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USAAML TECHNICAL REPORT 64-13

CYCLOIDAL CAM TRANSMISSION

PHASES II AND III

May 1965

JUN 30 1965

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

CONTRACT DA 44-177-TC-651
WESTERN GEAR CORPORATION
SYSTEMS MANAGEMENT DIVISION



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This report represents part of a continuing U. S. Army Transportation Research Command research program for the investigation of new concepts of high-speed reducers for use as main transmissions in helicopters. The basic objective of this program is to derive a speed reduction unit(s), with reduction ratios significantly higher than those currently used (40:1 and above), which would be more compatible with the high rotational speeds of aircraft turbine engines. With this objective in mind, the Cycloidal Cam Transmission investigation was undertaken.

The results of the instant study clearly indicate that the Cycloidal Cam Transmission concept is speed sensitive, is more complex, does not offer weight advantages, and is considerably less efficient (93.5 percent maximum at design speed and torque) than presently used planetary transmissions in turbine-powered helicopters.

No further development of this concept is anticipated by this command.

NOTE

On 1 March 1965, *after this report had been prepared*, the name of this command was changed from U.S. Army Transportation Research Command to:

U.S. ARMY AVIATION MATERIEL LABORATORIES

Task 1M121401A14414
Contract DA 44-177-TC-651
USAAML Technical Report 64-13
May 1965

CYCLOIDAL CAM TRANSMISSION

Phases II and III

Prepared by
Western Gear Corporation
Systems Management Division
Lynwood, California

for
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FORT EUSTIS, VIRGINIA

ABSTRACT

Previous work under this contract consisted of a parametric study of cycloidal cam transmissions and was reported in Cycloidal Cam Transmission, TREC Technical Report 61-38, ASTIA 270242.

This report covers a discussion of the basic theory of operation of the cycloidal cam reducer and the results of load testing on two 280-horsepower units. During testing of the first unit, excessive deflection in the reverter cage assembly resulted in improper load distribution and in the failure of bearings and other components. The failure was analyzed, and modifications were incorporated in the second unit. Failure was experienced in the early stages of testing on the second unit. This failure has been attributed to excessive bearing internal clearance and end play that caused skewing of the crank shaft bearings and severe abrasion of the shaft. Although the second unit was much more efficient than the original, power losses and excessive complexity remain as factors limiting the application of the cycloidal cam reducer for helicopter main transmissions.

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SUMMARY

The objective of this program was to develop an efficient, light weight, high-ratio, cycloidal cam-type speed reducer for helicopter drives. Phase I covered the feasibility study and state of the art survey. TREC Technical Report 61-38 was generated as a result of this study. During Phase II, various design studies of concepts proposed in Phase I were reviewed. One 840 horsepower unit was selected for engineering design and detailing. The selected design consisted of a three-module arrangement, each rated at 280 horsepower. As a result of this study, the following basic design parameters were established for the Phase II program:

Horsepower:	280	840
Speed:	6,000	6,000
Ratio:	19:1	30:1
Mast Loading:		
Vertical:	7,980	23,940
Horizontal:	1,400	4,200

The selected design for the 840 horsepower version consisted of three modules of the 280 horsepower configuration. After final design was complete, two 280 horsepower modules were detailed and manufactured for testing during Phase III of the program. This report covers a discussion of the basic theory of operation of the cycloidal cam reducer and the results of load testing.

The design and manufacturing phases of the cycloidal cam development were completed according to the conditions of the contract. During testing of the first unit, S/N 101, excessive deflection in the reverter cage assembly resulted in improper load distribution and premature failure of bearings and other components. The failure was analyzed, and modifications were incorporated in the second unit, S/N 102. Although the second unit was much more efficient than the original, premature failure was experienced in the early stages of testing. This failure has essentially been attributed to excessive bearing internal clearance and end play that caused skewing of the crankshaft bearings and severe abrasion of the shaft.

CONCLUSIONS

Based upon the results of the manufacture and testing to date, and with regard to the ultimate program objective of a light weight, high-ratio, compact, and highly efficient helicopter transmission, the following conclusions are reached:

1. The basic envelope size of the transmission is adequate for the 280-horsepower rating, provided the crank bearing problems can be resolved. The crank roller bearing design requires special consideration for two distinct cases:
 - a. The no-load or "free running" condition.
 - b. The normal-loading, full-speed condition.
2. The transmission efficiency (in excess of 90 percent) is above predicted values calculated during Phase I of the program.
3. The "pure" cycloidal type transmission is speed sensitive. The basic design uses a crankshaft; therefore, it is limited to normal crankshaft design and practice as applied in automotive engines.
4. The transmission will require further design effort to reduce the weight in order to compete with present-day helicopter geared transmissions.
5. The transmission is smooth and quiet in operation.
6. The transmission is complex in that there are a large number of rotating parts held to close tolerances.
7. The transmission is an in-line configuration readily mountable to a gas turbine accessory pad. Whenever an angular drive shaft reduction ratio is involved, the cycloidal cam transmission would require the addition of an angular gearbox.
8. Based upon the experience gained in this program, the transmission would be worthy of further development for concepts where the requirement for high input speeds (excess of 10,000 r. p. m.) is not required.

RECOMMENDATIONS

The tested efficiencies of the cycloidal cam speed reducer, its smoothness and quiet operation, dictate the desirability for continued development. Further development is recommended in the areas of the eccentric mechanism design, eccentric bearings and the fixed pin rollers. Effort should be directed toward advancing the state of the art for input speeds beyond 10,000 rpm.

DISCUSSION

Theory of Operation

Differential drives permitting high speed reduction do not operate in the same manner as the normal tangential load/radius proportioning used in conventional gearing. Essentially, the inclined plane or wedge principle is used, permitting large mechanical gains proportional to the wedging angle involved (Figure 1).

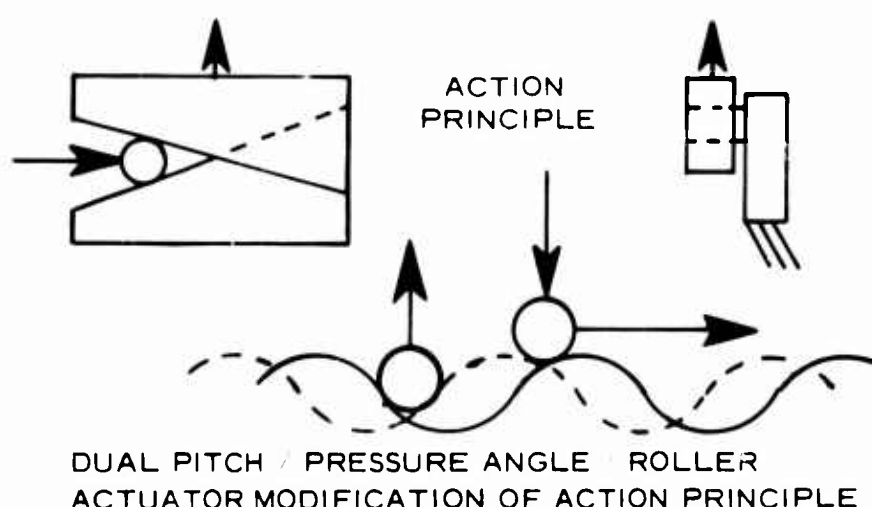


Figure 1. Basic Kinematics of Cam Action

Since the driving member in these arrangements carries the output force but must engage at input speed, gear tooth type drives have large power losses at high ratios. Teeth move into mesh at input velocity under full output forces, and efficiency goes down rapidly as the ratio goes up. By replacing the sliding contact of gear teeth with rolling contact then the efficiency will increase. This approach has culminated in the cycloidal type arrangement which permits rolling contact with all reaction members simultaneously, both fixed and moving.

The pure epicycloid/hypocycloid combination formerly used for gear teeth does not lend itself to continuous roller followers, and both epicycloid and hypocycloid curves have a sharp cusp which will give high instant acceleration to a follower; therefore, a modification of the follower path is advisable. The epitrochoid, which represents the path in space of a point at a lesser radius than

the radius generating an epicycloid and is generated by the same generating circle, rolling circle, and instant center as its companion epicycloid, is generally used. This curve eliminates the sharp cusp at zero pressure angle with the follower and greatly reduces peak angular acceleration of the followers or reaction rolling members. (Figure 2).

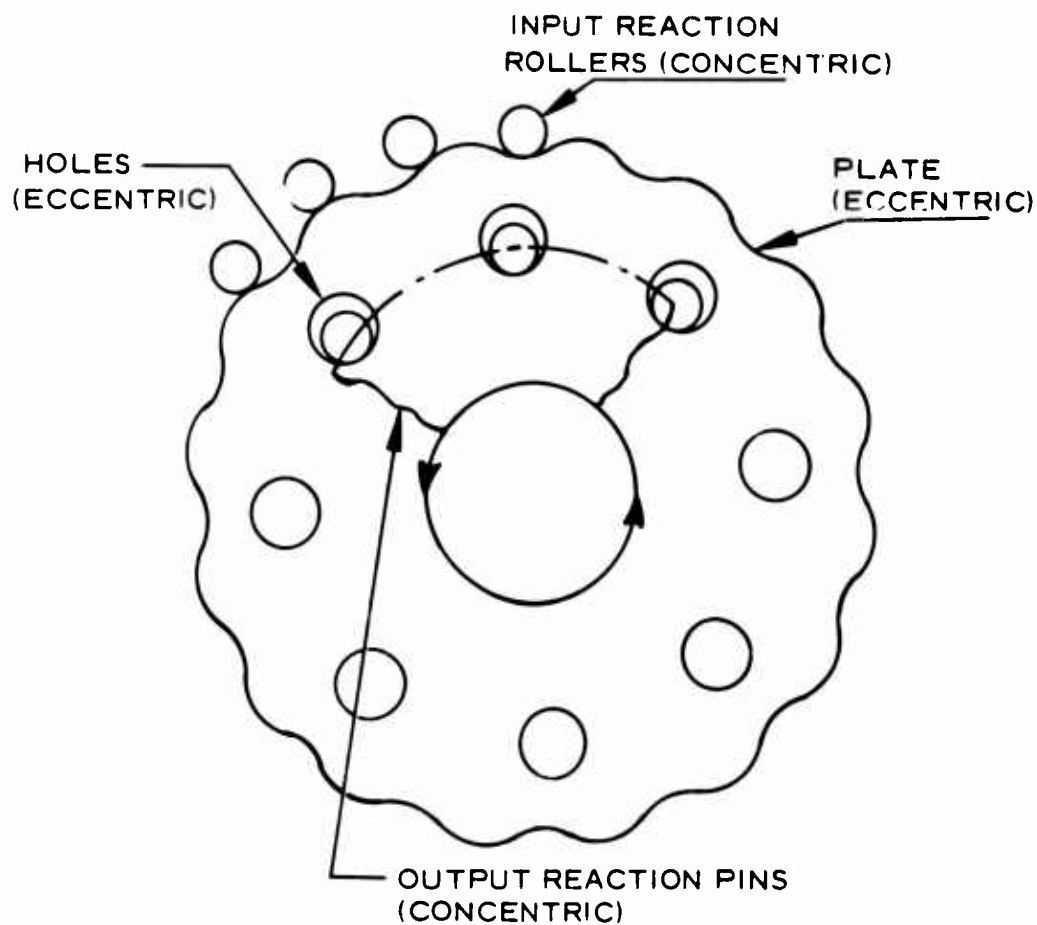


Figure 2. Cycloidal Cam Principle

The effective shape of the outside of the moving member therefore becomes a recurrent epitrochoid modified by the radius of the fixed reaction rollers. The displacement of the moving member is equal to the generating diameter of the epitrochoid used. The moving member then will reciprocate at input speed and will evolve at a speed determined by the number of modified epitrochoidal lobes on its periphery (N_1) and in a direction determined by the spacing (effective number N_2) of the fixed reaction members.

$$\text{Output Speed} = \text{Input Speed} \times \left(1 - \frac{N_2}{N_1}\right)$$

If the result is positive, input and output rotate in the same direction; if negative, in opposite directions. If the result is positive, a hypotrochoidal path for the follower centers is required.

The output motion is transmitted to the output shaft by a series of rollers whose circle of centers is concentric with the output shaft center line. These rollers are driven by holes in the moving member. The diameter of the holes must equal the roller diameter plus the full reciprocation distance of the input member. Since each roller can transmit tangential force effectively only through 90 degrees or less of input shaft rotation, six or more output rollers are necessary. If space permits, one output roller per input lobe will give good results; the output holes may then extend out into the lobes, and output tangential force occurs on a larger radius.

FIRST DESIGN CONCEPT (Figure 3, page 7 and Figure 4, page 8)

The crankshaft of the cycloidal cam transmission has two pairs of throws, each pair being 180 degrees out of phase with the other pair. Each throw is eccentric by 3/16 inches. The throws are in pairs in order to balance the centrifugal and bearing loads in the system. Each throw is attached to the geometric center of a cam plate through roller bearings. The outer periphery of the four cam plates have 19 lobes which are cycloidal in form. The lobes of the cam plates contact reaction pins at positions where the eccentric of the crank throw extends the cam plate to its maximum distance from the center of rotation of the crankshaft. Twelve reverter pins which are connected to the output shaft cage are positioned through holes in the cam plates. The ends of each reverter pin are connected to the output shaft cage member on either side of the cam plate assembly. The speed reducing effect is produced by rotating the crankshaft inside the four cam plates. The eccentrics on the crankshaft force the cam plates outwardly, causing the lobes to react against the reaction members in a wedging action.

Rollers on the 20 reaction members change the wedging effect to rolling, thus reducing friction. The reaction between the fixed members and the cam plate lobes turn the four cam plates on their

own centers by the amount of one lobe pitch per revolution of the crankshaft. Since there are 19 lobes on the cam plates and 20 reaction pins, the cam plates are advanced $1/19$ of a revolution for every revolution of the input shaft. Since the output shaft cage is driven through the reverter pins by the cam plates, the output shaft or mast therefore only rotates $1/19$ of a revolution for every revolution of the input shaft producing a 19:1 speed reduction.

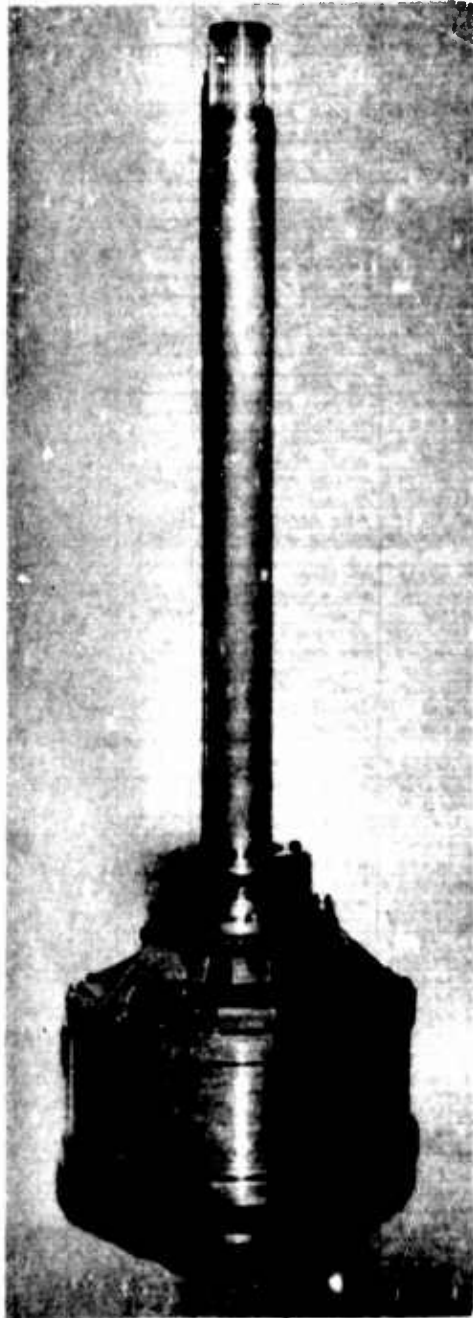
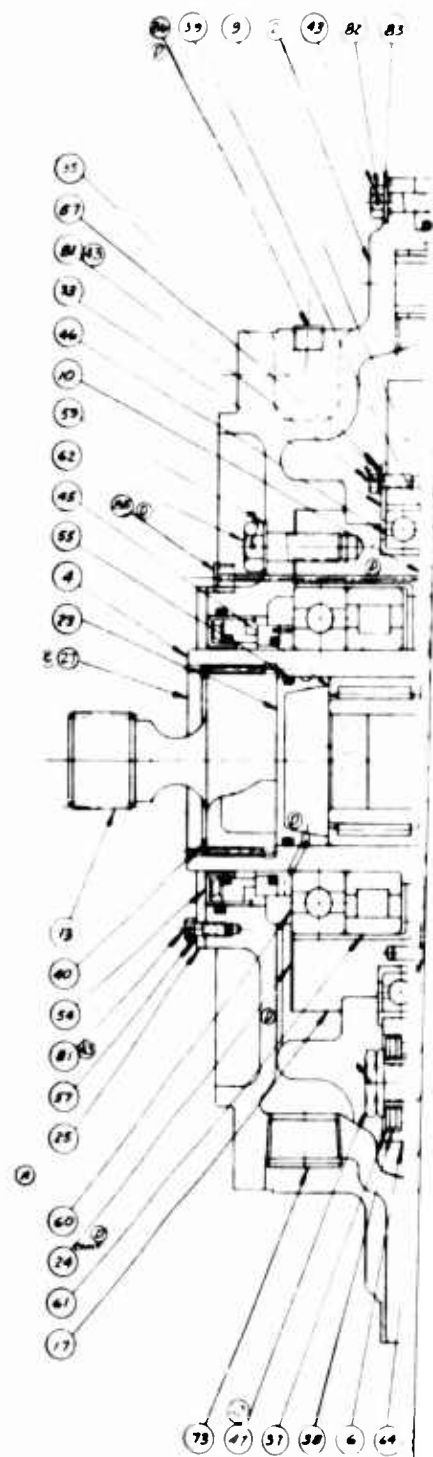
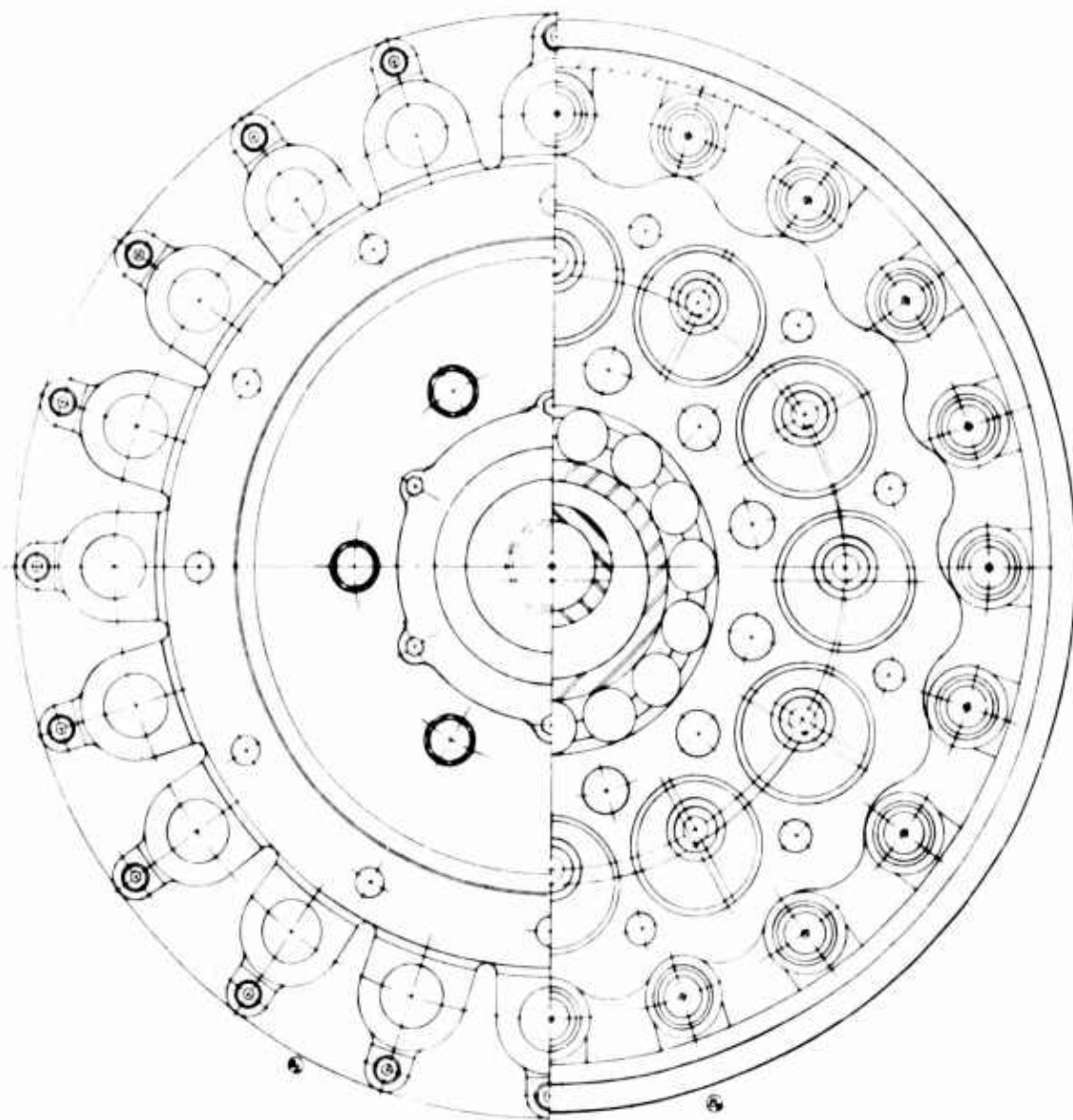


Figure 3.
Cycloidal Cam Transmission
S/N 101
Model 6144R149

The reverting system of shafts and holes translate the cam revolutions with uniform velocity. The four cam plates accomplish dynamic balance by having two cam plates traveling equal distances in opposite directions. The composite motion of all of the moving parts in the transmission is a rolling action creating a silent transmission.

The design objective of this unit was for a minimum bearing life of 1,000 hours. All bearings are standard catalog items with the exception of the reverter bearings and the crank bearings. The reverter bearings have a concentric O.D. but an eccentric I.D. The eccentricity is the amount of the crankshaft eccentricity, namely $3/16$ of an inch.

Standard oil seals are used. The input seal is a cartridge type carbon seal. The output seal is a lip type. Lubrication is supplied by an external force feed system. There are three inlet oil lines ($1/8$, $1/4$ and $1/2$ NPT) and one outlet line ($3/4$ NPT). Lubrication tests indicate that MIL-L-6086 lightweight aircraft oil and DTF lightweight turbine oil are satisfactory. A nominal oil flow of 3 gallons per minute with an oil pressure of 30 p.s.i. has proven marginal. A scavenge pump is used for the return oil. Also, an oil cooler is used. Three oil vents are located in the output housing. The transmission has been designed so that the lubrication system will function in any attitude.



