1986

Design and Test Results of a Sliding Vane Rotary Compressor for an Aircraft POD Cooling System

W. J. Godecker

W. C. Johnson

Godecker, W. J. and Johnson, W. C., "Design and Test Results of a Sliding Vane Rotary Compressor for an Aircraft POD Cooling System" (1986). *International Compressor Engineering Conference.* Paper 570.
https://docs.lib.purdue.edu/icec/570

DESIGN AND TEST RESULTS
OF A SLIDING VANE ROTARY COMPRESSOR
FOR AN AIRCRAFT POD COOLING SYSTEM

William J. Godecker
Group Engineer

William C. Johnson
Senior Project Engineer

Sundstrand Pneumatics Systems - Rockford, IL

ABSTRACT

This paper details the design and test results of a sliding vane rotary compressor for an aircraft pod cooling system.

The application for this compressor is the environmental control unit (ECU) for the Low Altitude Navigation and Targeting InfraRed for Night (LANTIRN) electro-optical pod system.

The ECU described below houses a fully contained R114 vapor compression cooling system for cooling electronic components at ambient extremes of +60°F to +220°F. The compressor was semihermetic, electric motor-driven, using discharge gas for motor cooling and of the sliding vane rotary compression-type, operating at 3700 rpm.

The final design and build of production compressors for a vapor compression system operating in a supersonic fighter aircraft environment, turned out to hold both interesting and formidable challenges. Traditional concepts for allowable superheats and refrigerant flow passages through a compressor had to be re-examined during development testing.

Specific areas discussed include compressor requirements for high speed aircraft-borne pods, R114 compressor design, and test results. Highlights of problem areas such as oil absorption and refrigerant condensing in the compression cycle were analyzed. Test results and theoretical predictions were used to show the effects of suction superheat, refrigerant flow passage, and heat transfer through the compressor shell, on refrigerant-oil solubility and ultimately on performance.
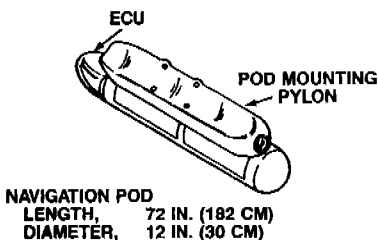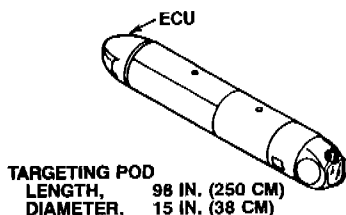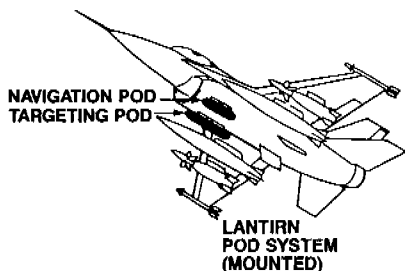
LANTIRN POD SYSTEM DESCRIPTION

LANTIRN is an electro-optical system being developed for high performance fighter aircraft by Martin Marietta Electronics Systems Division in Orlando Florida under contract with the United States Air Force. The system mounts to the fuselage of the aircraft in two separate pod packages (Figure 1). The system is currently envisioned for use on the F-15, F-16, and A-10 aircraft.

The navigation pod uses a forward looking infrared system transmitted to the cockpit via holographic display. Along with low level terrain following radar information, this pod allows flight at an altitude of 200 feet (61 meters) altitude in the dark under adverse weather conditions. The targeting pod acquires and tracks targets for laser-guided weapons and displays target images in the cockpit.

This program has reached the production phase after nearly three years of development testing. During testing, over 1,500 mock sortie missions

were flown, mostly at night with the system operating in excess of 2,000 hours in-flight time.

**LANTIRN Pods Showing
ECU Locations
Figure 1**

Environmental Control Unit
Description

Requirements to adapt this system to the host aircraft with a minimum amount of modification and to keep the pod-to-aircraft interface as simple as possible made it infeasible to provide temperature control of pod electronics using aircraft-mounted environmental control equipment. Therefore, each pod contains its own ECU. The LANTIRN ECU packages are complete, self-contained modular units comprising the aft section of the pods. The pod electronic packages are thermally coupled to the ECU through a liquid

cooling loop through which Coolanol 25, is circulated. The primary requirements of the ECUs are to circulate coolant throughout the pod's liquid-loop system and provide for thermal expansion of the system while maintaining coolant temperature being supplied to the pod between 40° and 86°F (4 and 30°C) over the complete range of LANTIRN missions and continuous ground operation.

