressure. The displacer assembly is mounted on a mechanical suspension system similar to that used in the compressor in order to maintain the non-contacting clearance seal. A clearance seal is also maintained between the displacer and the thin walled metal tube in which it oscillates. This clearance can be several times greater than that in the shaft seal since the pressure drop across the regenerator is small, and in any case there is some regenerative capacity in the annular gap around the displacer.

The displacer is driven by the varying pressure across the shaft seal. The correct phase angle between compressor piston and displacer is obtained by adjusting the resonant frequency of the displacer on its spring suspension and the stroke is controlled by the area of the shaft and the damping produced by the regenerator. A linear motor is attached to the displacer shaft and this allows detailed adjustment of phase angle

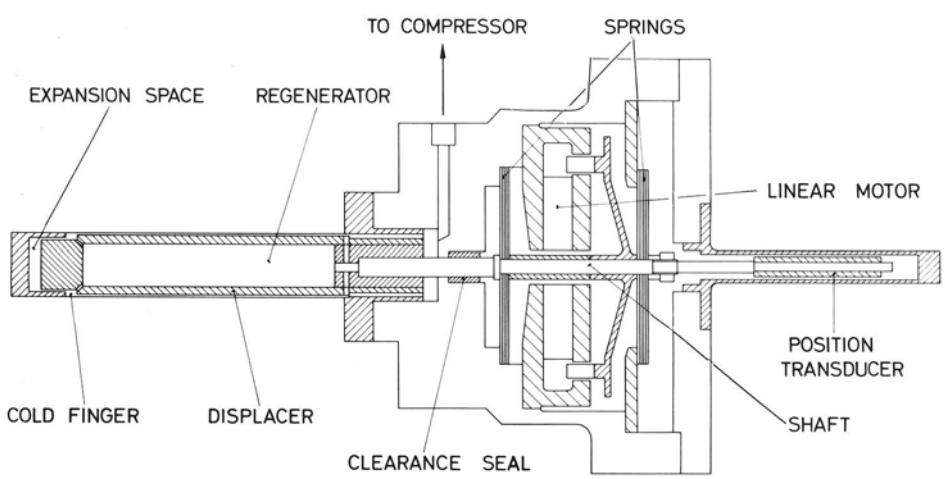

Fig. 2. Schematic diagram of displacer.

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and stroke. Both piston and displacer are fitted with a position transducer which can be used in a control system to maintain optimum stroke and phase angle during cooldown and during long term operation in difficult environments.

This cryocooler has a simple, robust construction with few parts and has a versatile configuration. However its most important attributes are what it has not got. It has no rubbing contacts, no friction, no lubrication, no wear and no fatigue problems.

DESIGN OPTIONS

The split cycle cooler described above is the most complicated, most versatile and most expensive version of the Oxford cryocooler but many variations are possible and have been produced. The simplest arrangement is to have the displacer driven entirely pneumatically with a simple control system on the compressor to prevent over-stroking during cooldown. It is also possible to make integral rather than split coolers with either a pure pneumatic displacer drive or with a linear motor control. Integral coolers generally are more compact for a given performance. Multistage coolers are, of course, possible in either format.

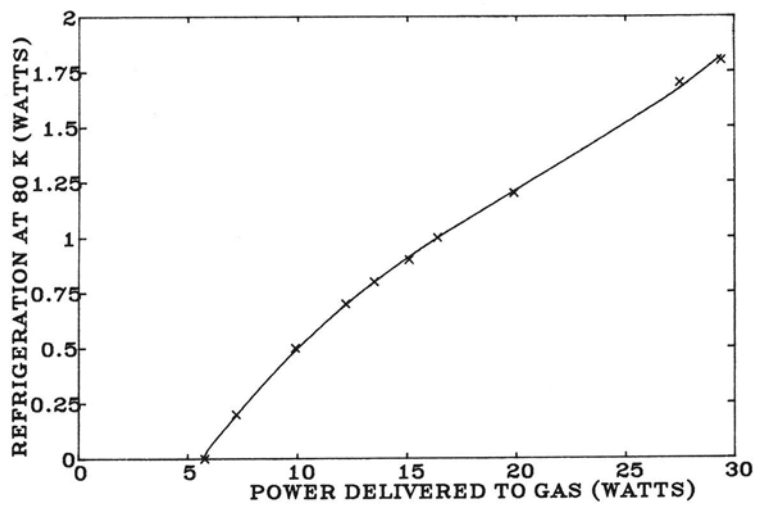

Fig. 3. Typical cryocooler performance.