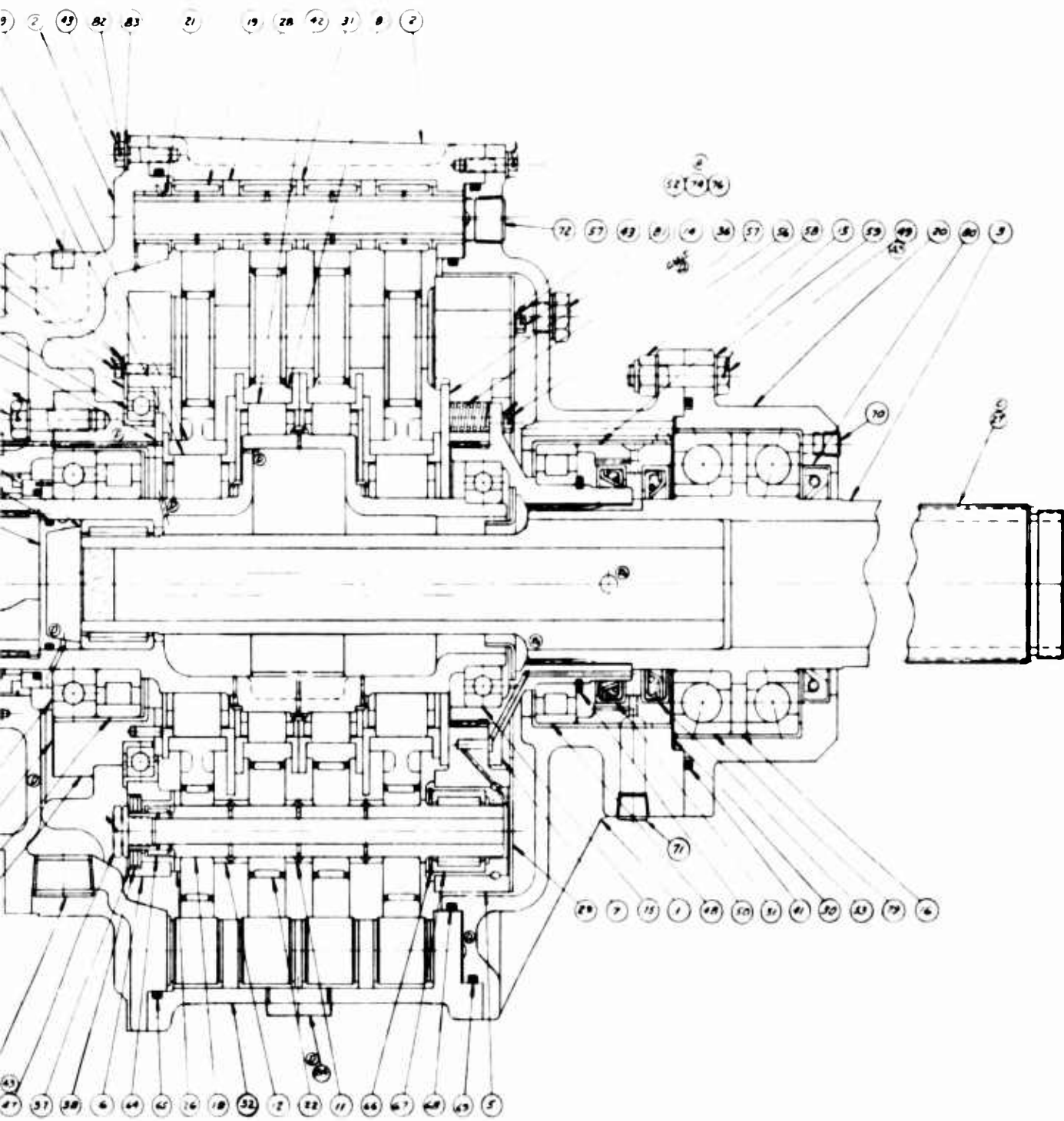


Figure 4

1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	25	26	27	28	29	30	31	32	33	34	35	36	37	38	39	40	41	42	43	44	45	46	47	48	49	50	51	52	53	54	55	56	57	58	59	60	61	62	63	64	65	66	67	68	69	70	71	72	73	74	75	76	77	78	79	80	81	82	83
---	---	---	---	---	---	---	---	---	----	----	----	----	----	----	----	----	----	----	----	----	----	----	----	----	----	----	----	----	----	----	----	----	----	----	----	----	----	----	----	----	----	----	----	----	----	----	----	----	----	----	----	----	----	----	----	----	----	----	----	----	----	----	----	----	----	----	----	----	----	----	----	----	----	----	----	----	----	----	----	----	----	----

B

86	1	8 NPT	PLUG PIPE HEX SOCK	COMM L
88	1	8 NPT	PLUG PIPE HEX SOCK	COMM L
89	1	8 NPT	PLUG PIPE HEX SOCK	COMM L
93	40	AN260C61	WASHER PLAIN	
94	40	AN261A	BOLT MACHINE	
91	12	AN261A	BOLT MACHINE	
90	1	55011	OIL SEAL	NAT'L VAL
79	24	725B16	BALL BEARING DUP CLASS 1	JFF
78				
77				
76	1	AN261A	WASHER LOCK	
75	1	CLS 2 1/2	BALL BEARING CLASS 1	ABEND HEAT
74	1	AN260C61	WASHER PLAIN	
73	1	1/2 NPT	PLUG PIPE HEX SOCK	COMM L
72	1	1/2 NPT	PLUG PIPE HEX SOCK	COMM L
71	1	1/2 NPT	PLUG PIPE HEX SOCK	COMM L
70	1	1/2 NPT	PLUG PIPE HEX SOCK	COMM L
69	1	2-27N1007	O RING	MARKER
68	1	2-27N1007	O RING	MARKER
67	12	NJ 12216	NEEDLE BEARING CLASS 1	TORRINGTON
66	12	SC 137	RETAINING RING	TORRINGTON
65	1	2280 NEW7	O RING	PACKER
64	12	3 108	NEEDLE BEARING CLASS 1	TORRINGTON
63				
62	6	AN261A	BOLT MACHINE	
61	1	CLS 2 1/2	ROLLER BEARING CLASS 1	NOEMA HEAT
60	1	CLS 2 1/2	BALL BEARING CLASS 1	NOEMA HEAT
59	20	AN260C61	WASHER PLAIN	
58	10	AN260C61	NUT SELF LOCKING	
57	10	AN260C61	WASHER PLAIN	
56	10	AN261A	BOLT MACHINE	
55	1	8 1/2	O RING	MINI 16
54	1	34 4475	OIL SEAL	NATIONAL
53	1	55011	OIL SEAL	NATIONAL
52	1	40078426	SUB ASY BEARING AND	
51	1	55562	OIL SEAL	NATIONAL
50	1	5500 334	RETAINING RING	TRUMEC
49	1	AN261A	BOLT MACHINE	
48	1	8 1/2 TYPE	BALL BEARING CLASS 1	MFC
47	12	MS205 03	PLUG STRAIGHT	
46	1	ED 55 CP	BALL BEARING CLASS 1	KAYDON
45	1	NJ26152	NEEDLE BEARING CLASS 1	TORRINGTON
44				
43	1	AN260C61	LOCKWEE	
42	2	6140C147	BRS ROLLER INNER CLASS 3	ROLLWAY
41	1	8258	O RING	MINI 16
40	1	5000-281	RETAINING RING	IN TRUCK
39	2	6140C148	BRS ROLLER OUTER CLASS 3	ROLLWAY
38	12	N741018	THRUST BEARING NEEDLE	TORRINGTON
37	12	161018	THRUST RACE	TORRINGTON
36	1	9 0808-2	SPRING COIL	DAILY MACH
35	1	AN261A	WASHER LOCK	
34				
33	1	6140C157	RETAINING RING	
32	1	6140C175	PLATE NAME	
31	20	6140C171	SPACER ROLLER	
30	1	6140C170	SHIM LOWER BRS	
29	1	6140C169	COVER REV. BORES	
28	1	6140C168	SUPPORT FIXED PIN	
27	1	6140C167	BASE PROCEURER	
26	12	6140C112	WASHER, THRUST	
25	1	6140C168	COVER OIL SEAL	
24	1	6140C167	SHIM, LOWER	NOTE: NOT REQUIRED
23	1	6140C166	SEAL CRANK SHAFT	
22	18	6140C165	BEARING, INVERTER	CLASS 3
21	20	6140C160	SHAFT FIXED PIN	
20	1	6140C159	CAP UPPER-INCL 6140C159, 1058 087	
19	1	6140C157	BASE ROLLER, FIXED PIN-INCL 6140C157, 1058 087	
18	12	6140C151	SHAFT, REVERTER	
17	1	6140C149	ADAPTER BEARING-INCL 6140C149, 1058 087	
16	1	6140C163	SPACER BEARING	
15	1	6140C162	ADAPTER, OIL SEAL	
14	1	6140C161	PLATE, PRESSURE	
13	1	6140C160	COUPLING, INPUT	
12	1	6140C159	OIL, EXHAUST OUTER	
11	1	6140C158	OIL, EXHAUST CRANK	
10	1	6140C150	SPACER CAN	
9	1	6140C118	PLATE, CAN OUTER-INCL 6140C118, 1058 087	
8	1	6140C116	PLATE, CAN INNER-INCL 6140C116, 1058 087	
7	1	6140C115	SPACER, OUTPUT-INCL 6140C115, 1058 087	
6	1	6140C114	FAST, OUTPUT RING-INCL 6140C114, 1058 087	
5	1	6140C113	FAST, OUTPUT CASE-INCL 6140C113, 1058 087	
4	1	6140C112	ADAPTER CRANK SHAFT-INCL 6140C112, 1058 087	
3	1	6140C111	ADAPTER, INCL 6140C111, 1058 087	
2	1	6140C110	ADAPTER, INCL 6140C110, 1058 087	
1	1	6140C108	ADAPTER, INCL 6140C108, 1058 087	

QTY	ITEM	DESCRIPTION	UNIT
LIST OF MATERIAL			

W

WESTERN GEAR

Lynch, California

ASSY. CYCLOIDAL CAM
TENS. CRANK
1058 087

FILE NO. 1058 087
REV. 1058 087

Figure 4

Horizontal lubrication testing is shown in Figure 5, page 9.

Assembly procedure (Figure 6, page 9) is based on starting from the input side of the unit and building it up cam plate by cam plate. The output mast is subassembled with its bearings and inserted into the transmission as the last item.

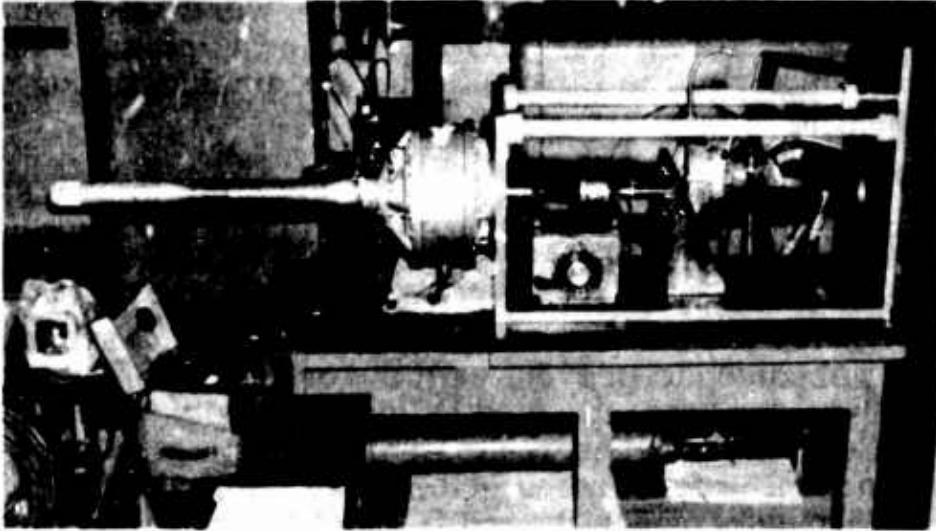


Figure 5. Run-in test stand with transmission S/N 101 mounted for horizontal operation.

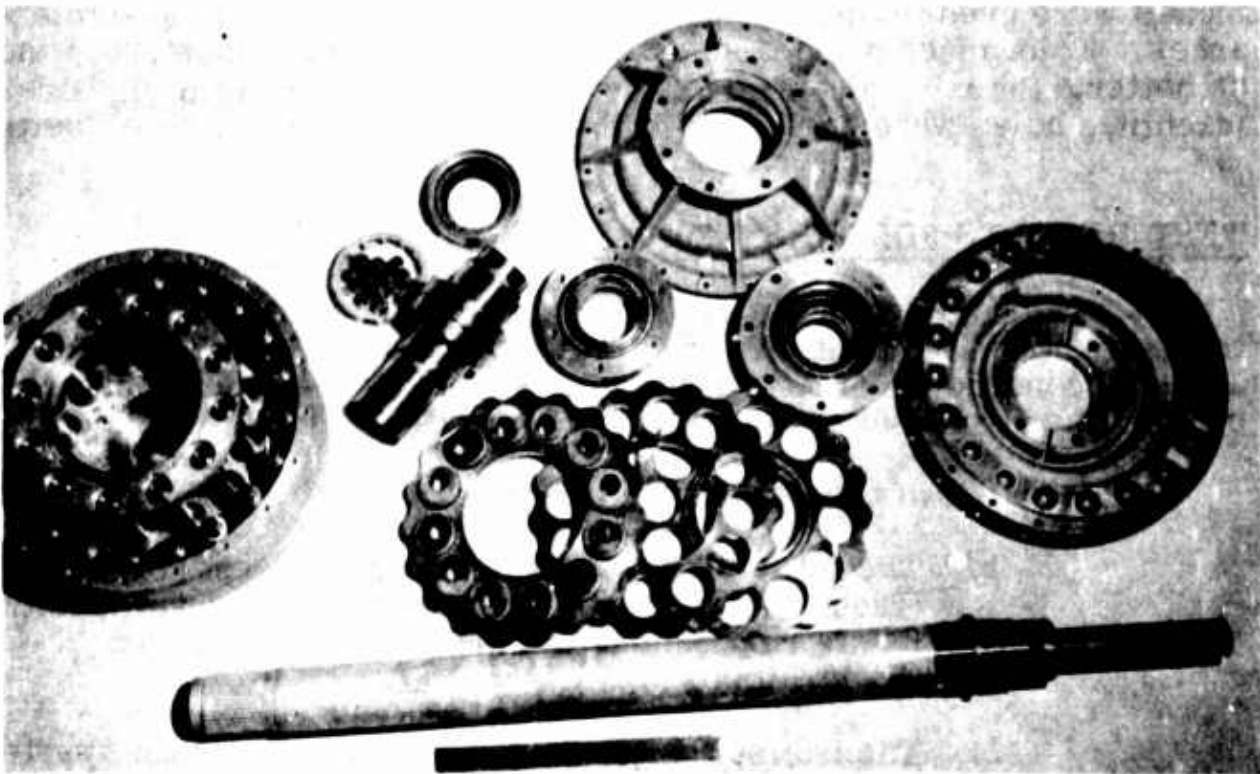


Figure 6. Partial disassembly of transmission S/N 101.

Special manufacturing techniques were developed for the crankshaft, the cam plates, and the reverter system. The balance of the parts followed standard aircraft methods of manufacturing. The crankshaft is a high strength SAE 4820 alloy steel casting. This steel was chosen for its toughness and impact properties. Experimental castings were made in order to secure a high quality casting. For the machining and grinding operations, soft plugs were inserted at each end. Centers were very accurately located in the plugs to maintain the spacing accuracy of the throws. The last manufacturing operation was balancing the crankshaft.

The cam plates, are of SAE 3310 alloy steel forgings. This steel was chosen for its ability to provide a tough and strong core and a hard wear-resistant, case-hardened surface.

After rough machining, the cam plates were case hardened. Quench dies were used to prevent cam distortion. Contour grinding of the cam lobes was accomplished by tooling designed for adaptation to a Detroit Gear Grinding machine. Two contour grinding operations were involved: one soft grinding before treatment and the finish grind after heat treatment. The I. D. and O. D. of the cam plates were checked using a surface plate and a height gage. The contour of each cam plate lobe was optically checked using an X-Y coordinate overlay at 10:1 magnification. The cam lobe profile accuracy was found to be within .001 inch of the base shape.

The functioning of the reverter system is very dependent on spacing the reverter holes accurately. Optical manufacturing techniques were used to locate, index, and grind the bores to close tolerances. Tolerances on hole size were held to plus or minus .0005 inch or better. By using an optical dividing head as a part of the jig bore machine, holes were positioned within .0002 inch of their true position.

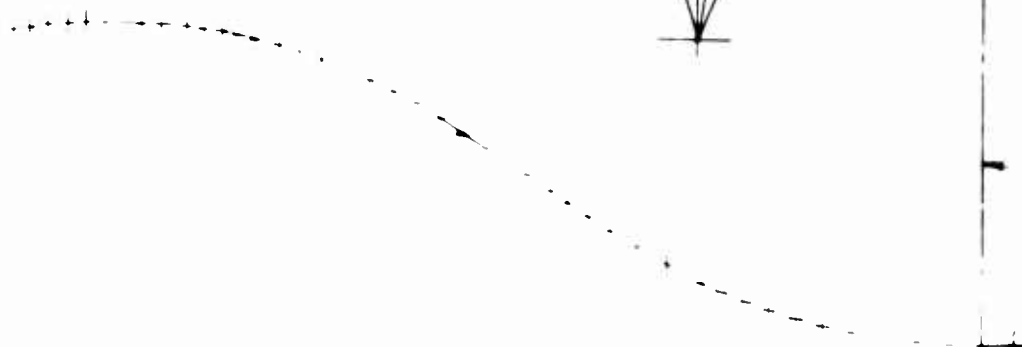
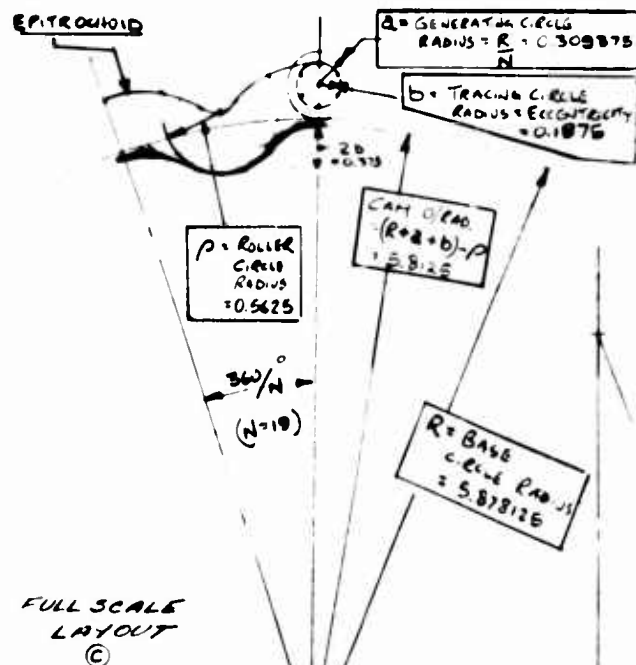
TEST RESULTS FOR FIRST TRANSMISSION

Upon completion of the manufacturing and assembly of the two transmissions, the first unit, S/N 101, was tested under "no-load" and load conditions. The second unit, S/N 102, was held in abeyance pending the results of testing the first unit.

The no-load test procedure for S/N 101 was as follows:

1. The transmission was mounted on a test fixture (Figure 7, page 11) in the vertical mounting position. It was also mounted in the horizontal position (Figure 5, page 9).
2. The transmission has an external lubricating system. Aircraft MIL-L-6086 lightweight oil was used.

COORDINATE NO.	X	Y
1	01120	5.43828
2	06341	5.44060
3	09419	5.44436
4	12395	5.44942
5	15239	5.45557
6	17938	5.46262
7	20472	5.47036
8	22842	5.47857
9	25045	5.48707
10	27088	5.49571
11	30712	5.51287
12	33192	5.52928
13	36416	5.54455
14	38665	5.55853
15	40616	5.57123
16	42333	5.58275
17	44577	5.59815
18	46544	5.61172
19	48334	5.62391
20	50574	5.63869
21	52757	5.65229
22	54970	5.66510
23	57274	5.67728
24	59707	5.68886
25	62297	5.69977
26	64351	5.70742
27	66501	5.71454
28	68749	5.72103
29	70990	5.72681
30	72700	5.73022
31	74348	5.73325
32	76031	5.73586
33	77745	5.73805
34	80372	5.74048
35	82153	5.74152
36	83956	5.74205
37	85776	5.74208
38	87609	5.74158
39	89451	5.74056
40	91298	5.73902
41	93146	5.73634
42	94623	5.73489
43	95648	5.73322
44	98173	5.72923
45	100297	5.72598



NOTE: 1

1. THE ABOVE POINTS ARE BASED ON THE COMPUTER TAB RUN (UNIVAC 120). THE TAB RUN HAS A GREAT MANY MORE POINTS, I.E. FOR EACH ANGULAR INCREMENT OF 0.01°

2. THE BASIC EQUATIONS FOR THE CURVE IS AS FOLLOWS:

$$X = (R+a) \sin \phi - b \sin[(N+1)\phi] + p \sin\left[\left(\tan^{-1} \frac{b \sin(N\phi)}{a - b \cos(N\phi)}\right) - \phi\right]$$

$$Y = (R+a) \cos \phi - b \cos[(N+1)\phi] - p \cos\left[\left(\tan^{-1} \frac{b \sin(N\phi)}{a - b \cos(N\phi)}\right) - \phi\right]$$

WHERE

$R = 5.878125$ - BASE CIRCLE

$a = 0.309375$ - GENERATING CIRCLE

3. $b = 0.18750$ - ECCENTRICITY (TRACING CIRCLE). THE PATH OF A POINT ON THE TRACING CIRCLE GENERATES THE EPITROCHOID.

$N = 19$ - NUMBER OF LOBES

4. $p = 0.56250$ - ROLLER CIRCLE. THIS RADIUS (USED AS A CUTTER RADIUS) & STRUCK ON EPITROCHOID, WHEN $X = 0.000000$ GENERATES THE CAM SURFACE.

$Y = 5.437500$ - ROOT RADIUS

$D = 10.8750$ - ROOT DIAMETER

5. AN OPTICAL OVERLAY IS REQUIRED FOR CHECKING THE CAM CONTOURS (AFTER GRINDING) FOR GIMMIES AND GIMMIE PARTS.

VENDOR: EXCELLO CORPORATION
OPTICAL GAGING PRODUCTS
12200 S. BELLFLOWER BLVD.
DOWNEY, CALIFORNIA

A

3. The three inlet oil lines were connected to the lubrication system as well as the drain line. Provisions were made to measure the oil pressures and the return oil flow. It was found to be necessary to use a scavenge pump.
4. Room temperature oil was circulated through the transmission (before rotating the unit) in order to observe the lubricating system.
5. A hydraulic power supply was used to drive the transmission.
6. Typical data recorded were input speed, output speed, oil in, oil out, case, bearings, ambient air temperatures, oil pressures, oil flow, hydraulic pressure, and sound level readings.
7. The transmission was checked for binding by rotating the input shaft by hand.
8. The friction torque was measured as well as backlash.
9. The transmission was slowly brought up to speed in increments:

<u>Input Speed, R. P. M.</u>	<u>Operating Time, Minutes</u>
600	10 Minimum
800	10 Minimum
1000	10 Minimum
1500	10 Minimum
2000	10 Minimum
2500	10 Minimum
3000	10 Minimum
4000	10 Minimum
5000	10 Minimum
6000	10 Minimum

The run-in or "no-load" testing of the first unit revealed a smooth, quiet transmission. Stabilized temperature runs were made for both the vertical and horizontal positions of mounting (Figures 8 and 9, page 13). The two positions of the transmission had little effect on the temperature or sound readings. The heat losses calculated were approximately 5 per cent. Lubrication testing consisted of varying the oil flow and pressure. It was optimized at about 3 g.p.m. and 30 p.s.i.

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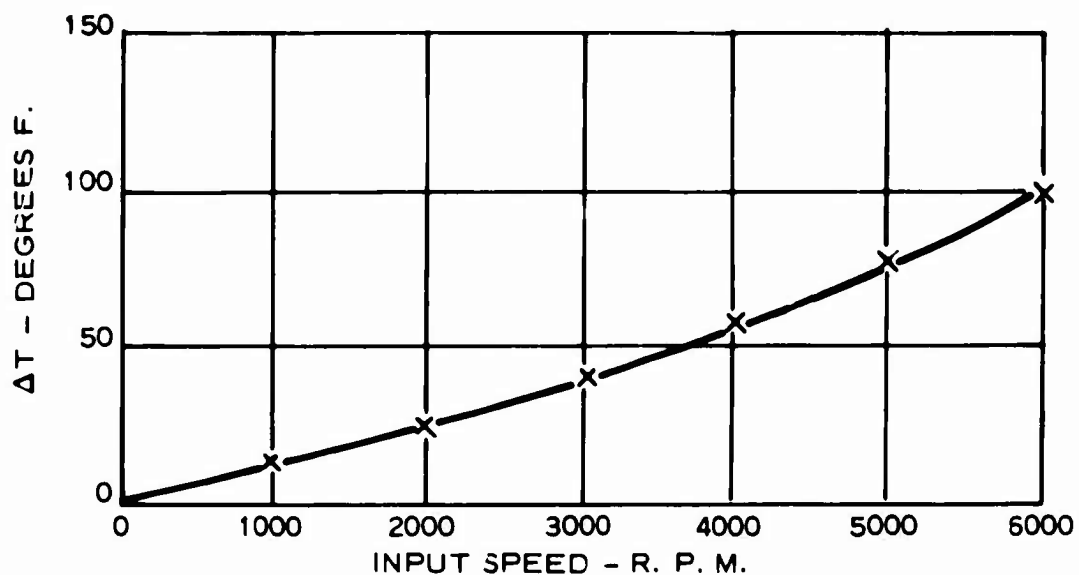


Figure 8. Stabilized temperature runs for horizontal operation with no external load applied to the output shaft. Lubricant: MIL-L-6086 Light, Flow: 2.0 gpm.

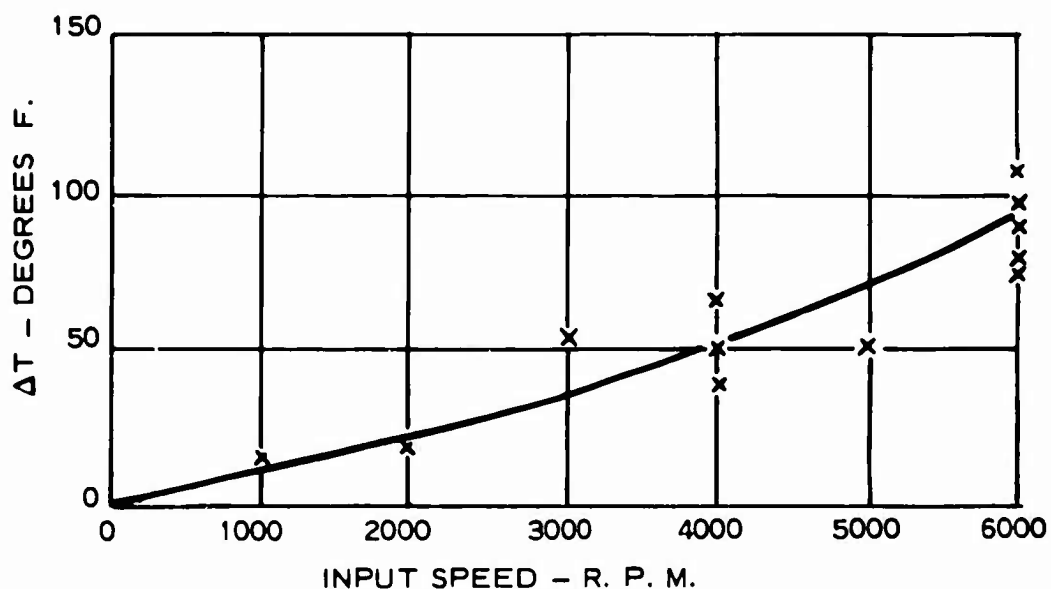


Figure 9. Stabilized temperature runs for vertical operation with no external load applied to the output shaft. Lubricant: MIL-L-6086 Light, Flow: 2.0 gpm

Four teardown inspections were accomplished during the no-load testing phase. The first inspection revealed:

1. The output torque cage was rubbing on the housing.
2. Fret corrosion had set up between the reverter shafts and the eccentric bearings.
3. The mast plug had no return oil hole.

