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DESIGN OF REFRIGERANT COMPRESSOR WITH HYDRAULICALLY COUPLED, HERMETICALLY SEALED LINEAR DRIVE

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ABSTRACT

Substitution of a metallic diaphragm for the power piston of a Stirling engine is shown to be a promising approach to transmitting power for a heat-actuated heat pump through a hermetic seal. Hydraulic fluid under high pressure transfers the power from the diaphragm to the compressor plunger, which is balanced by a counterweight. The discussion is divided into three parts which deal with the internal mechanical design of the compressor, development of the diaphragms, and the hydraulic oil management system.

INTRODUCTION

There is a need for heat pumps for applications where electricity is either not available or more expensive than combustible fuels. Work sponsored by the Department of Energy/Oak Ridge National Laboratory addresses this need.

The objective of the Heat Actuated Heat Pump (HARP) program at Mechanical Technology Incorporated, Latham, New York, is the development of a refrigerant compressor driven by a free-piston Stirling engine (FPSE). An approach has been identified which addresses the major problems of such machines. It holds promise as an economically manufacturable product.

The FPSE is a linearly oscillating heat engine with high efficiency, durability, and low emissions. MTI FPSE technology represents the state of the art. In previous efforts to couple refrigerant compressors to Stirling engines by direct mechanical means, a major problem has been inability to segregate refrigerant and engine working gas. The MTI approach uses a metallic diaphragm as the power piston. Power is then transferred to the compressor plunger through pressurized hydraulic fluid. The diaphragm is a hermetic seal between the engine's working gas (helium) and the hydraulic fluid, reducing the leakage problem to mixing of hydraulic fluid and refrigerant. This is not a severe problem. Fluorocarbon refrigerants do not affect the incompressibility of hydraulic oil, even at saturation.

Use of the diaphragms and hydraulic transmission system requires development of a new compressor configuration. An oil management system is also needed to maintain pressure balance across the diaphragms.

MECHANICAL DESIGN

Background

The major constraint on this design effort is the required compatibility with a free-piston Stirling engine (FPSE). Such engines are operated in dynamic resonance. The internal masses (displacer and power piston) oscillate against gas springs under the influence of the engine pressure wave and external control forces. Load is applied as damping between power piston and ground (Figure 1).

![Figure 1. Schematic Dynamics of FPSE and Compressor Load](image-url)
It is here that the compressor must fit. Because of the resonant condition, variation of load causes variation in stroke (resonant amplitude). The compressor design must accommodate this fact, either by operating as a variable-stroke machine or by converting the stroke variable to speed modulation. To avoid conversion losses and allow directly-coupled power transmission, the variable stroke concept was adopted.

Because the Stirling is a closed cycle, contaminants are not discharged with each stroke of the piston. They build up in the contained working fluid over the life of the engine and degrade performance. For successful long-term operation of a Stirling-driven heat pump, complete isolation of contaminants (including refrigerant) from the engine's working gas (helium) must be maintained. This has been accomplished by replacing the sliding power piston with a sealed metallic diaphragm deflecting elastically. Hydraulic power output is required as the diaphragms can support only pressure, not localized mechanical loading. The moving mass of hydraulic fluid (oil) then becomes a part of the piston mass. In all designs, the piston gas spring (charged with helium) is dynamically connected to and mechanically isolated from the oil-piston by a second diaphragm. Power (refrigerant compression) is extracted from the oscillating oil by interposing the compressor pistons in the flow path.

Earlier designs for such systems have stacked components in a coaxial assembly (Figure 2). Such units have typically been very long, causing mounting difficulties and preventing installation in conventional cabinets. The compressor components are necessarily imbedded into the hydraulic drive, making assembly and service difficult and expensive. For the HARF program, where extensive laboratory testing is required, such designs are very difficult to set up or alter.

Z-Flow Design Concept

In response to the problems of earlier units, a compressor concept has been developed (patents pending) at MTI which is more compact, lighter, and more accessible. The basic scheme is a folding of the oil-flow path by rotating the compressor axis to 90° with respect to the engine axis (Figure 3). The oil acts on either side of the transverse piston exactly as it did on axial ones. The only difference in flow is its curvature. This arrangement greatly reduces the length of the assembly, but without an increase in overall width (which is controlled by the diaphragms' diameter). Most important, especially for this program, is the location of the compressor heads external to the hydraulic drive system where the valves and plenums can be changed without affecting the drive. This capability allows for hardware optimization and improvement with minimum effort.

