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USAAVLABS TECHNICAL REPORT 70-41

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INVESTIGATION OF AN EXPERIMENTAL ANNULAR-SHAPED INTEGRATED TRANSMISSION OIL COOLER DESIGN

By

A. J. Lemanski

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September 1970

AD No. ___
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U. S. ARMY AVIATION MATERIEL LABORATORIES
FORT EUSTIS, VIRGINIA

CONTRACT DAAJ02-69-C-0055

THE BOEING COMPANY, VERTOL DIVISION

PHILADELPHIA, PENNSYLVANIA

U S ARMY MATERIEL COMMAND

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This report was prepared by The Boeing Company, Vertol Division, under the terms of Contract DAAJ02-69-C-0055. The objective of the contract was to design, fabricate, and environmentally test an annular-shaped, integrated transmission oil cooler with a radial airflow direction to determine its capability to reject the heat generated by a helicopter main rotor transmission, e.g., the CH-47C forward main rotor transmission.

It was found that it is both feasible and practical to produce such a cooler for both helicopter and V/STOL aircraft.

This command concurs with the conclusions presented in the report.

Task 1F162203A15003

Contract DAAJ02-69-C-0055

USAAVLABS Technical Report 70-41

September 1970

INVESTIGATION OF AN EXPERIMENTAL ANNULAR-SHAPED INTEGRATED TRANSMISSION OIL COOLER DESIGN

Final Report

D210-10119-1

by

A. J. Lemanski
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Prepared by

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Philadelphia, Pennsylvania

for

U.S. ARMY AVIATION MATERIEL LABORATORIES
FORT EUSTIS, VIRGINIA

SUMMARY

This program, conducted under U.S. Army Aviation Materiel Laboratories Contract DAAJ02-69-C-0055, consisted of the design, fabrication, and test of an experimental, aircraft-quality, annular-shaped oil cooler with a radial airflow direction. The program was undertaken as a follow-on to a previous study program under Contract DAAJ02-67-C-0112, titled "Relative Vulnerability and Cost-Effectiveness Study of Transmission Oil Heat Rejection Systems".

The oil cooler configuration evaluated by this program was conceptually designed for operation at the bottom of the CH-47C helicopter forward rotor transmission. This unit was designed, manufactured, and tested in accordance with the applicable CH-47C heat rejection and environmental conditions that prevail within the area of the forward rotor transmission.

The results of this program indicate that an annular-shaped oil cooler with a radial airflow direction is completely practical, from a producibility and manufacturing feasibility standpoint, for helicopter or V/STOL aircraft.

Based on the test results, the annular oil cooler fabricated during this program was highly successful in meeting all performance and environmental specifications. The testing included hydrostatic pressure evaluation, performance evaluation, pressure cycling, vibration evaluation, and cleaning.

FOREWORD

This final technical report concludes the manufacturing and engineering feasibility test program performed on an aircraft-quality, annular-type oil cooler with radial airflow direction. The program was conducted by the Advanced Drive System Technology Department of the Vertol Division of The Boeing Company under U.S. Army Aviation Materiel Laboratories Contract DAAJ02-69-C-0055, Task 1F162203A15003.

Technical direction was provided by Mr. James Robinson, Aerospace Engineer in the Safety and Survivability Division of the U.S. Army Aviation Materiel Laboratories.

The oil cooler was fabricated and tested by the Harrison Radiator Division of the General Motors Corporation, Lockport, New York. The effort at Harrison Radiator was directed by Mr. Robert Lockie of the Product Engineering Department, Industrial and Defense Section.

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A_D	direct surface area, ft^2
A_E	effective heat transfer surface area, ft^2
A_I	indirect surface area, ft^2
A_O	open flow area, ft^2
C_P	specific heat, Btu/lb-°F
D_H	hydraulic diameter, ft
E_f	fin efficiency, nondimensional
f	fanning friction factor, nondimensional
G	mass flow, lb/min-ft^2
h	convective heat transfer coefficient, Btu/min-ft^2-°F
itd	inlet temperature difference
j	Colburn factor, nondimensional
M	weight flow, lb/min
NTU	number of heat transfer units, nondimensional
Q	heat load, Btu/min
R	Reynolds number, nondimensional
T_{AVG}	average temperature, °F
T_F	film temperature, °F
T_{IN}	inlet temperature, °F
T_{OUT}	outlet temperature, °F
UA	overall heat transfer coefficient, Btu/min-°F
ε	effectiveness, nondimensional
μ	viscosity, lb/min-ft
ρ	density

INTRODUCTION

Operation of helicopters in the combat environment of Southeast Asia has revealed several distinct problem areas. One such area is the vulnerability of helicopter main rotor transmissions to oil loss as a result of combat hits in the lubrication system. In 1967, a study was initiated by USAAVLABS to investigate various methods of cooling helicopter transmissions which would demonstrate reduced vulnerability while still remaining cost-effective. The Vertol Division of The Boeing Company performed a study under Contract DAAJ02-67-C-0112 which used the CH-47 helicopter forward transmission as a baseline. This study analyzed nine systems, including current state-of-the-art methods and advanced methods of cooling the forward transmission of the CH-47 helicopter, and compared them to the existing oil/air-type cooling system. Each of the systems studied was evaluated with respect to vulnerability, weight, power consumption, reliability, maintainability, and system cost. The following systems were investigated during this initial study and are listed below in the order of the most to the least cost-effective.

1. Integral oil-air
2. Close-coupled oil-air
3. Oil-water/glycol-air
4. Oil-boiling refrigerant-air
5. Air-cycle heat pump
6. CH-47 production (baseline)
7. Heat pipe
8. Vapor cycle
9. Air cycle - air cooling
10. Ammonia absorption

The integral oil-air system listed above used an annular oil cooler of plate/fin-type construction with radial airflow and circumferential oil-flow directions. This type of oil cooler had not previously been fabricated by an aircraft oil cooler manufacturer; therefore, the feasibility of producing this type of oil cooler to aircraft-quality standards was questionable.

The objectives of the program reported herein were to:

- Establish criteria for the design of aircraft-quality annular heat exchangers and for the prediction of their performance.
- Verify or establish a fabrication method or process for an annular-type heat exchanger.
- Measure the annular heat exchanger performance and establish test data for use in future annular heat exchanger design work.

- Determine if a typical annular heat exchanger can pass production qualification testing as required on helicopter oil cooler assemblies.

TECHNICAL APPROACH

BACKGROUND

With the introduction of the helicopter into combat situations such as those encountered in Southeast Asia, the vulnerability of the aircraft has become of increasing interest to the user services and to the designer and manufacturer. Of particular concern are the large projected areas of the main transmission lubrication and cooling systems which are exposed to combat damage.

The loss of all oil from the main rotor transmission of a helicopter due to a projectile impact in the oil cooling system will usually occur in less than a minute. If the gearbox is required to carry the load for any amount of time without lubrication and cooling, a catastrophic transmission failure will result.

The investigation performed under Contract DAAJ02-67-C-0112 (Reference 1) indicated that an annular-type helicopter transmission oil cooler installation was less vulnerable to combat damage and was more cost-effective than present systems. This type of annular-shaped oil cooler had not previously been fabricated and was thus considered to be experimental. The intent of this program was to determine the performance characteristics and manufacturing feasibility of a typically sized, annular-shaped, aircraft-quality oil cooler.

STATEMENT OF PROBLEM

The annular oil cooler devised for reducing the vulnerability of helicopter transmission oil cooling systems is of an unusual design. This configuration represents a departure from the normal crossflow, plate-fin heat exchangers used in present helicopters. The annular oil cooler uses a radially directed airflow coupled with a circumferential oil flow which does not result in a true 90-degree crossflow heat exchanger. Annular-shaped oil coolers had previously been employed as heaters in some automotive applications; however, these heat exchangers used an axial airflow direction rather than radial.

Due to the fact that the construction of an annular-type oil cooler is substantially different from conventional configurations, a requirement was generated for evaluating the manufacturing feasibility and performance characteristics in order to establish the optimum design configuration required for production helicopter installations.

PROGRAM APPROACH

The approach employed for this program involved the design, fabrication, and test of a typically sized, annular-type oil cooler. The CH-47C helicopter's forward rotor transmission was selected as the baseline for this program in order to establish typical values of heat rejection rate, oil-flow rate, airflow rate, height/diameter ratio, vibrational environment tolerance, and applicable production qualification and procurement specifications. This approach was selected because it was felt that, if the experimental annular-type oil cooler met all of these requirements, it would be safe to assume that this type of cooler was suitable for production helicopter and V/STOL aircraft applications.

The annular oil cooler developed during this program was designed to meet the 2,200-Btu/min cooling requirement of the CH-47C forward rotor transmission. In order to meet these conditions, an oil cooler with a 20-inch outside diameter, a 10-inch inside diameter, and a 3.1-inch no-flow height (thickness) was designed. This oil cooler was of aluminum construction and was fabricated according to aircraft-quality standards.

OIL COOLER DESIGN SPECIFICATIONS

The heat exchanger was nominally designed to meet the following approximate conditions:

- Rated oil flow (MIL-L-7808) - 26 gal/min
- Rated airflow - 150 lb/min (maximum)
- Rated heat rejection - 2,200 Btu/min (minimum)
- Oil inlet temperature - 235°F
- Air inlet temperature - 125°F
- Oil inlet pressure - 90 psig
- Air inlet pressure - 14.7 psia
- Oil pressure drop - 30 psi (maximum)
- Air static pressure drop - 6 in. H_2O at inlet density of 0.679 lb/ft^3 (5.3 in. H_2O ρ ΔP).
- Air entrainment by volume in the oil - 20 pct (maximum)

OIL COOLER DESIGN

The experimental annular oil cooler was designed with an airflow normal to the cooler axis and a direction either from inner to outer diameter or from outer to inner diameter. The heat exchanger design calculations were based on standard rectangular heat exchanger cores. This was accomplished by assuming the annular cooler to be straightened out into a rectangular form. Average dimensions for this form were then used to calculate the cooler performance. Based on this, the calculated performance of the annular heat exchanger at rated conditions was found to be as follows:

	Air	Oil (MIL-L-7808)
Inlet Pressure	14.7 psia	90 psig
Inlet Temperature	125°F	235°F
Flow	150 lb/min	26 gpm (including entrained air)
		(20.8 gpm oil + 5.2 gpm air)
Pressure Drop	5.7 in. H_2O	15 psi
Heat Rejection (Maximum)	2,340 Btu/min	2,340 Btu/min

The calculated performance is for an equivalent rectangular heat exchanger and does not include any adjustments for possible higher air static drop from constricted airflow at the oil cooler inner diameter or reduced heat transfer that may result from possible uneven distribution of oil flow within the heat exchanger core. Oil pressure drop does include an allowance for inlet-outlet casting pressure drop and an allowance for entrained air. Calculated air static drop and oil pressure drop have been deliberately kept below the maximum specified, and heat transfer has been kept above that specified to allow for these unknown factors.

The oil cooler design (Figure 1) was accomplished through the use of standard heat transfer equations for forced convection. The method used follows, in general, that given in References 2 and 3. Design calculations for the cooler are shown in Tables I and II.

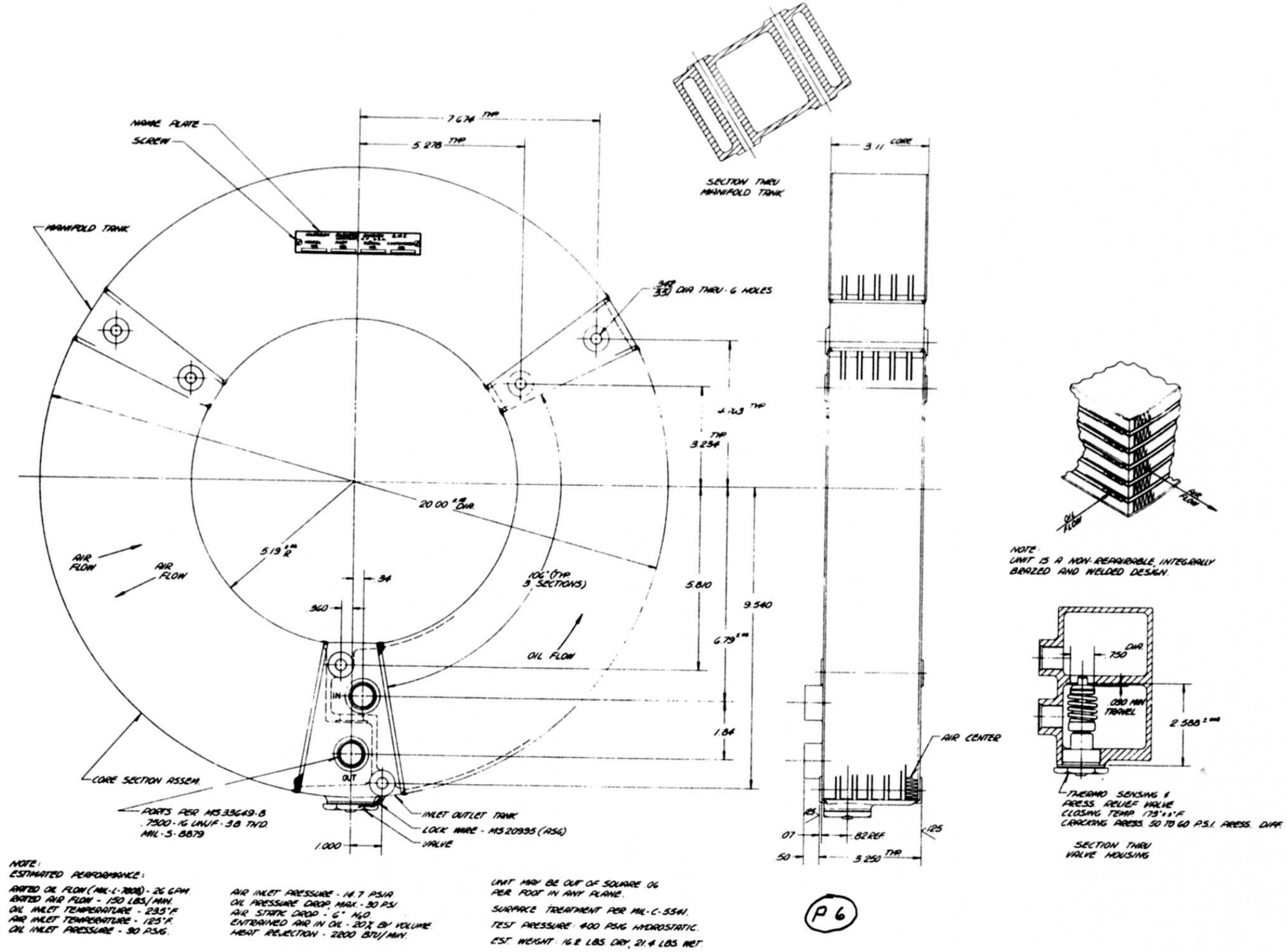


Figure 1. Annular Oil Cooler.

TABLE I. HEAT TRANSFER CALCULATIONS

Oil Side	Air Side
Oil Turbulator Fins: 7 convoluted fins per inch, 0.125 inch high x 0.014 inch thick	Air Fins: 17 fins per inch, 0.375 inch high x 0.005 inch thick
Oil Passageways: 5 total, 4.5 inches wide x 0.125 inch high x 42.15 inches long, based on oil cooler mean circumference	Air Passageways: 6 total, 0.375 inch high x 4.81 inches long x 41.01 inches wide, based on oil cooler mean circumference
Open Area, $A_{02} = 0.0164\ ft^2$	Open Area, $A_{01} = 0.602\ ft^2$
Direct Area, $A_{D2} = 14.93\ ft^2$	Direct Area, $A_{D1} = 14.26\ ft^2$
Indirect Area, $A_{I2} = 13.98\ ft^2$	Indirect Area, $A_{I1} = 106.25\ ft^2$
Hydraulic Diameter, $D_{H2} = 0.0080$ ft	Hydraulic Diameter, $D_{H1} = 0.0112$ ft
Oil Flow, M_2 = 20.8 gpm (146.5 lb/min) (MIL-L-7808 - 26 gpm + 20% air)	Airflow, M_1 = 150 lb/min
Inlet Oil Temperature, $T_{2\ IN} = 235°F$	Inlet Air Temperature, $T_{1\ IN} = 125°F$
Outlet Oil Temperature, $T_{2\ OUT} = 203.5°F$	Outlet Air Temperature, $T_{1\ OUT} = 189°F$
Average Temperature, $T_{2\ AVG} = 219.3°F$	Average Temperature, $T_{1\ AVG} = 157°F$
Film Temperature, $T_{2F} = 210.4°F$	Film Temperature, $T_{1F} = 179.2°F$
Specific Heat, C_{P2} = 0.508 Btu/lb-°F	Specific Heat, C_{P1} = 0.244 Btu/lb-°F
M_2C_{P2} = 74.36 Btu/min-°F	M_1C_{P1} = 36.57 Btu/min-°F
Viscosity, μ_2 = 0.1605 lb/min-ft	Viscosity, μ_1 = 0.00085 lb/min-ft
$\frac{C_{P2}}{P_r^{2/3}}$ = 0.0341 Btu/lb-°F	$\frac{C_{P1}}{P_r^{2/3}}$ = 0.3061 Btu/lb-°F
Mass Flow, $G_2 = \frac{M_2}{A_{02}}$ = 8,932 lb/min-ft^2	Mass Flow, $G_1 = \frac{M_1}{A_{01}}$ = 249 lb/min-ft^2
Reynolds Number, $R_2 = \frac{D_{H2}G_2}{\mu_2} = 448$	Reynolds Number, $R_1 = \frac{D_{H1}G_1}{\mu_1} = 3,290$
Colburn Factor, $j_2 = 0.0167$	Colburn Factor, $j_1 = 0.0074$
Convective Heat Transfer Coefficient, $h_2 = \frac{j_2G_2C_{P2}}{P_r^{2/3}}$ = 5.08 Btu/min-ft^2-°F	Convective Heat Transfer Coefficient, $h_1 = \frac{j_1G_1C_{P1}}{P_r^{2/3}}$ = 0.56 Btu/min-ft^2-°F
Fin Efficiency, $E_{f2} = 0.961$	Fin Efficiency, $E_{f1} = 0.893$
Effective Heat Transfer Surface Area, $A_{E2} = A_{D2} + E_{f2}\ A_{I2} = 28.36\ ft^2$	Effective Heat Transfer Surface Area, $A_{E1} = A_{D1} + E_{f1}\ A_{I1} = 109.14\ ft^2$
h_2A_{E2} = 144.2 Btu/min-°F	h_1A_{E1} = 57.8 Btu/min-°F