The mission profiles for this system include speeds up to Mach 1.2, altitudes from sea level to 72,000 feet (22,150 meters) and ram air temperatures of -40°F to 220°F (-40° to 104°C).

Coolant temperature control is accomplished by the ECU as follows:
1. At coolant temperatures below 37°F (3°C), two 1,000 watt heaters are energized to heat the coolant.
2. At coolant temperatures between 37°F and 67°F (3°C and 17°C), the ECU neither adds nor rejects heat from the coolant.
3. At coolant temperatures above 67°F (19°C) and with ram air temperatures below 62°F (16°C), heat from the coolant system is rejected directly to the airstream through an air-to-coolant heat exchanger.
4. At coolant temperatures above 67°F (19°C) and with ram air temperatures above 62°F (16°C), an R114 vapor-cycle system is energized to remove heat from the coolant system.

This system is shown schematically in Figure 2. The R114 system consists of a sliding vane rotary compressor, evaporator, condenser, and subcooler-superheater heat exchangers and thermal expansion valve. Refrigerant 114 was chosen for this system due to its stability and relatively low vapor pressures at the high condensing temperatures dictated by the LANTIRN application.

Compressor Requirements

Significant design parameters for this compressor included: thermal performance, power draw, weight, package and structural integrity. The compressor for this system has to operate anytime ram air entering the condenser is between 62° and 220°F
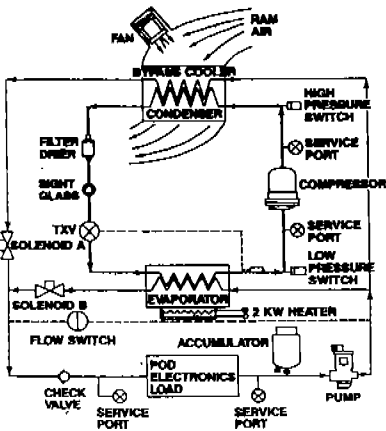
LANTIRN ECU
System Schematic: With SSHX
Figure 2

(20 and 104°C) and coolant temperatures greater than 67°F (16°C). Figure 3 dramatically shows the operating range of this compressor superimposed on an R114 pressure-enthalpy chart. Heat generated within the pod amounted to 6,142 Btu/h (1,800 watts) of thermal energy. This combined with 1,365 Btu/h (400 watts) of heat from the coolant pump, along with an estimated 1,023 Btu/h (300 watts) of internal losses gave a net cooling requirement for the compressor at the design point of 8,530 Btu/h (2,500 watts). Besides the thermal requirements, significant limitations on weight, package and power consumption were imposed. Table A summarizes the design requirements of this compressor. The compressor is also subjected to significant vibratory and structural loads.
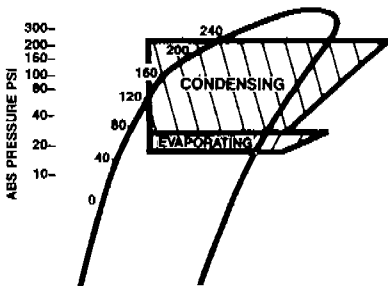

LANTIRN Operating Range
Figure 3

Table A Design Particulars

Cooling Requirement--	8,500 Btu/h 2,500 watts
Suction Pressure at Design Point	30 psia
Discharge Pressure	170 psia
Superheat	40°F
Maximum Continuous Power Draw	2300 watts
Minimum Condensing Temperature	70°F (21°C)
Maximum Condensing Temperature	230°F (110°C)
Minimum Evaporating Temperature	55°F (13°C)
Maximum Weight	21 lbs

