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**INTERIM TECHNICAL REPORT NO. 5**

on

**30,000-RPM VANE-PUMP DEMONSTRATION**

D. L. Thomas  
J. P. Dechow  
R. K. Catterson  
et al.

**BATTELLE MEMORIAL INSTITUTE**  
Columbus Laboratories

April 15, 1970

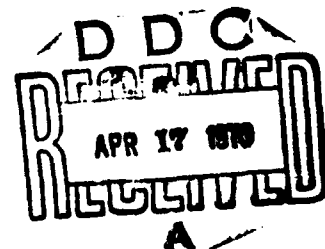
Contract No. F33615-69-C-1302

Project No. 3145

Period: January 1 to March 31, 1970

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## FOREWORD

This report was prepared by the Columbus Laboratories, Battelle Memorial Institute, 505 King Avenue, Columbus, Ohio 43201 under USAF Contract No. F33615-69-C-1302, Project No. 3145. The Battelle project number was G-9230. Administration was under the Air Force Aero Propulsion Laboratory, Air Force Systems Command, Wright-Patterson Air Force Base, with Mr. K. E. Binns (APIE-1) acting as project monitor.

The program is being conducted with Mr. D. L. Thomas as principal investigator. Mr. R. K. Catterson is the Division Chief in overall charge of the program. Other members of the professional staff who made significant contributions during the conduct of the project are Mr. J. P. Dechow, Mr. J. T. Herridge, Mr. H. T. Johnson, Mr. R. K. Mitchell, and Dr. J. P. Wilcox.

This report, which was submitted April 15, 1970, covers work conducted during the period of January 1 to March 31, 1970. Other Interim Technical Reports published on this contract are identified below:

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This report is being published and distributed prior to Air Force review. The publication of this report, therefore, does not constitute approval by the Air Force of the findings or conclusions contained herein. It is published for the exchange and stimulation of ideas.

### ABSTRACT

An answer to the start-up problem involving the pivoting tips and the soft indium-lead-silver-plated cam surface was confirmed experimentally. Changes include antirotation stops on the vanes and a chamfer on the leading edge of the pivoting tip. Successful speed runs were conducted with the variable-displacement pump to 29,000 rpm, but were interrupted by bearing failures. The high-speed drive was redesigned to eliminate the cause of the bearing failures. Considerable progress has been achieved with electron-beam welding and lamination-wrapping studies in the program to evolve a technique for developing deformable cam rings.

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SECTION I  
INTRODUCTION

This project is the continuation of the effort previously conducted under Contract AF33(615)-2723 for the investigation of a 30,000-rpm turbine-driven hydraulic pump. (1) During that project a vane pump was selected as the leading concept and preliminary experimental evaluations showed it to be very promising.

The purpose of the present project is to demonstrate the capability of the pump concept to meet future aircraft requirements. The specific objective is to demonstrate a variable-delivery pressure-compensated vane pump with the following characteristics:

|  |   |
|--|---|
| Rated Speed  | 30,000 rpm  |
| Rated Discharge Pressure   | 4000 psi  |
| Rated Inlet Pressure   | 125 psi   |
| Rated Delivery   | 50 gpm  |
| Rated Temperature at outlet, case drain, or intermediate point with 220 F at inlet | 275 F, maximum  |
| Efficiency at Rated Conditions   | 80 percent  |
| Hydraulic Fluid  | MIL-H-5606B   |
| Weight   | Not restricted at present -<br>eventual goal is 15 lb |
| Rated Endurance  | 1000 hr with no maintenance                           |

Major project tasks are:

- (1) Evaluate basic pump concepts to meet the design requirements
- (2) Experimentally qualify components such as vanes and variable-displacement mechanisms
- (3) Evaluate a fixed-displacement model of the pump
- (4) Design and fabricate a pressure-compensator control for the variable-displacement pump

(1) Thomas, D. L., et al., "Investigation of a 30,000-Rpm Turbine-Speed Hydraulic Pump", AFAPL-68-5, AD 834312, by Battelle Memorial Institute (June, 1968).

- (5) Evaluate the variable-displacement pump at the design conditions
- (6) Perform a 1000-hour endurance run.