Overall Heat Transfer Coefficient $= UA = \frac{h_1A_{E1} \times h_2A_{E2}}{h_1A_{E1} + h_2A_{E2}}$ = 41.2 Btu/min-°F

MC_P (Minimum) $= M_1C_{P1}$ = 36.57 Btu/min-°F

MC_P (Maximum) $= M_2C_{P2}$ = 74.36 Btu/min-°F

$$\frac{M_1C_{P1}}{M_2C_{P2}} = \frac{C_{MIN}}{C_{MAX}} = 0.492$$

$$NTU_{MAX} = \frac{UA}{C_{MIN}} = \frac{41.2}{36.57} = 1.127$$

From Figure 2

$\epsilon = 0.582$ (Crossflow Effectiveness) for $\frac{C_{MIN}}{C_{MAX}} = 0.492$ and $NTU_{MAX} = 1.127$

$$Q = \epsilon\ C_{MIN}\ (T_{2\ IN} - T_{1\ IN})$$

$$Q = 0.582 \times 36.57 \times 110 = 2,340\ \text{Btu/min}$$

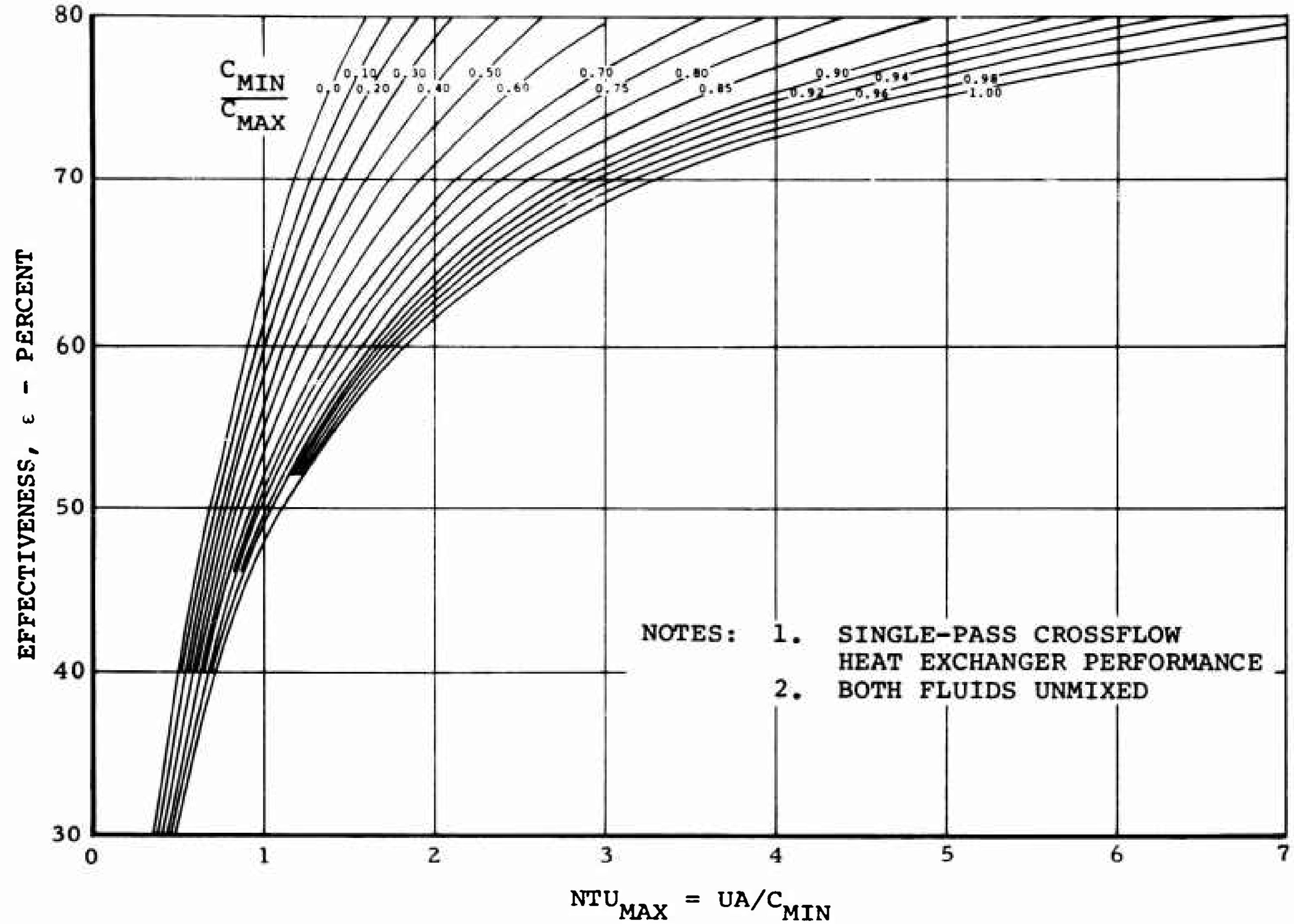


Figure 2. Crossflow Heat Exchanger Effectiveness.

TABLE II. PRESSURE DROP CALCULATIONS

Oil Side	Air Side
Open Flow Area, $A_{02} = 0.0164\ ft^2$	Open Flow Area, $A_{01} = 0.602\ ft^2$
Hydraulic Diameter, $D_{H2} = 0.0080\ ft^2$	Hydraulic Diameter, $D_{H1} = 0.0112\ ft^2$
Mass Flow, $G_2 = \frac{M_2}{A_{02}} = 8{,}932\ lb/min\text{-}ft^2$	Mass Flow, $G_1 = \frac{M_1}{A_{01}} = 249\ lb/min\text{-}ft^2$
Reynolds No., $R_2 = \frac{D_{H2} G_2}{\mu_2} = 448$	Reynolds No., $R_1 = \frac{D_{H1} G_1}{\mu_1} = 3{,}290$
Fanning Friction Factor, $f_2 = 0.102$	Fanning Friction Factor, $f_1 = 0\ 047$
Density, $\rho_{2\ AVG} = 53.2\ lb/ft^3$	Density, $\rho_{1\ AVG} = 0.064\ lb/ft^3$
$\Delta P_2 = 3(G_2)^2 \times 10^{-8} \frac{4\ f_2 L_2}{D_{H2}\ \rho_2} = 8.1$ psi	$\Delta P_1 = 83.1(G_1)^2 \times 10^{-8} \frac{4\ f_1 L_1}{D_{H1}\ \rho_1} = 5.4$ in. H_2O
In addition to this core pressure drop, an allowance for internal passage loss and entrained air must be added. Therefore, predicted oil pressure drop for 20 percent entrained air is approximately 15 psi.	To this core pressure drop must be added the entrance and exit losses and losses due to density changes. This results in a calculated air static drop of 5.7 in. H_2O.

FABRICATION

METHOD

Two complete heat exchangers were fabricated for this program. The design and construction were tailored for an operating temperature range of -65°F to +260°F. Each cooler was fabricated from three identical aluminum core sections, two cast aluminum manifold tanks, and one cast inlet-outlet tank. It was decided to make the cooler from three core sections in order to provide tank bolt holes to facilitate mounting and also to provide for adequate salt drainage of the core sections during the brazing operations.

The core sections were made up of alternate layers of air centers and oil centers (3003-0 aluminum). An average fin spacing of 17 fins per inch was maintained at the centerline of the inner and outer air centers. The oil and air centers were separated by tube sheets of No. 12 braze sheet. Side bars were made of 3003-0 aluminum, and header bars were of 3003-F aluminum. Top and bottom core plates were made from No. 11 braze sheet. Individual pieces and a partly stacked core section are shown in Figure 3. These pieces were brazed together to form a single core section. A core section, as removed from the brazing furnace, is shown in Figure 4. A brazed core section and two tanks are shown in Figure 5. Three brazed cores and three tanks of aluminum casting alloy 356-T6 were welded together as shown in Figure 6.

HYDROSTATIC PRESSURE TEST

At this point in the fabrication process, the cooler was given a preliminary hydrostatic pressure test to check for leaks before proceeding with the machining of the mounting holes. Refer to Figure 7 for the test setup. The completed oil cooler with a surface treatment per MIL-C-5541 is shown in Figure 8.

The final hydrostatic pressure test was accomplished by using MIL-L-7808 oil. A pressure of 400 psi was applied and held for one minute (see Figure 9). During this time, there were no leaks or permanent distortion.

CLEANING TEST

Each cooler was subjected to circulation of MIL-L-7808 oil at 250°F for 10 minutes in each direction. A 40-micron filter was installed at the cooler inlet to prevent possible contamination of the oil cooler from the oil system. A 60-micron filter was installed at the cooler outlet to catch foreign materials (such as brazing salts) if present in the cooler. After flushing for 10 minutes in the normal flow direction, the outlet filter was inspected for signs of foreign materials.

After finding the filter clean, the cooler and filters were reconnected to the oil system with the oil cooler reversed to obtain oil flow through the cooler from outlet to inlet. The outlet filter was examined after 10 minutes, and no foreign material was found; therefore, the cooler was judged to be clean and free of foreign material. This test was performed on both coolers prior to the start of the test program to verify that the normal manufacturing processes were capable of removing all the brazing salts from an oil cooler without straight oil-flow paths. The oil cleaning tests are shown in Figure 10.

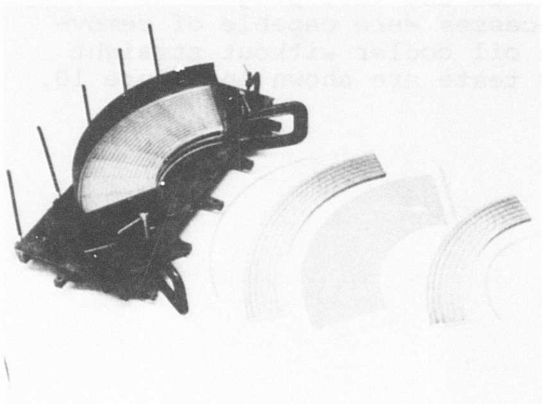


Figure 3. Cooler Components and Partly Stacked Core Section.

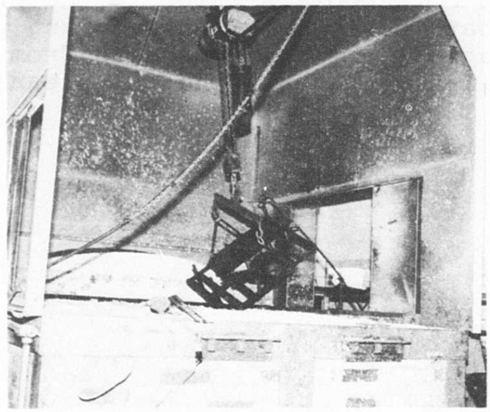


Figure 4. Core Section Emerging From Brazing Furnace.

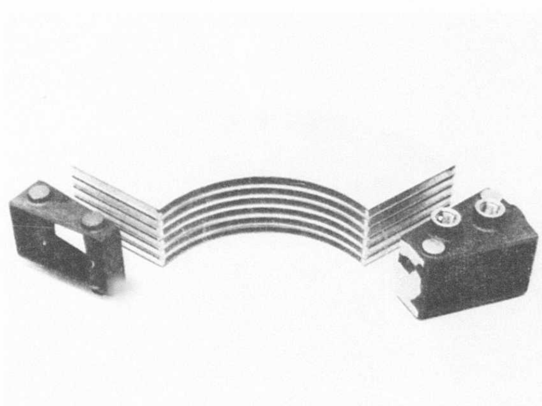


Figure 5. Core Section and Two Tanks.

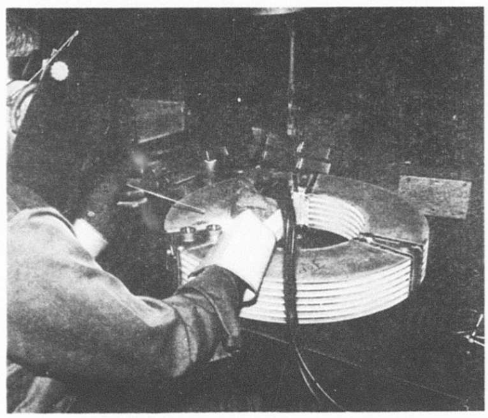


Figure 6. Cooler Being Assembled.

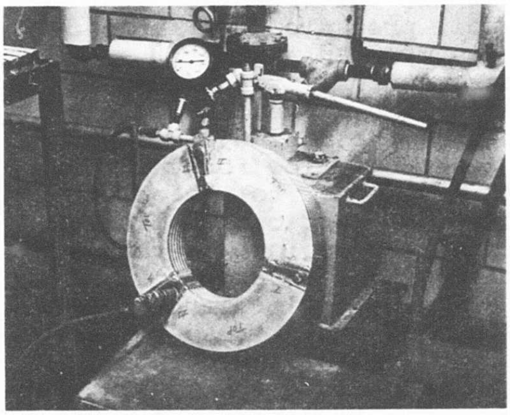


Figure 7. Hydrostatic Pressure Test of Cooler.

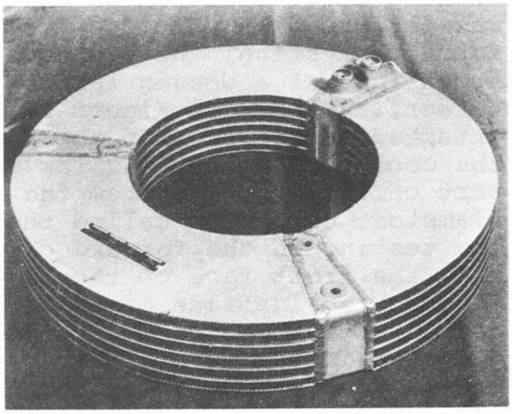


Figure 8. Finished Annular Oil Cooler.

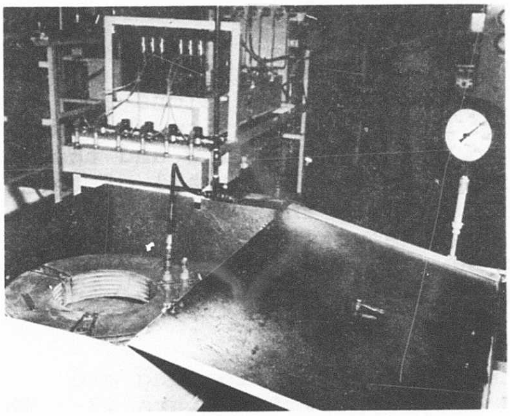


Figure 9. Final Hydrostatic Pressure Test.

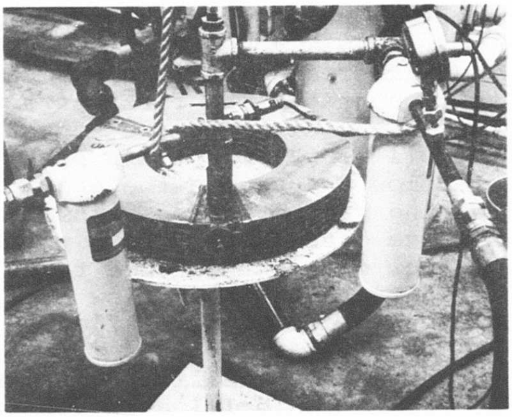


Figure 10. Cleaning Test of Oil Cooler.

TEST METHOD AND RESULTS

PERFORMANCE TEST

The oil cooler tests were divided between the two oil coolers fabricated for this program. The performance and pressure cycling tests were run on unit 1, while the vibration testing was performed on unit 2. For the oil pressure drop and performance testing, the oil cooler was mounted against the inside front face of a wooden test chamber, approximately 4 by 4 by 4 feet, as shown in Figure 11. A 10-inch-diameter duct was attached to the face of the chamber at the inside diameter of the cooler to supply air for the test. Airflow for the first part of the test was from the inside diameter to the outside diameter, hereafter called the forward direction. Following the testing in the forward direction, the cooler was removed from the front face of the test chamber and mounted against the back inside face of the chamber. The oil cooler was then tested with the airflow from the outside diameter to the inside diameter in the reverse direction. Figure 12 represents a schematic of the test setup and equipment location for the forward airflow direction testing. The oil cooler was moved to the opposite end of the test chamber, and various oil lines and instrumentation were relocated for testing in the reverse airflow direction. During the performance testing, the thermostatic control valve was removed from the oil cooler, and the bypass was positively blocked to preclude any inadvertent bypassing due to the wide range of test conditions.

The performance testing was divided into three parts for this test program. The first part used fixed oil conditions and varied the airflow; the second part fixed the airflow and varied the oil flow. The third part fixed both the airflow and the oil flow and varied the entrained air in the oil.' All of the testing was performed with airflows in both the forward and reverse directions. Data points for the variation in either airflow or oil flow were taken at 50, 75, 100, 125, and 150 percent of the rated flow conditions, which were 150 lb/min airflow and 26 gpm oil flow.

The data from the first part of the performance testing are listed in Table III and displayed graphically in Figure 13. The performance testing accomplished in the first part indicated that there was virtually no difference in the oil heat rejection for a given weight flow rate of air for either airflow direction. This testing indicated that the air side static pressure drop was approximately 32 to 35 percent higher in the reverse airflow direction than in the forward airflow direction.