The coolers so far described are unbalanced reciproccating machines and therefore unacceptable for some applications. Fully actively-balanced coolers can be produced, the best compressor design for this purpose being one in which two matched, opposed compressors operate pistons into the same compression space. In many non-space applications, the vibration caused by the compressor can be effectively isolated from the displacer and hence the element which it is cooling. The vibration caused by the displacer can then be considerably reduced by a simple, robust passive balancer.

Within these design options a large size range is available from miniature coolers with less than five watts of input power to larger cryocoolers with an upper limit on refrigeration power which is probably in excess of 500 W at 80 K.

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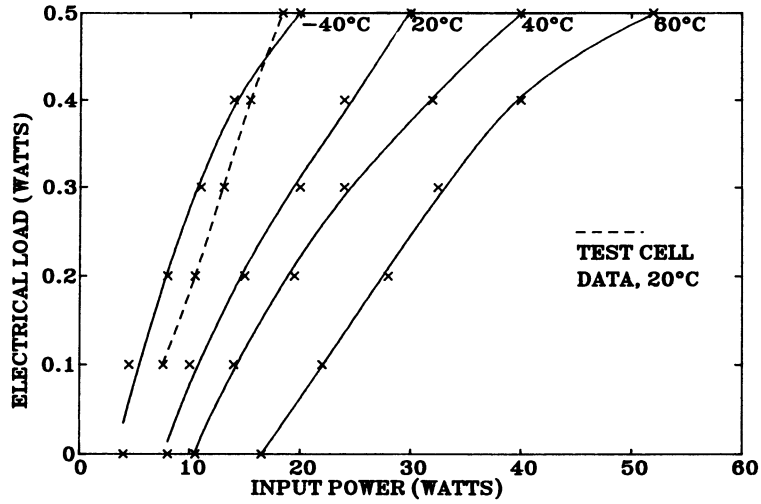

Fig. 4. Cryocooler performance in various ambient temperatures.

PERFORMANCE

Cryocooler applications, and hence their specifications, are extremely diverse and it is therefore difficult to give typical performance data. The two sets of data shown here are chosen to show the sort of results which can easily be achieved in small cryocoolers. Fig. 3 illustrates the performance of a cooler with a 10 mm diameter displacer and a nominal 1 W of refrigeration power at 80 K. The refrigeration power is plotted against power delivered to the gas rather than the total input power, the difference being due to the compressor motor inefficiency. This has been done because the efficiency of the motor is a function of its size, weight and cost, and is therefore very variable. However, compact linear motors with efficiencies of more than 85% can be produced, so that 1 W of refrigeration power at 80 K from 20 W of input power is a reasonable goal for coolers producing around 1 W of refrigeration.

The data of Fig. 4 was obtained from a cooler similar to that shown in Fig. 5 with a weight of 2 kg. The test cell data was obtained with the cold region in a radiation-shielded vacuum enclosure and a thermometer mounted directly on the cold end. The ambient air temperature was 20°C. The other results with ambient air temperatures from -40°C to +60°C were obtained with the cooler mounted in a dewar encapsulation containing a detector, the electrical refrigeration load being provided via the detector array. The results appear inferior to those obtained in the test cell because there is an additional heat conduction load from the dewar. More importantly, the thermal resistance between the cold region of the cooler and the detector substrate meant that the cooler was operating well below 80K to maintain the substrate at 80K, this being the place where the temperature was monitored. However, this is a realistic, practical situation in some cooler applications and the results illustrate the ability of this type of cryocooler to operate over a wide range of ambient temperatures and to deal with substantial refrigeration loads in a difficult application.

EXPERIMENTAL TECHNIQUES

One feature of the Oxford cryocooler is the ease with which it can be experimentally optimised. The first stage of very rapid optimisation is possible because of the electrical control of piston and displacer motion.