The lateral motion of the transverse piston causes a new mode of vibration. Previously, with in-line compressors (or engines alone) all vibration was axial. The transverse piston introduces a rocking, a moment about the system's center of mass. This vibratory mode, in addition to adding to noise (60 Hz hum), puts loads on the displacer bearings and the combustor assembly (mostly sheet metal). These combined loads are significantly more difficult to design for than the simple axial vibration. A balancing system is required to eliminate the problem by minimizing the compressor's contribution.

The chosen design (Figure 4) utilizes an annular balance ring, which, in fact, is the piston in the main resonant system. Driven from that ring through fixed-volume oil chambers is the main compressor piston. The arrangement is such that motion of the balance ring (counterweight) directly deflects both the gas spring and the compressor piston (opposite to the balance ring motion).
Figure 4. Annular Counterweight HAHP Compressor Schematic

The annular counterweight design minimizes complexity by its design. The single bore in the casting is made clear through. The counterweight and compressor piston with cylinder sleeves slide in and the heads and plenums cap the assembly. The unit actually being constructed for the HAHP program is much larger than a production unit would be, to allow for extensive instrumentation and exploration of parametric variations. A production unit would also be enclosed in a pressure vessel, hermetically sealed with the engine. The prototype machine (Figure 5) is free-standing, taking the high pressure loads in the casing walls. This makes the casing (cast iron) much heavier, but leaves the compressor components (heads, valves, etc.) accessible for laboratory work.

Centering Control

The hydraulic drive system implies a lack of positive location for the reciprocating elements (counterweight and compressor piston). It is possible for preferential leakage across the driving seals (close fit clearance) to occur. The net change in oil volumes will cause a drifting off-center. That is, the equal point in each cycle (when there is no ΔP across the element) will occur at other than the mid-stroke point. To correct this problem, centering ports are included. Such ports are small slots in the bores of the reciprocating elements. The slots are slightly longer than the seal lengths so that at (and very near to) the mid-stroke point, the seal is short-circuited by a fluid path. Should the pressure across the piston not be balanced at the mid-stroke, corrective flow will occur through the port, balancing the volumes on either side.

The center port system has proven very effective for dynamic centering in gas bearings and on test rigs with hydraulic oil. But during periods of shutdown, it is possible for a non-level machine to experience piston drift by slow seal leakage. If the drift is severe, the stroke of the unit when started may not even cross the mid-stroke centering port. No corrective flow can occur. In the extreme case, pistons could drift to their stops, effectively locking the unit from moving at all.

To address the quasi-static drift problem in a cost-effective manner, mechanical centering springs acting on both sides of the reciprocating elements have been included. The stiffness of the springs is required to be sufficient to hold fully leak-relaxed pistons (i.e., no support from surrounding oil) to within one quarter (1/4) of full stroke amplitude when the compressor is oriented such that pistons are upright. This assures that, when started, the pistons can and will cross over the center ports, and that if full-stroked in the maximum off-center condition, that the pistons would just touch the stops on the nearer side. In addition to mechanical spring centering, the prototype compressor allows individual control over each oil volume, giving total freedom to adjust and test the machine under all conditions.

The use of mechanical springs with a minimum required stiffness creates a potential fatigue failure. The compressor runs at about 60 Hz, so any springs being driven by the unit must have effectively infinite cyclic life. The geometry of the compressor determines where springs can fit, leaving stroke as the only free variable in the spring loading. To assure satisfactory spring life, compressor stroke is limited to about 20 mm. The actual maximum design stroke is .75 inches (19 mm). With the fixing of the stroke, the displaced volumes of diaphragms and compression spaces set the outer diameters of counterweight and piston. The
relative diameters (between these elements) are set for basic mechanical integrity (refer to Figure 4). The strokes of piston and counterweight are equal (for best spring life). Their masses are also equal such that balance is achieved by their equal and opposite momentums.