Corrective Action:

1. The housings were reworked for clearances. Annular oil groove for the fixed pins was installed. Leads to aid in disassembly were installed. Two more oil vents in output housing were added. Slot for return oil in input housing was added.
2. Return oil holes in mast plug were added.
3. The inner cam plate assembly was equalized with the outer cam plate assembly. One cam plate, plus one crank bearing, plus two side plates, plus twelve eccentric bearings equal 14.5 pounds.

Miscellaneous:

1. The transmission less the mast weighed 170 pounds.
2. The mast assembly including the thrust bearings weighed 38 pounds.
3. The initial static torque was 60 inch-pounds.
4. After reassembly, the static torque was 36 inch-pounds.

The second inspection revealed:

1. Cam plate side plates were discolored from heat.
2. Fret corrosion continued to show up between the reverter shafts and the eccentric bearings.

Corrective Action:

1. Added jack screw holes to aid the disassembly of the housings.
2. Added additional oil holes in the crankshaft for lubricating the crank bearings.

3. Removed the pressure plate springs depending on shims for location of the assembly.
4. Added a crown to the fixed pin rollers.
5. Enlarged oil holes in output seal adapter.
6. Added oil slots in fix pin roller spacers.
7. Enlarged oil holes in fix pin shafts.

The third inspection revealed:

No significant items were noted. The static torque after reassembly was 12 to 18 inch-pounds.

The fourth inspection revealed:

1. Two nuts backed out from reverter shafts.
2. Pins holding side plates on cam plates came loose.
3. Slight signs of scoring on crankshaft from crank bearings.

Corrective Action:

1. Squares added to reverter shafts for wrench in tightening reverter nuts on assembly. Also used Loctite and peening to keep nuts from backing out.
2. The reverter shafts were electrolyzed in order to reduce fret corrosion.
3. The fix pin shafts were oriented by keying in place so that the shafts would not rotate.
4. Additional housing clearance was added for lock-wiring the reverter cage system.
5. The straight pins were replaced with step pins in order to prevent the cam plate side plates from rotating relative to each other.
6. The output splines were heat treated for additional strength.

Using a B-L-H torque transducer, the "run-in" test stand was calibrated. The calibration consisted of measuring the input torque to the transmission under dynamic conditions relative to the hydraulic power supply (Figure 10, page 16).

Thermocouples

**Test Transmission
S/N 101**

Torque Transducer

Input Speed Pick-up

Step-up Gearbox

Hydraulic Motor

Figure 10. Run-in test for vertical operation.

A. Based on the calibration method, the power required to run at 6000 r.p.m. no load was:

$$\text{Horsepower} = \frac{135 \text{ inch-pounds} \times 6000 \text{ r.p.m.}}{63,000} = 12.9 \text{ HP}$$

B. Based on the maximum temperature rise between the inlet and outlet oil at 6,000 r.p.m. and at no load, the power loss was:

Heat loss = flow x oil weight x specific heat x temperature rise

Oil flow = 2 gpm

Oil weight = 7 pounds/gallon

Specific heat = .5 Btu/lb/°F

Temperature rise, $\Delta T = 62^\circ\text{F}$

Heat loss = 2 gpm x 7#/gal x .5 x 62 = 435 Btu/min

1 HP = 42.4 Btu/minute

$$\text{Horsepower} = \frac{435}{42.4} = 10.3$$

Comparing the two methods of calculating the transmission losses, it will be noted there is a pretty fair agreement. Method A is considered the more accurate since it includes all losses.

The transmission backlash was measured by locking the output shaft and measuring the angular displacement of the input shaft. At the fix pin radius of 5-3/4 inches the backlash was .016 inches.

The load testing procedure for the first unit, S/N 101, was as follows:

1. The transmission was mounted on the test fixture in the horizontal position with input power drive and output loading equipment (Figure 11, page 18).
2. The external lubricating systems with their pumps, filters, cooler, sumps and their associated plumbing, were connected. The oil reservoir

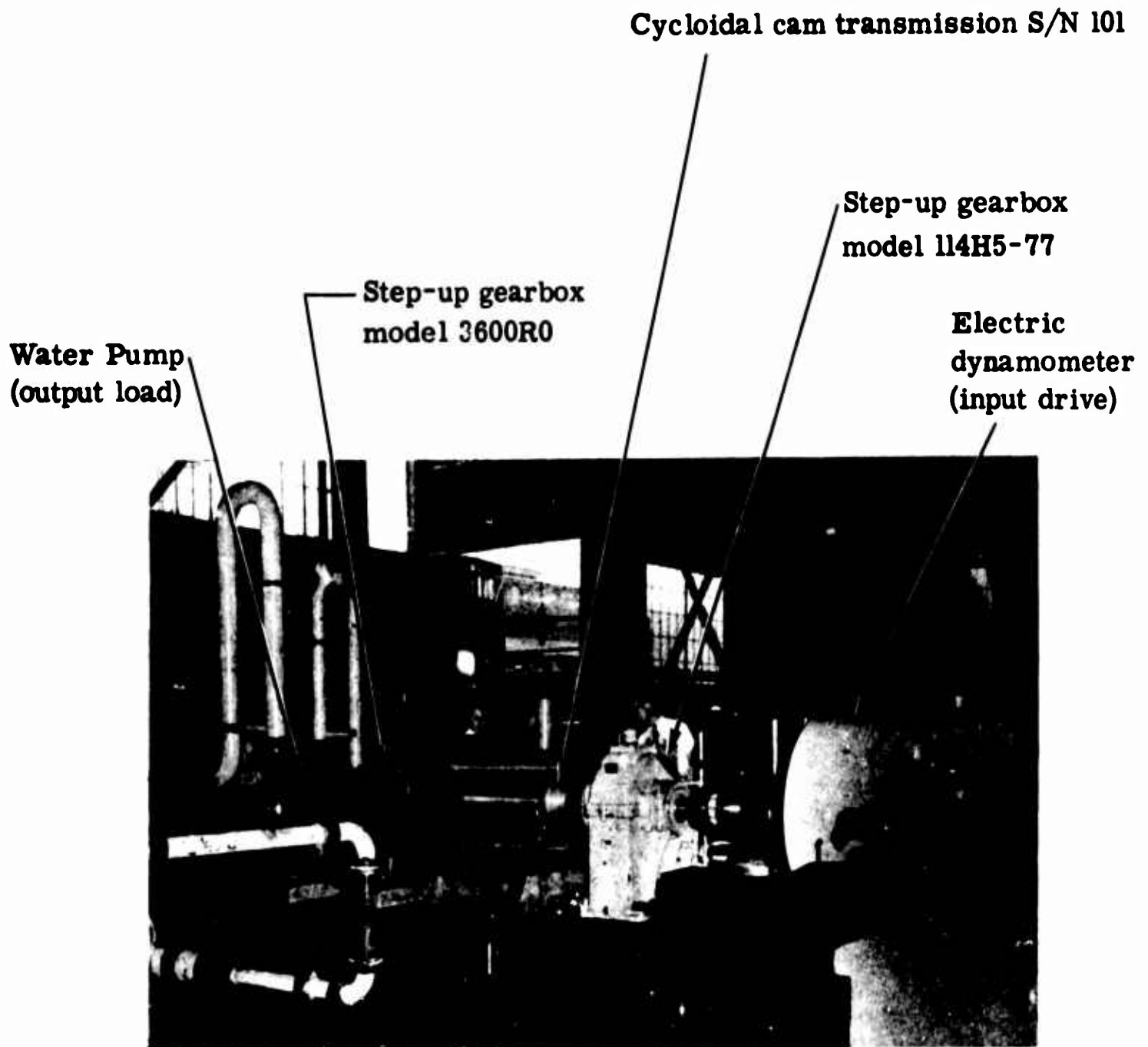


Figure 11. Load test stand for transmission S/N 101.

for the transmission was first filled with industrial spindle machine oil but later changed to aircraft MIL-L-6086 lightweight oil.

3. The three inlet oil lines and the center case drain line had provisions for varying the oil pressure, measuring the flow, and inspecting the return oil.
4. Room temperature oil was circulated through the transmission for one hour before rotating the unit.
5. The transmission was checked for binding by rotating the input shaft by hand (before the output coupling was connected to the load pump system).
6. Typical data recorded were time, dynamometer load, input speed, oil pressure, discharge pressure, pump suction pressure, oil flow, water flow, and transmission temperatures.
7. The transmission was slowly brought up to one half design speed with the load water pump disengaged. The run was continued until the transmission temperatures had stabilized.
8. With the load pump clutch engaged, the transmission was brought up to speed and load in the following increments:

<u>Input Speed</u> <u>R P M</u>	<u>Dynamometer</u> <u>Horsepower</u>	<u>Operating Time,</u> <u>Minutes</u>
1000	6	15 Minimum
1500	20	15 Minimum
2000	26	15 Minimum
3000	70	15 Minimum
4000	150	15 Minimum
4500	200	15 Minimum
5000	263	15 Minimum
5100	280	15 Minimum
5700	386	10 Minimum

9. A total of 50 hours of load testing was to have been accumulated but this was not accomplished. The testing was stopped after 20 hours because of chips detected in the return oil of the transmission.

Development testing under loading conditions revealed that the transmission was capable of transmitting the design horsepower of 280. The composite performance of the various load runs are shown on Figure 12, page 20. The mechanical efficiency during the load runs

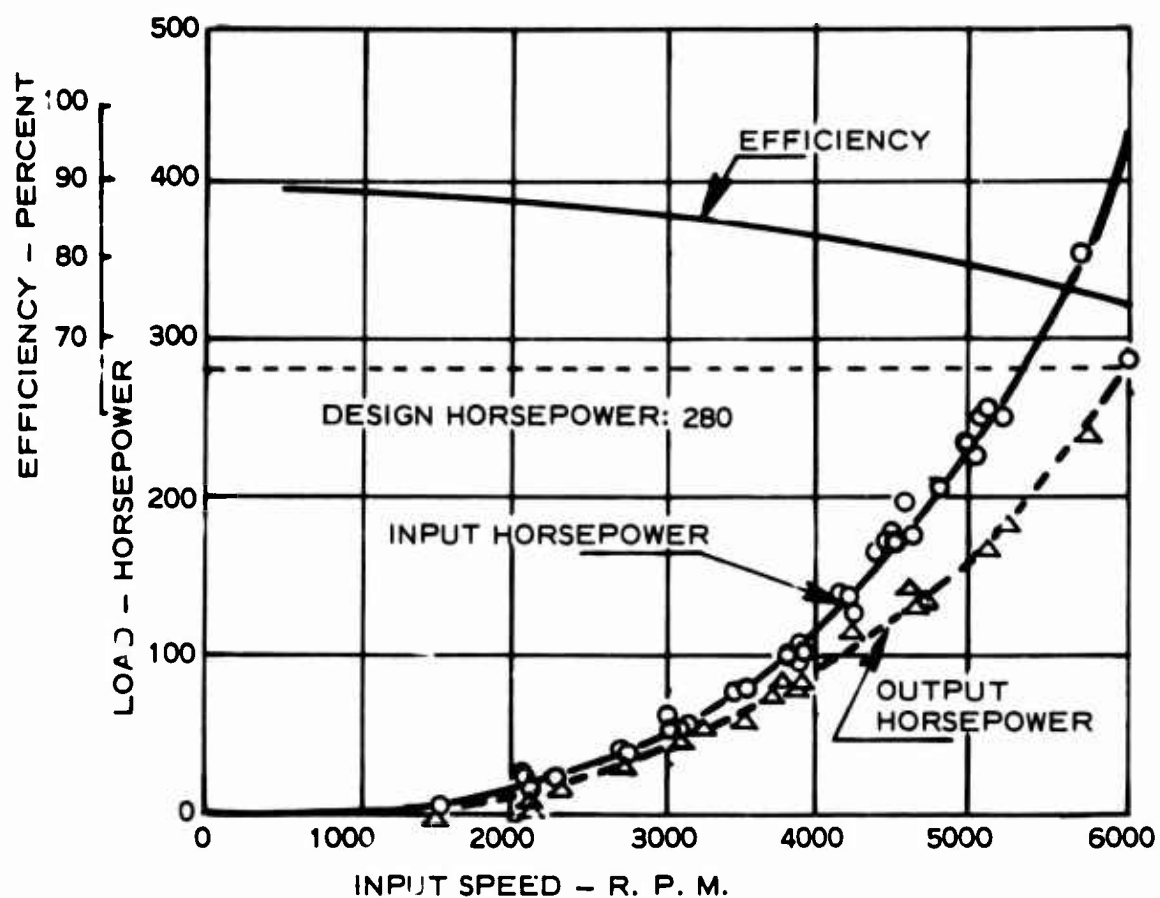


Figure 12. Composite Performance for Transmission S/N 101.

varied from 88 percent to 73 percent depending on speed and power settings. As the load spectrum of testing progressed, it could be determined that the efficiency could be changed by altering the quantity of oil flowing through the unit.

A sample set of horsepower loss calculations during the load runs was as follows:

Input speed = 5,190 RPM

Dynamometer HP = 280 HP

Temperature differential of oil flow = $t_{in}^{\circ F} - t_{out}^{\circ F} \Delta T$
= 115°

Surface case temperature differential ΔT = 133°F

Flow through transmission = 3.5 g. p. m.

HP = K Q ΔT (see page 17)

where $K = \frac{.5 \times 7}{42.4} = 1/12 = .0835$

Thermal oil flow loss = $.0835 \times Q \times \Delta T = .0931 \times 3.5$
 $\times 115 = 33.5$

Thermal radiation loss = $C \times \text{Surface area} \times T \times K$
 $= 3.931 \times 10^{-4} \times 6 \text{ square feet} \times 133 \times 2 = .627 \text{ HP}$

Total thermal loss = $33.5 + .6 = 34.1 \text{ HP}$

Thermal efficiency = $\frac{280 + 34.1}{280} = \frac{246}{280} = 88\%$

To better illustrate the heat balance approach, a graph (Figure 13) was plotted showing horsepower vs ΔT with efficiencies superimposed.

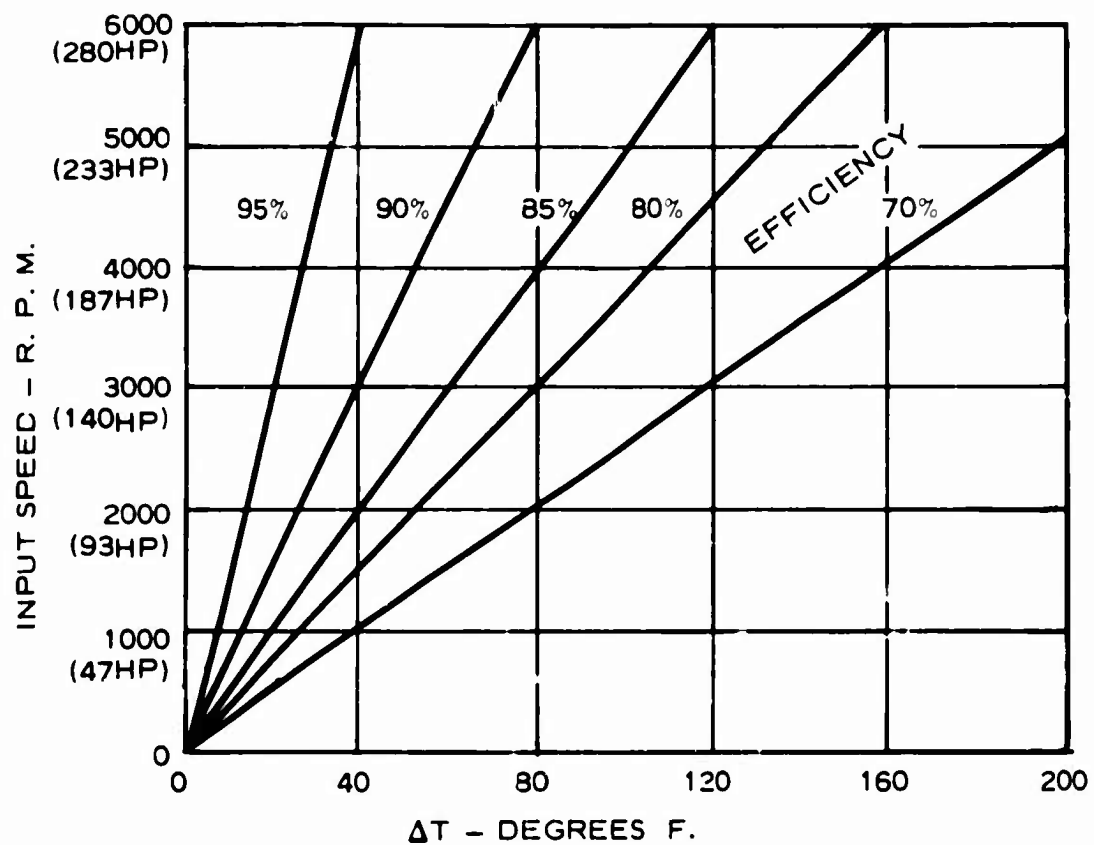


Figure 13. Theoretical Heat Balance Graph.
 This graph is based on the assumption that 80% of the heat loss is taken in the lubricating oil. The flow conditions were 3 gpm.

A stabilized temperature load graph for two types of transmission lubricants is shown in Figure 14. The MIL-L-6086 light oil is considered the better oil for the application involved. Therefore, testing with Chevron 1CH 11 was discontinued.

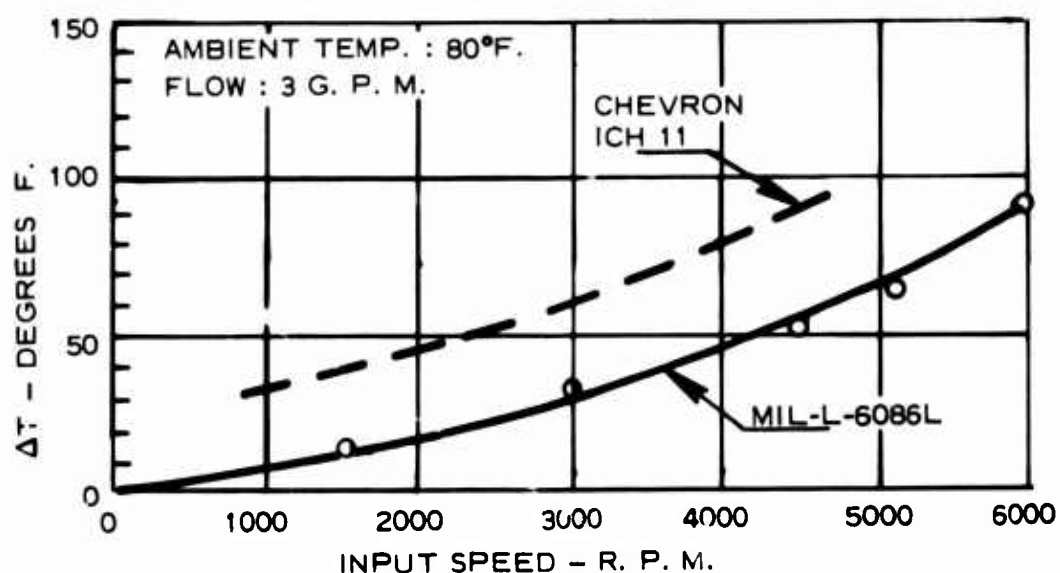


Figure 14. Stabilized temperature load runs for two types of transmission lubricants.

Mechanical loss in the test stand system

Loss between the dynamometer and the transmission

Loss = 31.5 HP

This value is from the calibration curve, Figure 15.

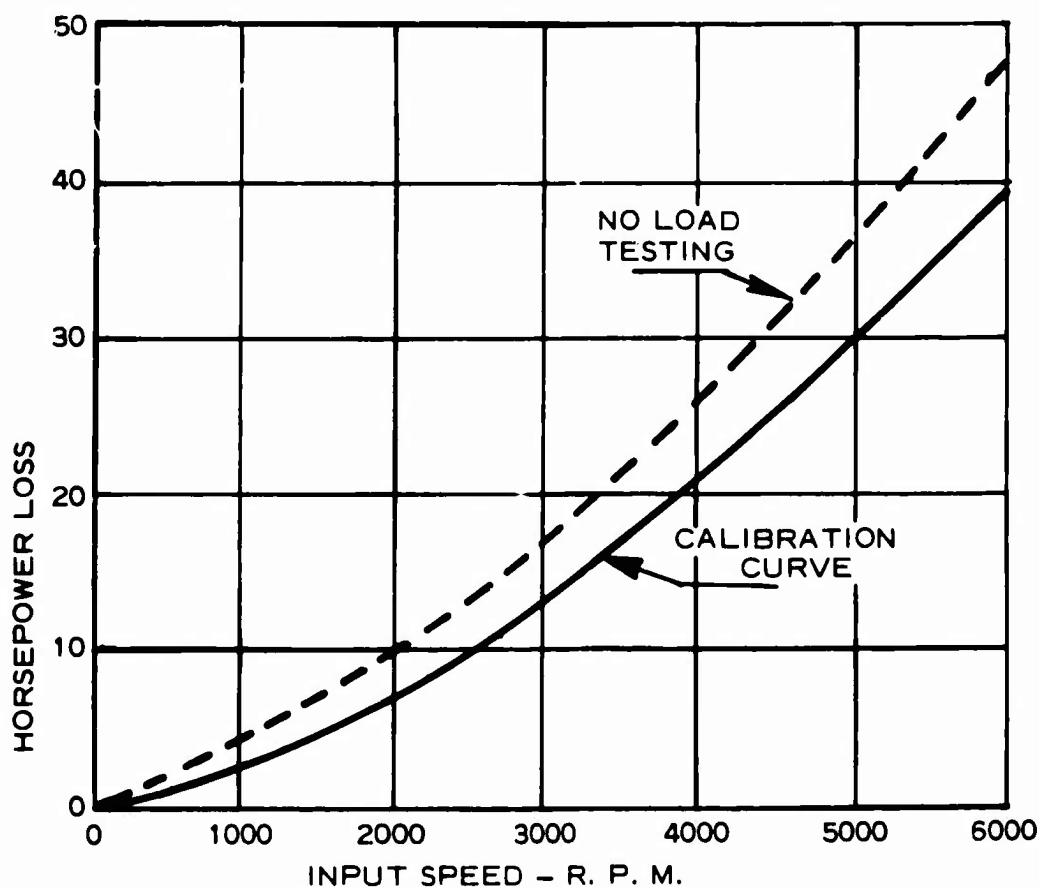


Figure 15. Calibration curve for step up gear box, Model 114H577, on input drive side. The graph is based on tests for the rated load condition of 1800 HP at 7200 RPM. The overall efficiency is approximately 90% which includes coupling loss.

Loss between the water pump and the transmission

Loss = 15 HP

This value is from the calibration curve, Figure 16.