The basic vane-pump concept selected for development has a laminated, deformable cam ring for varying displacement and pivoting-tip vanes that are hydrodynamically supported for high-speed operation. A computer analysis of the pivoting tips has been conducted and they have been experimentally qualified at the design conditions. Various cam-ring materials have been investigated in wear and fatigue tests. The fixed-displacement model pump has been designed and fabricated and evaluation has begun. A computer analysis of the pressure compensator has been completed. A program for evolving a deformable cam-ring fabrication technique and for providing cam rings for pump evaluation is being conducted.



## SECTION II

### TECHNICAL INVESTIGATION

#### 1. FIXED-DISPLACEMENT PUMP

##### a. Start-Up Problem

The start-up problem, discussed in Interim Technical Report No. 4, was investigated experimentally. The problem was revealed when a failure occurred seconds after start-up and before pump speed reached 2500 rpm. The pump had two vanes with tungsten-carbide pivoting tips and a one-half-displacement cam ring. One tip left its socket and was broken in many pieces, resulting in damage to the other parts.

The start-up problem apparently occurred because the sharp leading edge of a tip contacted the cam ring and dug in. The vane and tip vibrated radially, chattering against the soft indium-lead-silver-plated cam ring until a sufficiently high tangential force dragged the tip from the vane socket.

Two changes were made to correct this problem. A small chamfer was ground on the leading edge of the pivoting tip so it could contact the cam ring only with a smooth sloping surface. Stops were added to the vane to limit tip rotation. This ensured that radial vane force could overcome cam-surface frictional drag and allow the tip to pivot to its proper orientation with the cam surface. The stops were brass shim stock soldered in place and then machined.

The changes were evaluated experimentally using a solid cam ring with the circular indium-lead-silver-plated bore sufficiently eccentric to cause full vane stroking. Six starts were made with one pivoting-tip vane and seven starts were made with two pivoting-tip vanes. The initial position for the last ten starts was with the tips rotated backward so they would contact the cam ring with their leading edge. Thus the tip was forced to rotate in a direction counter to the frictional drag of the cam ring to achieve the proper orientation with the cam-ring surface. In the last nine starts, the vane was positioned radially inward at the bottom of its rotor slot. The cam ring was examined after each starting sequence. No distress was observed in any of the tests.

A new set of pivoting-tip vanes that have integral vane stops and chamfered pivoting tips were ordered. The set of tips was received late in March and the set of vanes is expected early in April.

b. Vane Cocking

Vane cocking in the radial direction, wherein diagonally opposite corners of a vane could jam between the end plates at start-up, is a potential problem. The possibility of jamming does not exist for vane end clearances greater than approximately 0.0043 inch because the rotor and cam ring limit radial motion. In four of the start-up experiments with two vanes described above, the vane end clearance was less than 0.0043 inch and as little as 0.0026 inch. No difficulty with vane jamming was noted.

c. Speed Runs

Speed runs were conducted using an indium-lead-silver-plated circular-bore eccentric cam ring, reworked vanes and tips, and essentially room-temperature hydraulic fluid. In a series of runs, six vanes were tested to approximately 29,000 rpm. At this speed a failure occurred in which the shaft-end pump bearing and the out-board drive bearing overheated.

The leading pivoting-tip vane stop was completely missing on one vane, partially missing on two vanes, and damaged slightly on the other three. The pivoting tips were unchanged. The cam surface showed some marks that were attributed to loss of the vane stops but showed no distress caused by speed or load. It is assumed that centrifugal force and/or fluid-inertial forces were great enough to cause the loss of one, perhaps insecurely soldered, vane stop, which then caused the other damage. As the new set of vanes will have integral vane stops, it was decided to not resume tests until they are available.

The speed tests were interrupted by several bearing failures which were due to high radial loads caused by the first critical speed for the pump-shaft drive-coupling arrangement. Subsequent to two successful rotor-only checkout runs to 30,000 rpm, the shaft-end pump bearing overheated while increasing speed to approximately 17,500 rpm. It was calculated that the first critical speed for the pump-shaft drive-coupling arrangement was less than 21,000 rpm. The drive coupling touched the magnetic speed pickup in a manner which indicated excessive deflection in a selected rotational orientation which confirmed that critical speed was the problem.

An outboard bearing was added to support the shaft coupling and preclude excessive deflection. The first outboard bearing tried had insufficient area and was inadequately lubricated. The second outboard bearing worked satisfactorily until it and the pump shaft-end bearing overheated at approximately 29,000 rpm, as discussed above.