**Internal Seals**

The hydraulic drive system depends on effective seals throughout for its function. The high-pressure oil-to-refrigerant seals are discussed elsewhere, but a description of the internal oil-to-oil seals is in order. These seals are all of close-tolerance clearance design. The outer diameter of the counterweight carries polytetrafluoroethylene (PTFE) pads, machined in place to give a close clearance fit in the bore. Radial gap is held to .001" (.025 mm) and the seal length is optimized based on laminar leakage flow and mechanical strength. The counterweight-to-cylinder seals (at the counterweight I.D.) are required to take up manufacturing tolerance between the cylinder sleeves' diameters and locations. To do this they are constructed with mild interference between the sleeves and pads of reinforced PTFE on the counterweight. The pads are wear-seated to give true-running, near-zero clearance seals. The seal between counterweight and piston is subject to the maximum tolerance stack-up. Because of possible error between cylinders, the piston itself is supported on stinger rods between the two compression faces. This eliminates binding from angular error between cylinders, but leaves the central piston with insufficient length-to-diameter to prevent jamming. This problem is eliminated by making the piston in spherical form. This way, rocking of the piston on its stingers does not affect the seal function or the mechanical freedom of the assembly.

**DIAPHRAGM DEVELOPMENT**

**Diaphragm Technology Review**

Providing a hermetic seal between the engine working gas and the refrigerant gas is central to the Heat Actuated Heat Pump concept, and it was intended from the beginning that either a bellows or a diaphragm would fulfill this function. In the early stages, the bellows concept was tabled by the perception that it would require a more involved development effort than diaphragms. To date, development effort has concentrated on diaphragms. The program has addressed problems in three areas:

1. Ratio of Displaced Volume to Diameter
2. Fatigue Life
3. Manufacturing Methods and Cost

A survey of existing technology in diaphragms revealed that they are primarily used in the high-speed flexible coupling industry. In this application the diaphragms are subjected to loadings which are quite different from those we were interested in. Diaphragms in couplings experience point loads, angular deflections, torque, and usually no pressure loading, the one type of loading which we would be applying.

Nonetheless, a number of coupling manufacturers were approached and it was found that there is a broad variety of technologies and a few inconsistencies. One company told us in no uncertain terms that good fatigue life was impossible without a large fillet at the boundary, while another was having good success with a simple mechanical clamp on the flat diaphragm sheet.

Diaphragms can be made in several configurations including flat, with constant thickness or with thickness varying as a function of radius (contoured), or convoluted, in which the diaphragm is formed with concentric waves (Figure 6). A flat diaphragm is the least expensive to make, but deflects the least, while a convoluted diaphragm offers the greatest displaced volume within a given stress limit, but can be the most expensive. The contoured diaphragm generally stands in the middle ground in these properties.

An appropriate clamping arrangement is necessary to minimize stress concentrations at the boundary edge (Figure 7). Various mechanical schemes are used, both rigid and pivoting, but welding, which is popular for bellows, is not used except outside of a mechanical clamp. The reason for this is the difficulty of producing a smooth, continuous weld fillet. Other manufacturers make a good fillet by taking a thick blank of material and removing material from the diaphragm area itself until the desired thickness is reached. This leaves a rigid outer ring with a smooth fillet, and it is one piece of metal with the diaphragm itself, which greatly reduces the chance (compared to a weld fillet) of a material defect near the critical boundary.

A flat diaphragm is simply cut from flat sheet stock, and most convoluted diaphragms are stamped.
from the same. A contoured diaphragm requires a more involved manufacturing method, especially if it is to have an integral fillet. Methods range from simple lathe turning to sophisticated electrochemical machining (ECM). Lathe turning is economical but can result in very undesirable residual stresses. The ECM process on the other hand is somewhat expensive, but can result in a scratch free finish with no process-induced residual stress. More recently, a vendor was found who will photo-etch the contour, but this does not draw from proven coupling technology. The photo-etch method cannot provide an integral fillet economically.

Materials for diaphragms range from the mundane to super alloy, but the use of steel is nearly universal. The use of elastomers may be on the horizon but is not current technology. For metals there is a consensus that the material must be quite clean and defect free for good fatigue life.

All of the manufacturers withheld their stress analysis methods as proprietary and none were sufficiently interested in our application to engage in design development. Therefore it was necessary to acquire the analytical capabilities to design a diaphragm for our compressor. For analyzing flat and contoured diaphragms a computer code was developed in-house (PLATE). The analysis for these planar diaphragms uses a simple finite differences iteration.

Inputs to the code are basic geometry and material properties and a series of parameters pertaining to boundary conditions. The code has the capability to do one of four options when it is run. First, for a central load and/or a distributed constant ΔP load across the plate the deflection and stresses can be found. This is a "once through" non-iterative application. Options two and three pertain to varying the load conditions in such a manner to solve for a particular center plate deflection or displaced volume or maximum principle stress. In the fourth option, the center loading of ΔP can be varied and iterated against maximum principle stress.