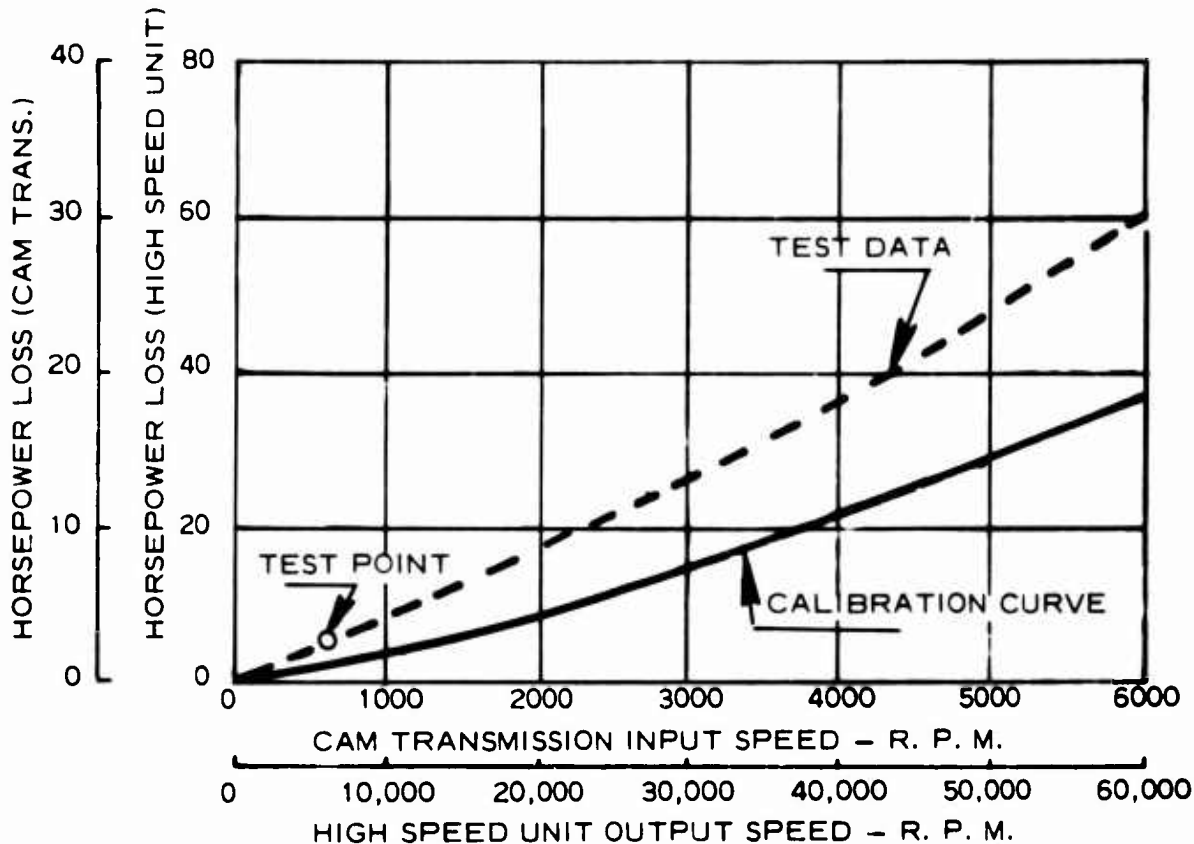


Figure 16. Calibration curve for high speed unit, Model 3600R0 on output load side. The graph is based on tests for the rated load condition of 600 HP at 60,000 RPM. The overall efficiency is approximately 97% which includes coupling loss.

Brake horsepower at the coupling

Output load pump = 166 HP

In order to arrive at this load, the following parameters were calculated or read from gauges.

Water pump speed = 3,640 RPM

Feet of water = 67.7 feet

Suction pressure = 29.3 p. s. i.

Discharge pressure = 404 p. s. i.

Water head = 935 feet

Total discharge head = 867.3 feet

Water flow = 95 g.p.m.

Pump efficiency = 12.5 percent

Therefore:

Net input HP = $280 - 31.5 = 248.5$

Net output HP = $166 + 15 = 181$

Net mechanical efficiency = $\frac{181}{248.5} = 73 \text{ percent}$

FAILURE ANALYSIS FOR FIRST TRANSMISSION

After completion of 46-1/2 hours of testing in July 1962 with transmission S/N 101, the program was stopped because of an excessive number of wear particles found in the return oil from the transmission. The wear particles were detected by examining the return oil filter. The transmission was removed from the stand and disassembled. A very complete failure analysis was performed. Briefly, the inspection revealed:

1. The torsional deflection in the reverter system was beyond the recommended design limits for the roller bearing deflections.
2. The torsional windup from the output load created an unequal load distribution across the cam plates such that one cam plate was overloaded about two and a half times its normal loading.
3. The main losses of the transmission were traced to the eccentric bearings in the reverter system. These losses developed from the bearings going through dead center position creating a wedging action or fighting condition (Figures 17 and 18, page 27).

The design deficiencies mentioned above were corrected and incorporated into the second transmission, S/N 102. Testing deficiencies were:

1. The method of measuring transmission efficiencies.

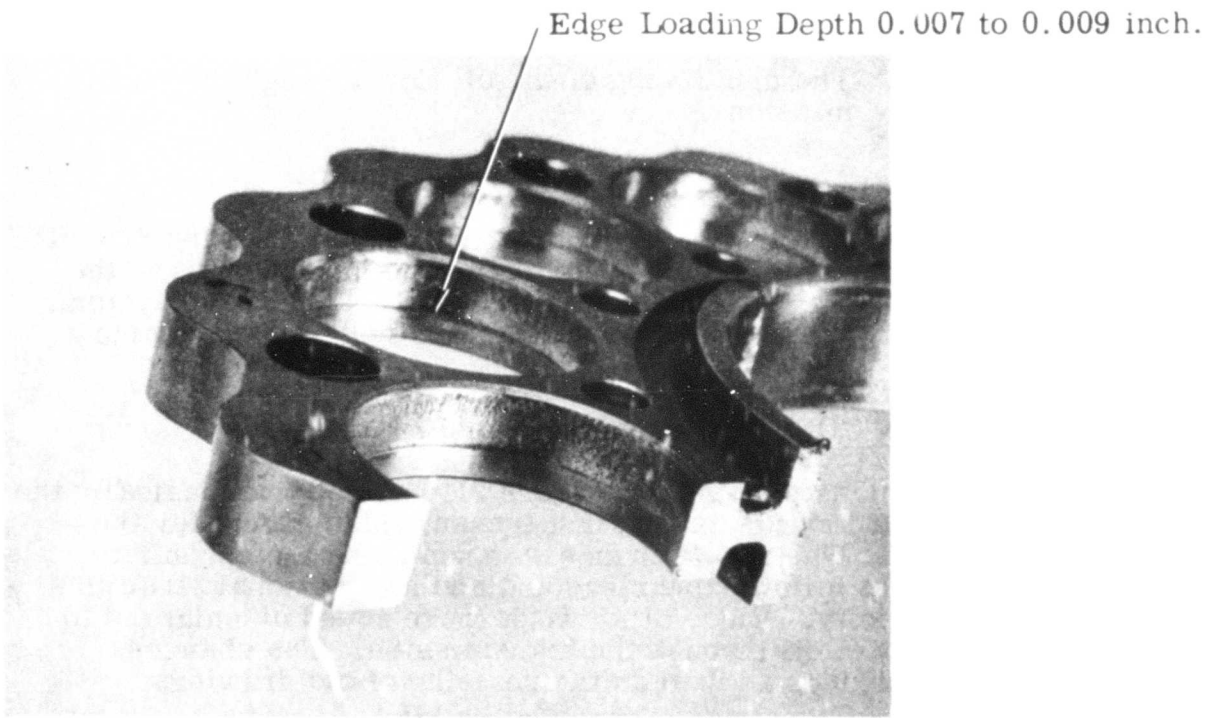


Figure 17. Cam plate #1 from test transmission S/N 101.
View showing reverter bearing load pattern.

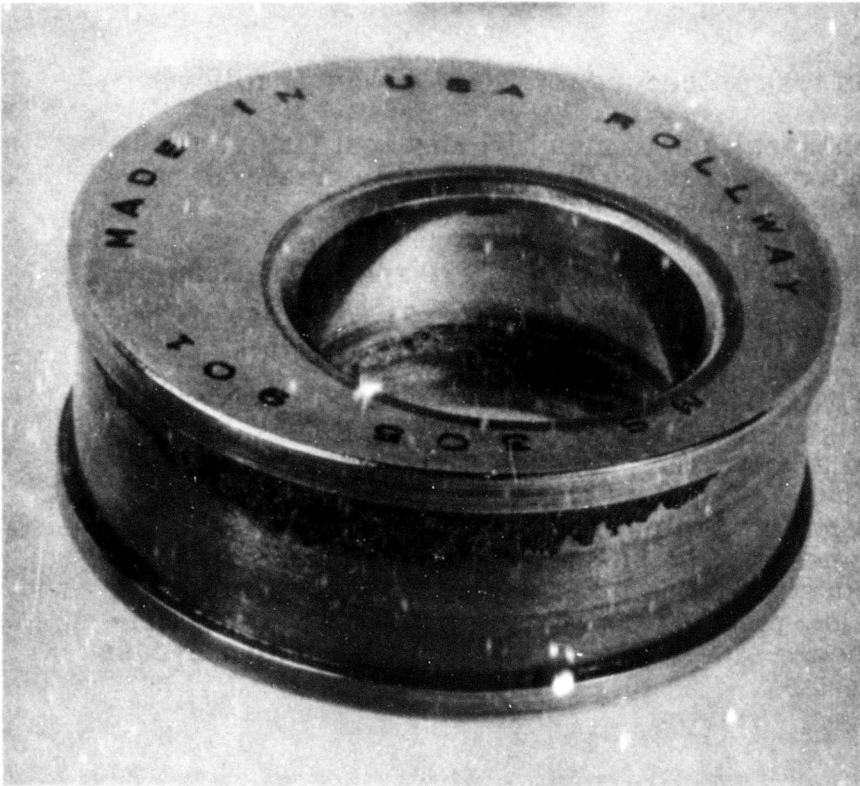


Figure 18. Reverter bearing from test transmission
S/N 101. View shows edge loading on inner
race of the bearing.

2. The monitoring of the oil flow through the transmission.
3. The wear or chip detector system.

These deficiencies were also corrected before testing S/N 102. The corrective action consisted of adding torque transducers to the test setup, adding magnetic chip detectors to the lubrication system, increasing the flow capacity of the lubrication system, and adding a flow meter.

SECOND DESIGN CONCEPT (Figures 19 and 20, page 29)

The second transmission (Figure 21, page 30) has basically the same elements and action as the first transmission except for the reverter system. The major change was replacing the eccentric bearings with cam follower bearings and adding additional structure for torsional rigidity. Also, oil orifices were added or enlarged to increase the flow of oil through the transmission. The changes required new drawings as well as the revising of old drawings.

TEST RESULTS FOR THE SECOND TRANSMISSION

Upon completion of the manufacturing and assembly, the second transmission, S/N 102, was tested under "no-load" and load conditions.

The no-load test procedure for S/N 102 was as follows:

1. The transmission was mounted on a test fixture for horizontal operation. The output stub mast was not connected to a load (Figure 22, page 31).
2. The transmission requires an external lubricating system. The system used during testing is shown in Figure 23, page 31.
3. The three inlet oil lines and the two drain lines were connected to the lubrication system. Each inlet oil line had a valve and pressure gage installed as close to the transmission as possible. Each line was of maximum size with a minimum of restrictions. The outlet lines were connected to the scavenge pump with provisions for three magnetic plugs, a temperature gage, flow meter, and heat exchanger. On the inlet or pressure side was a 10-micron filter with a bypass valve to tank.

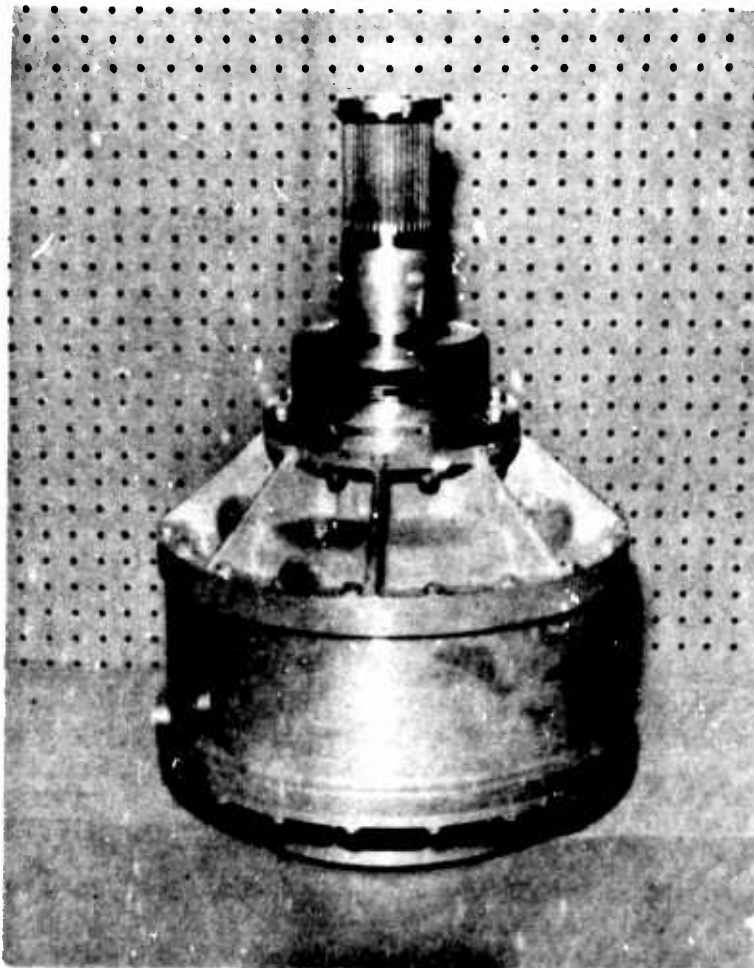


**Figure 19. Cycloidal cam transmission
model 6144R231 S/N 101.**

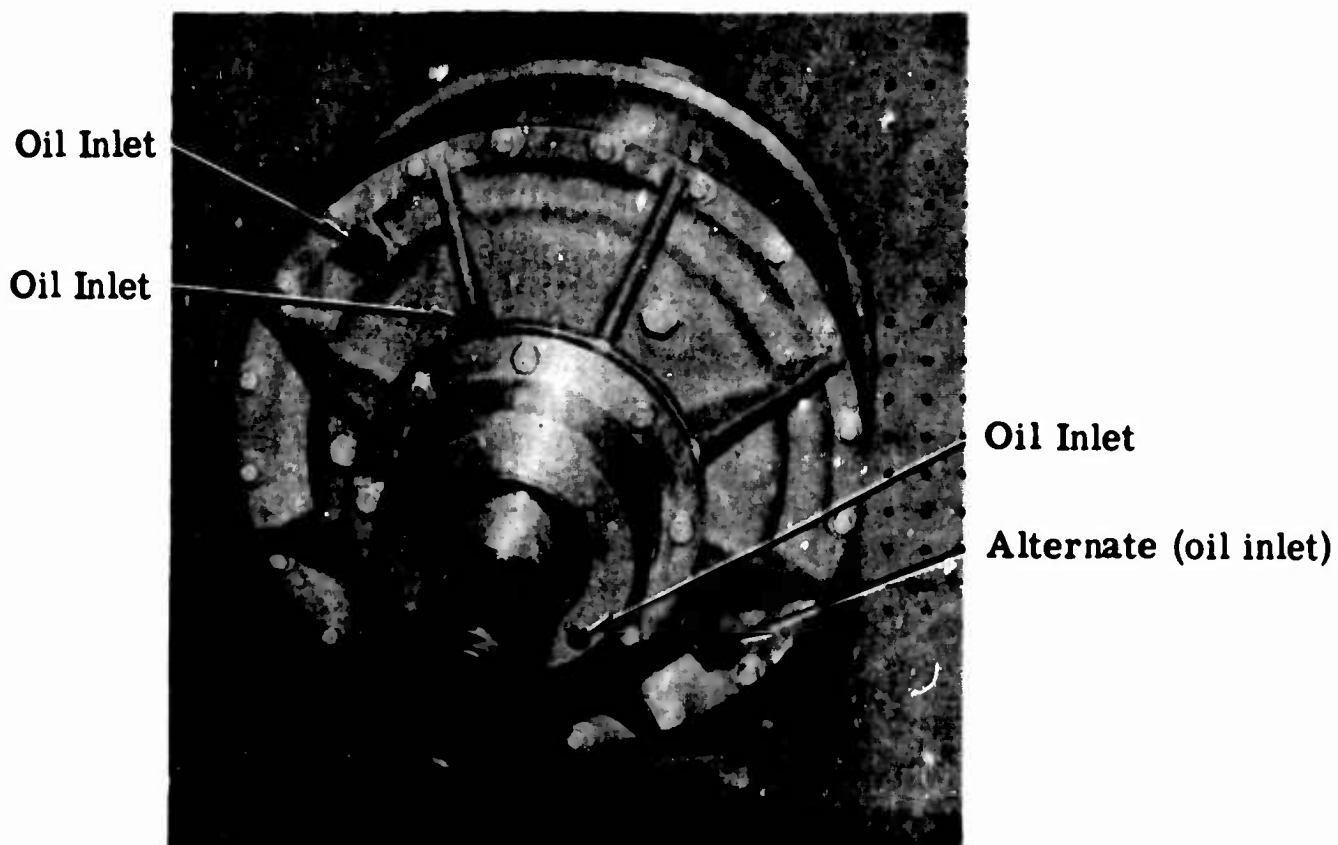


**Figure 20. End view of test transmission
S/N 102.**

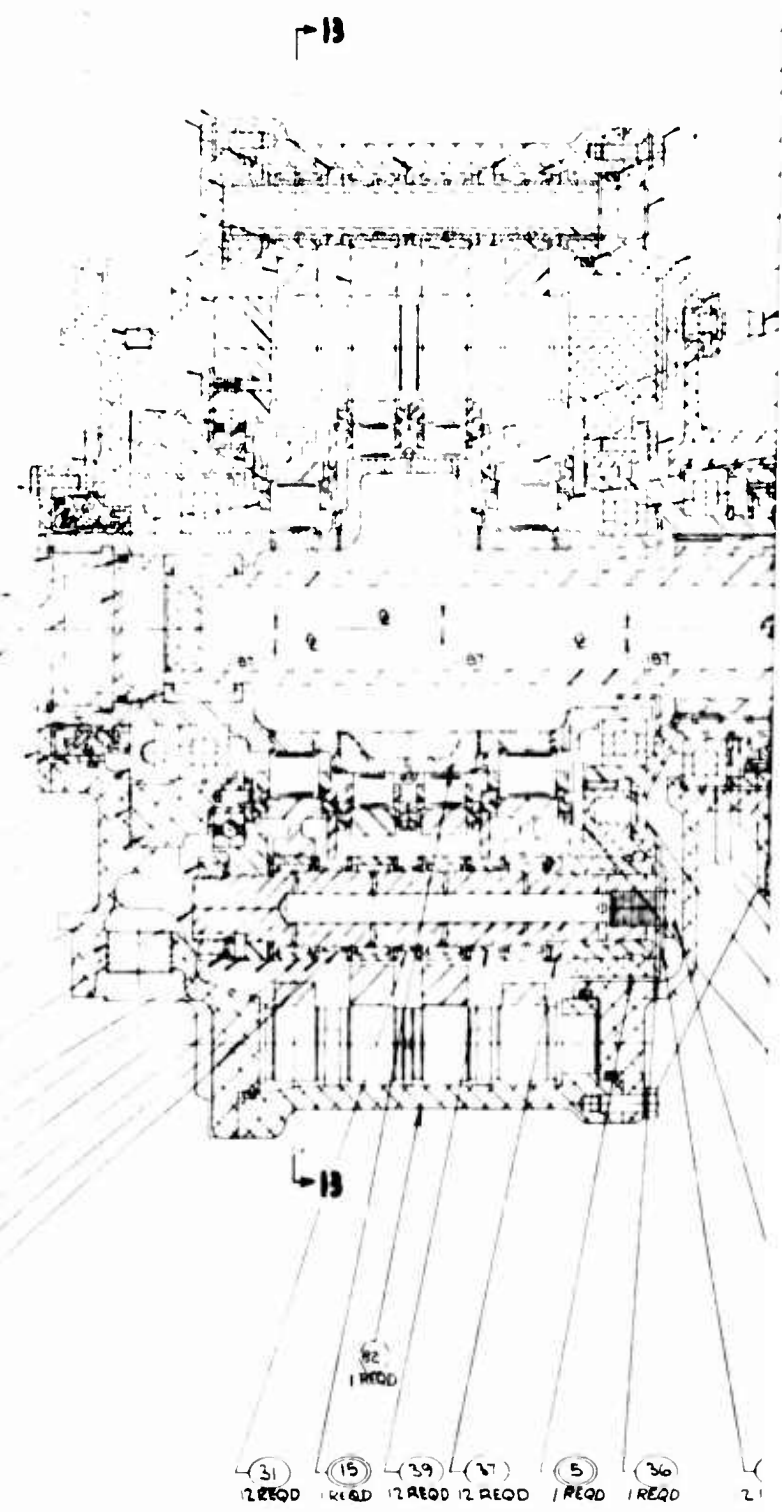
- 10 REQD (24)
- 20 REQD 3
- 1 REQD 2
- 6 67 71
- 20 REQD
- 1 REQD 56
- VENDOR ITEM
- 20 REQD 4
- 40 REQD 2
- 2 REQD 7
- 2 REQD 6
- 1 REQD 35
- 1 REQD 78
- 9 59 66 71
- 4 REQD
- 1 REQD 42
- 1 REQD 8
- 62 68 71
- 6 REQD
- 59 66 71
- 6 REQD
- 1 REQD 53
- VENDOR ITEM
- VENDOR ITEM
- 2 REQD
- 1 REQD 57
- VENDOR ITEM
- 1 REQD 51
- VENDOR ITEM
- 1 REQD 17
- 1 REQD 21
- 1 REQD 41
- 1 REQD 29
- 1 REQD 48
- 46 47
- 1 REQD VENDOR ITEM
- 1 REQD 22
- 1 REQD 19
- 12 REQD 73
- 1 REQD 81
- 12 REQD 30
- 12 REQD 74
- 12 REQD 72
- 12 REQD 34
- 48 REQD 27
- VENDOR ITEM
- 12 REQD 33
- 12 REQD 32

SEE NOTE 2

SEE NOTE 2

SEE NOTE 2

SEE NOTE 2



NOTES:

- 1 — THIS IS THE ASSEMBLY ASSIGNED SERIAL # 102 USED FOR TESTING (REF Q144R149 APPLY TO SERIAL # 101)
- 2 — LOCKWIRE PER M533540
- 3 — DOUBLE FIND NO FOR ASSY CALL OUTS

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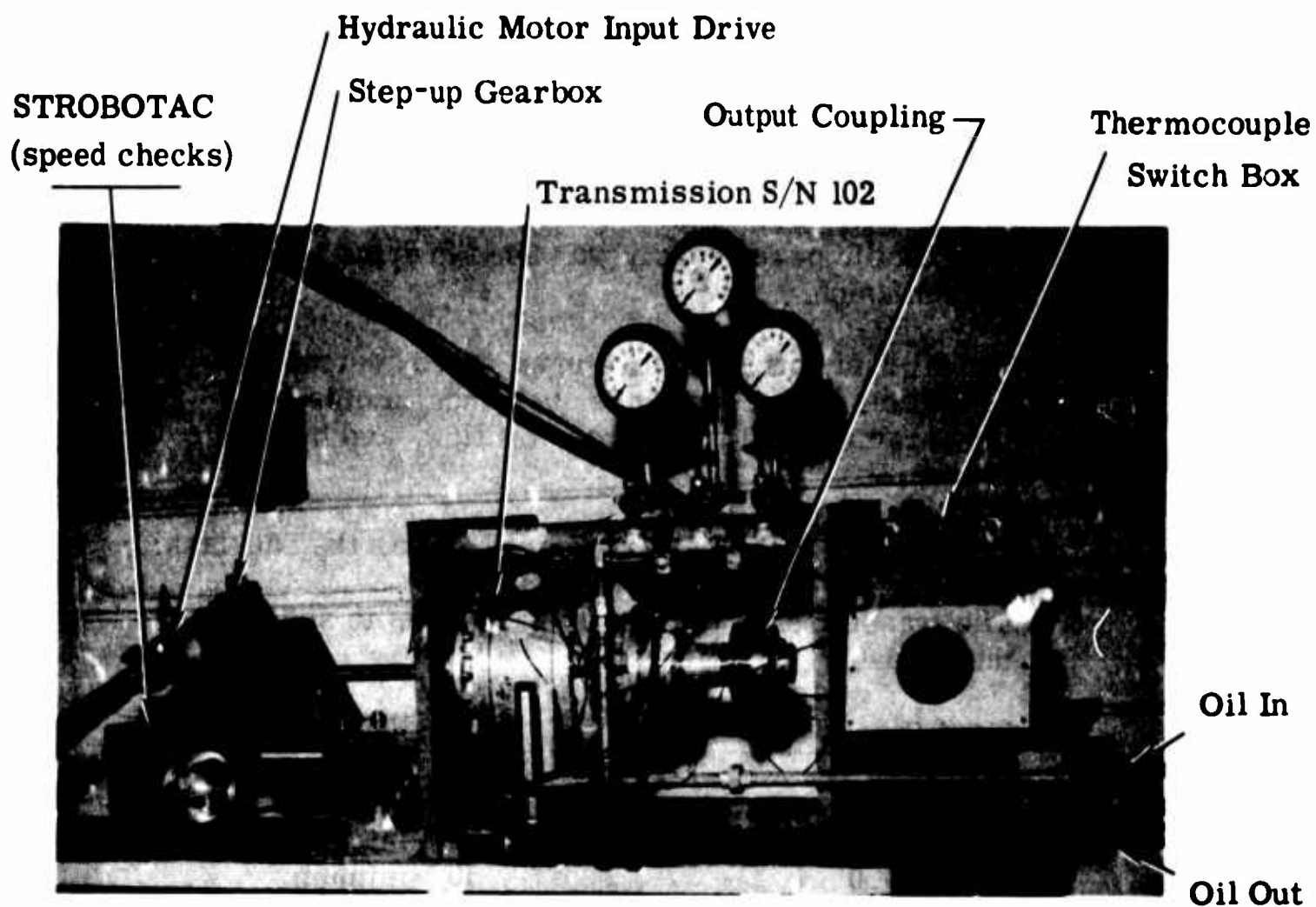


Figure 22. Run-in test stand for transmission S/N 102.