The second part of the performance test concerned the effect of varying the oil flow while holding all other parameters constant. This test was performed with MIL-L-7808 oil over a

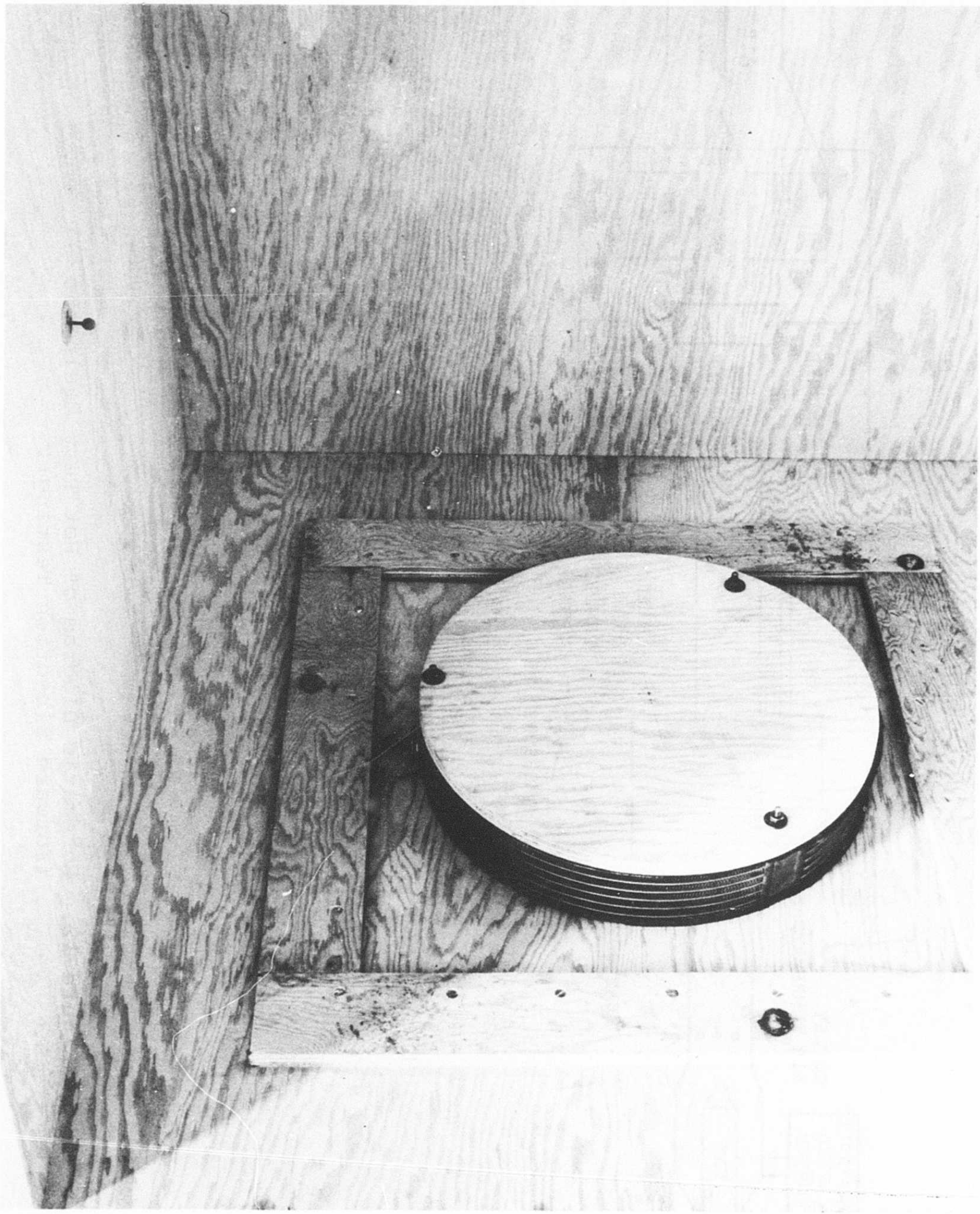


Figure 11. Cooler Mounting for Performance and Oil Pressure Drop Tests.

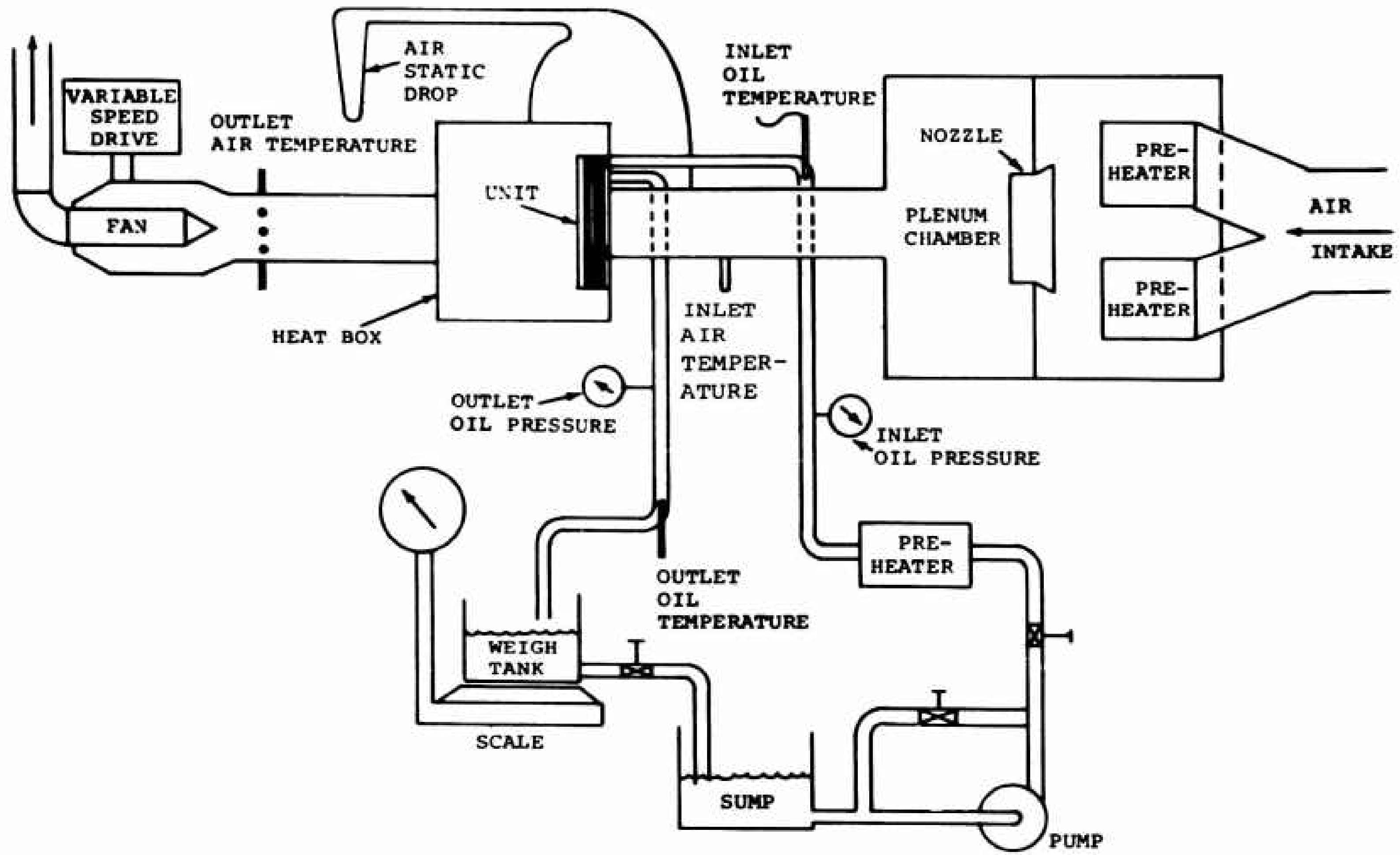


Figure 12. Schematic Diagram of Hot Test Equipment (Forward Direction Testing).

TABLE III. PERFORMANCE TEST DATA WITH AIRFLOW VARIED

Parameter	Forward Airflow Direction					Reverse Airflow Direction				
Airflow (lb/min)	225.6	187.5	150.7	112.5	75.6	225.1	187.5	150.3	112.5	75.6
Air Outlet Temp (°F)	179.7	184.8	189.9	196.2	204.4	181.5	186.2	192.9	200.0	209.9
Air Static Drop (in./H_2O X $\frac{\text{inlet density}}{0.0765\ \text{lb/ft}^3}$)	14.9	10.6	7.2	4.5	2.3	20.0	14.3	9.6	5.8	3.0
Oil Outlet Temp (°F)	202.6	206.3	209.6	213.9	219.6	203.1	205.8	209.5	213.7	218.8
Oil Side Heat Rejection (Btu/min/110°F)	3,059	2,761	2,425	2,020	1,504	3,046	2,774	2,435	2,024	1,534
Air Side Heat Rejection (Btu/min/110°F)	3,014	2,727	2,391	1,981	1,459	3,091	2,807	2,486	2,073	1,567

Constant Conditions:

Oil flow (MIL-L-7808) - 26 ± 0.1 gpm/187 ± 0.5 lb/min
Oil inlet pressure - 54 ± 4 psi
Oil side pressure drop - 18 ± 0.6 psi
Oil inlet temperature - 235 ± 0.5°F
Air inlet temperature - 125 ± 0.5°F
Inlet temperature difference - 110 ± 1°F

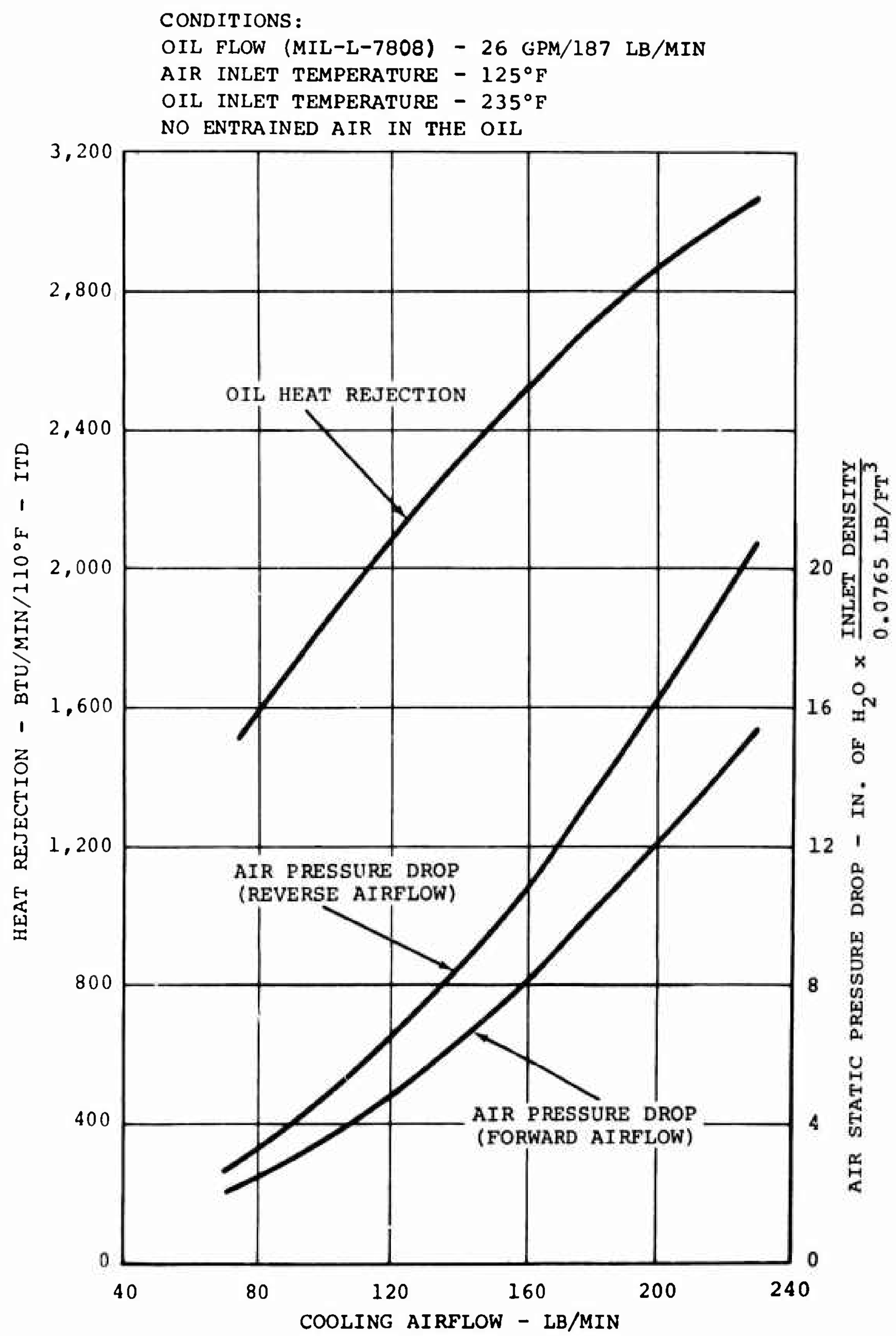


Figure 13. Performance Versus Airflow.

flow rate range of approximately 13 gpm to 39 gpm, while maintaining an airflow of 150 lb/min in both the forward and reverse directions for each oil-flow condition. The test results are presented in tabular form in Table IV and graphically in Figure 14. These results indicate that the oil heat rejection and oil pressure drop are independent of airflow direction for a constant air weight flow. At 13 gpm (50 percent of rated flow), the oil pressure drop was approximately 6 psi, while at 39 gpm (150 percent of rated flow), the oil pressure drop was 37 psi, with a 19-psi pressure drop at the rated oil-flow condition. The range of the oil pressure drop (6 psi to 37 psi) for an oil flow rate of 50 to 150 percent rated flow is an acceptable value for a helicopter transmission oil cooler.

The third test used a fixed airflow (150 lb/min) and a fixed oil flow (26 gpm) and varied the amount of entrained air in the oil from 0 (solid oil flow) to 50 percent by volume (13 gpm oil + 13 gpm air). The intent of this test was to determine the effect of entrained air in the oil on heat transfer performance and oil pressure drop. The data are presented in Table V and graphically displayed in Figure 15 along with the equivalent data for reduced oil flow; e.g., 13 gpm solid oil flow is plotted against 50 percent entrained air, which is 13 gpm oil + 13 gpm air. Thus, for the same weight flow of oil, the effect of air which causes an oil foam was compared to the effect of reducing the actual solid oil flow with no entrained air. This testing indicated that the addition of entrained air to the oil to form an oil foam through the cooler had virtually no effect on either oil pressure drop or heat transfer through a range of 0 to 20 percent entrained air by volume. Over the range of 20 to 50 percent entrained air, the oil pressure drop increased slightly (3 psi at 50 percent) while the heat transfer also increased (50 Btu/min at 50 percent). Therefore, it is concluded that, for normal design purposes, the effect of entrained air in the oil need only be considered from the oil flow rate reduction standpoint. A system with more than 20 percent entrained air by volume in the oil should be redesigned in certain areas to reduce oil foaming.

OIL PRESSURE DROP TEST

The oil pressure drop from the cooler inlet to the cooler outlet was measured with an airflow of 150 lb/min and an air inlet temperature of 125°F. The oil flow was established at 26 gpm of MIL-L-7808 oil with no entrained air, and the oil pressure drop was measured with inlet oil temperatures of 180°F, 200°F, 220°F, and 235°F. The oil pressure drop with entrained air in the oil was also measured with inlet oil temperatures of 180°F and 235°F and entrained air by volume of 10, 20, 30, and 50 percent based on a total flow rate of 26 gpm. Table VI presents the data for oil pressure drop with and without

TABLE IV. PERFORMANCE TEST DATA WITH OIL FLOW VARIED

Parameter	Forward Airflow Direction					Reverse Airflow Direction				
Oil Flow (gpm/lb/min)	38.7/276.0	33.0/236.0	26.1/187.5	19.7/141.7	13.0/94.4	39.2/280.0	32.3/230.0	26.1/187.5	19.5/140.5	13.1/94.7
Oil Outlet Temp (°F)	216.6	214.4	209.6	203.2	191.6	217.0	213.5	209.5	203.4	191.8
Oil Inlet Press. (psig)	64.0	77.0	58.5	30.5	15.0	67.0	74.5	51.0	31.5	15.5
Oil Press. Drop (psi)	33.4	26.3	18.0	10.3	5.0	35.5	27.0	18.4	12.0	6.5
Air Outlet Temp (°F)	193.9	193.0	189.9	186.5	179.6	196.7	194.9	192.9	188.7	181.3
Oil Side Heat Rejection (Btu/min/110°F)	2,620	2,527	2,425	2,302	2,055	2,601	2,508	2,435	2,253	2,055
Air Side Heat Rejection (Btu/min/110°F)	2,577	2,492	2,391	2,263	2,018	2,639	2,577	2,486	2,344	2,085

Constant Conditions:

Airflow - 150 ± 1 lb/min

Air inlet temperature - 125 ± 0.5°F

Oil inlet temperature - 235 ± 0.5°F

Inlet temperature difference - 110 ± 1°F

Air side static pressure drop (forward direction) - 7.3 ± 0.2 in. of H_2O

Air side static pressure drop (reverse direction) - 9.5 ± 0.2 in. of H_2O

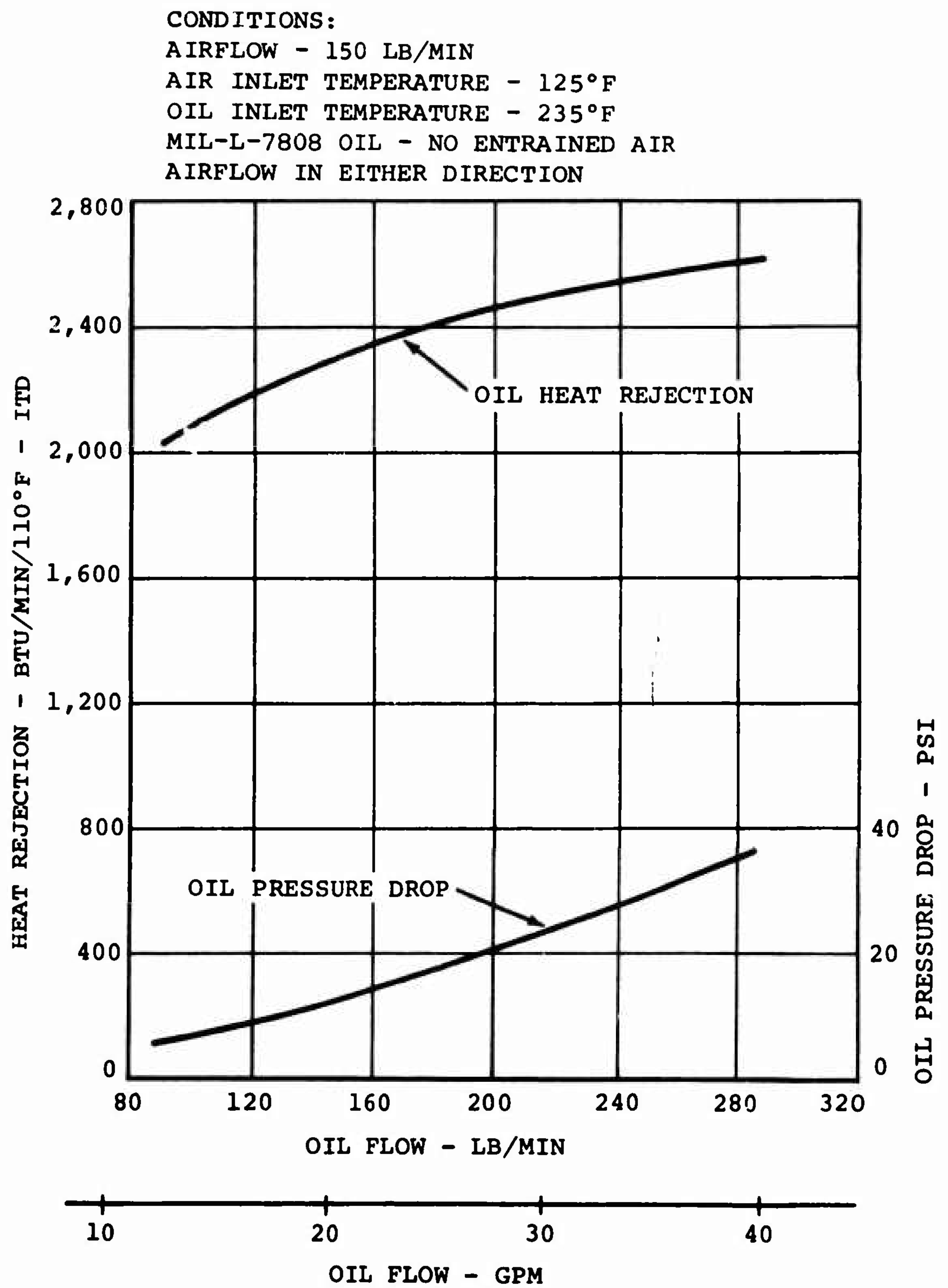


Figure 14. Performance Versus Oil Flow.

