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Further Results of a Rotary Compressor for an Aircraft Pod Cooling System

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The design of compressors for the fighter aircraft environment is a continuous challenge. The size, weight, and power draw must constantly be improved. This paper is a sequel to one delivered at the 1986 conference. It updates many of the changes in the LANTIRN compressor as this program enters the production phase.

The application of this compressor is for an Environmental Control Unit (ECU) used on the Low Altitude Navigation and Targeting Infrared for Night (LANTIRN) pod system. These electro-optical pods are flown on the F15 and F-16 fighter aircraft.

The ECU described previously houses a fully contained R114 vapor compression system for cooling of electronic components at ambient extremes of +48 to +205. The compressor is a semihermetic, motor driven, sliding vane, rotary piston type. The 15 built in the development phase of the program have proven successful despite the application.

Several major changes occurred between development and production. Most notable was an increase in capacity requirements and the addition of a microprocessor controller. These changes were required without increasing the package size and only 6 pounds of additional weight. To accomplish this the compressor speed was increased from 3700 RPM to 5600 RPM and the volume was reduced 25%.The speed increase and volume reduction were not accomplished without bringing on some unexpected problems.

Specific areas discussed in this paper include the updated design requirements, the second generation compressor development, oil inventory control problems with respect to package constraints, and the benefits and downfalls of higher speeds.

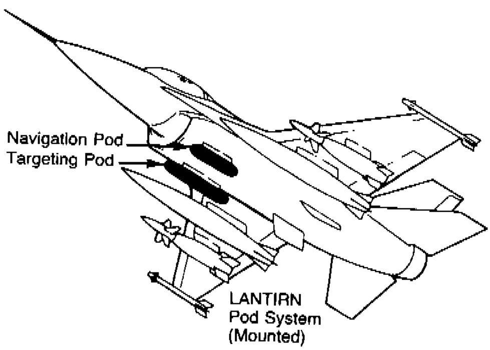


FIGURE 1

SYSTEM DESCRIPTION

Shown in Figure 1 is the LANTIRN pod set configured for use on an F-16 fighter aircraft. Martin Marietta Electronic Systems Division developed the system under contract with the United States Air Force. The current order is for 700 pod sets. This system allows fighter aircraft operation at low altitudes, in the dark, under adverse weather conditions.

The ECU has undergone an extensive redesign in preparation for production. Most significant to the ECU are:

1. Net cooling capacity increased from 8500 BTU/HR (1800 watts) to 11600 BTU/HR (2600 watts).

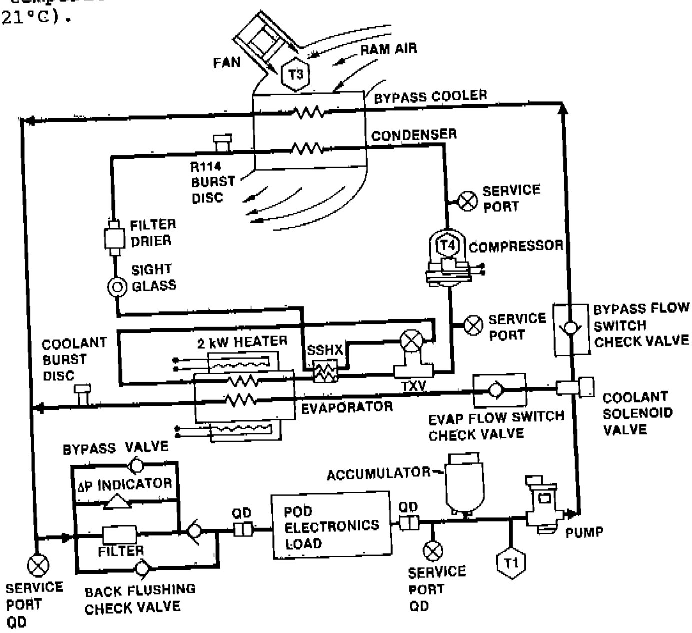
2. An increase in design point speed from Mach .81 to Mach .85.