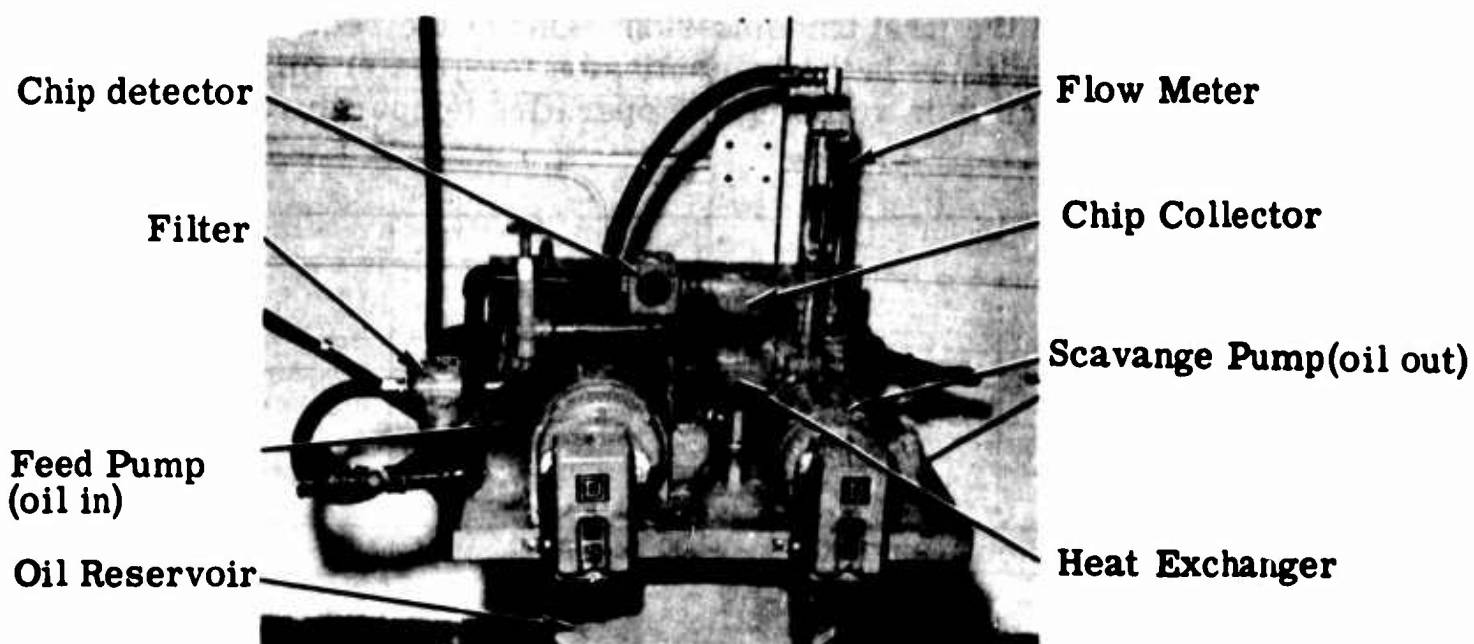


Figure 23. Lubrication system for test transmission S/N 102.

4. Room temperature oil was circulated through the transmission (before rotating the unit). The oil flow was observed at the flow meter. The transmission was rotated by hand to see if there were any binding conditions.
5. A hydraulic power supply was used to drive the transmission.
6. Typical data recorded were input speed, output speed, oil in, oil out, case, bearings, ambient air temperatures, oil pressures, oil flow, and hydraulic pressures.
7. The friction torque was measured as was the backlash.
8. The transmission was slowly brought up to speed in the following increments:

<u>Input Speed, RPM</u>	<u>Operating Time, Minutes</u>
1000	10 Minimum
2000	10 Minimum
3000	10 Minimum
4000	10 Minimum
5000	10 Minimum
6000	10 Minimum

The primary purpose of the "no-load" testing was to monitor the oil flow through the transmission to see the change of operating temperatures from the first transmission. One of the redesign changes was enlarging oil orifices in the transmission for greater oil flows. This was very effective in reducing the operating temperatures . (Figure 24).

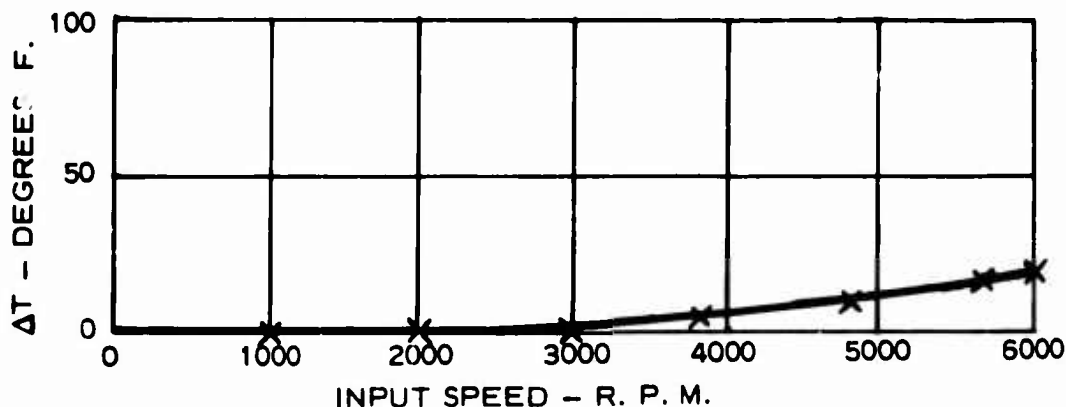


Figure 24. Stabilized temperature runs for horizontal operation of transmission S/N 102 with no load applied to the output shaft. Lubricant: MIL-L-7808D, Flow: 7.3 GPM.

Stabilized temperature runs were made for the horizontal position. The heat losses were about 5 percent. Oil flows were varied from 0 to 15 g.p.m. It was optimized at about 10 g.p.m. and 15 to 20 p.s.i.

$$\Delta \text{ HP} = \frac{Q}{12} T$$

$$\Delta \text{ HP} = \frac{10 \text{ gpm} \times 17}{12} = 14 \text{ HP}$$

The thermal loss was $\frac{14}{280} = 5\%$

The previous curves used for measuring the mechanical loss were also used for S/N 102.

$$\text{The mechanical loss} = \frac{150 \text{ inch-pounds} \times 6000 \text{ rpm}}{63,000} = 14.2 \text{ HP}$$

$$\text{Mechanical Loss} = \frac{14.2}{280} = 5\%$$

One teardown inspection was accomplished during the no-load testing phase. The inspection revealed:

1. Uniform contact markings on the fix pin rollers.
2. Slight sludge deposits were found on the magnetic plugs in the lubrication.

Corrective Action:

No corrective action was deemed necessary. The sludge was considered normal scuffing from running the unit in. The unit was reassembled. The static torque was considerably reduced. In fact the unit could be turned by hand, turning from the output side. This could not be done with the first unit.

The torque was 3 to 5 inch-pounds. The backlash was .090 inch measured at the fix pin roller radius. The backlash was about six times larger than that of the first unit. This comes about by the change in geometry of the reverter system.

The load testing procedure for the second unit, S/N 102, was as follows:

1. The transmission was mounted on the test fixture in the horizontal position with torque transducers aligned and mounted on the fixture (Figure 25, page 34).

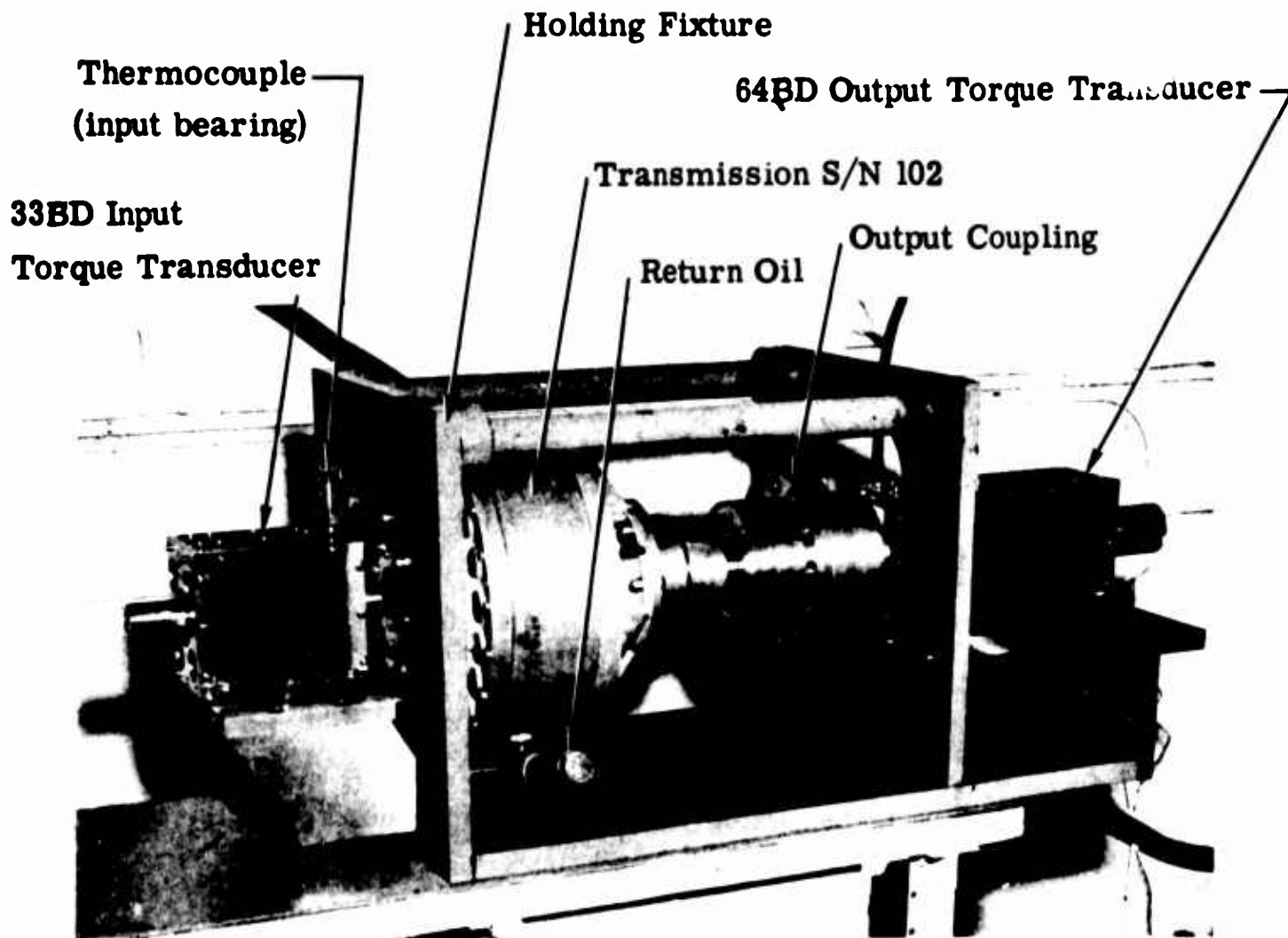


Figure 25. Assembled transmission load fixture.

- 2. The assembly was mounted in the load test stand with input power drive and output loading equipment.**
- 3. The external lubricating systems with their pumps, filters, coolers, sumps, and associated plumbing were connected. The oil reservoir for the transmission was filled with aircraft MIL-L-7808D oil (Figure 23, page 31).**
- 4. Room temperature oil was circulated through the transmission for 1 hour before rotating the unit. A similar procedure was followed for the other machinery used in the test setup.**
- 5. Before the output coupling was connected to the load water pump equipment, the transmission was checked for binding by rotating the input shaft by hand.**
- 6. Typical data monitored and recorded were dynamometer loads, speeds, temperatures, oil pressures, flows, torques, and wear detection checks.**

- 7
7. The transmission was slowly brought up to one-third design speed with the load water pump disengaged. The run was continued until the transmission temperatures had stabilized.
 8. With the load water pump clutch engaged, the transmission was brought up to speed and load, in the following increments:

<u>Input Speed RPM</u>	<u>Dynamometer HP</u>	<u>% Torque</u>	<u>Operating Time, Minutes</u>
1000	6	13	10 Minimum
2000	25	27	10 Minimum
3000	50	36	10 Minimum
4000	100	50	10 Minimum
5000	180	75	10 Minimum
6000	280	100	10 Minimum

Additional increments desired:

<u>Input Speed RPM</u>	<u>Dynamometer HP</u>	<u>% Torque</u>
3000	35	25
4000	47	25
5000	59	25
6000	71	25
5000	117	50
6000	140	50
5500	192	75
6000	210	75

NOTE: In order to secure the above increments, two load pumps were involved. One pump covered the light loads and the other pump covered the balance of the spectrum. (Figure 26, page 36).

9. A total of 50 hours of load testing was to have been accumulated but this was not accomplished. The testing was stopped after 10 - 1/2 hours because wear particles were found in the return oil of the transmission.

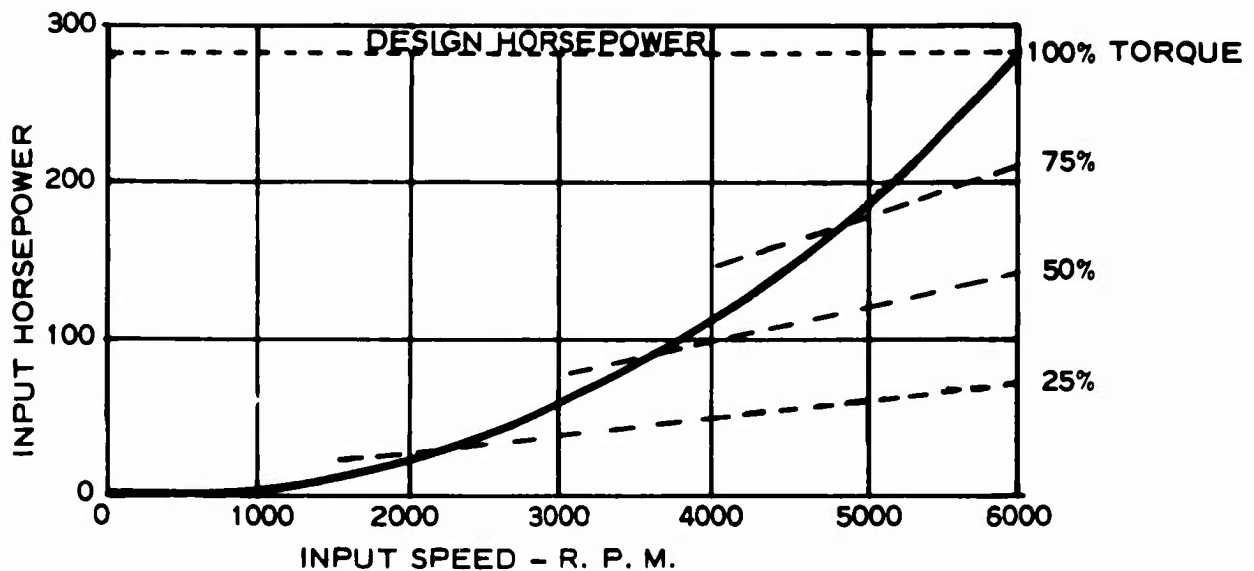


Figure 26. Loading Spectrum

Development testing under loading conditions revealed a decided increase in overall efficiency over the first transmission. By visual inspection after teardown, it was found that the desired stiffness in the reverter cage had been achieved. The four cam plates had equal load markings on the cam lobes.

Figures 27 and 28, page 37, show the relationship of oil temperatures to input speed. The differential temperature is the difference between inlet and outlet oil. The tremendous reduction in the T temperature between the two transmissions is attributed to the larger oil flow through the second transmission, rather than to differences between the two types of oil. The viscosity of the two oils are quite similar: at 100° F, 11 centistokes for MIL-L-7808D and 20 centistokes of MIL-L-6086. The lower operating temperatures for the second transmission is reflected by an increase in efficiency between the two units.

Figures 29, and 30, page 38 reflect the loading spectrum accomplished. Two loading pumps of different capacity were used in order to cover the range of torque settings and speeds desired, as outlined in the load test procedure. Torque settings of 25 percent, 50 percent, and 75 percent were achieved during the testing. The last setting of 100 percent was not accomplished because of excessive transmission wear requiring the testing be stopped.

Figures 31, 32, 33 and 34 pages 39 and 40, are torque curves generated from transducer data. Special attention is directed toward the transducers. The two transducers were developed specifically for this application. While commercial transducers now on the market could also have generated the same data, the βD transducers

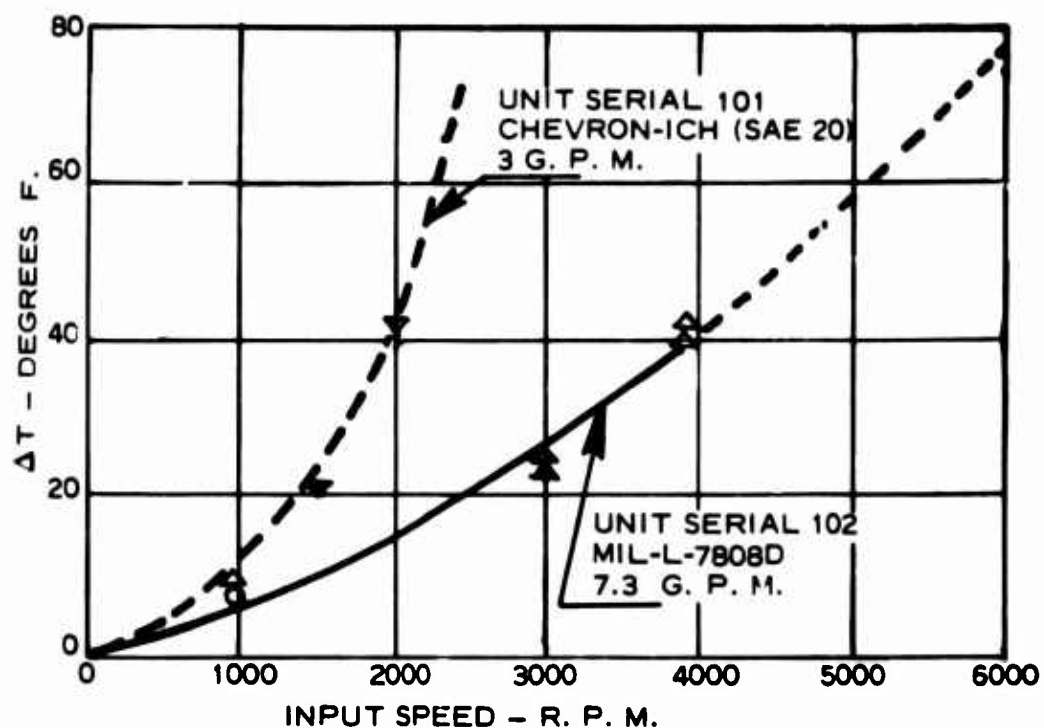


Figure 27. Stabilized temperature load runs for transmission in horizontal operation. Lubricants were not cooled. Loading: Water pump 2 x 3 x 10-1/2 H-SMU.

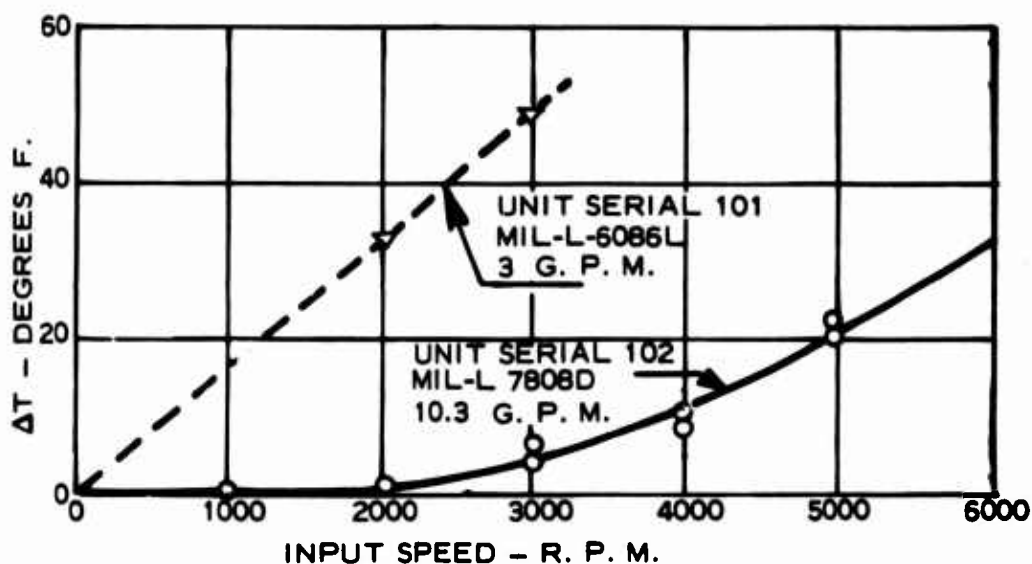


Figure 28. Stabilized temperature load runs for transmissions in horizontal operation. Lubricants were cooled. Loading: two stage water pump 3 x 4 x 11-1/2.

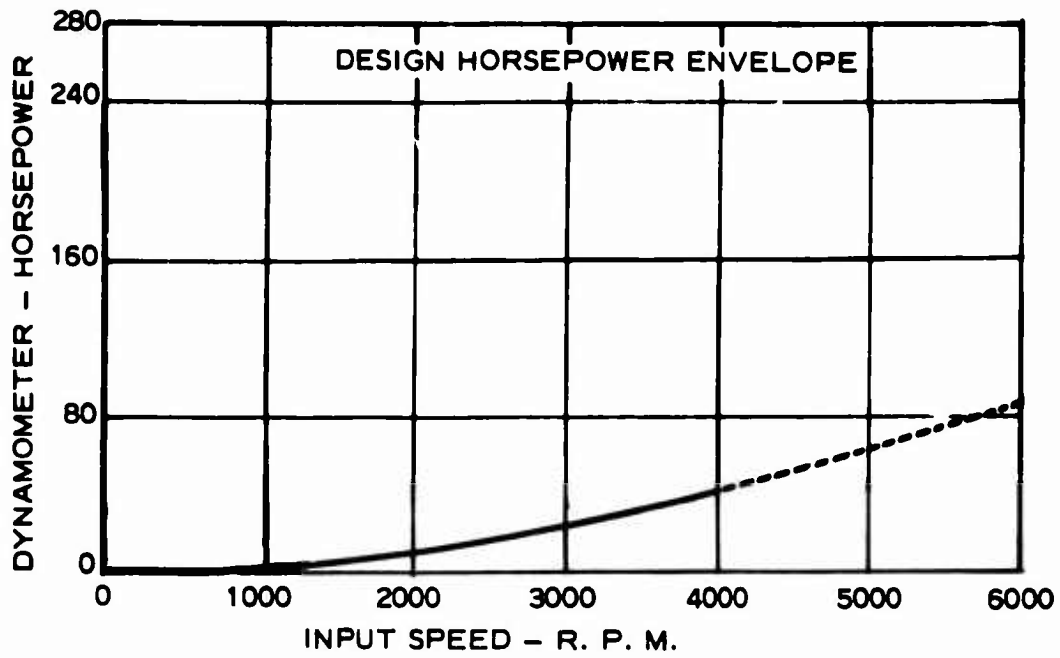


Figure 29. Input power. Load testing with water pump 2 x 3 x 10-1/2 H-SMU.

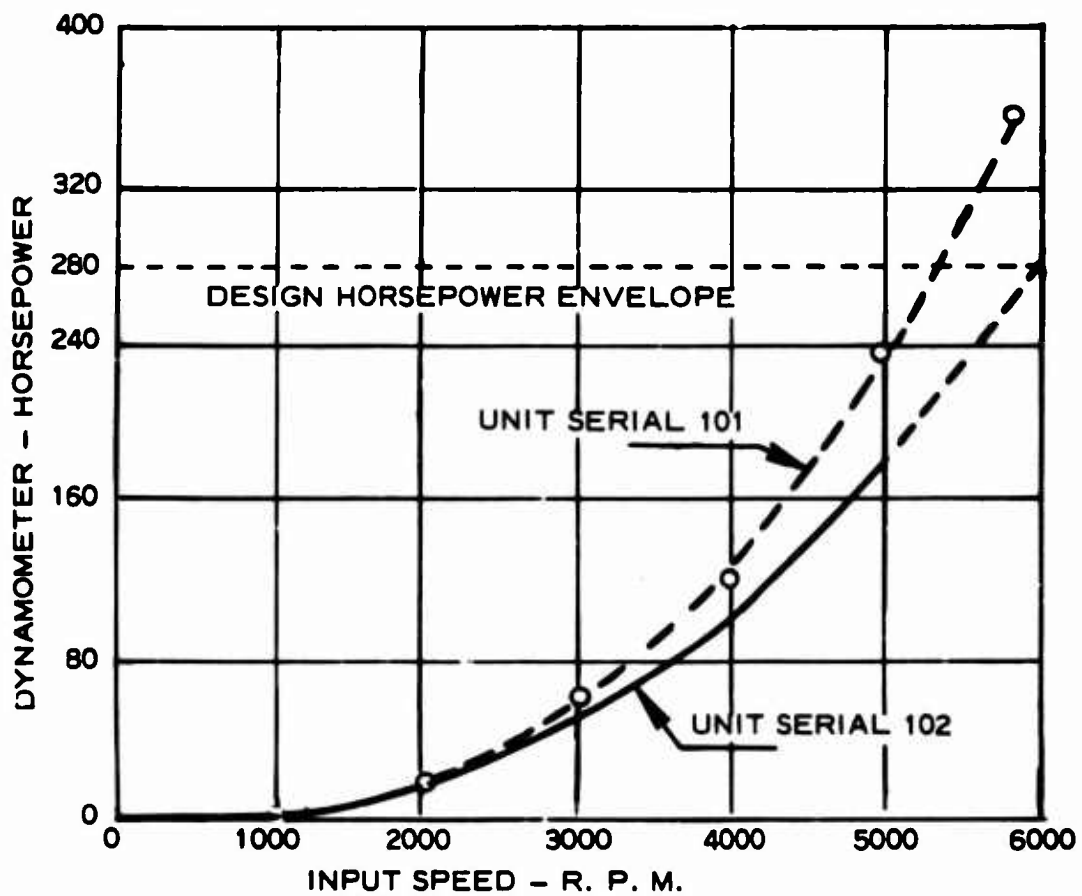


Figure 30. Input power. Load testing with two stage water pump 3 x 4 x 11-1/2.

