2000

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# A PROTOTYPE OIL-LESS COMPRESSOR FOR THE INTERNATIONAL SPACE STATION REFRIGERATED CENTRIFUGE

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## ABSTRACT

Under a subcontract from Lockheed-Martin Space Operations, Mainstream Engineering Corporation has successfully designed, fabricated, and tested a new oil-less hermetic reciprocating compressor that will operate in a zero-gravity environment. This compressor is used in a refrigerated centrifuge, which will be placed on the International Space Station. The original centrifuge used an oil-lubricated hermetic reciprocating compressor that was unsuitable for use in the zero-gravity space environment. The prototype oil-less compressor was a modified version of the hermetic reciprocating compressor used in the refrigerated centrifuge. Exhaustive orientation and performance testing of the modified compressor was performed. Life testing of the prototype compressors is underway.

## INTRODUCTION

Lockheed-Martin Space Operations (LMSO) is in the process of developing a refrigerated centrifuge for use on the International Space Station (ISS). A commercial off-the-shelf (COTS) refrigerated centrifuge has been selected for use. The COTS refrigerated centrifuge uses a R-404A vapor compression refrigeration system with a hermetic reciprocating compressor that has gravity-dependent lubrication unsuitable for use in space. Lockheed-Martin Space Operations commissioned a subcontract to Mainstream Engineering Corporation to design, fabricate, and test a prototype retrofitted compressor that will operate in a micro-gravity environment. Mainstream modified the existing compressor, performed orientation testing of the modified compressor, performed performance testing of the modified compressor, compared the performance of the modified compressor to the unmodified compressor, and initiated life testing of modified compressors to demonstrate 4000 hours of successful operation.

The prototype compressor had to meet several requirements related to performance, life, and reliability. First, the prototype compressor should have similar performance characteristics to the unmodified compressor, as measured by system capacity and power consumption. Second, the prototype compressor should be mounted in the centrifuge and operate satisfactorily for one hour on each of the six sides of the centrifuge. Third, the prototype compressor should demonstrate an operational life of 4000 hours with a 90% duty cycle.

## DESCRIPTION OF COMPRESSOR MODIFICATIONS

Many design modifications were incorporated into the prototype compressor to ensure suitable operation and performance in a micro-gravity, oil-less environment. A summary of the modifications is provided below, followed by a detailed discussion of the modifications.

- A self-lubricating polyimide sleeve was inserted into the cylinder bore (after opening the diameter of the bore to accommodate the sleeve).
- Polyimide thrust washers were used to replace the oil-lubricated thrust washers.
- Sealed roller bearings were installed on the crankshaft and connecting rod. These sealed, permanently lubricated bearings replaced the existing oil-lubricated bearings.
- An oil-impregnated bronze sleeve bearing was used on the wrist pin.
- The connecting rod and bearing plate were redesigned and fabricated to accommodate the sealed roller bearings.
- A modified two-piece crankshaft was fabricated to accommodate the connecting rod and bearings. The two-piece crankshaft was hardened, press fit together, and ground to final dimensions.
- Modifications to the compressor/enclosure mounts were made for suitability in various gravity orientations. The existing springs were replaced with hard mounts, which were welded into place.
- Modification of the discharge line was made to eliminate potential failure in a discharge line muffler joint. The discharge line muffler was eliminated.

### Polyimide Sleeve

The polyimide resin used for the sleeve and thrust washers contained 15% by weight graphite filler, which is added to reduce wear and friction. An analysis of the polyimide sleeve, which is press-fit inside the cylinder bore, was performed to confirm proper fit and dimensions in both high and low-temperature conditions. High temperature conditions will result in swelling of the sleeve, cylinder, and piston, and could cause interference at the sleeve/piston clearance. Low temperature conditions could cause the sleeve to shrink more than the cylinder bore, resulting in a sloppy fit of the sleeve in the bore. This analysis resulted in a range of allowable sleeve dimensions that would be suitable for both high and low temperatures.

### Crankshaft and Connecting Rod Modifications

The existing crankshaft and connecting rod of the compressor were sent to an outside lab for material identification. The crankshaft was fabricated from cast iron and the connecting rod was fabricated from 380 aluminum casting. Also the hardness of the crankshaft, piston, and wrist pin were measured and determined to be 88 HRB, 65 HRB, and 59 HRC, respectively. Mainstream selected a modified, vacuum-hardened, 4340 steel to fabricate the crankshaft and 6160-T6 aluminum to fabricate the connecting rod. No modifications to the piston were made.

In order to accommodate the modified connecting rod and sealed bearings, redesign of the crankshaft was required. A two-piece split crankshaft was necessary in order to adequately install the bearings and connecting rod on the shaft. The two-piece crankshaft was designed to have a profile that matched the existing crankshaft. The two shaft pieces are press fit with a class FN2, "medium-drive" interference fit. An alignment hole, pin, and connecting bolt were also installed to aid in proper alignment and fit. The crankshaft was machined to preliminary dimensions, hardened, and ground to final dimensions prior to final assembly. Finite element analysis of the crankshaft was performed to analyze the bending of the shaft at full load to confirm that it was in tolerable limits.