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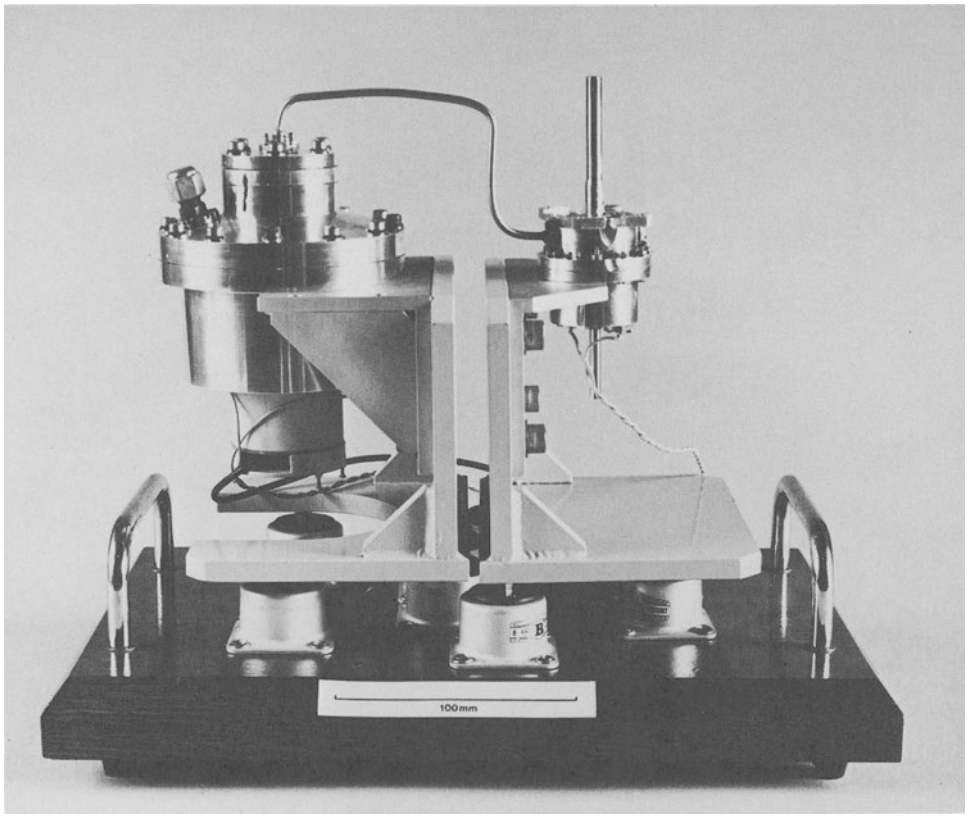

Fig. 5. A split cycle cryocooler.

Optimisation against piston and displacer strokes and their relative phase, drive frequency and filling pressure (which fine tunes the compressor resonance condition) can all be carried out quickly without any internal adjustments.

A second stage of optimisation is then possible because non-invasive experimental techniques have been developed to quantify all the significant power loss mechanisms in the cooler.[1] At this stage of final optimisation of a development model, it is possible to detect most fault conditions such as spurious pressure drops, seal leakage, etc. More importantly, a detailed experimental assessment of the regenerator performance can be obtained so that this important component can be modified to give best results.

These experimental techniques have another important application. It is possible to carry out an energy balance for the entire cooler, accounting for almost all the input power. The data can be used in simple, but very useful and accurate, semi-empirical models of Stirling cycle cryocoolers. A typical breakdown of the power consumption for a cooler producing 1 W of useful refrigeration at 80 K is as follows. Of the 20 W total input power, 3 W are lost due to linear motor inefficency and 9·7 Watts are dissipated because of the non-ideal compression process. Other losses due to pressure drop, seal leakage, windage, hysteresis and power required to drive the displacer, are in total hardly significent. This leaves 7.3 Watts available to produce refrigeration. With gas at the cold end at, say, 75 K, and at 310 K at the warm end, the Carnot factor is 3·1, so that 2·35 W of total refrigeration are produced. Of this, 0·45 W is consumed by conduction heat transfer, 0.4 W by shuttle heat transfer and 0.5 W by regenerator losses, leaving 1 W of useful refrigeration.