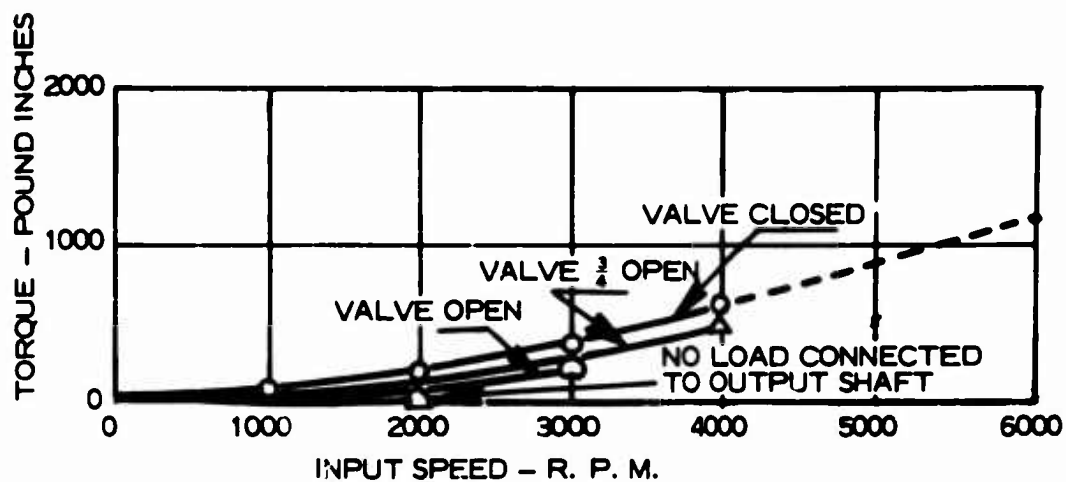


Figure 31. Input torque. Load testing with water pump 2 x 3 x 10-1/2 H-SMU. Torque measured with 33 β_D transducer.

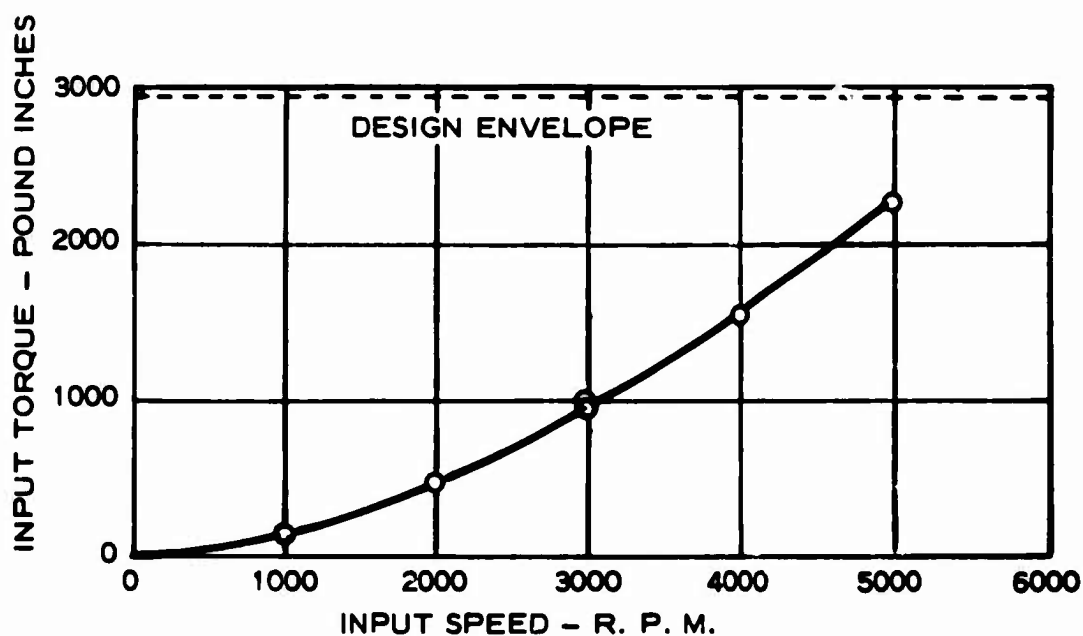


Figure 32. Input torque. Load testing with two stage water pump 3 x 4 x 11-1/2. Torque measured with 33 β_D transducer.

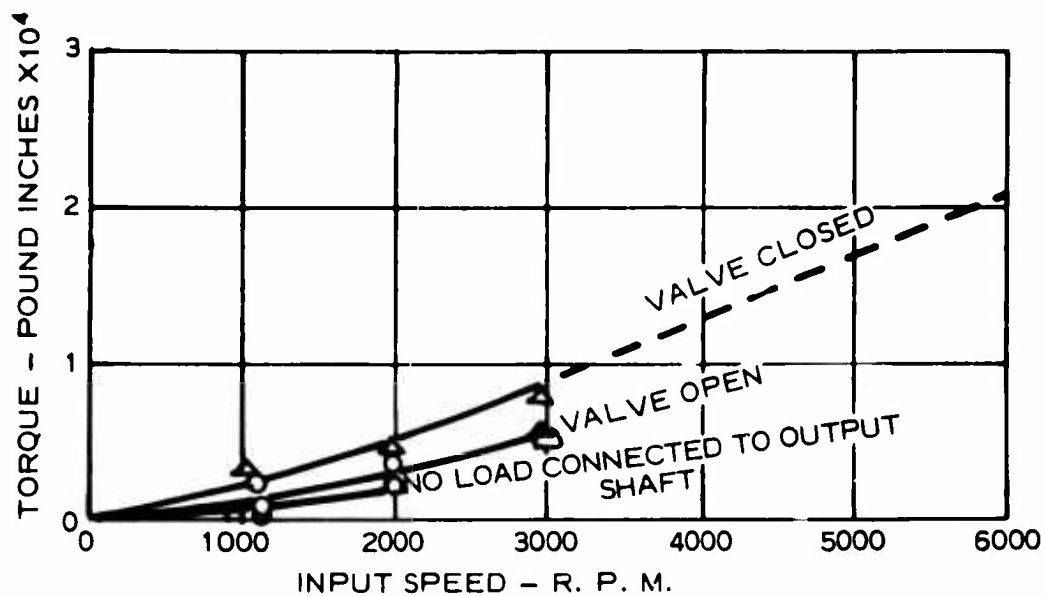


Figure 33. Output torque. Load testing with water pump 2 x 3 x 10-1/2 H-SMU. Torque measured with $64 \beta_D$ transducer.

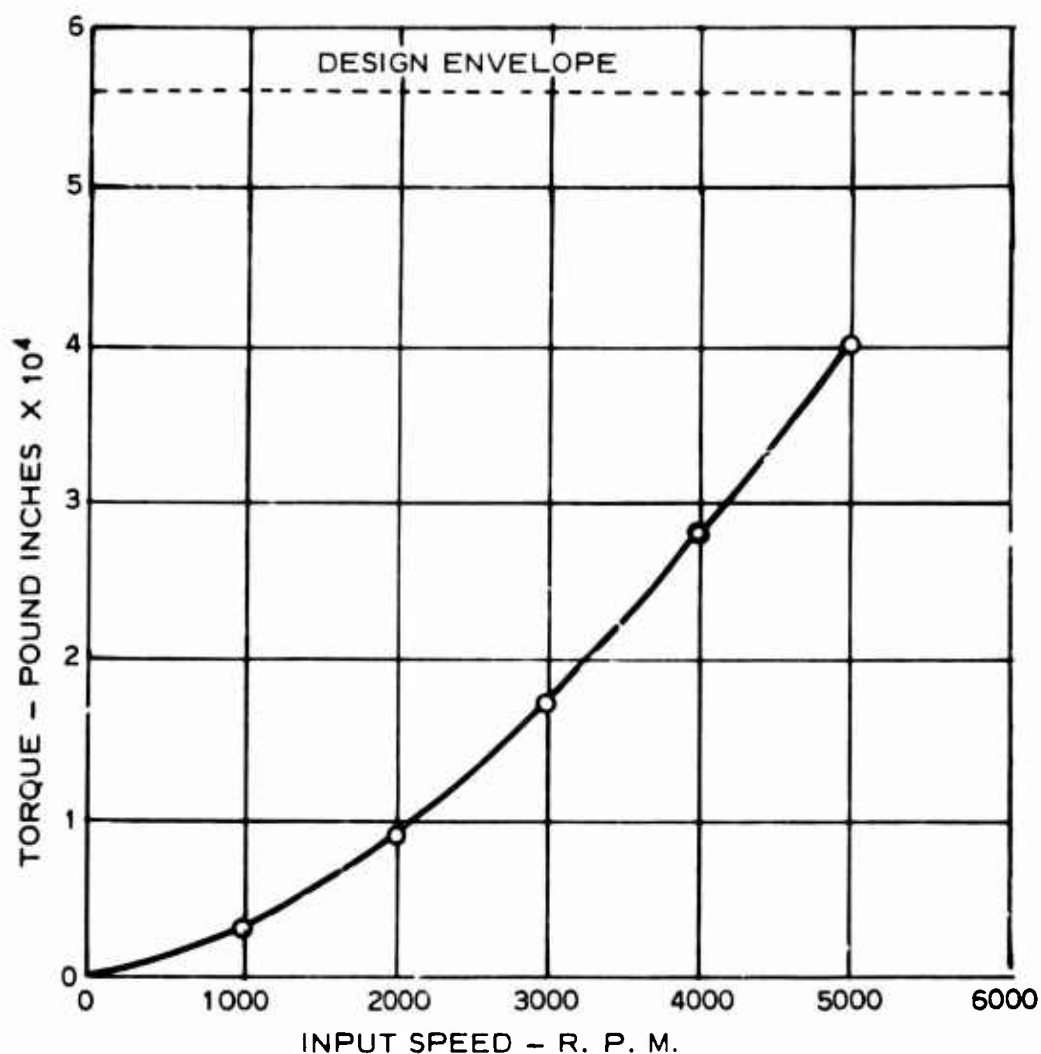


Figure 34. Output torque. Load testing with two stage water pump 3 x 4 x 11-1/2. Torque measured with $64 \beta_D$ transducer.

are unique in the field. Their uniqueness lies in their ability to translate torque directly into digital form. This form is "ideal" for modern transport tape computers such as IBM, Remington Rand, etc.

With the input and output torque curves, the transmission efficiencies were calculated as shown in Table 1.

TABLE 1
TRANSMISSION EFFICIENCIES

Input Speed (r.p.m.)	Input Torque (inch-pounds)	Output Torque (inch-pounds)	Ratio	Efficiency (percent)
1000	160	3000	19:1	98.8
2000	485	9000	19:1	97.8
3000	925	17000	19:1	96.5
4000	1525	27750	19:1	95.5
5000	2250	40000	19:1	93.5

Figures 35, 36, and 37, pages 42 , 43 , and 44 , are curves on efficiency. Direct comparisons with the two transmissions indicate a considerable improvement. The improvement was the product of design changes incorporated into the second unit. These design changes have been covered earlier in this section of this report.

After completion of 17 hours of testing with transmission S/N 102, the test program was stopped because of an excessive number of wear particles found in the return oil. The particles were detected by examining the magnetic plugs in the system. The transmission was removed from the stand and disassembled July 1963.

Briefly, the inspection revealed:

1. Heavy wear on the crankshaft.
2. Skidding and scuffing marks on the fixed pin rollers (reaction members).

FAILURE ANALYSIS FOR SECOND TRANSMISSION

S/N 102 was assembled incorporating additional structure and cam follower type reverter rollers. The roller design replaced the original eccentric bearings. Other minor modifications included enlarged oil passageways and replacement of the upper main ball bearing by a roller type.

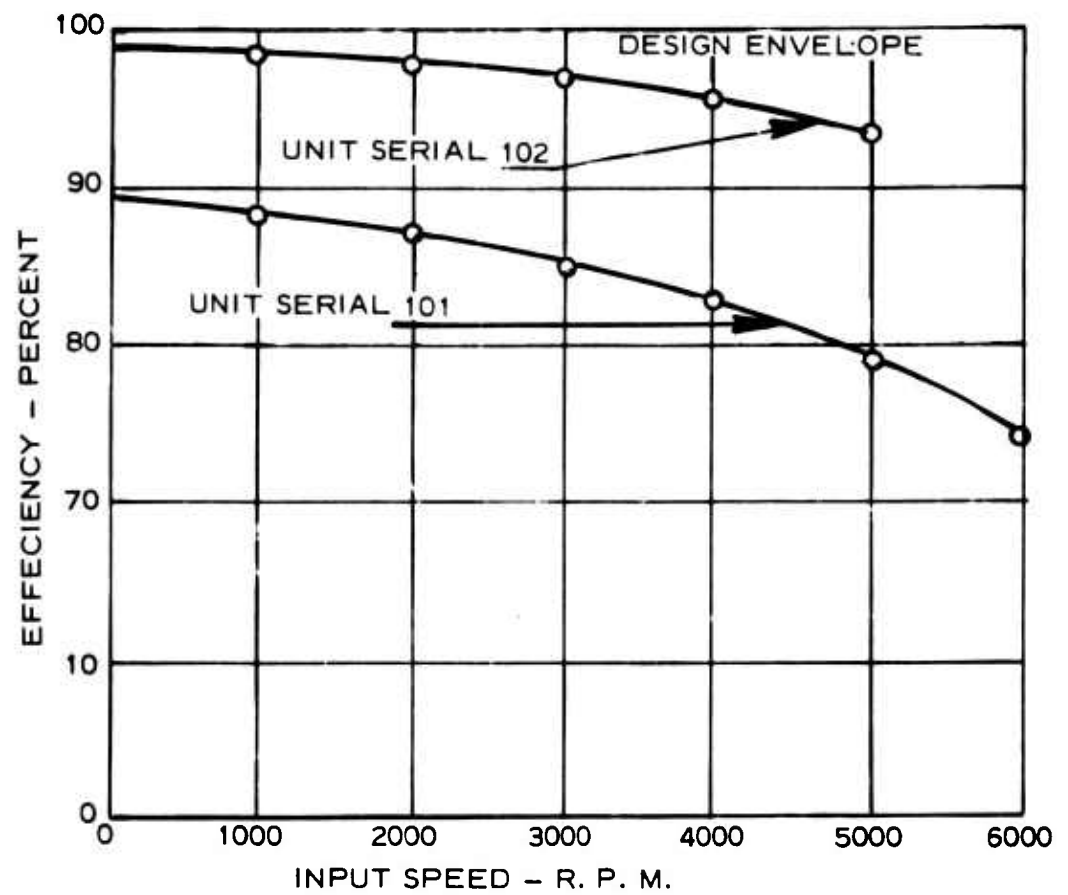


Figure 35. Comparison of efficiency under various loads and speed conditions.

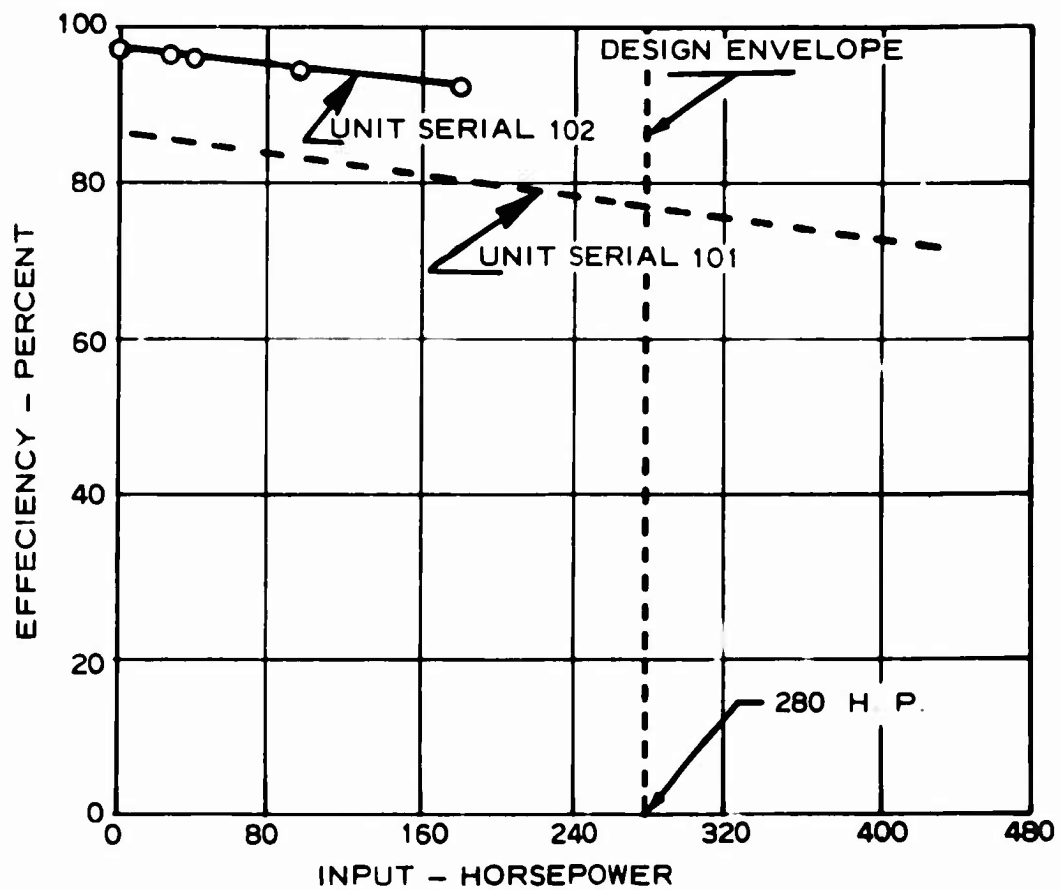


Figure 36. Comparison of efficiency with variable power conditions. Load testing with two stage water pump 3 x 4 x 11-1/2.

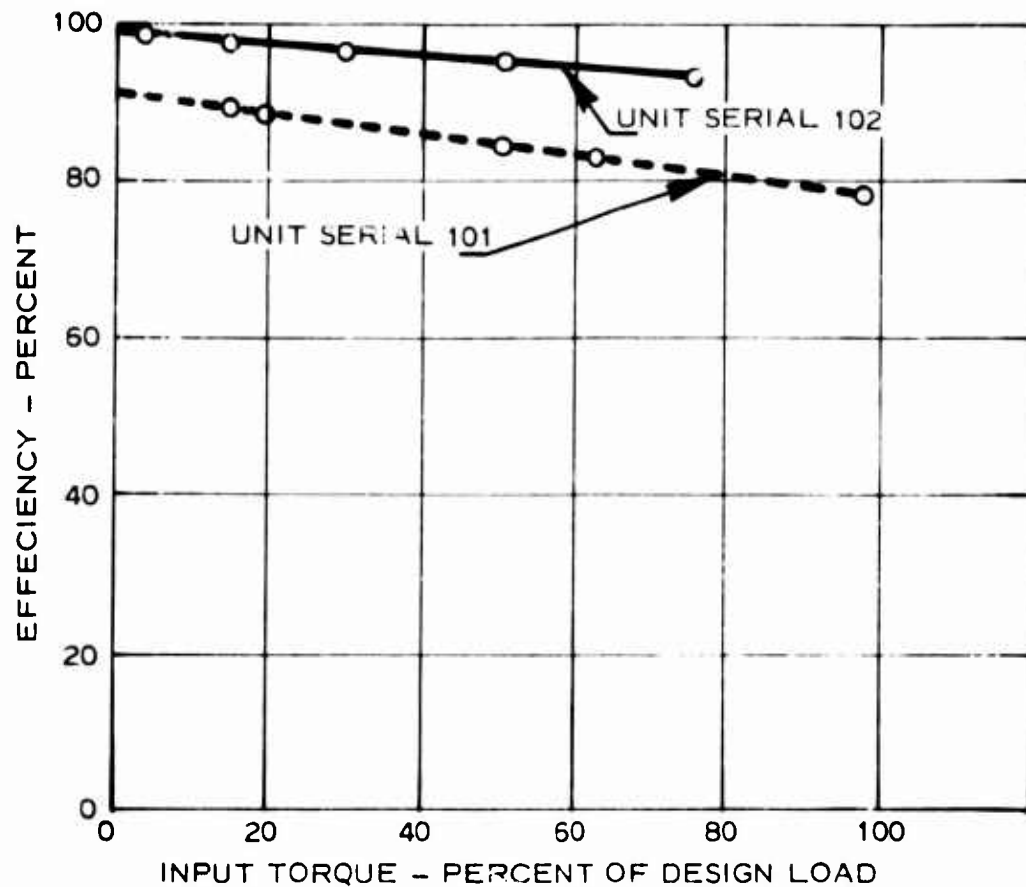


Figure 37. Comparison of efficiency with variable torque load testing with two stage water pump 3 x 4 x 11-1/2.

Metallic particles were discovered in the oil filter after 17-1/2 hours of test running on transmission S/N 102, the test stand was shut down. Upon disassembly of the transmission and visual inspection of the crank shaft throws at all four cam plate positions showed heavy wear. It might be noted that this type of heavy wear was non-existent with transmission S/N 101.

Further examination of the balance of the component parts of S/N 102 were in satisfactory condition. The heavy wear on the crank shaft is shown in Figure 38 on page 45. Metallurgical and X-ray tests of the crank shaft indicate an adequate amount of structure and sufficient hardness for the roller bearings. The crank shaft was sectioned for detail metallurgical examinations.