3. Inclusion of a microprocessor for control logic and Built In Test (BIT) sequencing.

4. Modular component packaging

5. A ducted ram air circuit to protect components from high velocity air flow.

The system (see Figure 2) is designed for continuous operation at ambients from -40°F to +192°F (-40°C to 89°C), while maintaining coolant temperatures being delivered to the pod between +40°F and +86°F (5°C and 30°C). The heart of the system, the vapor compression unit, remains an R114 system with a sliding vane rotary compressor, evaporator, condenser, subcooler superheater heat exchanger, and thermal expansion valve. The vapor compression system is on any time the coolant temperature is above 61°F (16°C) and ram air temperature entering the condensing heat exchanger is above 70°F (21°C).



ECU FLUID SCHEMATIC
FIGURE 2

DEVELOPMENT OF THE PRODUCTION COMPRESSOR

Five years of flight testing in the development program had provided the opportunity to refine the ECU into a remarkably dependable package. It was a logical decision to redesign for the new production requirements using the basic building blocks of the development ECU.

To meet the higher performance requirements only two options were available if the rotary compressor was to be used: Higher speeds or an increase in displacement. The pump used in the development program was already operating at its maximum displacement. Any increase would require a larger pump body.

The next readily available size pump would not be optimum in terms of displacement vs. pump size. An over size pump means extra weight and size of which there was none to spare. This led to increasing the speed.

The rotary compressor used in the development program could meet the performance requirements if the speed was increased from 3700 RPM to 5600 RPM. This accounts for the additional 3100 BTU/HR (800 watts) of heat load and the higher condensing pressure. Field inspections indicated that the moving parts had plenty of wear resistance operating at 3700 RPM and may be able to handle 5600 RPM. A prototype was needed to test the hardware at the higher speeds.

Three factors entered into sizing the motor: speed, higher discharge pressure, and a different motor efficiency. Besides the normal losses from higher speeds such as windage and friction, a larger than normal air gap would have to be used because of increased shaft deflection. At 3700 RPM the rotor would deflect .006". At 5600 RPM this deflection increases to .009".

Early results were only partially successful. The cylinder, roller and bearing plates appeared to handle the higher speeds quite well but the nose of the aluminum silicate vane wore out in 30 hours.

The wear problem was eventually cured by changing the vane material to 7330 Torlon. Torlon is a Thermoplastic made with a blend of carbon fiber and fluorocarbons. 7330 was developed specifically for sliding vanes. It offers a relatively high strength (26 kpsi) along with some lubricity.

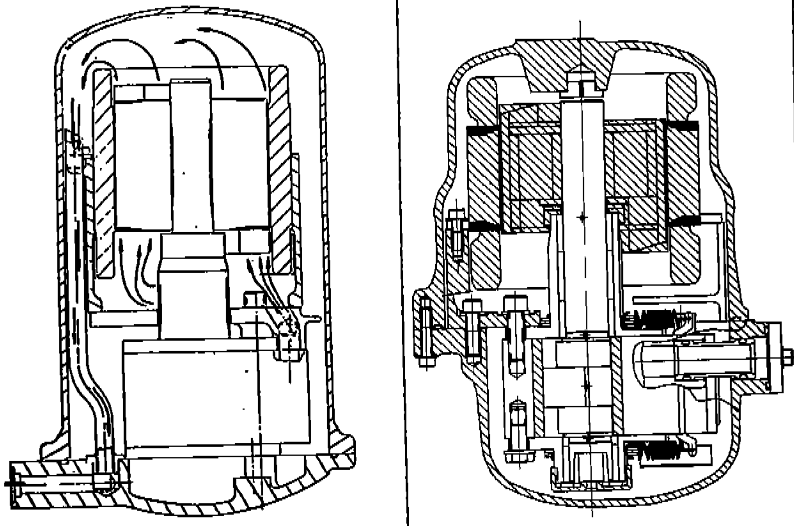
With the Torlon vane fitted into the prototype compressor, the wear problems went away. An endurance test was done to evaluate the long term life of the machine. After 500 hours of testing, a detailed inspection was done. The results were a complete success. Measured wear indicated that the dependability of the machine would not be compromised at these higher speeds.