The cause of this failure was not clear. Shredded material from the brass vane-stop could have passed through the pump shaft-end bearing in sufficient quantity to cause it to overheat. This could have caused sufficient shaft distortion and load to cause the outboard bearing to overheat. The other possibility was that the independent mounting of the pump and speed increaser by brackets from a common frame was inadequate for the precise alignment necessary. This drive arrangement was successfully used in the previous project, but, there, the pump shaft had a much higher natural frequency.

It was decided to mount the pump directly on the speed increaser in future runs. This will eliminate alignment problems, eliminate the outboard bearing, and increase the combined pump shaft-coupling critical speed to above 40,000 rpm. The new drive is completed and will be installed and tested early in April.

Solid cam rings having the cam-ground profiles of the one-half- and full-displacement cam rings are now ready for evaluation.

A center housing for evaluating a manually controlled deformable cam ring has been fabricated and is ready for use.

## 2. DEFORMABLE CAM RING

### a. Electron-Beam Welding Studies

During this period electron-beam welding studies were continued using solid cylinders which simulated the shape of the deformable cam ring but which did not have laminations. The purpose of the studies was to investigate possible means to limit the porosity and distortion introduced during welding. These investigations were completed during this period.

Previous efforts with electron-beam welding were made using hollow nested cylinders. These cylinders exhibited severe shrinkage problems, with the ID and OD at the axial center shrinking several mils more than the corresponding ID's and OD's at the edges.

Cylinders with stepped bores (small bore in the axial center with larger bores at the edges) were fabricated to increase the stiffness in the center and counteract the welding-shrinkage pattern. These stepped-bore cylinders reduced the bellmouthing problem, however, they still exhibited essentially the same distortion characteristics as a solid cylinder. Since the solid cylinder was easier to fabricate than the stepped-bore cylinder, a solid mandrel was chosen for the deformable cam ring.

During these welding studies, it was found that the electron-beam weld profile varied as the temperature of the part being welded changed. As the temperature increased during welding, the weld depth decreased and the weld width increased. To eliminate changes in weld profiles due to this cause, the feasibility of maintaining the interior temperature of the part at a nearly constant temperature was investigated.

Sample cylinders were preheated using a defocused, low-current electron beam to 300, 400, 500, 600, 700 and 800 F in the welding chamber and were electron-beam welded at these temperatures. Porosity and distortion in the different samples preheated to the temperature of 500 F and above was less than those at lower temperatures. The interior temperature during welding was maintained at the preheat temperature by adjusting the electron-beam welding rate so that the heat loss from the part to the welder chamber equaled the heat gain from the electron-beam during welding passes. The interior temperature of the part being welded was continuously monitored with thermocouples whose leads went through the electron-beam welding chamber walls to a strip-chart recorder.  $600 \pm 20$  F was selected as the preheat and welding temperature for welding the deformable cam ring.

A final investigation was made to determine the distortion and porosity effects on the cylinder from two, three, and six axial welds in a lap space. It was previously observed that the out-of-round condition was more severe at the OD where the electron-beam weld had a "nail head" than at interior diameters where the electron-beam was narrower and there was surrounding material to somewhat balance the shrinkage direction. In this study, it was found that twelve axial welds, three each in four lap-space-size areas spaced at 90-degree intervals on the sample cylinder, gave an out-of-round condition of 0.002 inch on R at the OD, and an axial straightness to within 0.0003 inch at the OD. These conditions were judged to be compatible with the desired tolerances for the deformable cam ring. The sample cylinder with six axial welds placed in a lap-space-size area had distortion that was thought to be marginal for the deformable cam ring.

Summary of Welding Studies. Methods that were successfully used to reduce porosity and distortion are:

- Increasing the part stiffness
- Preheating the part prior to welding
- Maintaining the interior of the part at the preheat temperature during welding
- Reducing the electron-beam energy input by limiting the depth, width, and number of welds.

b. Wrapping and Fabricating Studies

As stated in Interim Technical Report No. 4, the first sample deformable cam ring was fabricated using a mandrel with an 0.008-inch-deep spot-face profile placed in the mandrel in a lap-space-block location. An 0.008-inch-thick strip was fitted into this spot-face profile and inert-gas arc welded to the mandrel. The strip was then continuously wound on the mandrel using a low-speed lathe to turn the mandrel while a 250-pound pull was applied to the strip.