## COMPRESSOR PERFORMANCE TESTING

The performance of the unmodified and prototype compressors was measured on Mainstream's heat pump test stand, details of which have been described elsewhere (Ref. 1). Mainstream's test stand incorporates two independent temperature controlled water loops, one to supply the heat load to the evaporator and one to remove heat from the condenser. The heat load supplied to the evaporator is determined from the measured flow rate and temperature difference of the water loop. Compressor power is measured directly. In order to simulate the operating conditions of the refrigerated centrifuge, the water temperature supplied to the evaporator was controlled to 5°C and the water temperature supplied to the condenser was controlled to 32°C.

### Unmodified Compressor Performance

Eight unmodified compressors, identical to those in the COTS refrigerated centrifuge, were performance tested. Table 1 provides the average performance results for the compressors.

**Table 1 – Performance Results for Unmodified Compressors**

| Performance Parameter | Average Value |
|---|---|
| Suction Pressure | 478 kPa |
| Discharge Pressure | 1.65 MPa |
| Evaporator Exit Superheat | 6.2 °C |
| Condenser Exit Subcooling | 3.8 °C |
| Compressor Power | 553 W |
| Cooling Capacity | 879 W |
| COPc | 1.59 |

### Initial Verification Testing of Prototype Compressor

Initial verification testing of prototype compressor #1 was performed using an unhardened crankshaft that was machined to final dimensions (this decision was made to save time, since the hardening and grinding process required considerable time). During the initial verification testing, the compressor ran nominally for approximately 20-25 minutes. At that time, the compressor power rose significantly, resulting in the compressor overload protection engaging, stopping the compressor. An audible slowing of the compressor speed was observed during this event. It appeared that a significant amount of friction resulted in excessive heat generation and torque requirements.

Initial attempts to correct the problem focused on the piston/bore clearance. This clearance was gradually increased from an initial value of 0.0007 inches to 0.0017 inches. Increasing the piston/bore clearance, however, did not correct the problem.

The compressor was then disassembled. Inspection showed that the motor rotor and stator were rubbing. The friction and heat from this rubbing caused the compressor overload protection to shut the motor off. The potential causes of this were bearing misalignment, crankshaft deflection, and/or thermal expansion of the rotor/stator.

Further inspection of the compressor revealed some displacement of the crankshaft due to bearing play and misalignment. Misalignment of the bearings was corrected using precision shoulder screws to align the bearing plate. The bearing clearance is 0.0008-0.0016 inches, which contributes to the rotor touching the stator. Some deflection of the crankshaft at full load conditions also contributed to the problem. The rubbing of the rotor and stator was confirmed using temperature indicators on the compressor components. Temperature indicators on the stator showed temperatures in excess of 215°C. Temperature sensors on the cylinder head and housing indicated temperatures of approximately 120°C.

Methods to minimize the bearing clearance were investigated. This could be accomplished by grinding the OD of the crankshaft to match the bearing ID. However, this method would require an exact measurement of the bearing ID using a plug gauge, which would require selection of the appropriate plug gauge with the correct diameter. This is not practical due to the variation in bearing tolerance. It was also believed that this would not likely diminish the shaft displacement.

We next investigated modifications to the stator. We added alignment pins where the stator fastened to the compressor housing to minimize the play between the stator and housing. We investigated opening the ID of the stator magnets using a honing stone. Our approach was to open the stator ID in 0.002 inch increments radially. Significant improvement was seen in the first 0.002 inch increment. The second 0.002 inch increment appeared to eliminate the problem, resulting in several experiments of duration of 2-3 hours each, without motor shut-down.

Upon further verification testing and inspection, we noticed that the connecting rod had cracked due to the piston hitting the head. Inspection of the machined crankshaft showed significant deflection, contributing to the piston hitting the head. Replacement of the machined crankshaft with the hardened, ground crankshaft eliminated this problem.

## Performance Testing of Prototype Compressors

A second prototype compressor was fabricated using the same modification as the first prototype compressor. Prototype compressors #1 and #2 were then tested on the heat pump test stand using the same water supply temperatures as the unmodified compressors. Table 2 summarizes the performance of the prototype compressors along with the average performance of the unmodified compressors.

| Table 2 - Compressor Performance Comparison | | | |
|---|---|---|---|
| | **Unmodified (Avg. of 8)** | **Prototype #1** | **Prototype #2** |
| Power Consumption | 533 W | 551 W | 575 W |
| Cooling Capacity | 879 W | 705 W | 832 W |
| COPc | 1.59 | 1.28 | 1.45 |
| Suction Pressure | 478 kPa | 503 kPa | 502 kPa |
| Discharge Pressure | 1.65 MPa | 1.52 MPa | 1.63 MPa |
| Evap. Superheat | 6.2 °C | 4.8 °C | 5.1 °C |
| Cond. Subcooling | 3.8 °C | 0.4 °C | 3.2 °C |