For analysis of convoluted diaphragms, an in-house program was considered too costly and also duplicative of existing knowledge, so a computer code (NONLIN) was purchased from Battelle Memorial Institute. It uses direct numerical integration to solve the differential equations which represent the physical configuration of the diaphragm. The geometry is represented by conical and toroidal shells of revolution, and the program is capable of linear elastic axisymmetric and non-symmetric deformations, and of nonlinear axisymmetric deformations. Very good correlation between analytical prediction and strain gage testing has been achieved by Battelle in their evaluation of the method. It is also able to analyze flat and contoured diaphragms, and when compared with PLATE for analyzing the same diaphragm, good agreement was found.

Design for Hydraulic Drive

Several factors contribute in consideration of design specifications for a diaphragm in our application. For overall packaging and cost considerations, minimizing diameter of the diaphragm while providing the desired volume displacement is important. The convoluted diaphragm is the best for this, but it was felt that this type of design could introduce problems with standing waves in the diaphragm itself. Also, the convoluted diaphragm has a very low stiffness and since it was thought that the oil management system would measure ΔP across the diaphragm to control the diaphragm centering, a convoluted diaphragm would not be stiff enough to produce an easily measurable ΔP even when far off center. This is also a reason that non-metallic materials were taken out of consideration.

Early experimental work with a mechanically clamped, stamped, convoluted diaphragm revealed that fatigue due to fretting at the clamped boundary was a significant limiting factor. This problem was anticipated, so it had earlier been decided to specify an integral fillet for the diaphragm for the compressor. Diaphragms were made using two methods of fabrication: lathe turning and ECM. The lathe turned diaphragms suffer from what is known as oil canning, where the diaphragm can take one of two different positions when unloaded, neither of which is on center. The ECM diaphragms do not exhibit this problem and furthermore have no process-induced residual stresses. The surface finish is very good except that there is some difficulty in avoiding corrosion pitting.

The diaphragms are made from 4340 low alloy steel, vacuum melted and cross-rolled. This material has a minimum yield stress of 100,000 psi and a nominal fatigue limit of 70,000 psi. Diaphragm manufacturers recommend against designing for more than half of the nominal fatigue limit, so we have used 35,000 psi as our maximum stress.

The diaphragms have a 10 inch diameter at the fillet boundary with a 3/16 inch flange thickness. This geometry affords a 3.75 in³ displacement volume at the design stress limit.

Because of long lead time for contoured diaphragm manufacture, flat diaphragms were made from 0.020" stainless steel sheet with mechanical clamping. These gave us an opportunity to further evaluate mechanical clamping and the effect of surface finish on fatigue life.

Test Experience

Diaphragm testing has been conducted with two objectives:

1. Verification of computer stress analysis
2. Demonstration and development of long life

Priority has been on the latter; this and instrumentation difficulties have precluded any reliable strain data as yet, but cursory investigation has
shown that both computer codes are quite good.

Testing with flat sheet diaphragms showed that many of our empirical assumptions were correct with regard to fatigue life. The diaphragms were over-stressed and one suffered a fatigue crack after 3.24 million cycles. Subsequent metallographic inspection revealed that the crack initiated site was a large surface defect. This experience reinforced efforts to assure good surface finish for long life.

No other flat diaphragms failed during the tests. Some reached as many as 6.24 million cycles before the program moved on to contoured diaphragms.

Experience with contoured diaphragms has been undramatic so far. Test rig capability has limited peak stress in the diaphragms, but modifications to the rig will raise this limit in the future. At design stress one pair of diaphragms has accumulated 35.0 million cycles with no sign of fatigue. No contoured diaphragm has yet experienced a fatigue failure in this program.

The opportunity was seized to install diaphragms with pronounced oil can to see how, if at all, they affected the stability of the resonant linear driver. It was impossible to operate the rig at all, which illustrates the need to avoid this effect.

HYDRAULIC TRANSMISSION DESIGN

Hydraulic Transmission losses

At the initiation of the HARP program there was no information on the magnitude of the power which would be lost to churning the hydraulic fluid. An evaluation of this has been one of the first undertakings of the program. If the hydraulic power were large, then this type of drive system may not be worth pursuing. That has not been the case.