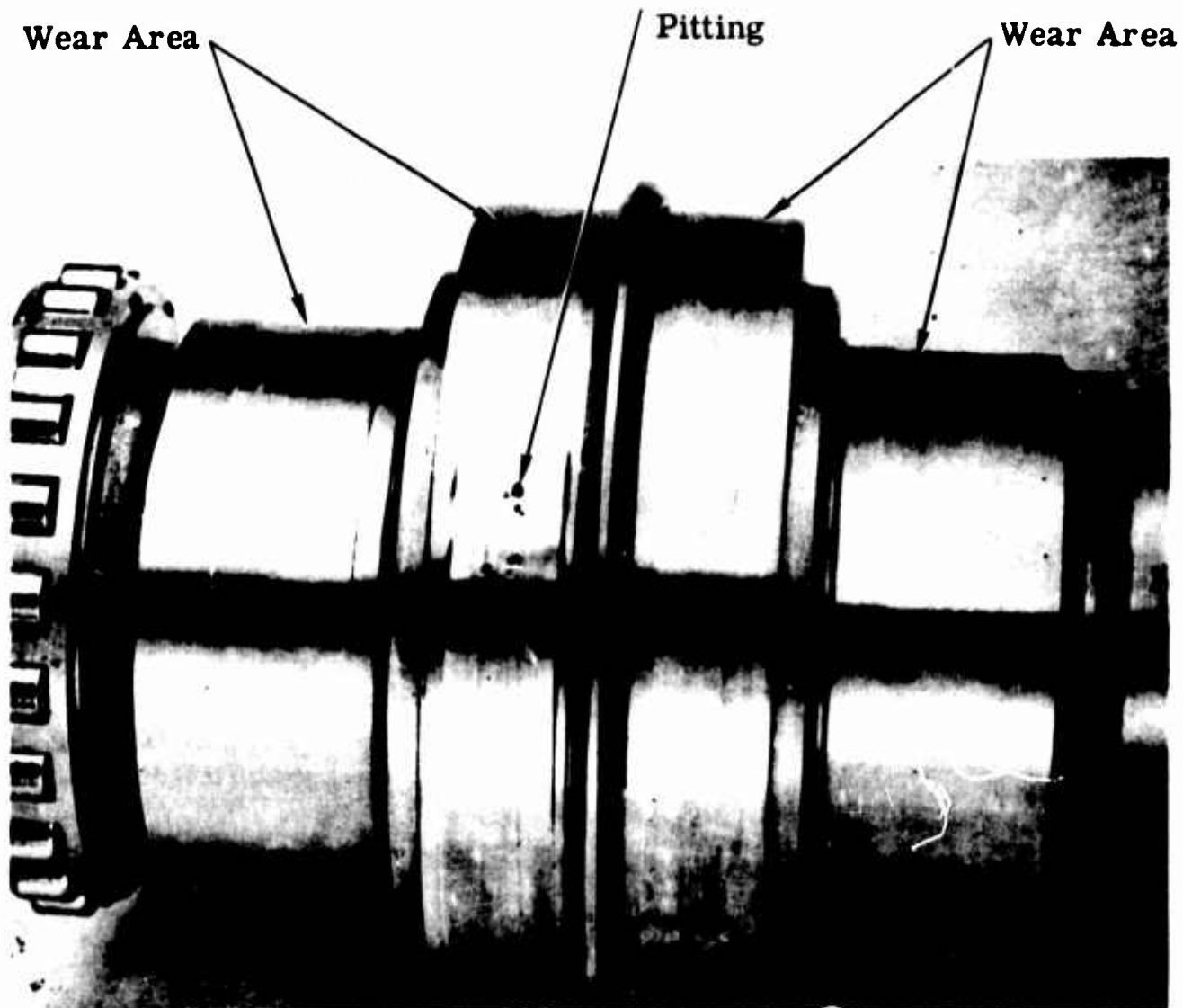
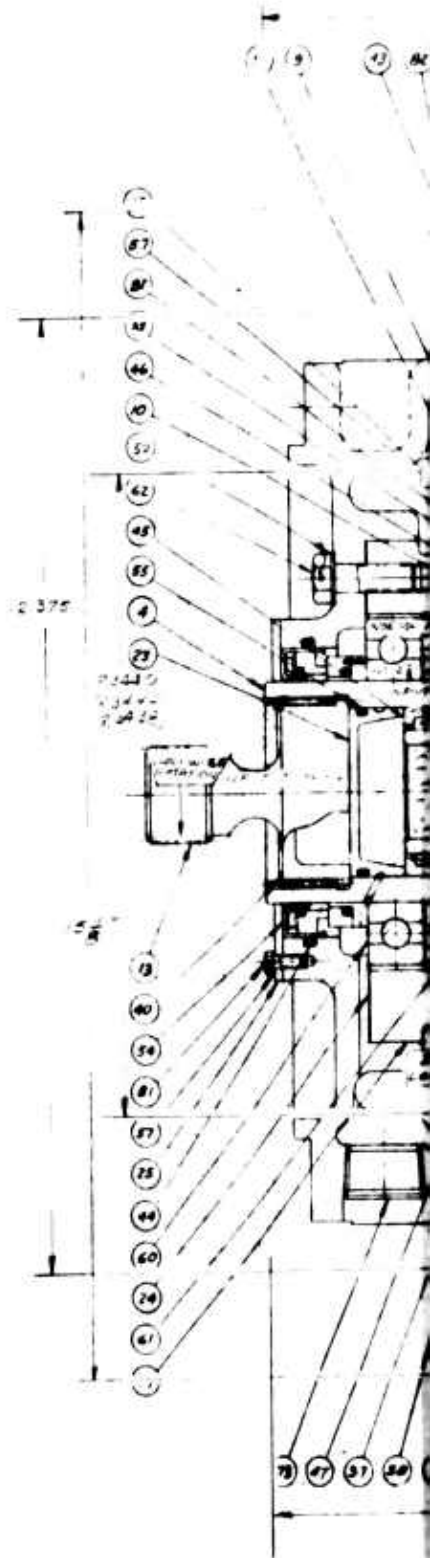
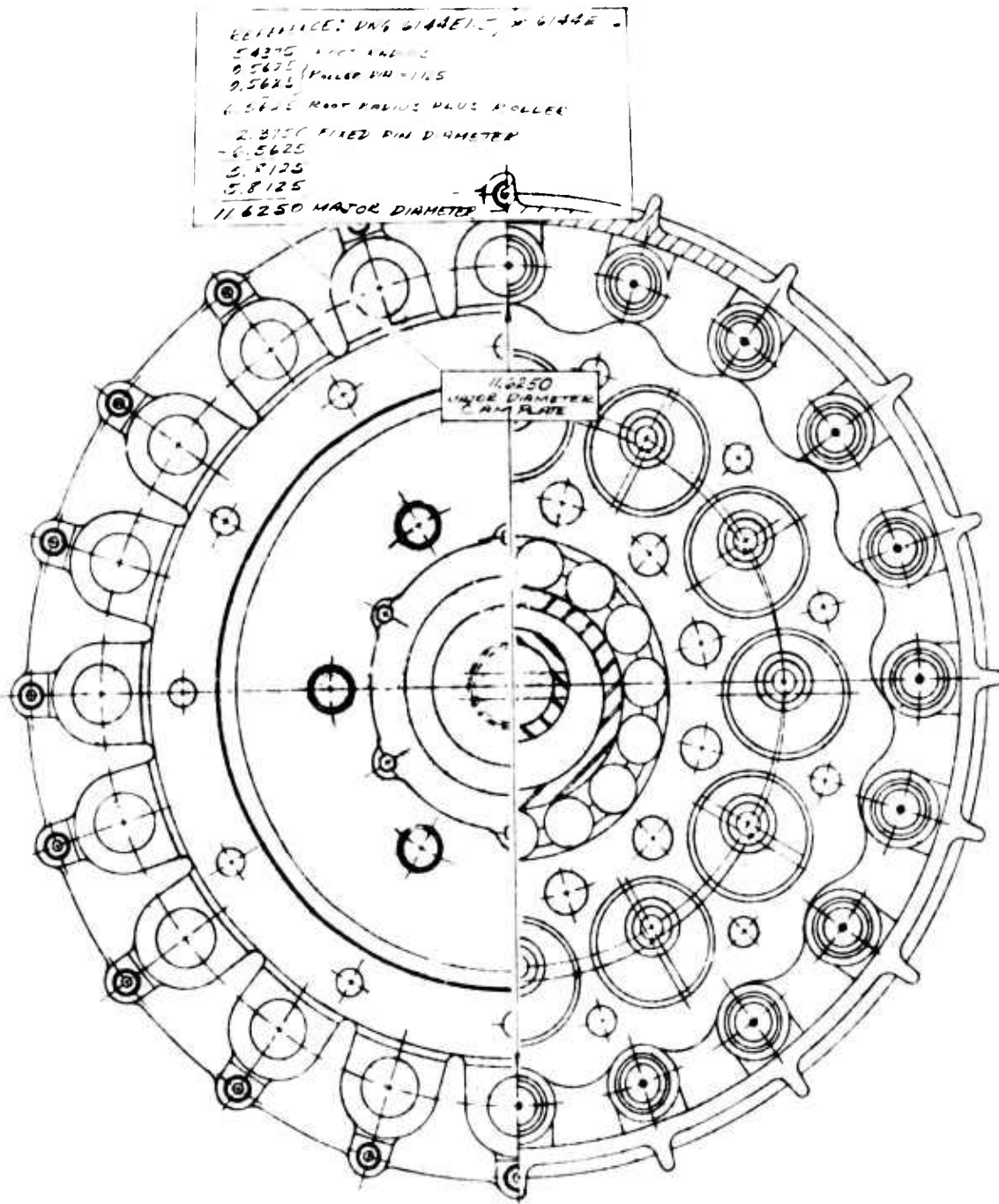


Figure 38. Crank shaft for transmission S/N 102 showing wear pattern from roller bearings.

Re-examination of the theoretical bearing loads (Figure 39, page 46) indicates that under normal running conditions, overloading cannot occur. Subsequent trial assembly tests proved that the cam plates cannot be mistimed during assembly, thus eliminating the possibility of overload being built into the unit on initial assembly.

A study of the two units has, however, produced a theory which explains why the crankshaft of S/N 101 operated normally and accounts for the wear condition of crankshaft No. 2. The basis of the theory is that reverter eccentrics were fitted in each cam plate of the first unit, while cam follower type reverter bearings are used in the second unit. The reverter eccentric bearings "fill" the cam plate holes, while the cam follower bearings do not.

It must be borne in mind that due to the novel nature of the Cycloidal Cam transmission, no previous running data for large horsepower units is available; consequently, optimum clearances between components have not yet been established. Retrospective analysis will show that when eccentric reverter bearings are used, each



NOTES

1. REVERTER LOADING

$$F = \frac{282 \text{ HP} \times 63000}{125 \text{ MM} \times 815 \text{ RPM} \times 4.156 \text{ RAD}} = 930^{\circ}$$

2. FIXED PIN LOADING

$$N = \frac{2N}{100} = \frac{185 \times 60000}{100} = 800 \text{ PSI}$$

$$P = \frac{N}{LQ} = \frac{400^{\circ}}{.9 \times 18} = 600 \text{ PSI}$$

$$\text{CALCULATED LOAD} = 388^{\circ}$$

$$\text{BEARING CLEARANCE} = .001$$

3. REVERTER BEARING CLEARANCE = .001 TO .0015

4. CRANK BEARING CLEARANCE = .0025 TO .0035

5. LUBRICATION

PRESSURIZED SYSTEM SIMILAR TO MILLER TRANSMISSION (1982 R11)

$$\text{OIL FLOW} = 1 \text{ GPM} \approx 60 \text{ MLIT}$$

$$\text{LUBE} = \text{MIL-C-6529A TYPE 2J/AND 02}$$

$$\text{AN-0-3 LIGHT}$$

$$\text{OIL PRESSURE} = 50 \text{ PSI}$$

$$\text{TEMPERATURE RSE} = 30^{\circ}\text{F}$$

$$\text{OIL IN} = 210^{\circ}\text{F}$$

$$\text{OIL OUT} = 220^{\circ}\text{F}$$

6. MILLER RATING = 216 MP

$$\text{TAIL ROTOR} = 34 \text{ MP}$$

$$\text{TESTING SPEC, MIL-T-8679}$$

7. BEARING LIFE CALCULATIONS

$$\text{B10} = 1000 \text{ HOURS}$$

8. FIXED PIN BEARING CLEARANCE = .001

9. REVERTER SHAFT CLEARANCE = .001

83	40	AN560C616	WASHER, PLAIN		
82	40	AN146A	BOLT, MACHINE		
81	22	AN243A	BOLT, MACHINE		
80	1	2-10	OIL SEAL		
79	20	725B1/2	BALL BEARING DUP	CLASS 1	SKP
78					
77					
76					
75	1	KLS 24	BALL BEARING	CLASS 1	ADAMAS HMT
74					
73	1	1/4 NPT	PLUG, PIPE, HEX SOCK		CONNEL
72	1	1/4 NPT	PLUG, PIPE, HEX SOCK		CONNEL
71	1	1/4 NPT	PLUG, PIPE, HEX SOCK		CONNEL
70	1	1/4 NPT	PLUG, PIPE, HEX SOCK		CONNEL
69	1	1340D1/2	O RING		PARKER
68	1	1100-1/8	O RING		PARKER
67	12	NJ 10216	NEEDLE BEARING	CLASS 1	TORRENTON
66	12	SC 37	RETAINING RING		TORRENTON
65	1	1400-1/8	O RING		PARKER
64	12	J 100	NEEDLE BEARING	CLASS 1	TORRENTON
63					
62	6	AN611H	BOLT, MACHINE		
61	1	RPLS 27/8	ROLLER BEARING	CLASS 1	ADAMAS HMT
60	1	KLS 27/8	BALL BEARING	CLASS 1	ADAMAS HMT
59	20	AN560C616	WASHER, PLAIN		
58	8	AN560C620	NUT, SELF LOCKING		
57	10	AN560C616	WASHER, PLAIN		
56	8	AN560C616	BOLT, MACHINE		
55	1	8156	O RING		ADAMAS HMT
54	1	SK 4470	OIL SEAL		CATFISH
53	1		OIL SEAL		
52	1	AN1339A	O RING		
51	1	555	OIL SEAL		
50	1	500-334	RETAINING RING		TRUACE
49	8	AN6-3A	BOLT, MACHINE		
48	1	R 1917 TMC	ROLLER BEARING	CLASS 1	MRC
47	18	MS20508	PLUG, STRAIGHT		
46	1	KD 55CP	BALL BEARING	CLASS 1	CAYDON
45	1	NJ26550	NEEDLE BRG	CLASS 1	TORRENTON
44					
43	1	AN560C616	LOCKWEE		
42	2	6144C147	RPS ROLLER, INNER	CLASS 3	ROLLWAY
41	1	8258	O RING		MAIN CLARKE
40	1	5000-231	RETAINING RING		M. TRUACE
39	2	6144C148	BRS, ROLLER, OUTER	CLASS 3	EDLEY
38	12	NTH 1018	THRUST BEARING		TORRENTON
37	20	TED 018	THRUST RING		TORRENTON
36	4	9 0808-21	SPRING, COIL		DANLY HMT
35	4	AN335-10	WASHER, LOCK		
34					
33	4	6144B17	RETAINED RING		
32	1	6144B175	PLATE, NAME		
31	20	6144B171	SPACER, ROLLER		
30	1	6144C170	SHIM, LOWER REG		
29	1	6144C169	COVER, REV. DOCK		
28	10	6144B18	SUPPORT FIXED PIN		
27					
26	12	6144B112	WASHER, THRUST		
25	1	6144C148	COVER, OIL SEAL		
24	1	6144D167	SHIM, LOWER		
23	1	6144C166	SEAL, CRANK SHAFT		
22	48	6144C146	BEARING, REVERTER	CLASS 3	ROLLWAY
21	20	6144C140	SHAFT, FIXED PIN		
20	1	6144B165	CAP, UPPER		
19	20	6144C17	ASSY, ROLLER, FIXED PIN		
18	12	6144C151	SMALL, REVERTER		
17	1	6144B161	ADAPTER BEARING		
16	1	6144C163	SPACER, BEARING		
15	1	6144B162	ADAPTER, OIL SEAL		
14	1	6144D161	PLATE, REVERSE		
13	1	6144D167	COUPLING, INPUT		
12	24	6144C157	OIL, REVERTER, OUTER		
11	12	6144C154	OIL, REVERTER, CENTER		
10	1	6144D130	SPACER, CAM		
9	2	6144E115	PLATE, CAM, OUTER		
8	2	6144E116	PLATE, CAM, INNER		
7	1	6144E113	POD, OUTPUT		
6	1	6144E114	ASSY, OUTPUT RING		
5	1	6144E111	ASSY, OUTPUT CASE		
4	1	6144E117	ASSY, CRANK, SHIRT		
3	1	6144E118	ASSY, NUT		
2	1	6144E112	ASSEMBLY HOUSINGS		
1	1	6144E115	HOUSING, OUTPUT		

Figure 39

LIST OF MATERIAL			
QTY	DESCRIPTION	DATE	UNIT
1	WASHER, PLAIN		
1	BOLT, MACHINE		
1	BOLT, MACHINE		
1	OIL SEAL		
1	BALL BEARING DUP		
1	BALL BEARING		
1	PLUG, PIPE, HEX SOCK		
1	PLUG, PIPE, HEX SOCK		
1	PLUG, PIPE, HEX SOCK		
1	PLUG, PIPE, HEX SOCK		
1	O RING		
1	O RING		
12	NEEDLE BEARING		
12	RETAINING RING		
1	O RING		
12	NEEDLE BEARING		
6	BOLT, MACHINE		
1	RPLS 27/8		
1	KLS 27/8		
20	WASHER, PLAIN		
8	NUT, SELF LOCKING		
10	WASHER, PLAIN		
8	BOLT, MACHINE		
1	O RING		
1	OIL SEAL		
1	O RING		
1	RETAINING RING		
8	BOLT, MACHINE		
1	ROLLER BEARING		
18	PLUG, STRAIGHT		
1	KD 55CP		
1	NEEDLE BRG		
1	LOCKWEE		
2	RPS ROLLER, INNER		
1	O RING		
1	RETAINING RING		
2	BRS, ROLLER, OUTER		
12	THRUST BEARING		
20	THRUST RING		
4	SPRING, COIL		
4	WASHER, LOCK		
4	RETAINED RING		
1	PLATE, NAME		
20	SPACER, ROLLER		
1	SHIM, LOWER REG		
1	COVER, REV. DOCK		
10	SUPPORT FIXED PIN		
12	WASHER, THRUST		
1	COVER, OIL SEAL		
1	SHIM, LOWER		
1	SEAL, CRANK SHAFT		
48	BEARING, REVERTER		
20	SHAFT, FIXED PIN		
1	CAP, UPPER		
20	ASSY, ROLLER, FIXED PIN		
12	SMALL, REVERTER		
1	ADAPTER BEARING		
1	SPACER, BEARING		
1	ADAPTER, OIL SEAL		
1	PLATE, REVERSE		
1	COUPLING, INPUT		
24	OIL, REVERTER, OUTER		
12	OIL, REVERTER, CENTER		
1	SPACER, CAM		
2	PLATE, CAM, OUTER		
2	PLATE, CAM, INNER		
1	POD, OUTPUT		
1	ASSY, OUTPUT RING		
1	ASSY, OUTPUT CASE		
1	ASSY, CRANK, SHIRT		
1	ASSY, NUT		
1	ASSEMBLY HOUSINGS		
1	HOUSING, OUTPUT		

C

individual cam plate is located radially by the associated crank bearings and a set of 12 reverter bearings. When cam follower bearings are used, and assuming clearance exists between the cam profiles and the reaction rollers, then the crank bearing alone affords radial location of the cam plate.

Thus, two separate conditions exist in the two units. In each case, the clearance between the cam lobes and the reaction rollers amounts to approximately 0.004 inch, while 0.0025/0.0035-inch diametral clearance exists in each set of crank bearings.

In the case of S/N 101, the eccentric reverter bearings had approximately .0005 - inch clearance with the cam plate reverter bores. Thus it can be seen that in S/N 101 the clearance between the cam profiles and the reaction rollers and the clearance in the crank bearings were of little importance as regards radial location of each cam plate. In this case, each cam plate was well located by the reverter bearings.

However, in S/N 102 with the cam follower reverter bearings installed, the clearance between the cam profiles and the reaction rollers ("A" dimension in Figure 40, page 48) can allow each cam plate to wander off center by the amount of the crankshaft bearing clearance.

When this is realized, it is obvious that reaction to the drive load at the reverter pins can cause displacement of the cam plates until the crank bearing clearance "C" is taken up. Rotation and displacement of the cam plates then occurs as the remaining clearance (A-C) is taken up. The plates are not displaced with respect to the crankshaft.

The cam-plate "orbit" as the crankshaft revolves will now be dictated by the difference in diameter between the cam follower type reverter rollers and the reverter holes in the plates. Although this has essentially the same magnitude as the crankshaft throw, the cam-plate orbit is now offset to the crank throw orbit by an amount equal to the initial clearance "A".

Figure 40, page 48, and Figure 41, page 49, illustrate the "at rest" and "running" conditions which were obtained with S/N 102. The two diagrams are drawn for the same basic instantaneous position of a cam plate, but Figure 41, page 49, illustrates the displacement which can occur in the cam plate under driving conditions.

Typically, when drive load reaction is present, the cam plates are forced to oscillate around the reverter rollers in an offset condition with respect to the crankshaft. The rigidity of the reverter pins maintains the offset condition and thus the cam plate and crankshaft "orbits" interfere.

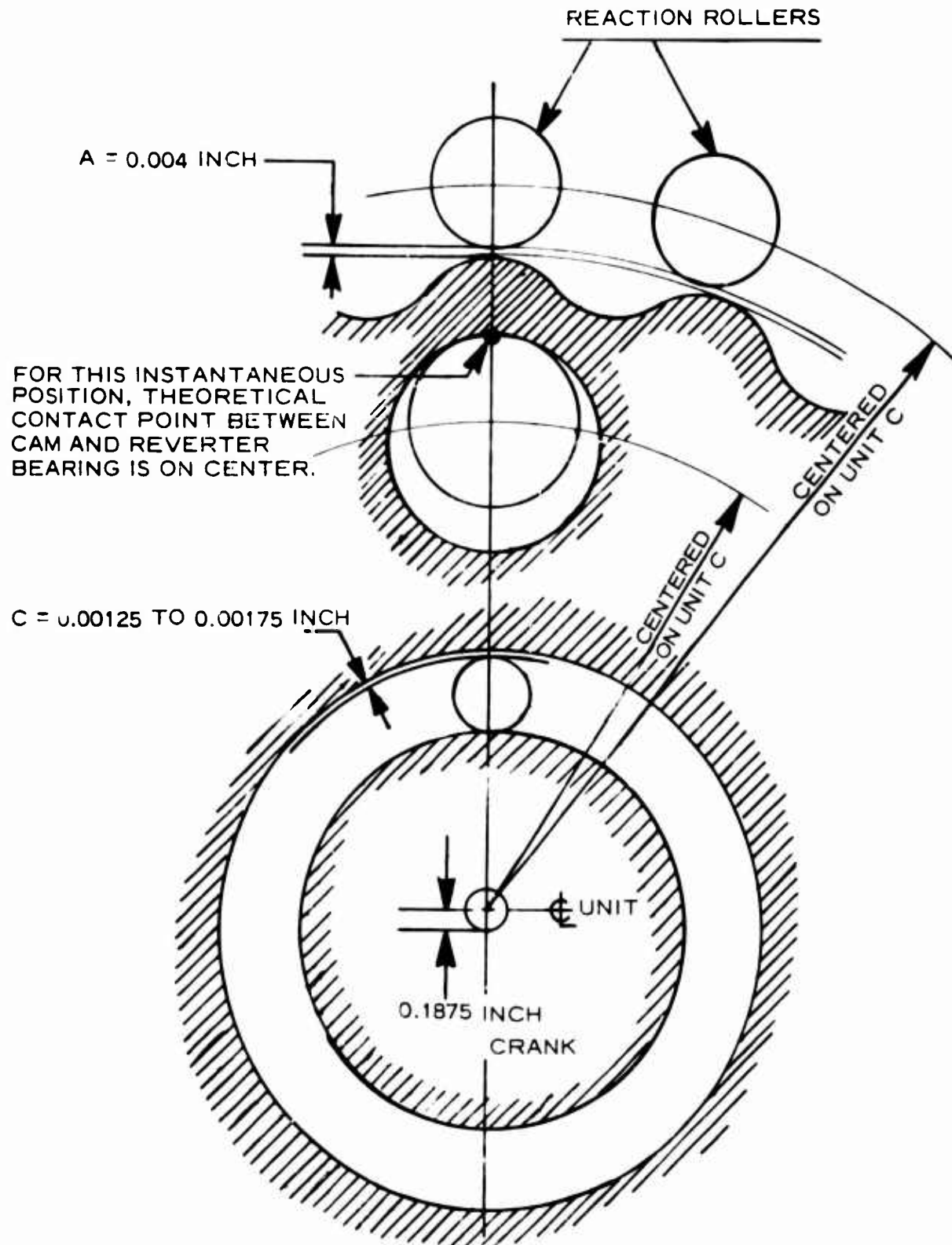


Figure 40. Bearing Clearances in Relation to Cam Plate in At "Rest" Position.

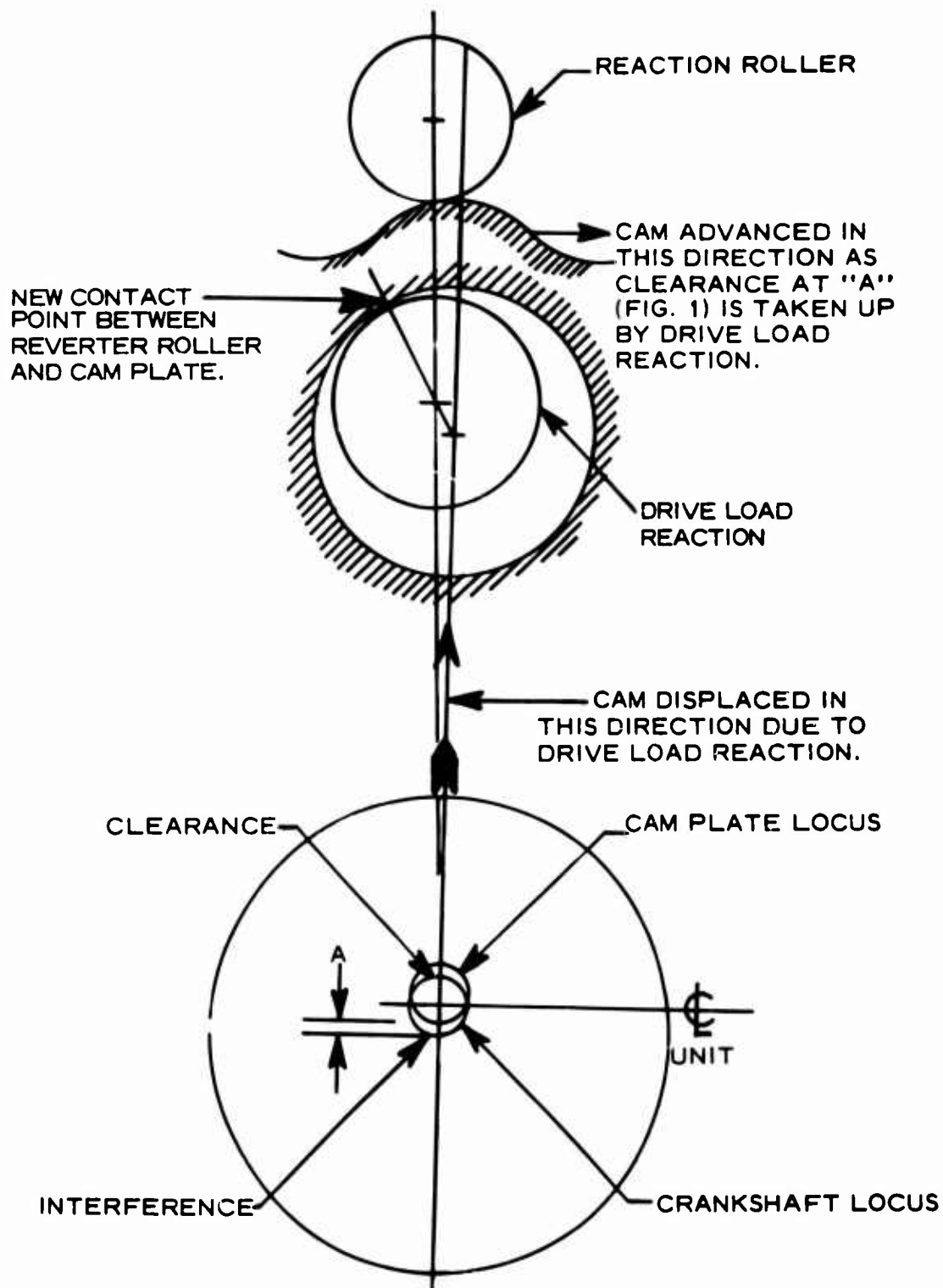


Figure 41. Displacement of Cam Plate in The "Running"/Operating Position.