Table 2 shows that the performance of the modified compressor is only slightly lower than the unmodified compressor. Power consumption rose 3.4%, which is considered negligible since it is within 2 standard deviations of the average power consumption of the unmodified compressors (see Table 1). Cooling capacity dropped 19.5%. The drop is cooling capacity is likely the result of a reduction in compression efficiency, caused by two phenomena:

1. The piston/bore clearance in the modified compressor, which was increased from 0.0007 inches to 0.0017 inches during verification testing. The piston/bore clearance in the unmodified compressors is 0.0007 inches. This increased clearance results in leakage around the piston during the compression process, resulting in less gas getting compressed with each stroke, lowering cooling capacity.
2. The discharge temperature of the modified compressor was higher than the unmodified compressor (approximately 190°F versus 132°F). This is an indication of increased friction in the modified compressor, converting some of the power supplied to the compressor to heat, thus lowering the compression efficiency. This is not a surprising result, since oil lubrication is preferred over the sealed bearings, a polyimide lubricated piston, and a polyimide thrust washer used in the modified compressor.
3. The clearance volume when the piston is at top-dead-center may be greater in the modified compressor. During verification testing, some material was taken off the piston, which would increase this clearance volume. This would decrease the volumetric efficiency of the compressor.

## COMPRESSOR ORIENTATION TESTING

The modified compressor was installed in the COTS centrifuge for orientation testing. The refrigeration system was charged with 170 grams of R-404A refrigerant (the factory charge). For orientation testing, the set point of the centrifuge was set at 6°C. The centrifuge was operated for one hour on each of the six sides of the centrifuge. The test objective was to confirm proper operation at all orientations and to monitor the total compressor run time and duty cycle (the compressor cycled on and off during the test). After each test, the centrifuge was allowed to warm up to room temperature prior to initiation of the following test.

Table 3 shows the results of the orientation tests, showing the total compressor run times and duty cycle for each orientation. The variation in the total run time and duty cycle seen in Table 3 for the various orientations is the result of the effect of gravity orientation on the performance of the condenser and capillary tube expansion device. The condenser is designed to operate in an orientation with gravity where the vapor enters at the highest location and the liquid leaves at the lowest location. Thus, liquid flows in a direction where it is assisted by gravity, supplying subcooled liquid to the capillary tube. As the centrifuge was place in other gravity orientations, liquid was forced to flow against gravity. This could result in hold-up of the liquid at a low spot in the condenser, resulting in a two-phase liquid-vapor mixture entering the capillary tube (since vapor flows against gravity much easier than liquid). This would decrease the cooling capacity of the refrigeration system, since the refrigerant entering the evaporator would have a higher vapor quality than intended. This explains difference in compressor run times and duty cycles for the various orientations with gravity.

**Table 3 – Orientation Testing Results**

| Orientation | Total Run Time | Duty Cycle |
|---|---|---|
| Upright | 22 min. | 36.7 % |
| Left Side | 45 min. | 75.0 % |
| Right Side | 46 min. | 76.7 % |
| Top Down | 34 min. | 56.7 % |
| Back Down | 60 min. | 100 % |
| Front Down | 22 min. | 36.7 % |

## COMPRESSOR LIFE TESTING

The purpose of compressor life testing was to demonstrate that the modified compressors could operate for 4000 hours at a 90% duty cycle. The life test stands consist of a modified compressor, an air-cooled heat exchanger, a commercial filter-drier, and a hand expansion valve. Thus, the life test cycle is a vapor cycle. The test stand is instrumented with thermocouples on the compressor suction line, discharge line, and compressor body. Pressure gauges are installed on the compressor suction and discharge lines. The refrigerant charge and hand expansion valve are adjusted to provide a suction pressure of approximately 480 kPa and a discharge pressure of approximately 1.55 MPa. An hour meter and timer are installed on the compressor's electrical supply to monitor and control the compressor run time. The timer is set so that the compressor is on for 15 minutes, 40 seconds and off for 1 minute, 43 seconds. This is a 90.12% duty cycle, and will provide 15,318 cycles over the 4438 hours of testing. Life testing of the compressors is currently underway.

## CONCLUSION

Mainstream successfully modified and tested an oil-less compressor for operation in a zero-gravity environment. Performance testing of the modified compressors showed negligible difference in power consumption compared to the unmodified compressors and only a 19-20 % decrease in cooling capacity and COPc. Orientation testing was successful in demonstrating operation of the compressor in all six gravity orientations. Life testing, to demonstrate 4000 hours of operation at a 90% duty cycle, is underway.

## REFERENCES


1. Grzyll, L. R., Scaringe, R. P., and Gottschlich, J. M., "The Development of a Performance-Enhancing Additive for Vapor-Compression Heat Pumps," *Proceedings of the 32nd Intersociety Energy Conversion Engineering Conference,* 1252-7, 1997.