An analytical approach was taken first. The wall geometry is very irregular and asymmetric, but since this analysis was intended to be an approximation, the geometry was simply represented by straight pipe sections and elbows. For this type of geometry there is much documentation of flow analysis.

The losses can be broken down into two parts: viscous drag in steady or unsteady flow; and velocity head losses from flow reversal and turning corners.

The Reynolds number indicates that the flow is well into the turbulent range. By using the Darcy-Weisbach equation for viscous drag in steady flow:

\[ h_f = f \frac{L v^2}{D g} \]

where:
- \( h_f \) = Pressure Head Loss
- \( f \) = Friction Factor (from Moody Diagram)
- \( L \) = Equivalent Length of Pipe
- \( D \) = Equivalent Diameter of Pipe
- \( V \) = Flow Velocity
- \( g \) = Acceleration of Gravity

total fluid losses of 4.0 watts were calculated for the design condition. Initially this appeared to be extremely low compared to the 3 kW of transmitted power, and so other approaches were looked at.

A method by Trikha for simulating frequency dependent friction in transient flow was tried, but this yielded results on the order of a kilowatt which was believed too high. Likewise an analysis of velocity head losses from flow reversal produced unreasonably high loss values.

The extreme results of the analytical approach fostered increased interest in producing experimental results. A test rig called the hydraulic simulator, Figure 8, was built and has been used to evaluate hydraulic transmission losses. This is the same device used for testing diaphragms. The simulator consists of a double-ended hydraulic piston which is the armature of a linearly oscillating motor. The motor drives the piston which in turn displaces hydraulic fluid against a pair of diaphragms, backed by helium gas springs at high pressure. The simulator is equipped with instrumentation to evaluate the drag induced by piston rings and gas spring hysteresis. When these are subtracted from the total power input, the remainder is the hydraulic loss. Figure 9 shows a comparison of predicted and measured losses. The predicted hydraulic loss is based only on viscous drag in steady flow. The experimental results indicate that this is in the correct order of magnitude. This is encouraging because it shows that hydraulic transmission losses are reasonable, and the basic concept is feasible for an efficient compressor drive.

\[ \text{Equation for Darcy-Weisbach equation} \]

Operating Frequency – 60 Hz
Gas Spring: Helium at 40°C
Oil Type: A ATF at 60°C

Experimental Data:
- With Cast Iron Piston Sealing Rings
- With No Sealing Rings

Experimental Oil Loss
- Gas Spring Loss
- Piston Ring Loss

Figure 9. Hydraulic Simulator Loss Comparison

Testing with the hydraulic simulator has also shown that there is no cavitation problem and that introduction of refrigerant in solution has no ill effects.

Oil Management System

Because of the high pressure involved and the delicateness of the diaphragms, it is important to keep close control of the oil charge. There will be leakage of oil from the high pressure cavities through the seals at the piston rods. This oil must be reinjected into the high pressure cavities to keep the diaphragms on center.

The compressor has been designed with two alternative oil seal designs which will be evaluated and compared when compressor testing begins. One design is a straightforward two-stage chevron seal, Figure 10a, while the other is a pumping ring, Figure 10b. The pumping ring uses a viscous hydrodynamic principle to carry oil from the low pressure side to the high side when the rod is moving in that direction. On the return stroke, a pressurized elastic backing ring clamps the pumping ring down to prevent backflow. Although the net pumping flow is very small, the low pressure side must be kept flooded with oil because the ring will be damaged if it is run dry.

One method of oil management which will be tried is volume control by measuring the amount of leaked oil in a receiving tank and pumping the same amount back in. Such a level sensor device has been built and tested, and it works very well. There is some question however about accumulation of oil in the drain lines and distribution of oil mist throughout the refrigerant loop. It would be impossible for the level sensor to account for these, and until we are able to test the whole system it is unknown whether these are significant quantities. Figures 11a and 11b show schematic diagrams of proposed oil circuits for use with chevron seals and pumping rings respectively.

Since the pumping rings are doing the high pressure pumping, the external pump need only be a low pressure pump for that system. The pumping ring flow is self regulating. The oil pumped into the high pressure cavities is bled through a pressure regulating valve back to the pumping ring supply side.

A system that controls oil by measuring the mean pressure difference across the diaphragms would assure better diaphragm centering. Recently a system has been proposed which uses control valves actuated by pressure difference across a bellows, but this idea has not yet been developed.

Figures 11a and 11b. Oil Management Schematics for Chevron Seals and Pumping Rings.