Compressor Design

A sliding vane rotary compressor was chosen for this application because of its relatively high efficiency and overall rugged and reliable design. Figure 4 shows the cross-section of the rolling elements of this compressor. A 2.856 cubic-inch-per-revolution displacement was chosen. Figure 5 shows a cross-section of the compressor as it was originally configured.
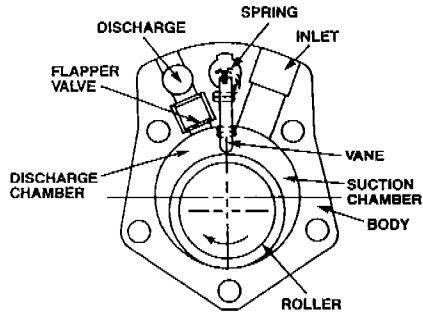
Rolling Element Cross-Section
Figure 4

The housing was semihermetic, using a cast aluminum base with a hydroformed aluminum dome. The motor was a 12-pole, 400 hertz design operating at 3,700 rpm. The motor was cooled using discharge gas.

The compressor subassembly used upper and lower carbon journal
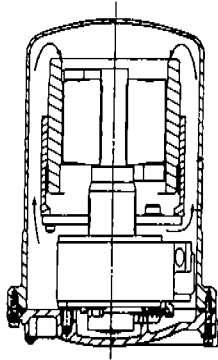
Phase I Compressor Cross-Section
Figure 5

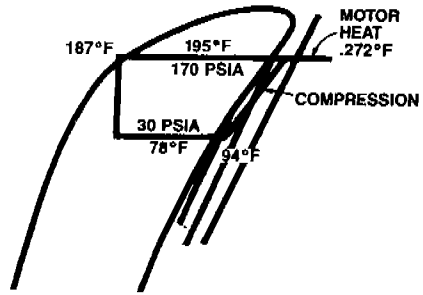

Cycle Diagram
Figure 6

bearings. Lubrication was accomplished through an oil-filled sump with oil being drawn up the center of the drilled shaft by centrifugal action.

Because of package constraints, it was necessary to have the suction and discharge fittings at the bottom of the compressor. Refrigerant flows up through the motor and then down to an internally mounted discharge tube. Oil return to the sump was accomplished within the dome as flow velocities reduced after the motor passage.

The design point for the ECU performance was flight at Mach .81 at sea level on a 110°F (43°C) day. This condition yields an inlet air temperature to the condenser of 185°F (85°C). Matching the heat exchangers with the compressor yielded requirements of 8,530 Btu/h (2,500 watts) at 170 psia discharge and 30 psia suction pressures.

Development: Phase I

The system as originally configured did not have a suction-to-liquid heat exchanger. Figure 6 shows the operating cycle of this configuration thermodynamically. As modelled, the system was intended to run with a maximum of 15°F (8°C) of superheat entering the compressor inlet. The system was intended to be run very much like an R12 system. By keeping inlet superheat down as close to saturation as possible, it was believed optimum system performance would be obtained since the evaporator would be

used primarily for boiling and secondarily for superheating the gas.

Because of a tight development schedule for the system, the compressor was installed with only minimal prior testing and no instrumentation. Significant system problems were found in early testing. Performance output was less than 50 percent of rated value, but more significant was the inability to control refrigerant inventories.

The refrigerant volume calculations estimated that 1.5 lb. of refrigerant was needed to fill the system; yet, at times, charging the system with as much as 5 lb. would not yield a clear sight glass. In early testing with only a few hours on the system, two compressor failures were encountered. Teardown investigation revealed journal bearing failures which appeared to be due to lack of lubrication, in spite of the fact that a full oil sight glass had been observed during operation.

A closer analysis of the cycle was begun. Looking at the thermodynamic cycle more closely, refrigerant vapor at the compressor inlet was compressed, in the case of the design point, from 30 psia to 170 psia at something less than a fully isentropic process. The vapor was then routed through the motor "air" gap where it was heated at nearly constant pressure. See figure 6. Looking at the compression process closely, it was noted how vertical the isentropic lines are for R114 compared to common refrigerants.

The rotary compressor being used was thought to be approximately 50 percent efficient in the compression phase. Looking at the inlet conditions

803

for 30 psia and 15°F superheat, a truly ideal compression process would have resulted in an end point well to the left of the saturated vapor line. The enthalpy change for such a process would be approximately 10 Btu/lb. Using standard calculations, then the enthalpy change for a process at 50 percent efficiency is 20 Btu/lb. This moves the discharge of the compressor body to the right of the saturated vapor line, but only by 25°F.