The next chore was to package this compressor more efficiently to make room for the controller and reduce the weight. All previously designed compressors, including the one used in the development program, were made with a dome having a large enough diameter to house the entire assembly. This dome then bolted to a base plate that also secured the pump/motor assembly. Although simple to construct, it was not the most space conscious.

The production configuration housing molds around the items. The sump is molded around the pump body which drastically reduces the size at the bottom. The upper section features a patented stator design that eliminates the large stator support housing.

The stator is supported by oversizing the bottom 10 laminations and using them as a bolting flange. This type of stator is lighter and allows the dome to have a reduced diameter in the upper section.

The production compressor, in its final configuration, weighs 1.8 lbs. This is only 1.1 lbs heavier than FSED but 25% smaller.



ENGINEERING DEVELOPMENT

PRODUCTION

ECU COMPRESSORS
FIGURE 3

LUBRICATION INVENTORIES AND SEPARATION TECHNIQUES

When the first production ECU builds came out some problems developed. Repeated lock-ups of the compressor after only a short time of operation drew much concern. Disassembly of the failed compressors revealed large amounts of galling and pitting of the bearing plates where the roller passes. A lack of lubrication was the most highly suspected cause of these lock-ups.

During operation the oil level would always be at the bottom of the lower bearing plate regardless of fill level or quantity added. This had always been noted in other compressors but not as severe.

A comparison was made between the engineering development design and the production design (see figure 3). Refrigerant in the production compressor is discharged from the pumping cylinder into the stator support cup. From here it travels up the rotor stator gap to be discharged out the opening in the top of the dome. Oil separation was intended to occur at the top of the rotor. The spinning action of the rotor would force the oil outward, toward the wall, to drain back to the sump while the lighter refrigerant gas stays in the center and discharges out the top.

The engineering development compressor has a more natural flow path with respect to oil return. The refrigerant goes up the rotor stator gap as in the production design but then turns and comes down the outside wall. The large open passage gives a long residence time and low velocities to allow for oil drop out.

Table 1 shows a comparison of some of the pertinent parameters relative to oil-refrigerant flow between the two designs. It was noted that velocities in areas critical for oil drop out were considerably higher in the production design. But what criteria should be used to determine the velocities needed for oil carry over?

COMPRESSOR	ENGINEERING DEVELOPMENT	PRODUCTION	MODIFIED PRODUCTION
MASS FLOW (lb/min)	3.7	5.5	5.5
FLOW AREA (in^2)	3.93	.54	1.23
MASS FLUX (lb/sec/ft^2)	2.25	24.4	10.6
Min Mass FLux (lb/sec/ft^2)	34.6	21.0	25.9
OIL CARRY OVER	NO	YES	NO

TABLE 1

The technique used for this compressor was to employ standard oil carry over techniques for refrigerant piping. Specifically, a paper written for ASHRAE by Jacobs, Scheideman, Kazan and Macker (No.2423) was referenced whereby a "mass flux" or flow rate per unit of area is calculated. In this paper Jacobs et al give a method of calculating a minimum mass flux which they believe will ensure oil carry over. Their interest in such a calculation is the opposite of our desire here. Their assumption is that oil is pushed upward along the walls and held in suspension by the gas flow as long as the gas momentum is high enough to support the weight of the lube. The lube, of course, is a mixture of oil and condensed refrigerant which has been absorbed. The ability of this oil to move with the gas stream is a function of the mass flux given as:

$$G = \frac{\dot{m}}{A} \tag{1}$$

And the density of liquid lube mix which Jacobs et al gives as:

$$\varrho_f = \frac{\varrho_0}{F} \cdot \frac{1}{1 + R\left(\frac{\varrho_0}{\varrho_l} - 1\right)} \tag{2}$$