During this report period the above sample ring was electron-beam welded. The cam ring had six overlapping axial electron-beam welds placed in the area containing the spot face in the mandrel. This welded the transition zone from one lamination to the next into a solid block, while all other elements of the cam-ring laminations outside this transition zone were continuous circular arcs. To maintain the straightness of axial elements of the cam ring after welding, six counter-balancing axial welds were placed at 180 degrees to the spot-face profile. With all of the above welding as well as the required axial welds in the other lap space block areas and circumferential welds at the ends of the cam ring, the distortion was greater than desired even using the above-mentioned preheating techniques, etc. Out-of-round conditions on the OD of the sample ring were measured at about 0.004 inch on R. Some porosity of the electron-beam welds in the spot-face-profile area was also found. Because of the distortion and porosity, efforts on this type of cam ring were discontinued.

Another method of fabricating the deformable cam ring was investigated, in which a round mandrel was used. The strip stock was inert-gas arc welded to the round mandrel and then continuously wrapped with 250 pounds pull on the strip. With this technique, a triangular-shaped gap occurs at the transition of the first wrap to the

second wrap in a lap-space area. After the electron-beam welding, etc., the mandrel will be removed and this triangular-shaped region will be plated with silver when the cam-ring bore is plated.

Currently three deformable cam rings are being fabricated per this second method. After the wrapping on the round mandrel mentioned above, the first deformable cam ring (SN-1) had three initial axial welds placed in each lap space and one circumferential weld put in each end of the cam ring. These welds were put only through the lamination stack and into the mandrel. No outer ring was used, and the electron beam had to penetrate only the lamination stack and into the mandrel. Since no solid ring was located over the laminations, there was less cross section to weld through. This permitted the minimum energy input and the minimum weld depth, i. e., just through the lamination stack. The OD of the electron-beam welds were ground down to the OD lamination surface and an outer ring, from which the lap space blocks are to be fabricated, was heat shrunk onto the lamination stack. This outer ring was relieved (using a milling operation) in the areas where the axial welds defining the lap-space-block edges were placed. These cut-out or relieved areas provided a reduction in the cross section that the second electron-beam welds must penetrate. SN-1 has been heat treated and is presently being faced down to length. It will then have the lap space blocks ground to size.

For the second deformable cam ring (SN-2) being fabricated, the number of electron-beam welds was reduced. Only one axial weld (instead of three as for SN-1) was put in each lap space. The axial welds, one in the approximate center of each lap space, and two circumferential welds, one on each end of the cam ring, were put only through the lamination stack, as before. Some porosity was found in the exterior surface of the axial weld in the lap space with the triangular gap at the mandrel/lamination interface. It appears that part of the electron-beam weld penetrated through this gap. A second welding pass was made which eliminated the exterior porosity. However, the extent of porosity remaining deeper in the weld cannot be determined until the center mandrel is removed. This cam ring now has the lap-space-block ring (with areas relieved for electron beam welding) heat shrunk on the lamination stack. The electron-beam welds, defining the edges of the lap-space blocks, will now be put in through the edge of the blocks, through the lamination stack, and into the mandrel.

A third deformable cam ring (SN-3) has been started. The processing of this cam ring will be essentially the same as for SN-2

except that the center electron-beam weld in the lap space with the triangular gap will be offset by the amount necessary to miss the edge of the gap. It is thought that this will eliminate the weld porosity found in SN-2. This cam ring is now ready for the first electron-beam welding.

### 3. VARIABLE-DISPLACEMENT PUMP

Decisions regarding the configuration of the variable-displacement pump have been made. It will have all of the rotating parts required in flight hardware (including shaft seals, bearings, etc.), but it will not be designed for minimum weight. It will be designed to fit the accessory pad on an auxiliary power unit for demonstration purposes. The design will reasonably simulate flight hardware, but its experimental nature will not be compromised and it will be instrumented as fully as necessary for laboratory evaluation.

### SECTION III

#### FUTURE WORK

For the coming 3-month period, the following work is planned:

- (1) Complete laboratory experiments with the fixed-displacement pump involving solid cam rings and mechanically positioned deformable cam rings
- (2) Complete the development of a process for fabricating deformable cam rings
- (3) Design and begin fabrication of the variable-displacement pump.



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