TABLE V. PERFORMANCE TEST DATA WITH AIR ENTRAINED IN OIL

Parameter	Forward Airflow Direction				Reverse Airflow Direction			
Entrained Air in Oil (% by volume)	50.0	20.4	10.1	0.0	50.0	20.0	10.0	0.0
Actual Real Oil Flow (gpm/lb/min)	13.1/94.8	20.8/150.0	23.5/168.5	26.1/187.5	13.1/94.7	20.8/150.0	23.4/167.5	26.1/187.5
Entrained Airflow (gpm)	13.2	5.4	2.5	0.0	13.0	5.2	2.6	0.0
Oil Foam Inlet Pressure (psi)	24.0	36.5	26.0	58.5	24.0	36.0	41.0	51.0
Oil Foam Pressure Drop (psi)	9.[illegible]	13.2	13.6	18.0	8.5	12.2	13.9	18.4
Oil Outlet Temp (°F)	190.8	204.4	207.1	209.6	190.9	204.5	207.1	209.5
Air Outlet Temp (°F)	182.1	186.9	187.7	189.9	183.3	189.4	190.8	192.9
Oil Side Heat Rejection (Btu/min/110°F itd)	2,103	2,329	2,338	2,425	2,106	2,334	2,387	2,435
Air Side Heat Rejection (Btu/min/110°F itd)	2,113	2,294	2,337	2,391	2,142	2,366	2,424	2,486

Constant Conditions:

Airflow - 150 ± 1 lb/min (total oil foam flow rate - 26 ± 0.2 gpm)

Air inlet temperature - 125 ± 0.5°F

Oil inlet temperature - 235 ± 0.5°F

Inlet temperature difference - 110 ± 1°F

Air side static pressure drop (forward direction) - 7.2 ± 0.2 in. of H_2O

Air side static pressure drop (reverse direction) - 9.5 ± 0.1 in. of H_2O

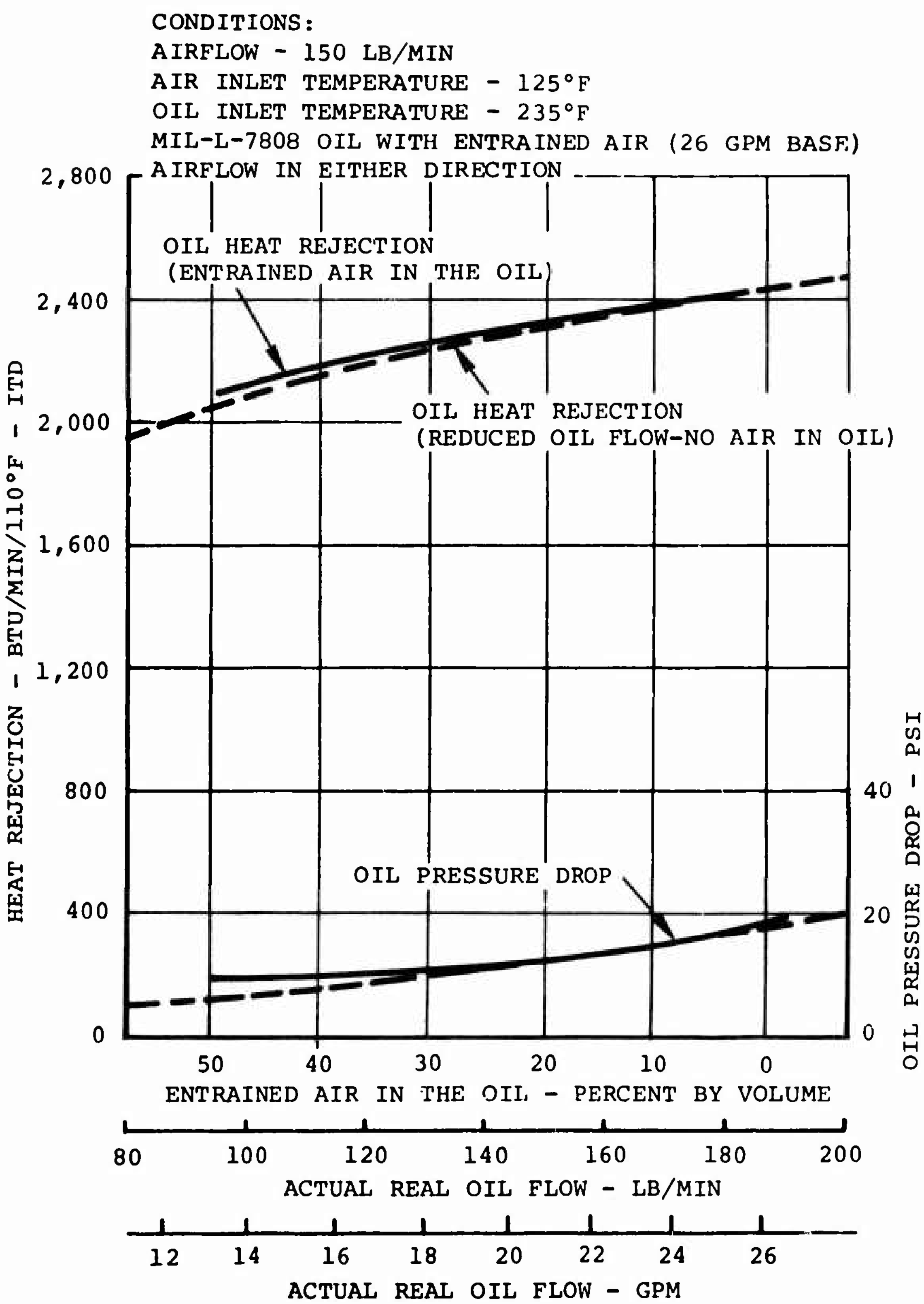


Figure 15. Performance With Air in the Oil.

TABLE VI. TEMPERATURE EFFECT ON OIL PRESSURE DROP

Oil Inlet Temperature (°F)	Oil Flow Rate (gpm)	Oil Pressure Drop (psi)
180	26.3	19.0
200	26.3	19.0
220	26.3	19.0
235	26.1	18.0

Entrained Air (26 gpm baseline) (%)	Actual Real Oil Flow (gpm)	Oil Pressure Drop (psi) Oil Inlet Temperature 180°F	Oil Pressure Drop (psi) Oil Inlet Temperature 235°F
50.7	13.1	9.1	9.0
30.1	18.2	12.8	9.6
20.2	20.7	14.0	13.2
10.1	23.4	16.1	13.6
0.0	26.1	19.0	18.0

Constant Conditions:

Airflow rate - 150 $\pm$ 1 lb/min

Air inlet temperature - 125°F

Forward airflow direction

entrained air at various oil inlet temperatures. These data indicate that there is virtually no effect on oil pressure drop due to inlet oil temperature changes from 180°F to 235°F, either with or without entrained air in the oil. The data in Table VII show the results of the oil pressure drop testing without entrained air with various oil flow rates and constant airflow and inlet temperatures. These data are graphically presented in Figure 14, showing oil flow rate versus oil pressure drop and heat rejection at constant airflow. Table VIII shows the results of the oil pressure drop testing with entrained air, which is presented graphically in Figure 15. The data presented in Figure 15 indicate that entrained air in the oil has no more of an effect on oil cooler performance and pressure drop than reducing the oil flow rate between the ranges of 0 to 20 percent entrained air by volume in the oil. However, above a 20-percent entrained air content and up to the 50-percent entrained air test data limit, the entrained air actually caused a slight increase in the heat transfer rate and oil pressure drop over that of comparable conditions with reduced oil flow. This situation is a result of the air homogeneously mixing with the oil and forming a foam that, although possessing the same real oil flow rate (lb/min) as the data for reduced oil flow, has a higher velocity and thus a higher heat transfer film coefficient on the oil side. Although this is an interesting point, an oil-cooling system that contains more than 20 percent entrained air by volume in the oil is incorrectly designed and should have the foam-producing area modified to eliminate the problem. Therefore, under normal conditions (20 percent entrained air or less), there is virtually no effect on the system's heat transfer rate or oil pressure drop other than the result of reducing oil flow by displacing a given volume of oil with air.

BLOWER LOCATION OPTIMIZATION TEST

The object of the blower location optimization testing was to determine the best arrangement for a typical aircraft-type blower in a blower/oil cooler package. The testing used a Joy Manufacturing Company aircraft quality blower, Model AVF100-60D2124 (see Figures 16 and 17), which delivered the amount of air required for all phases of the testing. The blower was tested at four different locations from the cooler face (0, 1, 2, and 3 inches) and for both forward and reverse airflow directions. Figure 18 shows the relative location of the blower and annular oil cooler for the various test locations and airflow directions. Airflow for these tests was measured through calibrated nozzles before entering the blower/cooler package inlet duct. Inlet air temperature was measured with thermocouples located in the airstream between the blower and the cooler, inside the turning vanes. Outlet air temperature was measured with thermocouples located around the outer circumference of the cooler. Figure 19 is a schematic diagram

TABLE VII. OIL PRESSURE DROP DATA WITHOUT ENTRAINED AIR		
Oil Flow Rate (gpm)	Oil Pressure Drop (psi)	Heat Rejection (Btu/min)
Forward Airflow Direction		
38.7	33.4	2,600
33.0	26.3	2,510
26.1	18.0	2,410
19.7	10.3	2,280
13.0	5.0	2,040
Reverse Airflow Direction		
39.2	35.5	2,620
32.3	27.0	2,540
26.1	18.4	2,460
19.5	12.0	2,300
13.1	6.5	2,070

Constant Conditions:

Oil inlet temperature (MIL-L-7808 oil) - 235°F

Air inlet temperature - 125°F

Airflow rate - 150 ±1 lb/min

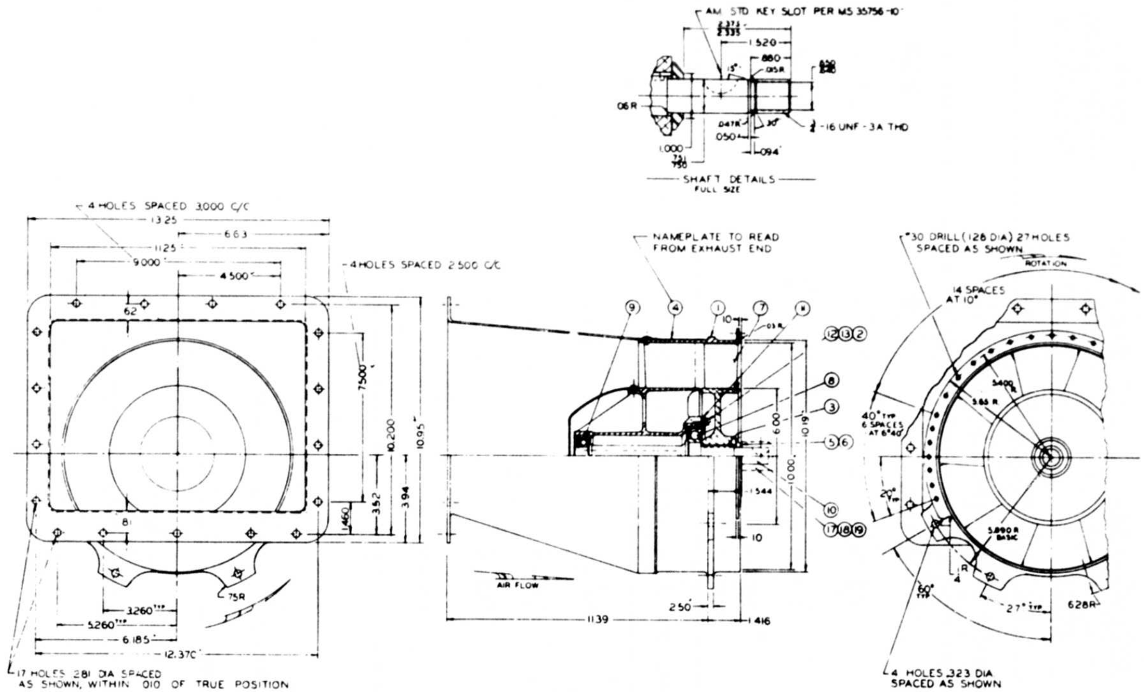
TABLE VIII. OIL PRESSURE DROP DATA WITH ENTRAINED AIR			
Entrained Air (26 gpm baseline) (%)	Actual Real Oil Flow (gpm)	Oil Pressure Drop (psi)	Heat Rejection (Btu/min)
Forward Airflow Direction			
50.0	13.1	9.0	2,110
20.4	20.8	13.2	2,310
10.1	23.5	13.6	2,340
0.0	26.1	18.0	2,410
Reverse Airflow Direction			
50.0	13.1	8.5	2,120
20.0	20.8	12.2	2,350
10.0	23.4	13.9	2,410
0.0	26.1	18.4	2,460

Constant Conditions:

Oil inlet temperature (MIL-L-7808 oil) - 235°F

Air inlet temperature - 125°F

Airflow rate - 150 ±1 lb/min



FAN RATING MINIMUM
2550 CFM AT 10.6" Ps AT .075#/CU FT CURVE C-4410 (A)

POWER REQUIREMENT
8.2 HP 6500 RPM AT .071#/CU FT

FAN WEIGHT EST
10.10 LBS MAGNESIUM CASTINGS, FIBER GLASS TRANSITION

FINISH
MAGNESIUM PARTS DICHROMATED PER MIL-M-3171A, TYPE III, PAINTED 1 COAT GLIDDEN NUBELON 'S', LIGHT GRAY COLOR #16473 PER FED STD #595 (4)
TRANSITION PAINTED ONE COAT EPOXY PRIMER PER MIL-P-23377, AND 2 COATS ACRYLIC LACQUER, FIELD GREEN COLOR ANA 627 PER MIL-L-19538. (15) (16)
STEEL PARTS CADMIUM PLATED, EXCEPT BEARING DIA'S ON SHAFT.

Part Number — 500702 - 4980
Model Number — AVF 100 - 60D2124
Design Rating CFM — 2530
Pressure(Inches)W. G. — 10.6
Weight (Pounds) — 10.2
Construction — Magnesium
Horsepower — 8.2
Speed R.P.M. — 6,500

Figure 16. Blower Used for Test Program.