Could a portion of the refrigerant flow condense at the compressor body outlet before it was sent through the motor for additional heating? An examination of the flow path in the compressor revealed that refrigerant was routed out of the compressor discharge port, around the outside of the stator. From here, it traveled over the end turns, down through the motor gap and into the discharge tube, (Figure 5).

Supporting evidence to this possible "condensing in the dome" theory is as follows:

1) The missing 3 lbs. could have easily been stored in the compressor dome.
2) The journal bearing failures could have been explained by liquid refrigerant displacing the oil in the sump and thus lubricating the bearings with freon instead.

The compressor dome itself, while not directly in the airflow path, was in the same compartment as the condensor, where 20 lb/min to 200 lbs/min of air was circulated depending on the flight condition being simulated. While it was realized the calculated process should preclude condensing refrigerant within the dome there were conditions that might lead to it. These include:

1) Refrigerant gas passing up the dome would be cooled on the wall.
2) In the actual system superheat would drop below 15°F. This would have the effect of moving the discharge point further to the left.
3) Within the discharge gas flow path, refrigerant would be absorbed into the oil which is also being pumped in this path. The ability of the oil to absorb refrigerant is directly related to how close to saturated vapor the

refrigerant is. In the absorption process, Refrigerant vapor is condensed to a liquid.

Development Phase II

At this point, the schedule for producing 12 working units for flight testing dictated that a fix was required rather than a deliberate development program.

The configuration of the compressor was changed to that shown in Figure 7. Of significance here is the refrigerant flow path, which now has the discharge gas routed immediately up through the motor gap and then down the shell and into a discharge tube. In addition, the compressor shell was insulated to further limit heat transfer.

Limited testing of the compressor itself was then done to determine in gross terms, the effects of inlet superheat and shell heat transfer on performance. Table B shows some significant points from this testing:
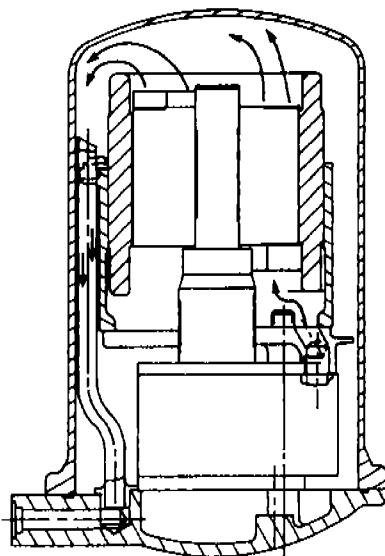
Phase II Compressor Cross-Section
Figure 7

Based on these results, the system was configured with the following changes:
1) The compressor flow path was as shown in Figure 7.
2) The shell was insulated.
3) A suction-to-liquid heat exchanger was installed to raise the superheat temperature, while allowing the evaporator to be used primarily for evaporating. By design, this would provide for 40°F of inlet superheat at design point.

The system operated as planned, delivering capacity within power draw requirements.

Data Point 4 is perhaps as close as the new system could be simulated. Volumetric efficiency improved over Data Point 3.

Table C shows a comparison of some of the relevant mass flow calculations for these points. An attempt was made at this point to account for the differences in performance for these four points.

The rotary compressor is a constant volume pumping machine where volumetric efficiency is defined as:

$$\eta_{VOL} = \frac{\dot{W}_{NET} + \dot{W}_{LOST} + \dot{W}_{OIL}}{\dot{W}_{THEO}}$$

Table B
Phase II Test Results

Run No.	Suction Press Psia	Disch Press Psia	Inlet Supht F	Disch Temp. F	Volum Effic. %	Air Over
1	30	168	10	223	80.0	No
2	30	171	41	253	85.2	No
3	30	163	7	211	72.9	Yes
4	30	170	37	245	78.5	Yes

Table C
Phase II Test Results - Continued

Run No.	Massflow Calcul lbs/hr	Massflow Theore lbs/hr	Solub- ility %	Massflow Oil lbs/hr	Massflow Absorbed lbs/hr	Volum Effic. %
1	282.3	352.8	40	28.2	18.8	85.3
2	281.6	330.2	10	28.1	3.1	86.2
3	264.1	362.5	50	26.4	26.4	80.1
4	261.5	333.2	20	26.1	6.5	80.4

Condensing in the Dome and Oil-Freon Solubility

The system has since proven itself to the point where further analysis and testing are not warranted. However some further ideas were advanced.