Jacobs et al then give the following as a means of determining the minimum mass flux needed to keep oil in suspension:

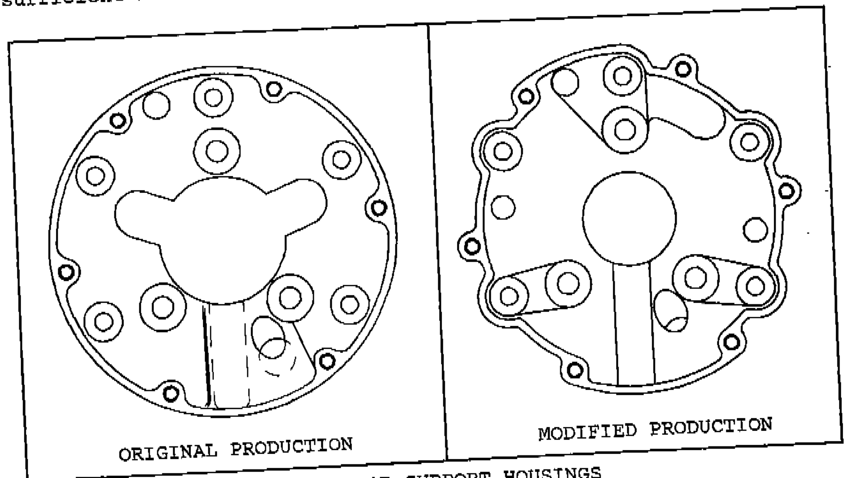
$$G = .723\sqrt{\varrho_g \cdot g \cdot D_{eg} \cdot (\varrho_f - \varrho_g)} \tag{3}$$

Sample calculations are shown in Appendix 1.

Mass flux densities within the compressor were calculated using an equivalent diameter. As shown in Table 1, the mass flux densities calculated are such that good oil return should occur in the development compressor but not necessarily the production compressor which, in fact, was the occurrence.

Figure 4 shows the redesigned stator support housing that features a revised flow path. This design increases flow area while not increasing the outside package. Drain holes were added to the stator support cup. This allowed for both oil drainage of the cup and also allowed for some gas to go up the outside of the stator. The stator cup was also cut back and reshaped to allow for additional flow and drain back area.

As can be seen from table 1, the mass flux was changed so that oil drop out should occur. Some concern was raised about stator cooling at this point, but with the stator mounting method used as described, it was assumed that the flow impinging directly on the outside of the stator and over the top end turns, would be sufficient for cooling.



STATOR SUPPORT HOUSINGS
FIGURE 4

Rebuild of the production unit showed that oil control did return. The oil level in the sump could be controlled at any level during operation.

SPEED EFFECTS ON ROTATING HARDWARE

Despite the tremendous headway made on oil level control the lock-ups did not go away. Since the failed compressors would always show galling on the bearing plates where the roller wears, the solution seemed to be increase the roller end clearance. This approach still did not relieve the problems even using roller end clearances as high as .004". Mostly by trial and error, it was discovered that these higher speed compressors were far more sensitive to operating clearances than the 3700 RPM compressors.

The roller to cylinder wall clearance seemed to be the single biggest cause of the lock-ups. All compressors previously built were set to have .0005" to .001" roller to cylinder wall clearance. Setting these high speed compressors at anything less than .0010" would guarantee a lock-up. When set to .0015"-.0020" no galling would appear even with roller end clearances as tight as .0014.

Handling of the parts has become intensely critical. Small dings and burrs that would have been ignored in the past, are catastrophic to this machine. Deburring done by the manufacturer must all be rechecked carefully prior to assembly. Corner breaks on this compressor are double of other compressors.

Despite the larger roller to cylinder wall clearance and generous corner brakes used, this compressor has better volumetric efficiency. It appears that the higher speed more than compensates for the larger leak paths. Table 2 provides a comparison of performance between the ED and the production compressor.