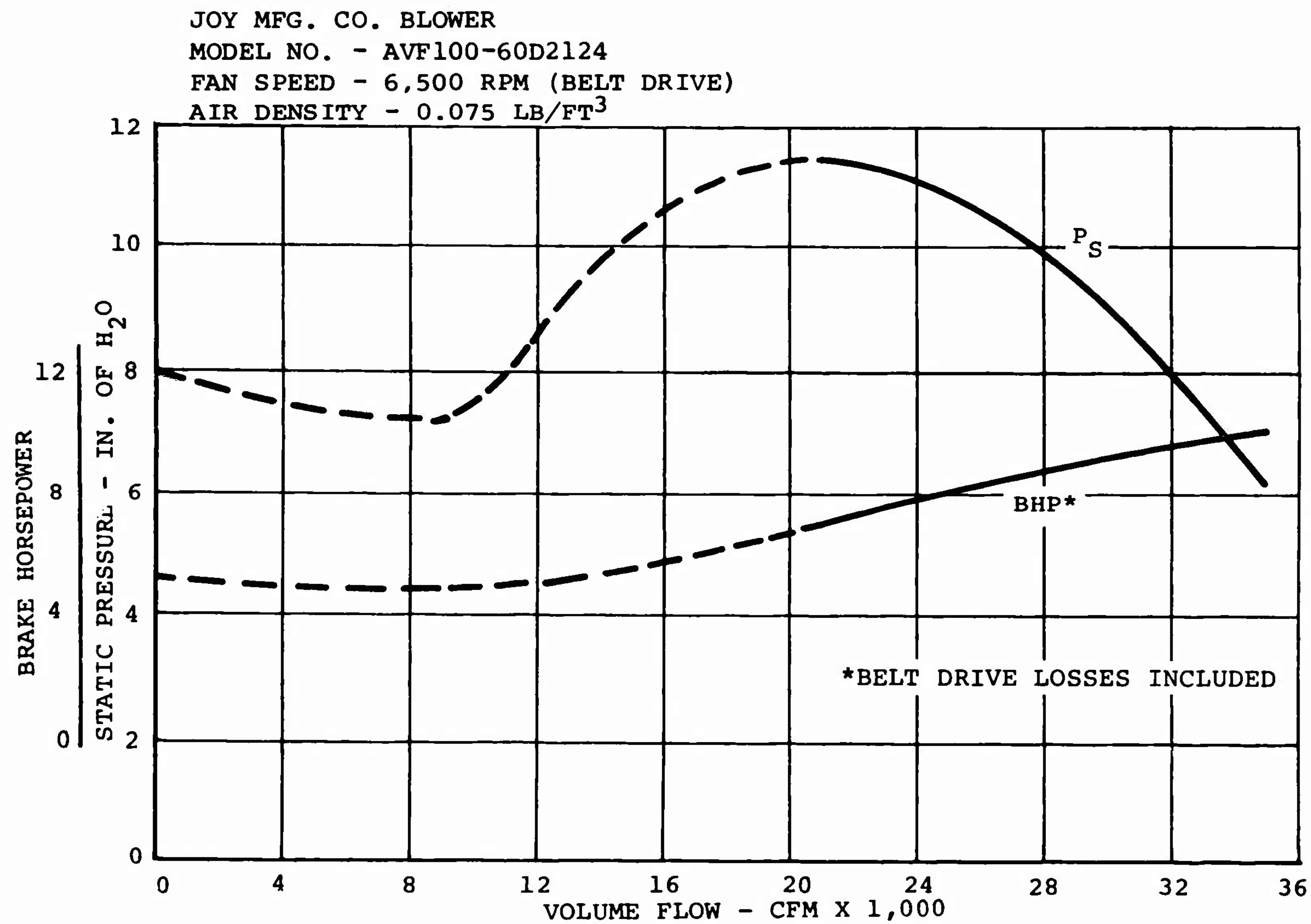
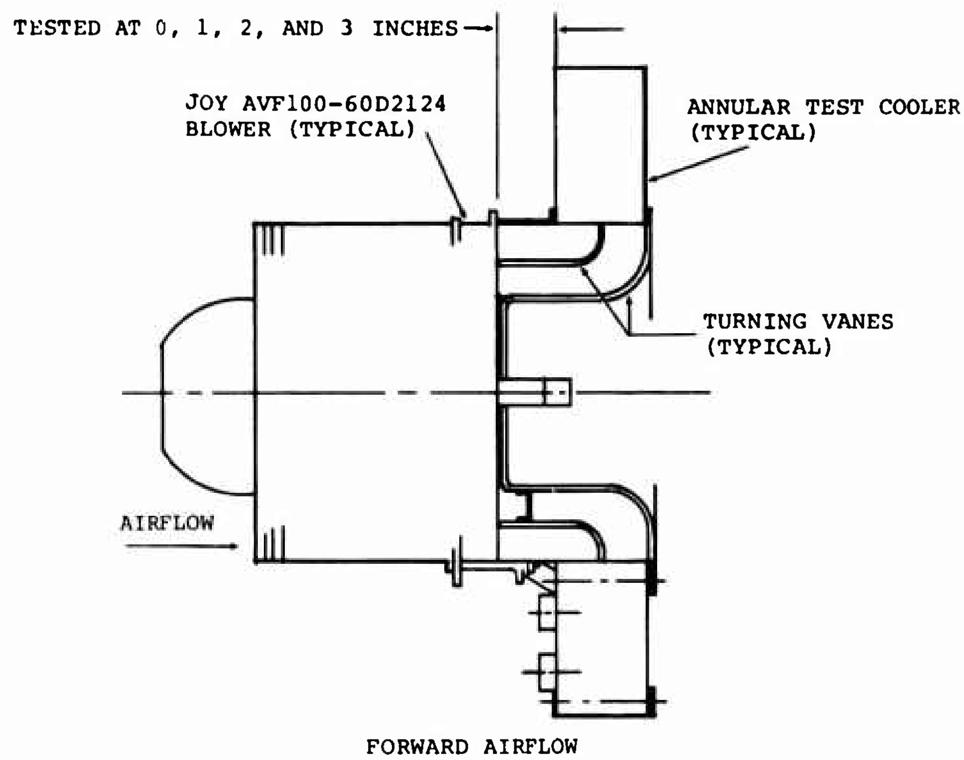


Figure 17. Blower Performance Curve.



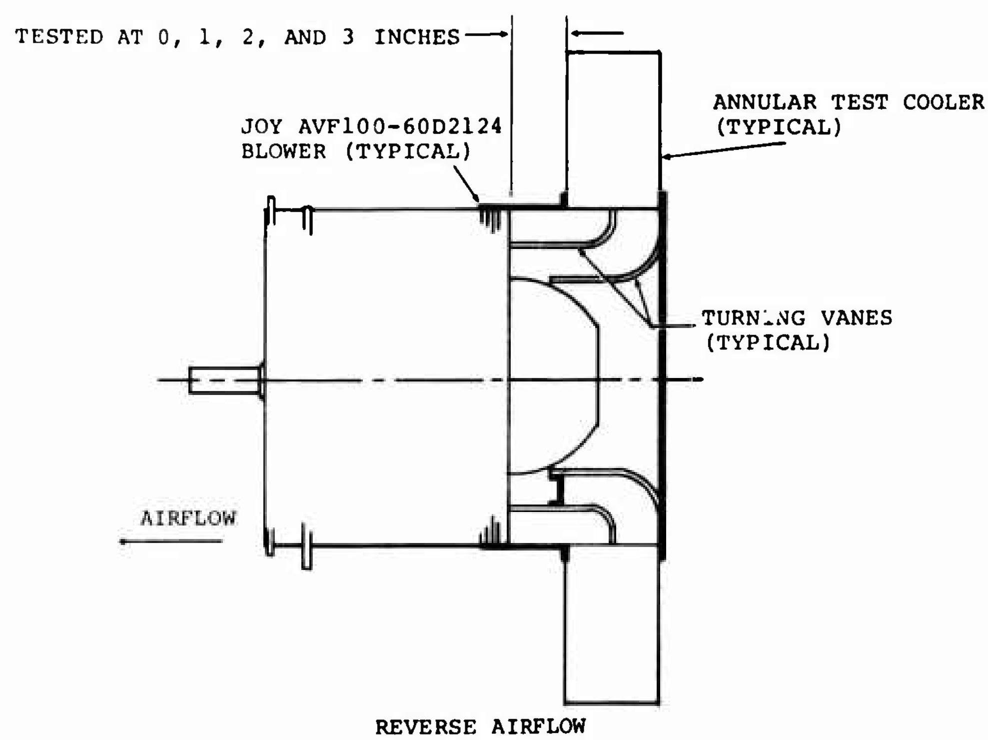


Figure 18. Blower Location Optimization Test Setup.

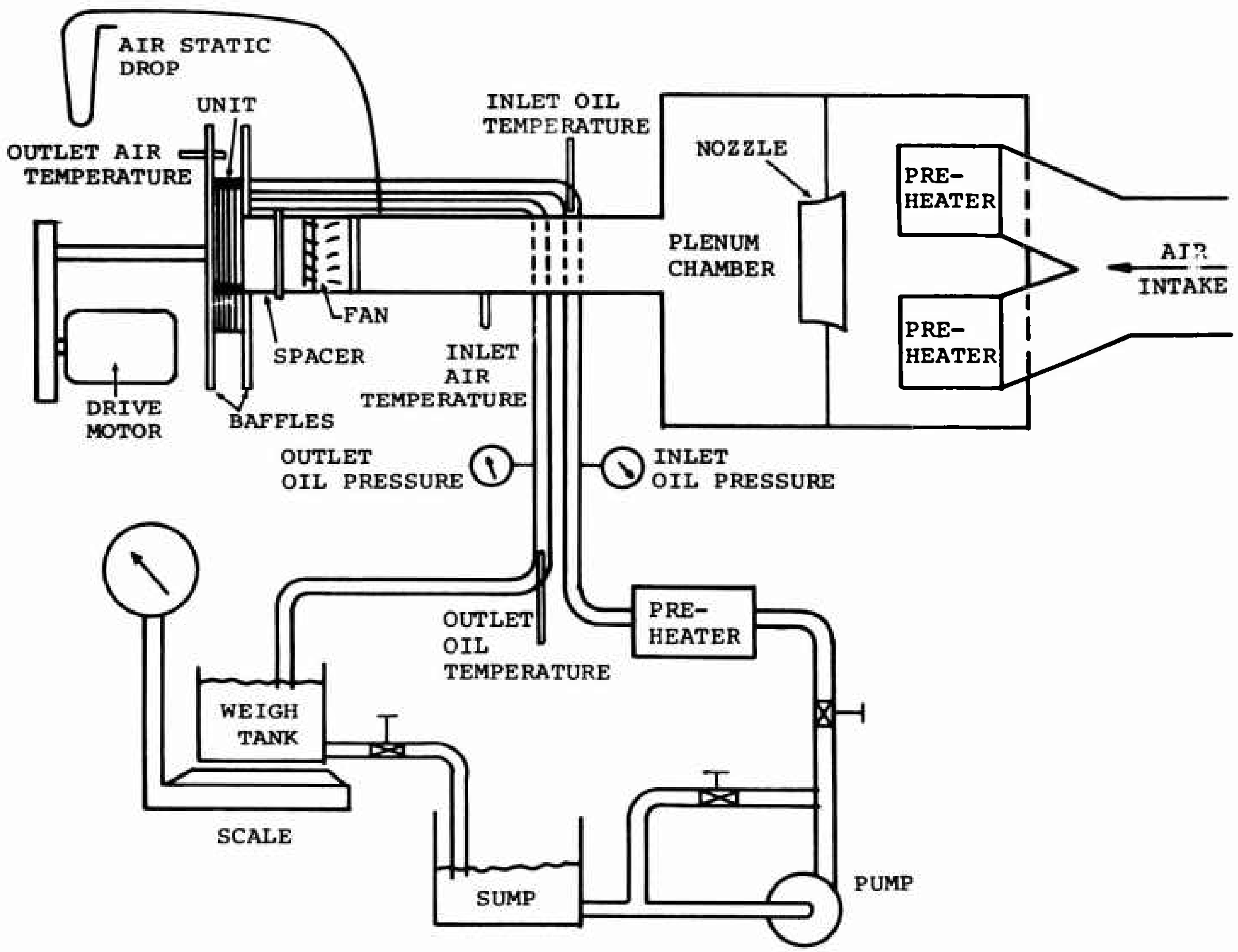


Figure 19. Blower Optimization Test Setup (Forward Direction).

of the test setup employed for the forward airflow direction and is typical of the setup for the reverse airflow direction, except for the relocation of the various inlet and outlet temperature monitors and reversal of the blower position. A view of the inner diameter of the oil cooler showing a turning vane assembly is presented in Figure 20, while the total test setup is shown in Figures 21 and 22.

The results of the blower location optimization testing as shown in Table IX and Figure 23 indicate that the axial position of the blower has no effect on the overall system static pressure, with the blower either pushing or pulling the air through the oil cooler. Preliminary estimates indicated that air stagnation might develop with the blower located directly against the cooler in the zero-inch position. The testing indicated that this condition did not exist and that the blower can be mounted directly against the oil cooler to provide the smallest overall system envelope and, thus, the lowest vulnerable surface area possible.

A second design concept that was investigated during this program included the addition of turning vanes within the oil cooler inner diameter to reduce possible stagnation and thus decrease the overall system static pressure drop due to the smoother airflow passages. These turning vanes were designed for each of the blower locations and airflow directions tested. The testing indicated that the turning vanes were not required for any of the blower locations tested and that they only served to raise the overall system static pressure drop (see Table X and Figure 24). Therefore, the best system design appears to be an oil cooler with the blower mounted directly against the cooler without turning vanes and with the airflow from the inner diameter of the cooler to the outer diameter.

VIBRATION TEST

The vibration testing was accomplished by mounting the oil cooler inlet side up on a 1-inch-thick aluminum plate with six 5/16-inch steel bolts through the mounting holes in the tank assemblies. This aluminum plate was then mounted on a slip table on the vibration test machine for the lateral and longitudinal testing, as shown in Figures 25 and 26. The plate/cooler assembly was mounted directly on top of the shaker machine for testing in the vertical mode, as shown in Figure 27. During the vibration testing, the oil cooler was filled with MIL-L-7808 oil and pressurized to 90 psig.

The oil cooler was subjected to frequency sweeps in accordance with MIL-E-5272C to determine the four predominant resonant frequencies to be used for further testing. The vibration testing was performed in accordance with MIL-E-5272C, Paragraph 4.7, Procedure XII, with the substitution of the more

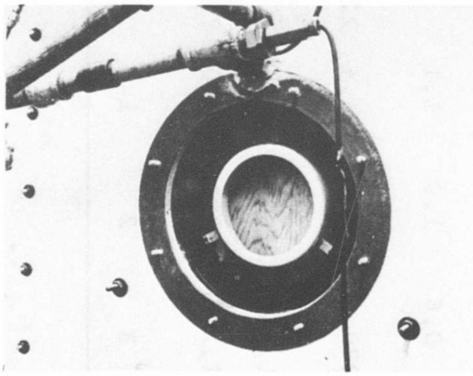


Figure 20. Oil Cooler Mounting With Turning Vane Assembly.

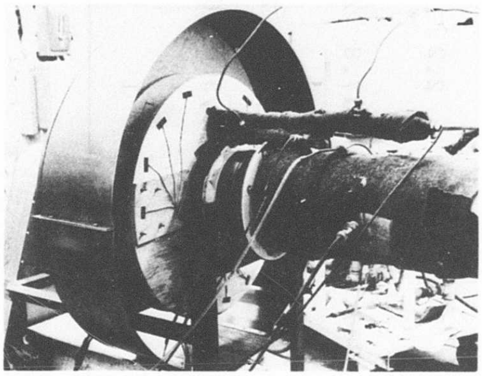


Figure 21. Blower Location Optimization Test Setup.

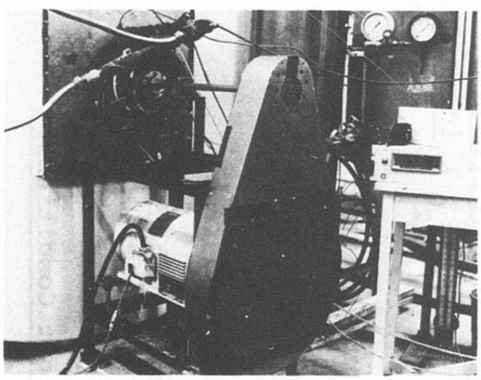


Figure 22. Blower Location Optimization Test Setup.

TABLE IX. BLOWER LOCATION OPTIMIZATION TEST DATA

Parameter	Forward Airflow				Reverse Airflow			
Airflow Rate (lb/min)	148.2	148.2	147.3	146.8	66.1	92.9	91.9	73.5
Air Outlet Temp (°F)	190.0	190.1	190.3	189.0	212.9	205.3	205.1	210.5
Air Side Static Pressure Drop (in. of $H_2O \times \frac{\text{inlet density}}{0.0765 \text{ lb/ft}^3}$)	2.1	2.2	2.2	2.2	0.6	1.0	1.1	0.7
Oil Outlet Temp (°F)	207.8	207.8	207.8	207.7	220.2	216.1	216.4	219.2
Oil Heat Rejection (Btu/min/110°F)	2,599	2,605	2,614	2,611	1,442	1,820	1,784	1,527
Blower Speed (rpm)	6,700	6,560	6,560	6,600	7,200	7,200	7,200	7.150
Blower Spacing (in. - see Figure 18)	3	2	1	0	3	2	1	0

<u>Constant Conditions:</u>

Oil flow rate (MIL-L-7808 oil) - 186 ±0.5 lb/min

Oil inlet temperature - 235 ±0.5°F

Oil pressure drop - 19 psi

Air inlet temperature - 125 ±0.5°F

Inlet temperature difference - 110 ±1°F

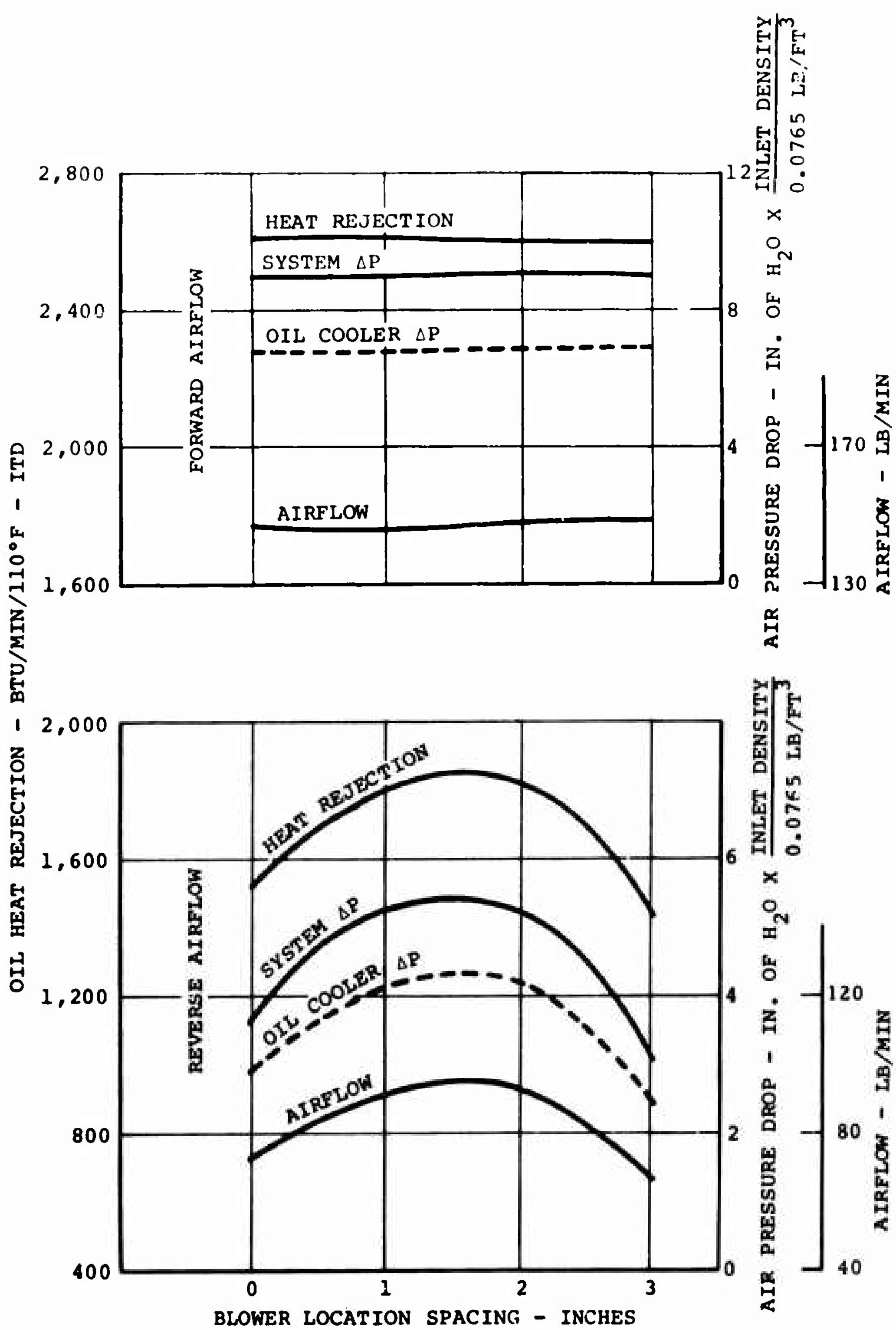


Figure 23. Blower Location Optimization Results.