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A cryocooler with full electrical control of piston and displacer is a powerful development tool. Coolers which are intended to operate with a purely pneumatic displacer drive are normally fitted with a linear motor drive in the development stage so that optimum operating parameters can be determined. Displacer dynamics are then adjusted to reproduce these values without electrical control.

RECENT DEVELOPMENTS

Earlier designs of the Oxford cryocooler were suitable for the production of small quantities of high-cost units over a limited size range. Clearance seals were obtained by using a considerable amount of high-precision machining, together with slow and difficult assembly methods. New designs of linear motors and suspension systems, and improved manufacturing and assembly methods, have largely removed these limitations.

The simple linear motor already illustrated is not suitable for larger coolers because the moving mass is too large to allow operation at resonance. Modified designs allow air-gap flux densities in excess of 1·3 T, so that much smaller coils are needed for a given power output. In addition, this allows long-stroke capability for high efficiency whilst the permanent magnet is used at its optimum working point and is therefore of minimum size. This is an important consideration because of the high cost of magnets.

Clearance seals with small gaps and long-stroke capability are difficult to produce, but new manufacturing and assembly methods now allow much more rapid production of high-quality seals. One problem with any free-piston machine is that, due to differential gas leakage past the piston seal, the piston tends to move towards one end of the cylinder instead of oscillating about the mid position. This problem can be overcome in several ways, but all add complication and reduce efficiency. We have long suspected that in practical cryocoolers the problem was not due to fundamental scientific reasons, but was caused by poor seal manufacture. Clearance seals made and assembled by our present methods show no tendency to this fault. Indeed, it appears that the quality of a clearance seal (the uniformity of the annular gap and the co-axiality of piston motion and cylinder) can be judged from the symmetry of the piston motion.

Early spring suspension systems were adequate for limited applications. However, the resonant frequency of individual spring arms was around 100 Hz and easily excited by the second harmonic of the drive frequency. This produced acoustic noise and the risk of spring failure. Also, radial stiffness allowed a maximum of only 3g radial acceleration before piston and cylinder touched. Since until recently, although true clearance seals were produced, there was limited control of co-axiality, contact between piston and cylinder was possible with quite small radial forces. Spring design is complicated by the necessity to control spring-arm resonance, radial stiffness and, for the displacer suspension, axial stiffness as well. Long stroke is also necessary for efficient operation and to meet compressor resonance conditions. All these conditions must be controlled within the paramount requirement of infinite fatigue life. A recently-developed computer program for spring design sets criteria for infinite life and calculates all other parameters. It allows manipulation of the design to give optimum properties for a given application. Higher spring-arm resonance can now be obtained and radial stiffnesses up to 8 times greater than previously possible. The programme plots the chosen design complete with all holes, slots, etc. This plot is then photographically reduced to produce the photo-tooling necessary to cheaply and accurately manufacture the finished item by chemical etching.

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OPERATING LIFETIME

With cryocoolers which are intended to operate for many years, it takes a long time to establish life expectancy from operating experience. However, there are other considerations which help to engender confidence in reliability over very long operating periods. Fatigue life can be established using standard methods without very long test schedules. Fatigue testing of sample components for relatively short periods with considerable overstroke and hence greater peak stresses will give confidence in infinite fatigue life under normal operating conditions.

It is possible with simple test procedures to establish that a cooler has been assembled with true clearance seals, and hence will not wear. Given freedom from fatigue failure and no wear, few possible modes of failure remain. With no lubrication, simple bake-out procedures during and after assembly will ensure freedom from failure or degradation due to contamination by condensable gases. Particulate contamination is the remaining possibility. All but very small particles can be eliminated by careful cleaning procedures and clean assembly conditions, and the clearance seal is designed with self-protection against such remaining contamination.

These considerations, together with several extended life tests ranging from 6000 hours to 26,000 hours without failure or degradation in performance, seem to have established very considerable life expectancy. It is still not clear in what manner these cryocoolers will eventually fail.

REFERENCE

1. A. H. Orlowska and G. Davey, Measurement of losses in a Stirling cycle cooler, Cryogenics 27:645 (1987)

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