Looking at Data Points 1 and 2 in Table B it is noted that a 5 percent improvement in volumetric efficiency was obtained by increasing the superheat from 10 to 41°F at the inlet.

Data Point 3 was performed to determine the effects of some forced cooling of the shell with low inlet superheats. It is noted that an additional 5 percent volumetric efficiency was lost as compared to Data Point 1.

For this exercise $\dot{W}_{LOST}$ was assumed to include:

1) Losses due to leakage and
2) Losses due to refrigerant condensing in the shell, a portion is due to oil refrigerant absorption, and a portion is due to refrigerant condensing on the cooler shell wall.

$\dot{W}_{OIL}$ is that volume flow rate of the pumping stroke which is taken up by the volume of the oil which gets pumped for lubrication purposes.

It is the volume flow rate lost to condensing in which we were interested here.

```
                              ---- 40%
                              ---- 30%
                              ---- 20%
                              ---- 10%
```
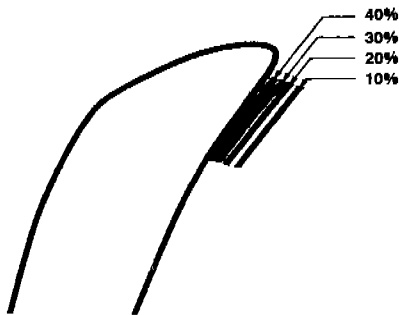
Oil-Refrigerant Solubility
Figure 8

Figure 8 shows a cross plot of oil-refrigerant solubility onto the pressure-enthalpy curve for refrigerant 114. Note that as the saturated vapor line is approached, solubility is increased. Table C also shows an approximate value for the solubility for each of the test points for the measured discharge gas conditions. For Data Point 1 gas exiting the motor gap is assumed to be 223°F (106°C) and 168 psia. The solubility is assumed to be in the range of 40 percent. Also shown in Table C is a value estimated for oil flow through the discharge circuit. Based on previous testing it was assumed that the oil flow was 10 percent of the total mass flow being pumped. Since the oil and the refrigerant are flowing in the same path and have a high affinity for each other, absorption is the natural consequence. This absorption will essentially be lost refrigerant mass flow since the refrigerant would condense as it was absorbed. The liquid refrigerant would then be drawn up into the bearings with the lubrication where it would be returned to the suction cavity as a vapor, completing the cycle.

Looking again at Data Point 1 in Table C it is noted 28.2 lb/hr is the projected oil flow rate in the circuit. At 40 percent solubility 18.8 lb per hour of refrigerant would be absorbed in the oil. Adding in this "missing" mass flow and recalculating the volumetric efficiency yields 85.3 percent. Using the same technique for data point 2 yields a recalculated volumetric efficiency of 86.2 percent for a value nearly the same as Data Point 1.

Data Points 1 and 2 are the static air cases. The attempt here was to account for some of the missing mass flow in the low superheat case as refrigerant being absorbed into the oil where it is effectively lost performance.

Data Points 3 and 4 are the dynamic air cases meant to simulate Mach 0.8. Using the same technique for these two points shows volumetric efficiencies are nearly identical to each other. When compared to Data Point 2 however, it is noted another 15 lb/hr of refrigerant flow is missing. Further analysis was not done to determine this.

Conclusions

The primary conclusion drawn from this exercise was that some of the losses in this particular R114 system could be accounted for as refrigerant condensing inside the compressor at least in gross terms. Ultimately, 15 of these systems were built and put into various flight test programs. Some 5,000 operating hours have been logged to date in what has turned out to be a rugged test for a vapor compression system. Flight test environments are not a haven for "good refrigeration practices" servicing.

Two failures have occurred to date in the field. One was due to a motor failure believed to be caused by improper installation of a motor phase separation insulator, the other due to loss of oil resulting from system leakage. This compressor has proven itself to be reliable and rugged enough for the highly dynamic fighter aircraft environment.

For the production version of this compressor for which some 1,400 compressors could be built two major changes will be made:

1) System capacity has been increased with no increases in package allowed for. In order to meet this requirement compressor speed has been increased to 5,650 rpm. and
2) The compressor will be mounted remote from the condenser air circuit to further reduce shell heat transfer.