TABLE X. AIR PRESSURE DROP (COOLER + ZERO-INCH TURNING VANES)

Parameter	Forward Airflow			Reverse Airflow		
Airflow Rate (lb/min)	225.2	153.0	75.4	225.2	150.8	75.7
Air Outlet Temp (°F)	179.4	189.6	203.6	180.7	191.6	209.1
Oil Outlet Temp (°F)	202.7	209.3	218.6	202.8	209.2	218.8
Air Side Static Pressure Drop (in. of $H_2O \times \frac{\text{inlet density}}{0.0765\ \text{lb/ft}^3}$)	18.0	9.0	2.7	23.0	10.8	3.7
Oil Heat Rejection (Btu/min/110°F itd)	3,077	2,455	1,582	3,054	2,446	1,550

Constant Conditions:

Oil flow rate (MIL-L-7808 oil) - 186 ±0.5 lb/min

Oil inlet temperature - 235 ±0.5°F

Oil pressure drop - 18 ±1 psi

Air inlet temperature - 125 ±0.5°F

Inlet temperature difference - 110 ±1°F

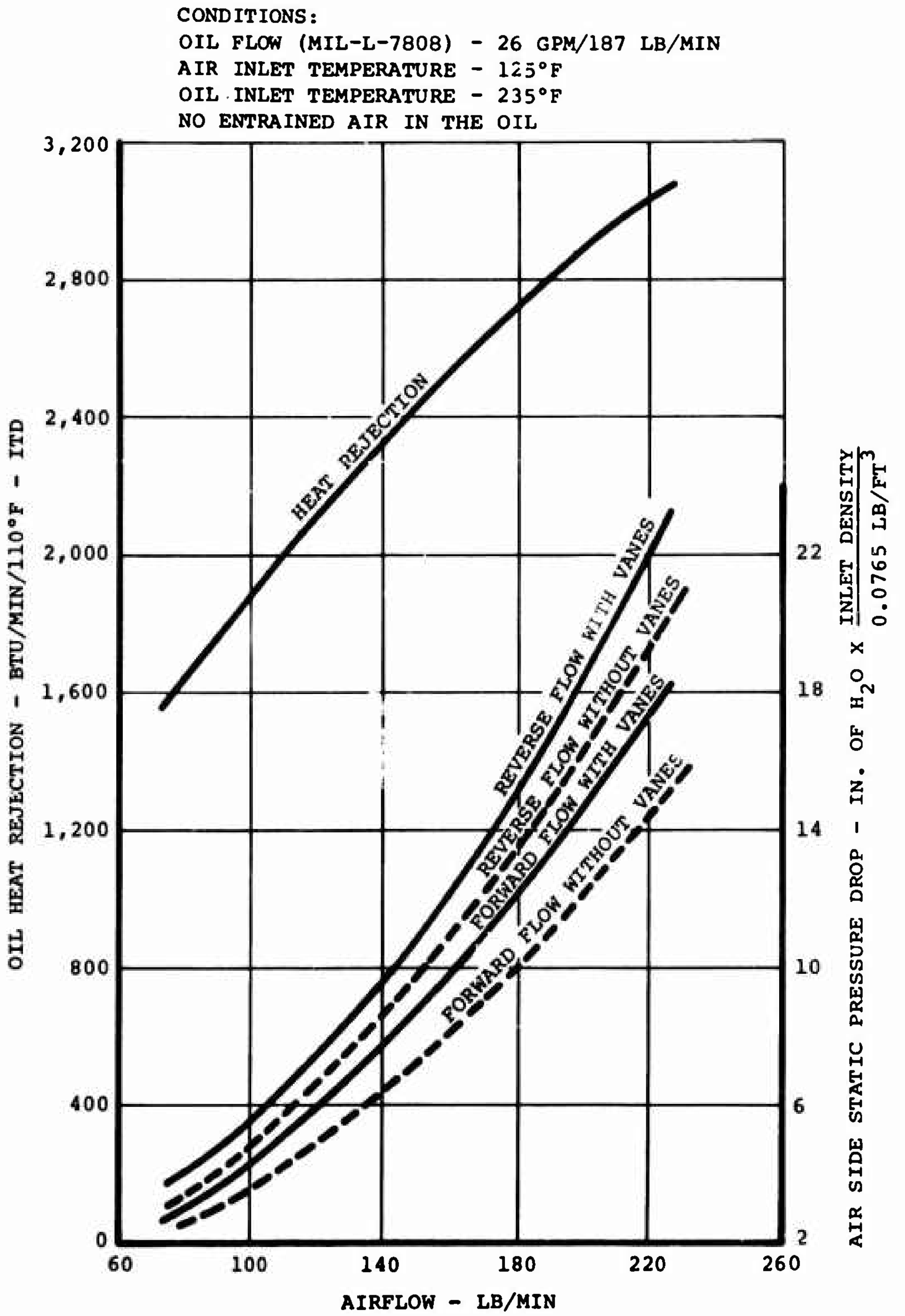


Figure 24. Effect of Zero-Inch Turning Vanes on Air Pressure Drop.

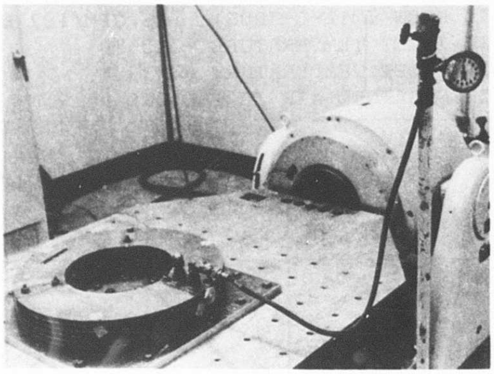


Figure 25. Longitudinal Vibration Test Setup.

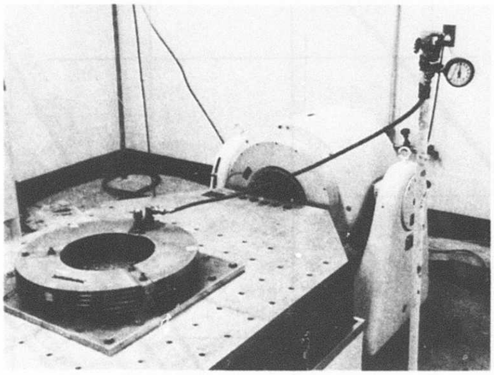


Figure 26. Lateral Vibration Test Setup.

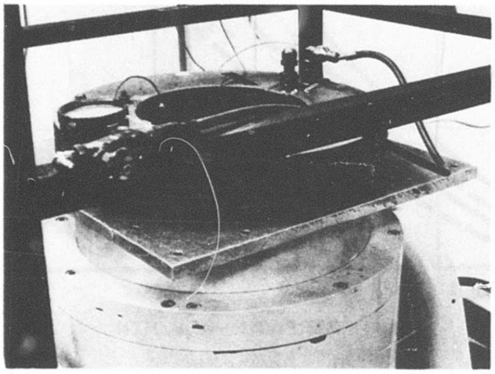


Figure 27. Vertical Vibration Test Setup.

stringent Boeing-Vertol range curve shown in Figure 28. In addition, the cooler was tested at approximately 1,563 Hz with an acceleration of $\pm$10g for 50 hours in the lateral, longitudinal, and vertical directions. This frequency is the predominant frequency present at the bottom of the CH-47C forward rotor transmission and is typical of helicopter transmission gear meshing frequencies.

Table XI contains the results of the vibration testing, while Figure 29 defines the various accelerometer locations used during testing. The tests were performed to ensure that an oil cooler of an annular design could be hard-mounted to a typical helicopter transmission without suffering damage from vibration. The CH-47C helicopter's forward rotor transmission was chosen as a baseline for the vibrational environment. Resonance searches were performed in each of the three mutually perpendicular axes, followed by 30-minute dwell tests at the four most-severe resonances in each axis. One-hour cycling tests from 5 to 3,500 Hz and 50-hour dwells at 1,563 Hz were also conducted in each axis. The 1,563 Hz is typical of the predominant vibrational frequencies of the CH-47 helicopter's forward rotor transmission. In general, no severe resonances were found below 375 Hz. The cooler satisfactorily passed all vibration test requirements typical of production CH-47 helicopter oil coolers.

The design of the test cooler was considered to be excellent since the lowest significant resonance was above the rotor harmonic and unbalance speeds of all dynamic components of the CH-47 helicopter. Because the test specification under which the cooler was tested exceeded the requirements of MIL-E-5272C, Procedure XII, this oil cooler has also been vibration-qualified for use on all gas turbine engines, on structures of aircraft powered by reciprocating, turbojet, or turboshaft engines, and on missiles powered by turbojet engines. The above testing was performed on an oil cooler filled with MIL-L-7808 oil and pressurized to 90 psig.

PRESSURE CYCLING TEST

Upon completion of the performance, oil pressure drop, and blower location optimization testing on unit 1, the oil cooler was subjected to a pressure cycling test. This test was designed to simulate the cycling of oil pressure that occurs during the startup and shutdown of the aircraft throughout a normal operational life. During testing, the oil cooler was filled with MIL-L-7808 oil at 300° + 0° - 10°F and cycled from 3 $\pm$ 3 psig to 60 $\pm$ 1 psig. A pressure cycling rate of 12 cycles per minute was used for a total of 50,000 cycles. Figure 30 is a photograph of the oil cooler during the pressure cycle testing. At the conclusion of this test, the oil cooler was subjected to a 400-psig hydrostatic pressure test and checked

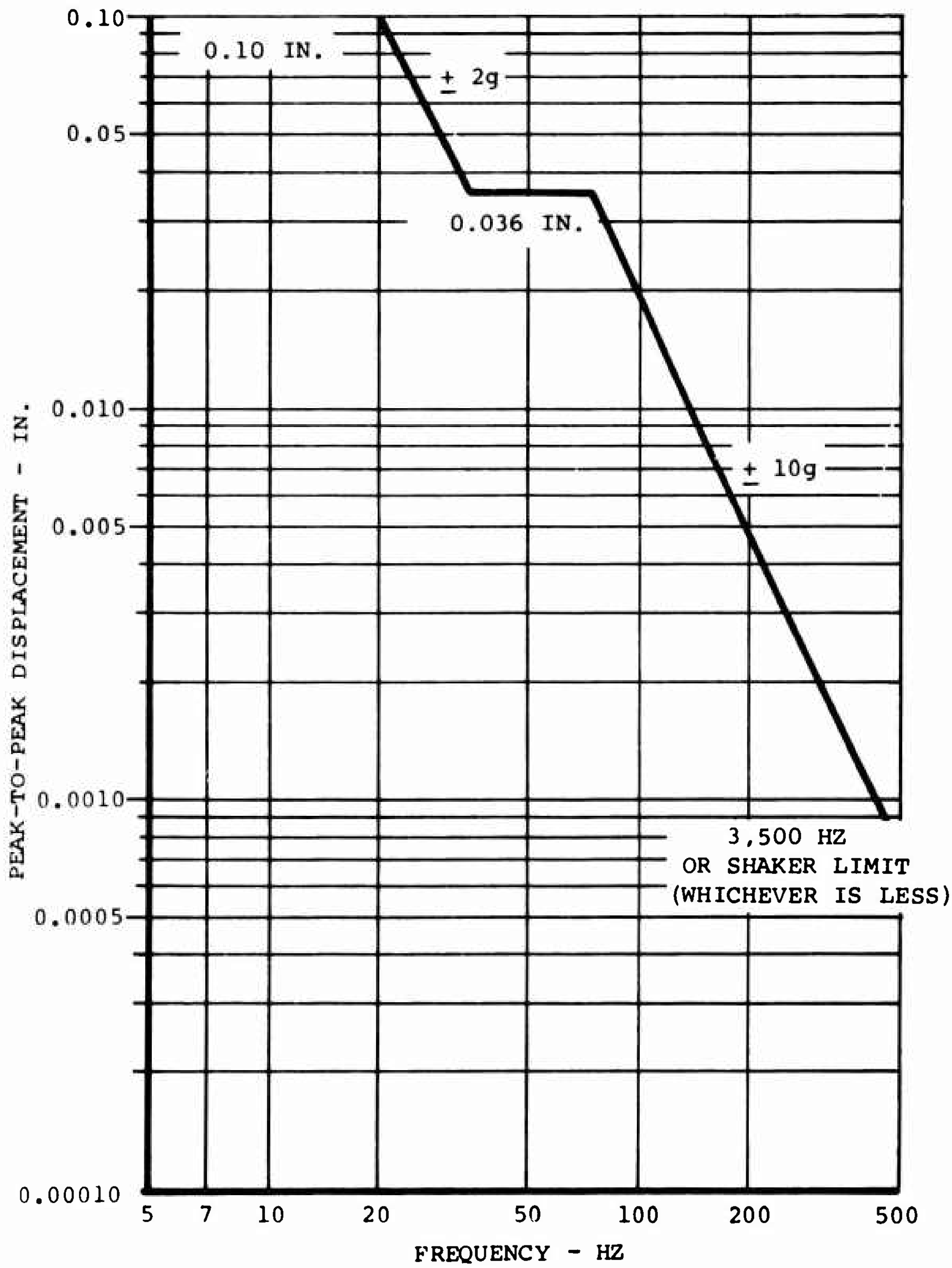


Figure 28. Boeing-Vertol Component Qualification Range Curve for Vibration Tests.

TABLE XI. VIBRATION TEST DATA

Horizontal Plane — Longitudinal Axis

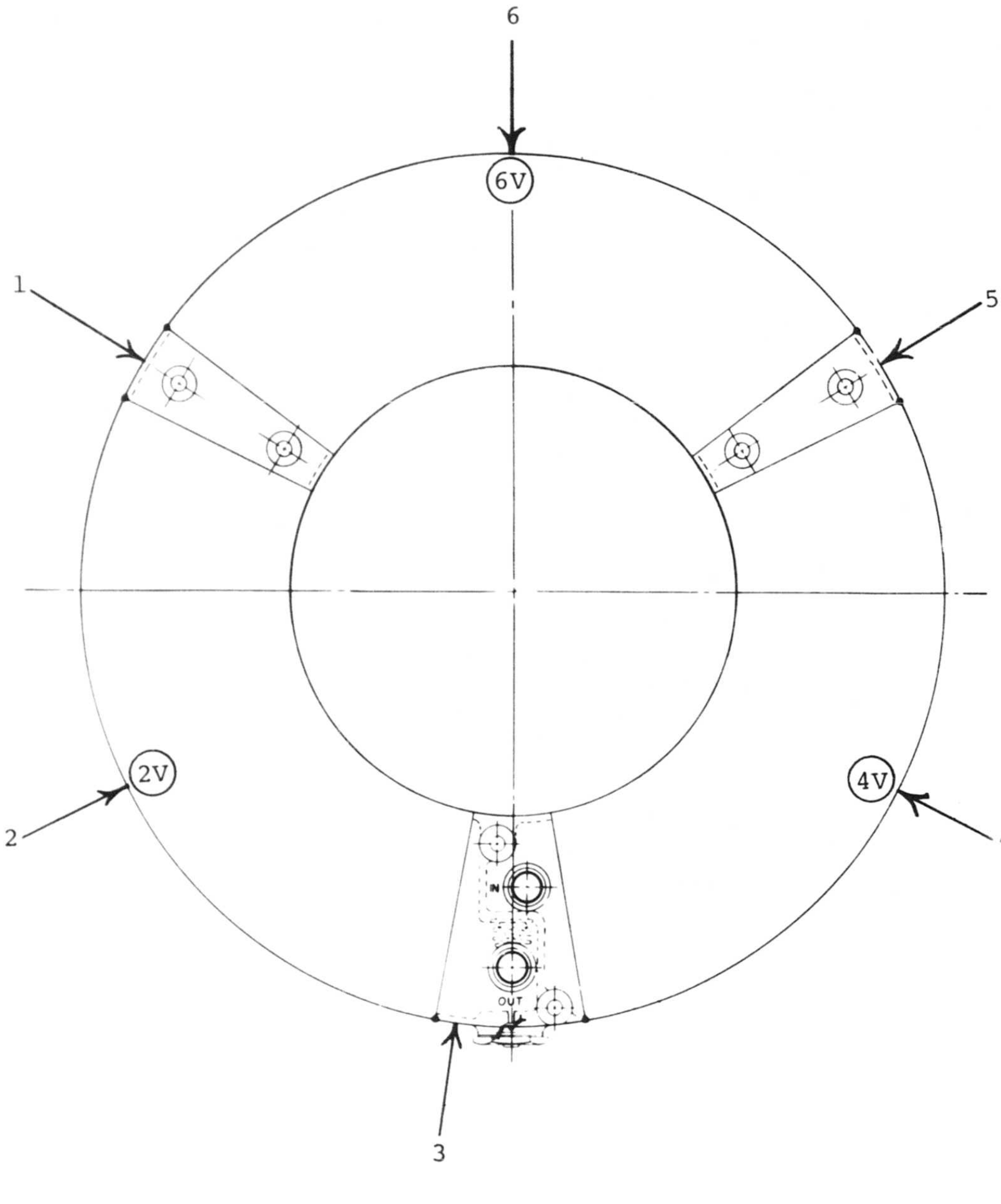
Vibrational Frequency (Hz)	283	430	570	858	1,133	1,735	2,309	1,563
Input Acceleration (g)	10	10	10	10	10	10	10	10
Measurement Location	Output Acceleration (g)							
1	6	12	16	15	10	12	8	6
2	32	56	84	86	12	84	4	10
2V	22	70	94	90	50	62	54	30
3	22	20	72	16	22	16	8	4
4	20	28	62	32	6	26	6	14
4V	30	68	58	100	24	72	12	20
5	30	32	60	28	20	14	10	6
6	18	24	62	44	10	44	4	8
6V	20	56	80	92	40	78	52	42
Test Time (hours)	-	0.5	0.5	0.5	-	0.5	-	50

Horizontal Plane — Transverse Axis

Vibrational Frequency (Hz)	375	496	648	818	1,144	1,717	2,292	1,563
Input Acceleration (g)	10	10	10	10	10	10	10	10
Measurement Location	Output Acceleration (g)							
1	40	48	33	22	25	7	23	10
2	18	26	74	55	10	24	4	18
2V	25	58	58	87	83	22	87	23
3	20	40	25	18	22	5	24	3
4	22	30	80	80	18	18	28	35
4V	50	36	100	-	86	20	68	55
5	7	18	24	18	28	6	26	3
6	22	56	24	140	7	6	6	8
6V	32	43	30	-	65	12	68	10
Test Time (hours)	-	-	0.5	0.5	0.5	-	0.5	50

Vertical Axis

Vibrational Frequency (Hz)	375	722	1,045	1,299	1,380	2,085	2,742	2,919	1,563
Input Acceleration (g)	10	10	10	10	10	10	10	10	10
Measurement Location	Output Acceleration (g)								
1	4	8	8	12	10	6	18	8	3
2	12	20	10	6	11	4	18	11	4
2V	35	80	40	10	10	12	18	10	8
3	10	12	10	26	16	5	26	11	7
4	10	10	8	14	8	6	11	6	4
4V	30	80	32	20	16	20	22	15	56
5	10	10	10	6	11	12	25	14	5
6	8	10	8	20	10	4	12	10	8
6V	18	26	18	40	16	20	26	15	18
Test Time (hours)	0.5	0.5	0.5	0.5	-	-	-	-	50



1 THROUGH 6 - LATERAL AND LONGITUDINAL MEASUREMENTS
2V, 4V, and 6V - VERTICAL MEASUREMENTS

Figure 29. Acceleration Measurement Locations.