COMPRESSOR	CYLINDER SET	ROLLER END CLEARANCE	CORNER BREAKS	VOLUMETRIC EFFICIENCY	OVERALL EFFICIENCY
DEVELOPMENT	.0010"	.0006"	.0020"	85.2%	38.2%
PRODUCTION	.0018"	.0018"	.0080"	92.5%	45.7%

* Efficiencies measured at 15 psig suction pressure with 40°F superheat and 155 psig discharge pressure.

TABLE 2

CONCLUSIONS AND FUTURE PLANS

The primary conclusions of this compressor program are as follows:

1. The commercially available compressor parts used for this application are adaptable to higher speed rotation, specifically 5600 RPM. The assembly of parts, finishes and clearances becomes even more critical, but it is an acceptable approach to reduction of package volumes for use on fighter aircraft.

2. Package volume reduction can bring on problems with oil carry over, but traditional methods for calculating oil carry over in evaporators and piping systems can be used to also prevent oil carry over in compressors.

3. The rotary compressor has proven itself the right choice for pod mounted aircraft cooling.

Compressor optimization continues to be an active effort in spite of ongoing production. The compressor represents 25% of the total ECU weight. Weight is always a high priority on fighter aircraft, and efforts are underway to reduce its weight. Alternate materials for bearing plates are being investigated to improve wear resistance for high speed application.

SYMBOLS

G = *Mass flux*
G_{min} = *Mass flux needed* for *oil carry over*
m = *refrigerant mass flow*
A_g = *airgap flow area*
A_w = *wall flow area*
ϱ_f = *density of oil liquid refrigerant – oil mixture*
ϱ_o = *density of oil*
F = *density correction factor*
R = *frigerant – oil solubility*
ϱ_l = *density of liquid refrigerant*
ϱ_g = *density of refrigerant gas*
D_g = *nominal diameter of rotor gap*
X_g = *air gap*
D_w = *nominal diameter of gap at wall*
X_w = *nominal gap at wall*
A_t = *total area*
g = *gravitational constant*

APPENDIX I

The calculations shown here are for the as-built configuration assuming best flow split. Part of the refrigerant flow goes up between the rotor air gap and part of the flow goes up the outside of the stator. The area of the air gap is:

$$A_g = \pi \cdot D_b \cdot X_g = \pi(3.361)(.009) = .095 in^2$$

$$A_w = \pi \cdot D_w \cdot w_w = \pi(5.625)(.025) = .442 in^2$$

$$A_t = A_g + A_w = .537 in^2$$

$$\dot{m} = 5.47 lb/min$$

$$G = \frac{m}{A_t} = \frac{(5.47)\ (144)}{(60)(.537)} = 24.4 lb/sec\ ft^2$$

$$\varrho_F = \frac{\varrho_0}{F} \cdot \frac{1}{1 + R\left(\frac{\varrho_0}{\varrho_l} - 1\right)}$$

$$\varrho_0 = 53.0 lb/ft^3$$

$$R = .60 (for\ 210\ psia \text{ and } 250°F)$$

$$F = .92\ (From\ Jacobs)$$

$$\varrho_l = 73.5 lb/ft^3\ (for\ 210\ psia)$$

$$\varrho_f = 69.2 lb/ft^3$$

$$G_{min} = .723\sqrt{\varrho_g \cdot g \cdot D_{eg} \cdot (\varrho_f - \varrho_g)}$$

$$D_{eg} = \sqrt{\frac{4A_t}{\pi}} = \sqrt{\frac{4\ (.537)}{\pi}} = .827\ in = .069\ ft$$

$$\varrho_g = 6.06\ lb/ft^3 \text{ for } 210\ psia \text{ and } 250\ degreeF$$

$$G_{min} = .723\sqrt{6.06\ (32.17)\ (.069)\ (69.2\ -\ 6.06)}\ =\ 21.0$$

$$G > G_{min}$$

$\therefore$ *Oil Should Carry Over*