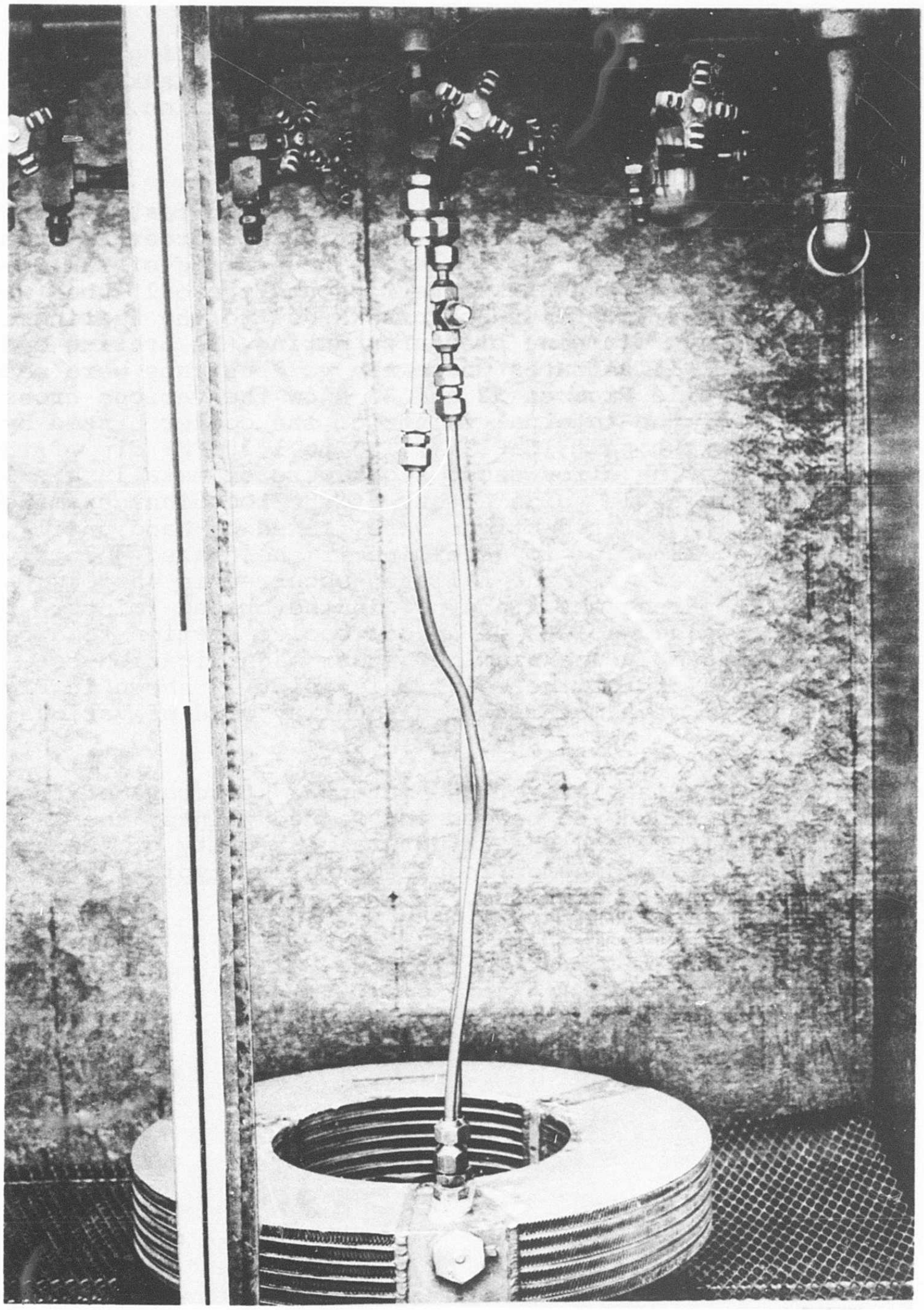


Figure 30. Pressure Cycle Test of Cooler.

for signs of leakage or permanent distortion. During the pressure testing, the oil cooler did not develop leaks or permanent distortions and was in excellent condition.

OIL COOLER INSPECTION

After the successful completion of the vibration testing, the test unit was sectioned at various locations in order to examine the internal condition of the cooler. The brazing of the internal oil turbulator and air fin strips and the oil tubes was examined at various locations for cracks due to the testing or voids due to uneven clamping pressures during the brazing operation. Figure 31 illustrates where the various cuts were made in the cooler, while Figures 32 and 33 show the various cross sections. All of the internal joints of the cooler brazed by the high-temperature salt (NaCl) (1,100 to 1,200°F) dip of the aluminum-clad brazing alloy sheet stock were of metallurgical quality with no visible voids at any of the locations examined. In addition, thin cooler sections were flexed by hand until failure occurred in order to determine if the brazed joints would fail. However, all the failures occurred in various sections of the parent metal and not in the brazed joints. One of the heliarc-welded joints of an oil cooler section to a cast aluminum tank assembly was examined under magnification to determine the quality of the weld; this joint is shown in Figure 33 and is typical of a metallurgical-quality weld of various aluminum alloys.

Based on these examinations of the internal structure of the annular oil cooler and the brazed and welded joints, the method of manufacture and the structural integrity of the unit were found to meet the design requirements for typical helicopter environmental conditions.

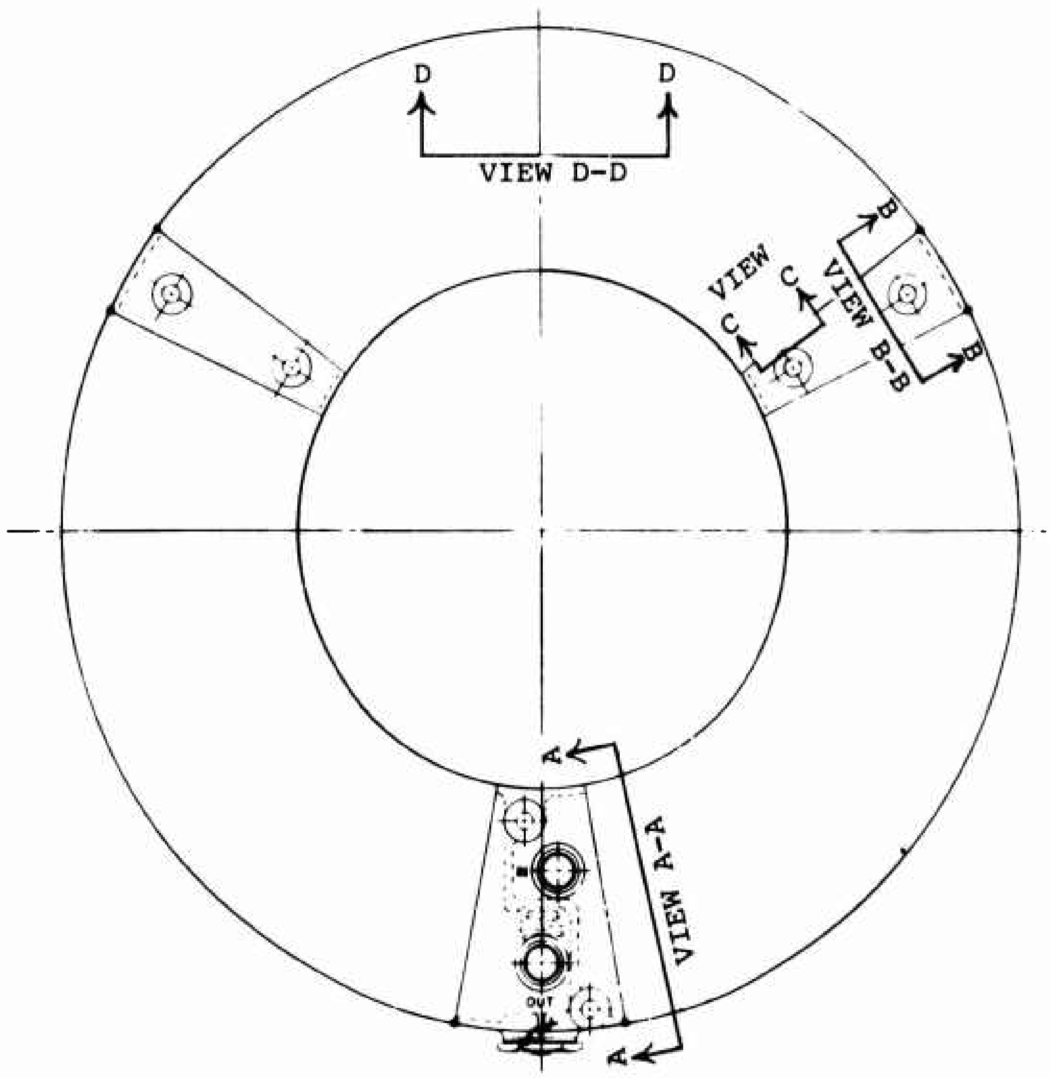
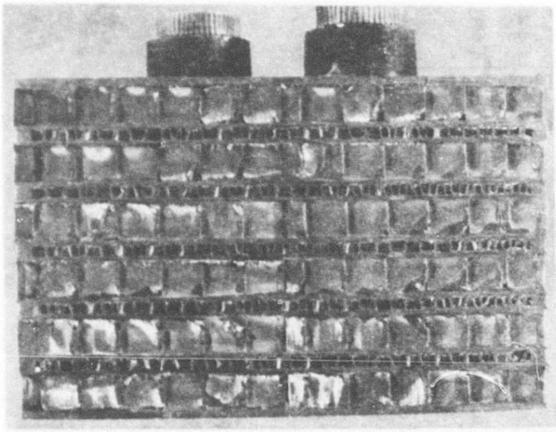
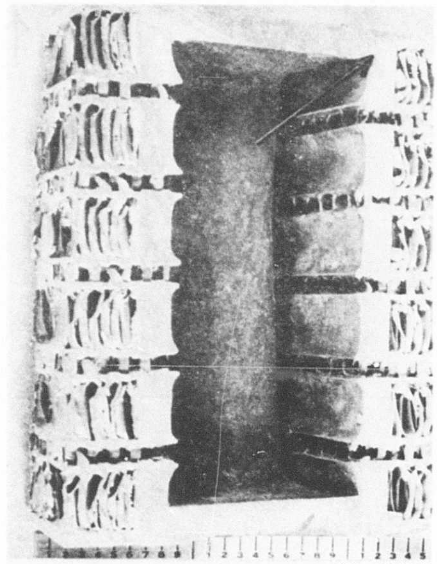


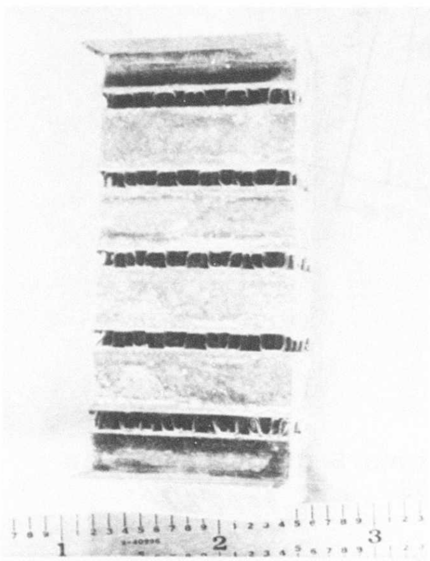
Figure 31. Location of Various Sections Shown in Figures 32 and 33.



(0.71X)
VIEW A-A



VIEW B-B (0.89X)



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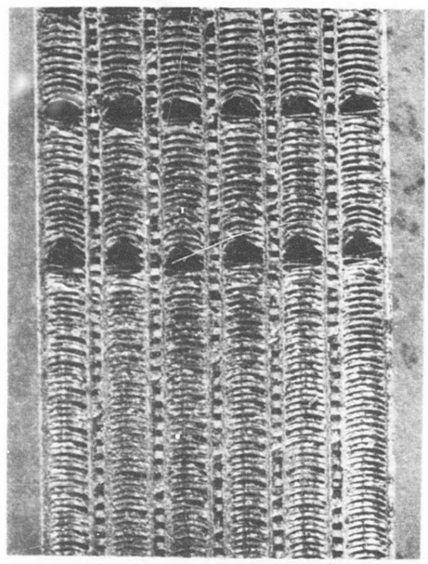
VIEW C-C



VIEW D-D (0.71X)

Figure 32. Oil Cooler Inspection Photographs.

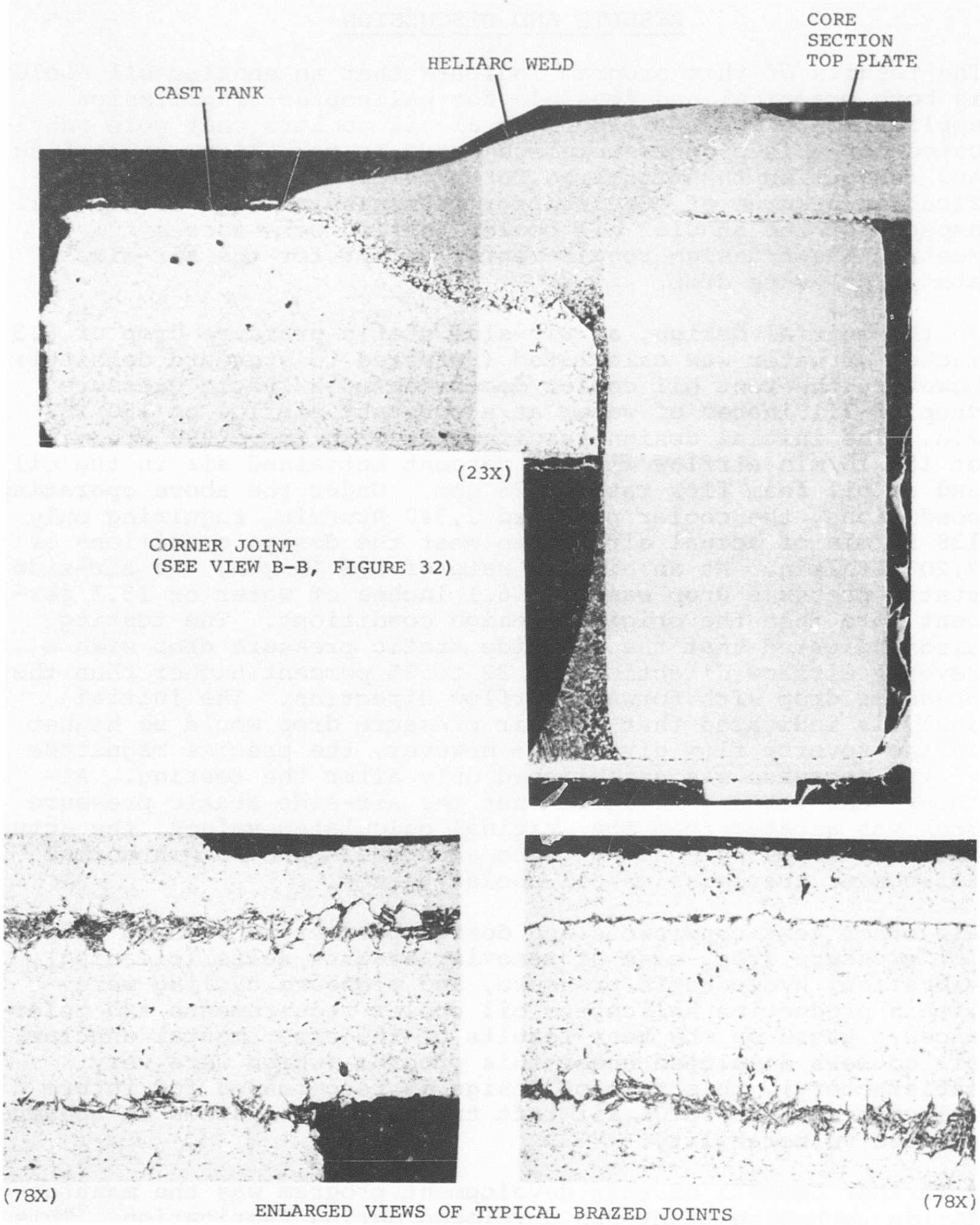


Figure 33. Typical Brazed and Welded Joints.

RESULTS AND DISCUSSION

The results of this program indicate that an annular oil cooler is both practical and feasible for helicopter transmission application. The two experimental oil coolers that were fabricated for this program were subjected to more rigorous testing and inspection than required for a complete production qualification program of a helicopter transmission oil cooler. All aspects of the annular oil cooler testing were successful in meeting basic design requirements, except for the air-side static pressure drop.

In the initial design, an air-side static pressure drop of 5.3 inches of water was calculated (referred to standard density); however, the test oil cooler demonstrated a static pressure drop of 7.1 inches of water at a constant airflow of 150 lb/min. The initial design requirements were for 2,200 Btu/min at 150 lb/min airflow with 20 percent entrained air in the oil and an oil foam flow rate of 26 gpm. Under the above operating conditions, the cooler produced 2,340 Btu/min, requiring only 138 lb/min of actual airflow to meet the design conditions of 2,200 Btu/min. At an airflow rate of 138 lb/min, the air-side static pressure drop was only 6.1 inches of water or 15.1 percent more than the original design conditions. The testing also indicated that the air-side static pressure drop with a reverse airflow direction was 32 to 35 percent higher than the pressure drop with forward airflow direction. The initial analysis indicated that the air pressure drop would be higher in the reverse flow direction; however, the precise magnitude of the increase was established only after the testing. Although the testing revealed that the air-side static pressure drop was greater than the original calculated values, the actual pressure drops were found to be acceptable and within normal helicopter transmission oil cooler values.

All other test conditions and design parameters such as weight, oil pressure drop, ease of removing brazing salts (cleaning), vibration, hydrostatic pressure, and pressure cycling were within production helicopter oil cooler requirements and tolerances. Based on the test results of the experimental annular oil coolers developed under this program (which were very satisfactory), this type of design is recommended for future helicopters and V/STOL aircraft transmission systems to achieve reduced vulnerability.

A further benefit of this development program was the manufacturing technology that was developed during fabrication. This experience can also be used to accurately determine the cost of production quantities of oil coolers of this type. In the initial study contract (DAAJ02-67-C-0112), the most cost-effective system turned out to be the integral transmission oil cooling system using an annular-type oil cooler of the same size as the units fabricated under this program. A system cost rating and reliability cost value of $1,800 was estimated for this cooler in the study contract. This figure was approximately 2.5 to 3 times higher than for an equivalent rectangular unit. At that time, this value appeared to be reasonable when compared to data from various aircraft oil cooler manufacturers. With the knowledge gained from this program, it is currently estimated that an annular oil cooler manufactured in production quantities will be in the same cost range as the present rectangular oil coolers and, thus, directly competitive from an initial cost standpoint.

CONCLUSIONS

The following conclusions are drawn from this program as a result of the work described herein:

1. Based upon the results of the heat transfer performance testing conducted during this program, an annular-type, plate-fin oil cooler is feasible for cooling the main transmissions of present and future helicopter and V/STOL aircraft configurations.

2. The annular-type oil cooler fabricated for this program did not present manufacturing problems and is therefore considered to be a producible configuration.

3. The axial location of the blower in an annular cooler/blower package has virtually no effect on system static pressure drop within the limits of the test program conditions. This indicates that the best overall blower location is directly against the annular oil cooler with no intermediate ducting.

4. The addition of the turning vanes within the inner diameter of the annular oil cooler to aid in the required 90-degree change in airflow direction has no beneficial effect and only tends to increase the system's static pressure drop.

5. The annular oil cooler design fabricated during this program is considered to be excellent from a vibration standpoint since the lowest significant resonance was found to be above the rotor harmonic and unbalance speeds of all dynamic components on current Boeing-Vertol helicopters. Based on the vibration test results, an annular-shaped oil cooler with the same type of construction and three-point mounting method as the unit fabricated during this program should be capable of withstanding the vibrational environment existing on the transmissions of present and future helicopters and V/STOL aircraft.

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APPENDIX
DESIGN SIZING CHARTS

Detailed design information for aircraft-type, plate-fin, liquid-to-air heat exchangers is not readily available to the helicopter transmission design engineer. The unusual shape of the annular-type oil cooler used in this program adds more complexity to the design procedure of plate-fin-type heat exchangers. To facilitate the incorporation of annular oil coolers into the preliminary designs of new V/STOL aircraft power transmission systems so that the designer may readily investigate the advantage of this type of oil cooler, the following design charts are included in this report.

The design sizing charts presented in Figures 34, 35, and 36 are plotted for a constant air-side static pressure drop at inlet density conditions based on the actual air static drop obtained from the tests conducted in this program. The charts are generalized by plotting heat transfer and airflow as a function of oil cooler no-flow (axial thickness) height.

To use these annular cooler size selection charts, an inner cooler diameter must be selected. This inner diameter is usually determined by the minimum blower diameter necessary to move the required amount of cooling air at the blower drive shaft speed available. Select a chart for the air static drop desired (the higher the static drop, the smaller the cooler); then, for a given inner diameter and airflow length (radial thickness), the airflow and heat rejection per inch of no-flow height can be determined. To obtain the cooler height required, divide the desired heat transfer by heat transfer per inch. Airflow required by the heat exchanger is obtained by multiplying the heat exchanger height by the airflow per inch. Oil flow required to meet the heat transfer requirements can be found by multiplying oil passage width by the number of oil passages in the heat exchanger. Each oil passage and air center combination is approximately 0.53 inch high and thus requires that the oil cooler no-flow height measurement be in increments of 0.53 inch.

For example, let us assume that an annular heat exchanger is required to have an inside diameter of 8 inches, an outside diameter of 18 inches, and a heat rejection capacity of 1,430 Btu/min/110°F inlet temperature difference (itd), with a maximum air-side static pressure drop at inlet conditions of 6 inches of H_2O. From the 6-inch static pressure drop chart and an airflow length of 5 inches, it is found that the heat rejection is 565 Btu/min/110°F itd per inch of no-flow height, while the airflow is 31 lb/min per inch of no-flow height. Now, divide the 1,430 Btu/min by the 565 Btu/min, and the required no-flow height is 2.53 inches; since the nearest multiple of 0.53 inch is 5, the actual no-flow height will have to be 2.65 inches. The required airflow will then be 2.53 inches times the 31 lb/min per inch of no-flow height, or approximately 79 lb/min.

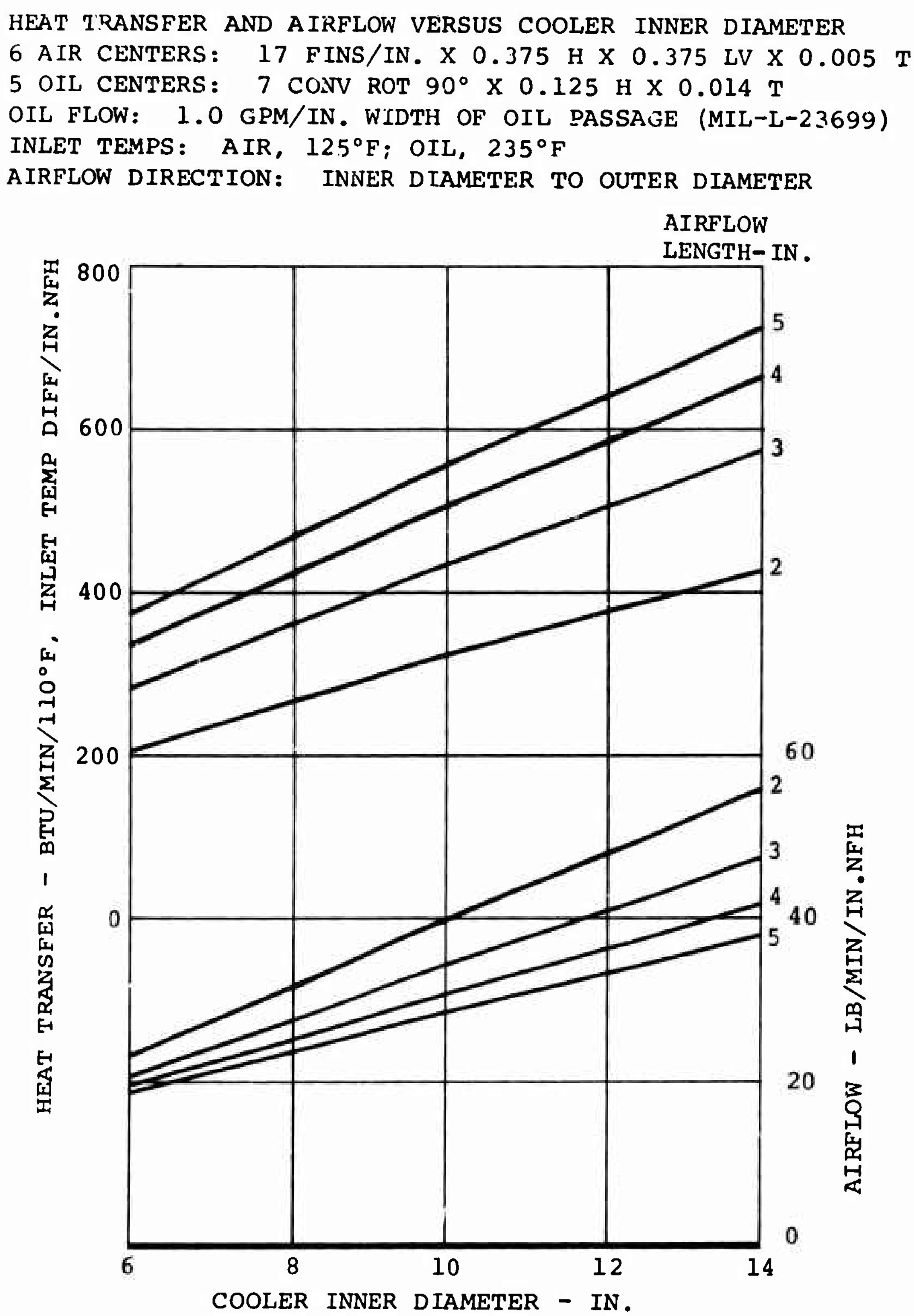

Figure 34. Annular Oil Cooler Size Selection Chart, Static Drop of 4 Inches of Water at 0.0679 Lb/Ft^3 Density.

HEAT TRANSFER AND AIRFLOW VERSUS COOLER INNER DIAMETER
6 AIR CENTERS: 17 FINS/IN. X 0.375 H X 0.375 LV X 0.005 T
5 OIL CENTERS: 7 CONV POT 90° X 0.125 H X 0.014 T
OIL FLOW: 1.0 GPM/IN. WIDTH OF OIL PASSAGE (MIL-L-23699)
INLET TEMPS: AIR, 125°F; OIL, 235°F
AIRFLOW DIRECTION: INNER DIAMETER TO OUTER DIAMETER

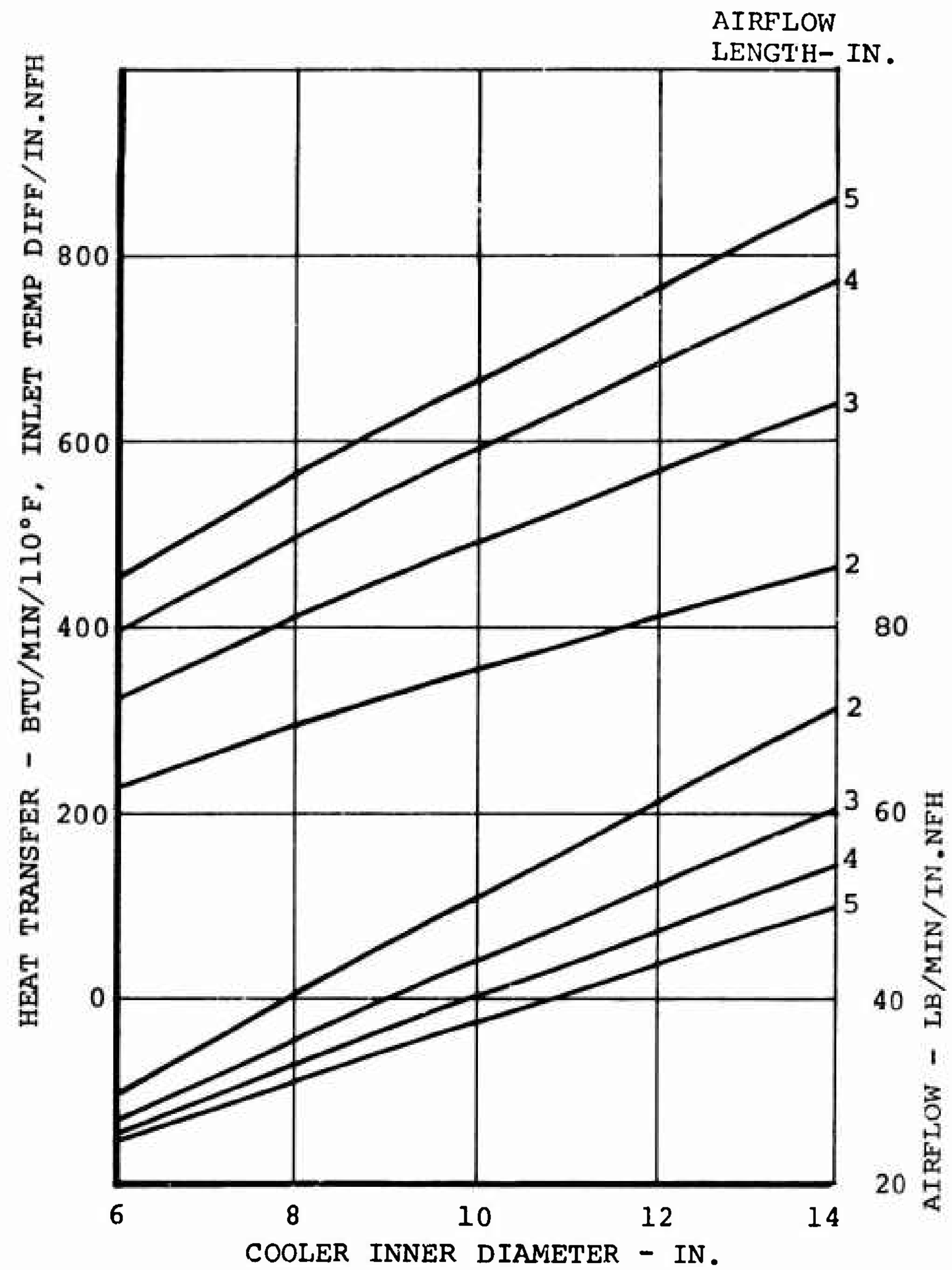

Figure 35. Annular Oil Cooler Size Selection Chart, Static Drop of 6 Inches of Water at 0.0679 Lb/Ft^3 Density.

HEAT TRANSFER AND AIRFLOW VERSUS COOLER INNER DIAMETER
6 AIR CENTERS: 17 FINS/IN. X 0.375 H X 0.375 LV X 0.005 T
5 OIL CENTERS: 7 CONV ROT 90° X 0.125 H X 0.014 T
OIL FLOW: 1.0 GPM/IN. WIDTH OF OIL PASSAGE (MIL-L-23699)
INLET TEMPS: AIR, 125°F; OIL, 235°F
AIRFLOW DIRECTION: INNER DIAMETER TO OUTER DIAMETER

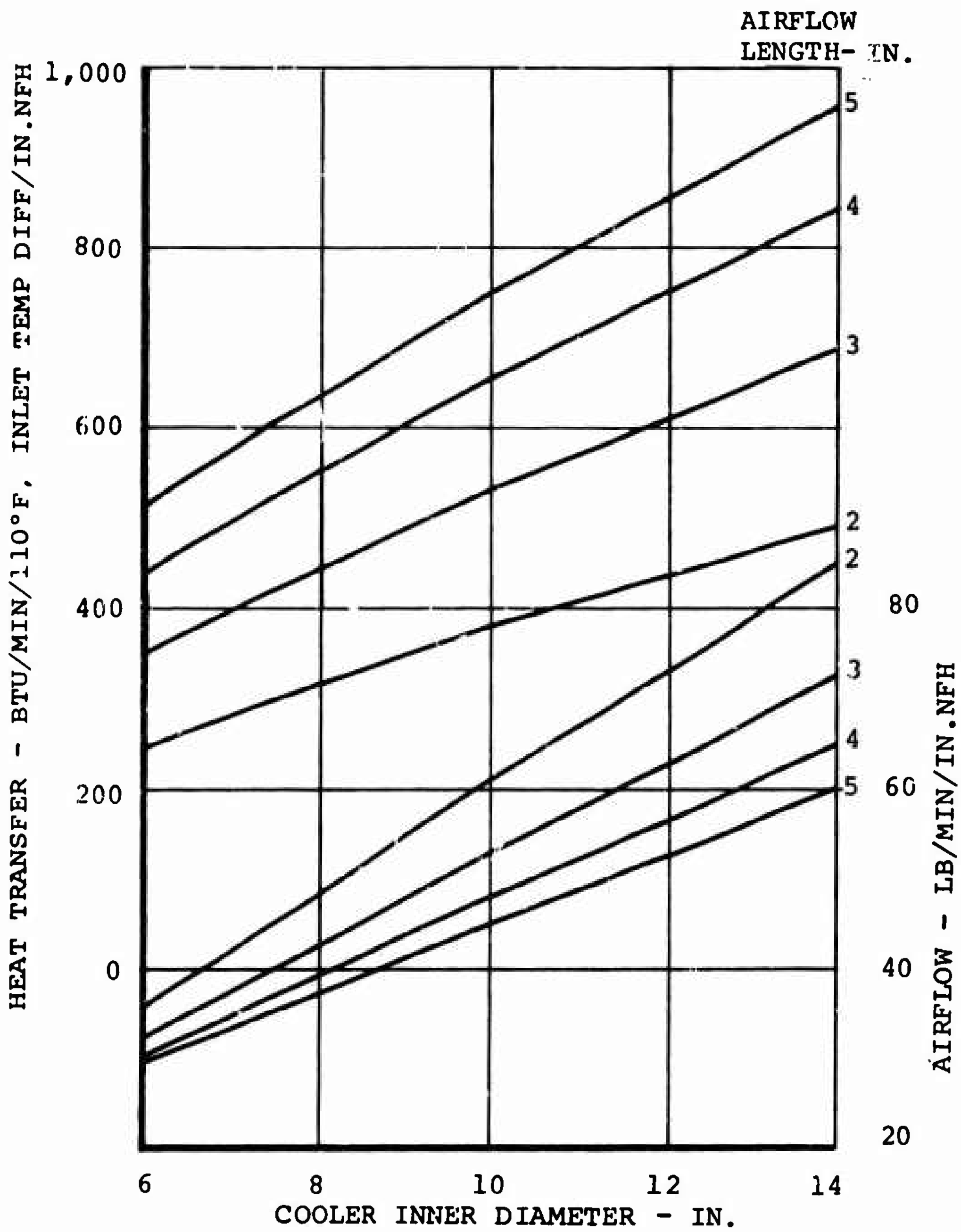

Figure 36. Annular Oil Cooler Size Selection Chart, Static Drop of 8 Inches of Water at 0.0679 Lb/Ft^3 Density.

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13. ABSTRACT

Under this contract, Boeing designed and built an experimental, annular-shaped transmission oil cooler for use on the bottom of the forward-rotor transmission of the CH-47C helicopter. Subsequent tests indicate that this annular-shaped cooler with a radial airflow direction is practical, from a producibility and manufacturing feasibility standpoint, for helicopter and V/STOL aircraft.

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Oil Cooler Heat Exchanger Annular-Shaped Oil Cooler CH-47C Forward-Rotor Transmission Radial Airflow Direction Vulnerability of Rotor Transmissions						