The initial clearance "A" amounts to some 0.004 inch. Thus 0.004 inch of interference can occur between each cam plate and crankshaft "orbit" as the system revolves. Clearance in the crankshaft support bearings will tend to reduce this interference.

Figure 41, page 49 , shows one instantaneous position of cam plate and crank. It can be demonstrated that the "interference" in respective orbits travels around the crankshaft as the crankshaft makes one revolution.

Conclusions

It seems obvious that the clearances in the system must be reduced. Ideally, no clearance should exist in the crank bearings or between the cam profiles and the reaction rollers. However, if the initial clearance at "A" can be held to an amount less than the radial clearance which exists in the crankshaft support bearings, then the system should tend to self-centralize under load and relieve any wear at the crank throws.

Clearance "A" may be reduced by plating the reaction rollers (Figure 42).

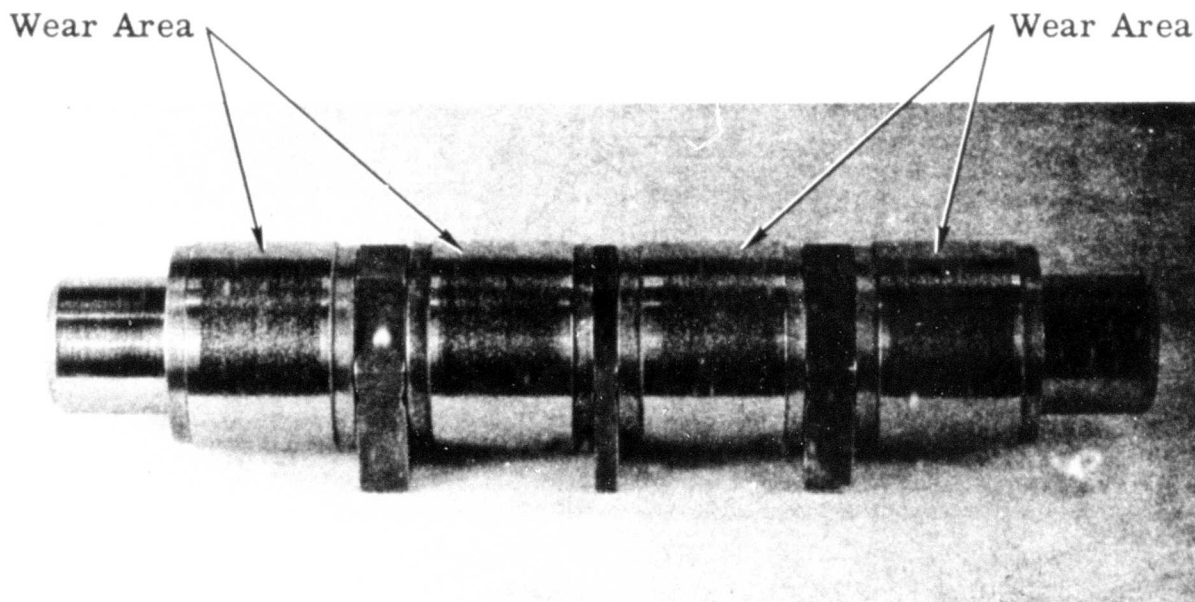


Figure 42. Fixed pin rollers from transmission S/N 102 showing skidding and scuffing.

INVESTIGATION OF POSSIBLE CAUSES OF CRANKSHAFT WEAR FOR S/N 102 TRANSMISSION

1. Crankshaft Load Evaluation

An examination of the cycloidal cam after testing revealed severe wear on those crankshaft surfaces which served as roller bearing inner races. It was thought that possibly some loads existed during operation that had not been anticipated during the design examination. Several possible causes of extra loads are listed below with conclusions.

Possible Causes of Extra Load

- a. Differential thermal expansion - cam plates growing with respect to housing and outer pins.

Conclusion: If this loading existed, the forces involved were not significant. The net effect should have been to cause loading on the side of the crank opposite the point of maximum throw. The main wear did not occur there.

- b. Centrifugal force to keep cam plate in 3/16 inch orbit.

Conclusion: This loading is a function of speed only, and is (at full speed) about 1/4 of the transmitted load (at full power). The wear pattern should be the same as in "a" above. This loading was anticipated and included in the design calculations.

- c. Inclination of line of action of force on crankshaft.

Conclusions: Observation of the geometry of the cycloidal cam model indicates that the effective line of action of the housing reaction force on each cam plate is inclined about 50 degrees above, and not at right angles to, the crank throw. A load chart is shown in Figure 43, page 52. Such a condition causes the bearing load to be about one and one-half times that obtained if the line of action is assumed perpendicular to the crank throw. The additional force is not sufficient to have caused the bearing wear.

General Conclusions Concerning the Extra Loads

No extremely large forces capable of causing the observed wear can be found among the forces examined above.

2. Material Compatibility

Since the severe wear occurred on the crankshaft bearing inner race and none on the outer race, a study was made on the inner and

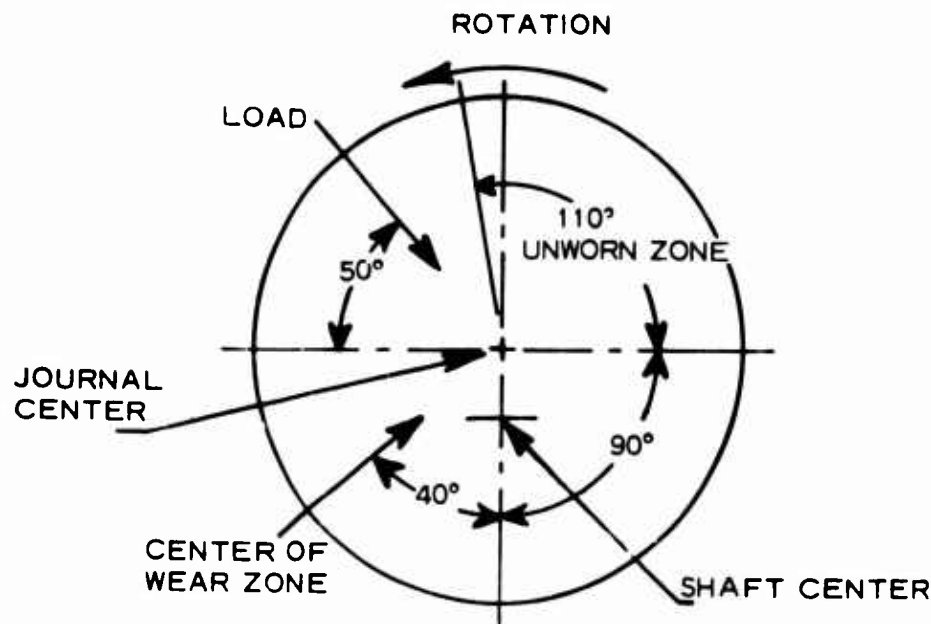


Figure 43. Loading and wear zones for crankshaft.

outer race stresses and of the material compatibility. The following stress calculations show that the contact stress on the inner race is only about 10 percent greater than that on the outer race.

Wear Tests

In order to test the hypothesis that the material combination of the crankshaft and rollers resulted in more wear than that of the cam plate and rollers, wear tests were made. The tests were run on a Macmillan Wear Test Machine. The Macmillan Wear Test Machine rubs a steel cylinder on the specimen to be wear tested. Loading is by dead weights acting through a compound lever system. Lubricant placed in the aluminum cup is carried to the wear zone by rotation of the cylinder.

The outer surface of the cylinder which rubs on the test specimen is a Timken cup T-54148, composed of 52100 steel. Approximately cubical wear specimens were cut from the crankshaft and cam plate. The bearing rollers fit into the machine with little alteration. Load weights were adjusted to compensate for the different curvatures of the rollers, cam lobes, and crankshaft, so that 100,000 p.s.i. contact stress was maintained, (Figure 44, page 53).

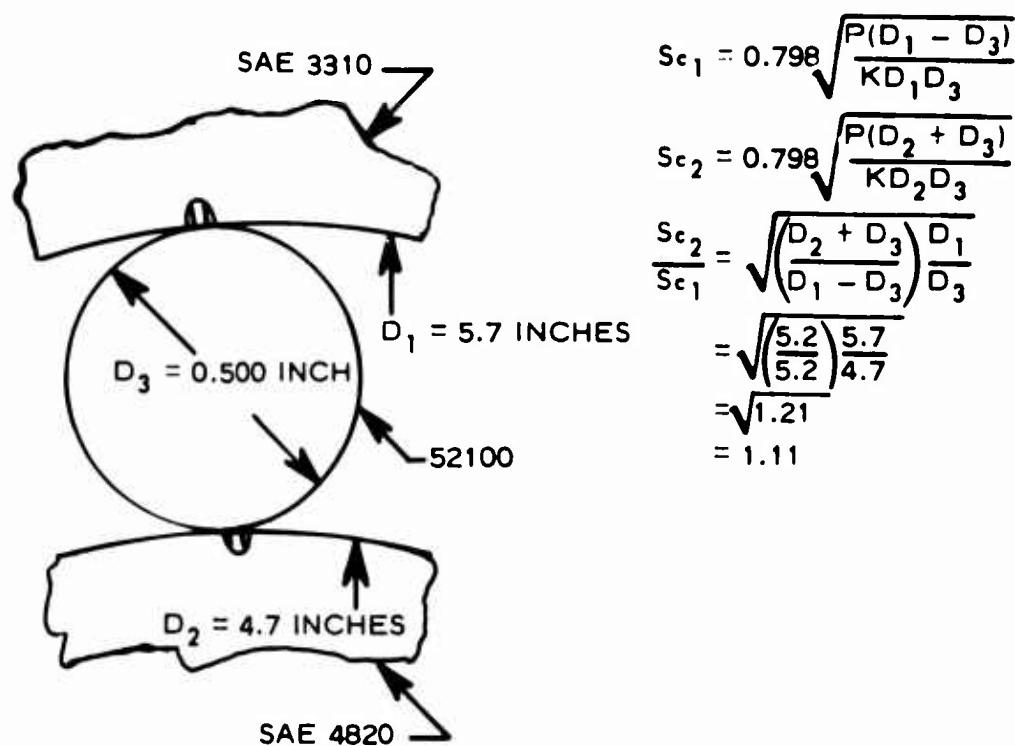


Figure 44. Roller crank bearing Hertz (contact) Stress ratio relationship.

The test conditions were as follows:

1. Lubrication - MIL-L-7808 (both fresh oil and used oil from the cycloidal cam unit test).
2. Wear Ring Material - 52100 Steel (Rc 60).
3. Sliding Velocity - 209 feet per minute.
4. Contact Stress - 100, 000 p. s. i.
5. Time of Test - 8 minutes per test.

The following observations were made based on three cam plate, three bearing, and eight crankshaft tests (Figure 45, page 54).

- A. No significant difference was noted between the wear of the crank and of the cam plate.
- B. Bearing rollers seemed to wear slightly less than either of the above members.
- C. In all cases, the wear patterns due to sliding did not resemble the smooth, polished appearance of the worn crankshaft.

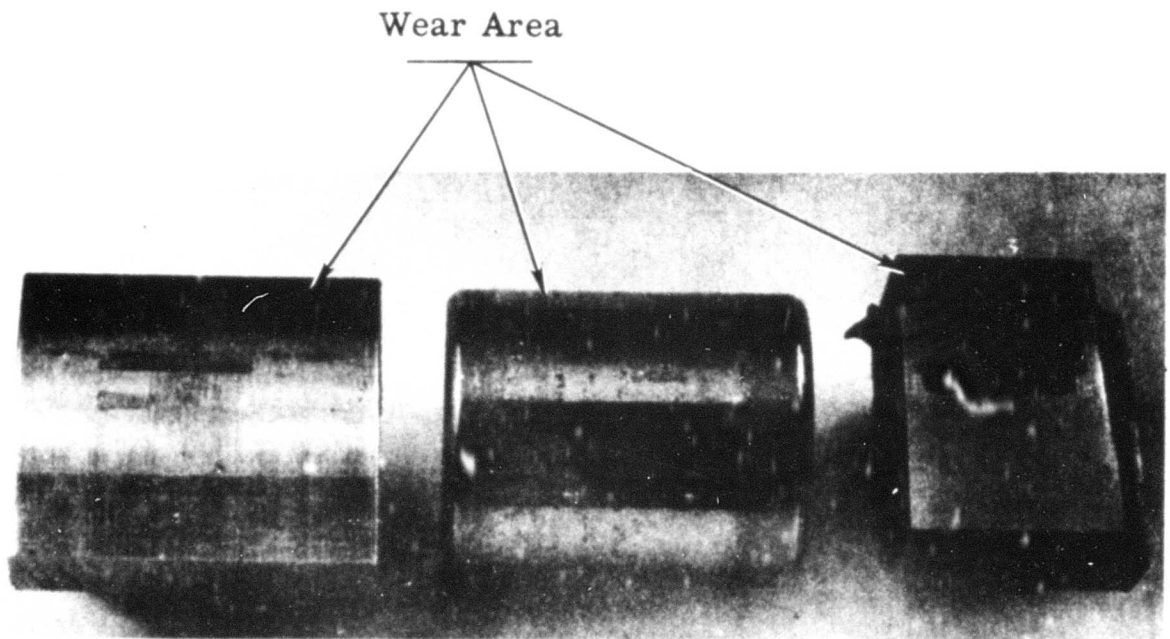
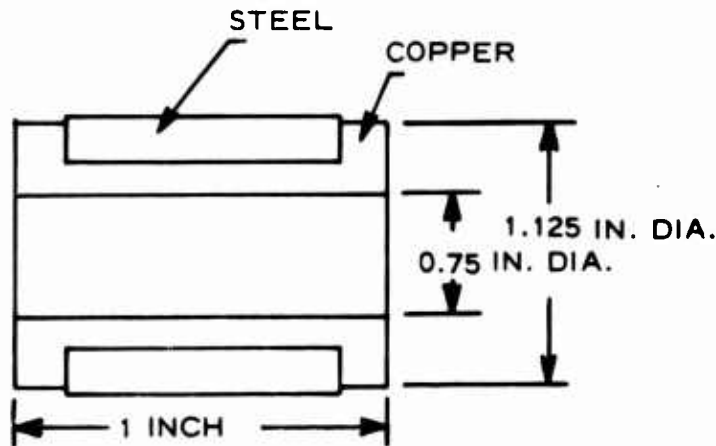


Figure 45. Wear test specimens with normal wear patterns from a cam plate, crank roller bearing, and the crankshaft.

Conclusions:

There is no indication that the combination of crankshaft and roller material results in radically greater wear than the combination of roller and cam plate material.

3. Calculation Relevant to Wear on the Outer Lobes of the Cam Plate
 - a. In Figure 46 a calculation is shown of the net tractive force necessary to accelerate the outer rollers from rest to 6,000 r.p.m. in the time required for the abraided zone to pass by. This tractive force is the net force after friction with its supporting pin accounted for.



$I = \text{Moment of Inertia} = .98 \times 10^{-4} \text{ lbs-in-sec}^2$
Calculation of Net Tractive Force

$t_1 = \text{time during which acceleration through abraided zone takes place} = .0015 \text{ sec.}$

$V_1 = \text{peripheral velocity of roller} = 177 \text{ in/sec.}$

$W_1 = \text{angular velocity of roller} = V/.562 = 314 \text{ radians/sec.}^2$

Net torque necessary to accelerate roller $= I W_1 / t_1 = 20.5 \text{ inch-pounds.}$
 Net tractive force required $= 20.5 / .562 = 36.4 \text{ lbs.}$

Calculation of Normal Acceleration Tractive Force

$t_2 = \left(\frac{1}{6000} \text{ minutes/rev} \right) (60 \text{ seconds/minute})$
 $(1/20 \text{ rev/cycle}) = 5 \times 10^{-4} \text{ seconds/cycle}$

$V_2 = \text{peripheral velocity of roller} = 26 \text{ in/sec.}$

$W_2 = \text{angular velocity of roller} = .26 / .562 = 46.2 \text{ radians/sec.}$

Normal Acceleration Tractive Force Required

$$F = \frac{(.98 \times 10^{-4}) 46.2}{5 \times 10^{-4} \times .562} = 16 \text{ lbs.}$$

Figure 46. Calculations for acceleration tractive force on fixed pin rollers.

The force called normal tractive force is the maximum net tractive force required to accelerate the roller during its normal cycle of speed change from rest to 6,000 r.p.m. and back to rest during each input shaft rotation.

- b. Figure 47 is a quick calculation to the allowable normal force that can be reacted by an outer roller if the contact stress is limited to 300,000 p.s.i. and the width to 1/2 inch. Since the maximum load is about 1,000 pounds, it is concluded that 1/2 inch is sufficient width for these rollers. This is a net reduction of length of 1/2 inch.
- c. Figure 43, page 52, illustrates the relationship between the direction of rotation of the crank within the cam plate, the journal throw, the unworn zone, the center of wear on the crankshaft, and approximate direction of loading from the sum of the outer roller loads. This latter load direction is only approximate. The exact loading on the crankshaft depends also on the reverter pin loading, which is assumed here to be very nearly a pure torque load.

METALLURGICAL EXAMINATION OF CRANKSHAFT FOR S/N 102 TRANSMISSION

A section was removed from the 3-inch-diameter bearing-throw surface. This was the throw furthest away from the splined area. The bearing surface showed a large amount of wear (approximately 0.007 inch). A photomicrograph, Figure 48, page 57 of the transitional area between the worn and unworn surfaces did not show anything unusual.

Consider maximum contact stress,
 S_c , of 300,000 psi.

$$S_{c, \max} = .591 \sqrt{\frac{PE}{D_1 - D_2}} = 300,000$$

Effective radius diagram

(Reference: Formulas for Stress & Strain
 Roark, R.J. - p. 289)

where

$E = 30 \times 10^6$ psi = Youngs Modulus

P = Allowable load in lbs/inch

D = Diameter in inches

$$P = \frac{300,000^2}{.591} \quad D_1 - D_2 = \frac{E (D_1 - D_2)}{}$$

$$\frac{25.64 \times 10^{10}}{30 \times 10^6} \quad \frac{1.125 \times 1.26}{1.125 \times 1.26}$$

$$= .509 \times 10^4 = 5090 \text{ lbs/inch}$$

For Roller = 1/2 inch wide

$P_{\text{allowable}} = 2545 \text{ lbs.}$

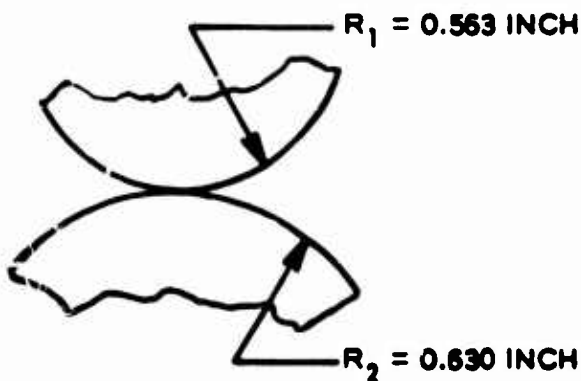


Figure 47. Contact stresses for fixed pin rollers.

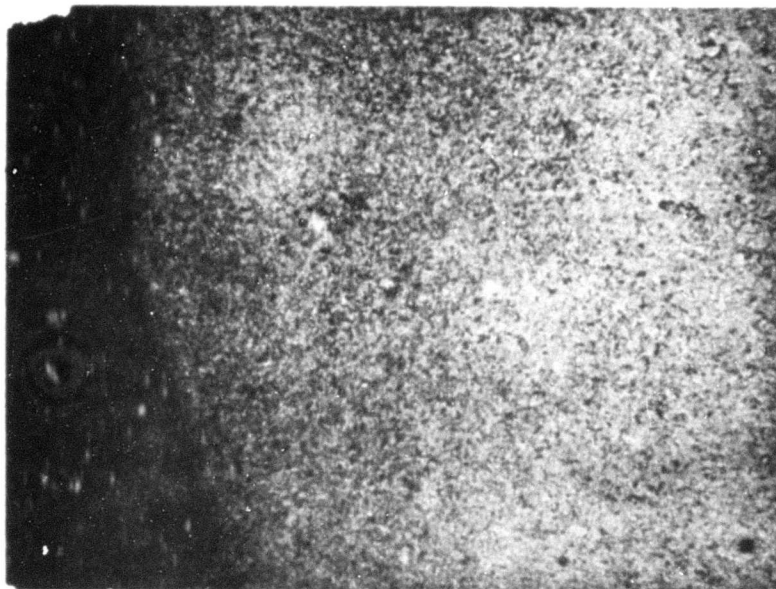


Figure 48. Transition area between worn and unworn surfaces. 100X

A photomicrograph, Figure 49, of the core area revealed a typical core structure for the type of heat treatment. The core structure consisted of a low-carbon, tempered martensite. No free ferrite was noted in the core structure; thus, it may be concluded that the quench rate was above the critical rate.

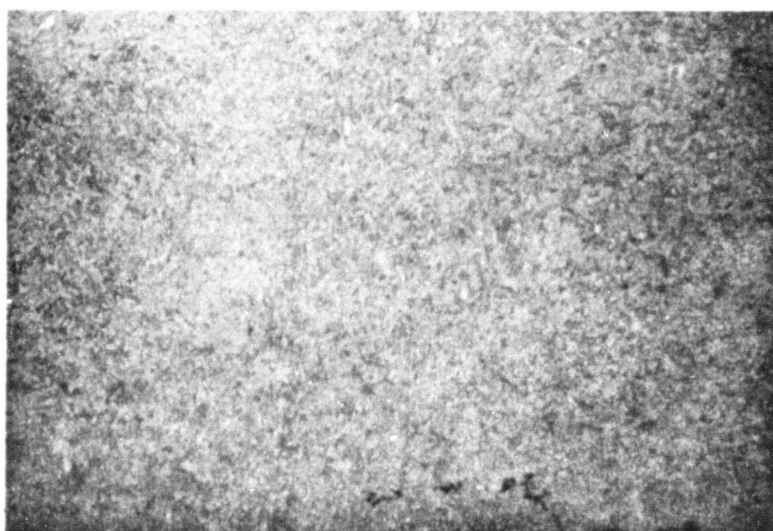


Figure 49. Core area 100X.

The case, Figures 50 and 51, consisted of very fine tempered martensite platelets. A hardness survey was made using a Tukon hardness testing machine with a Knoop indenter and a 500-gram load.

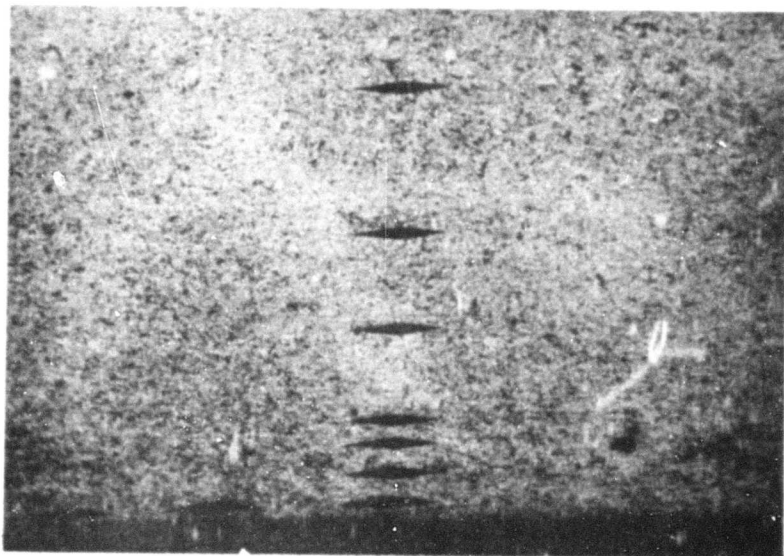


Figure 50. Unworn surface 100X.

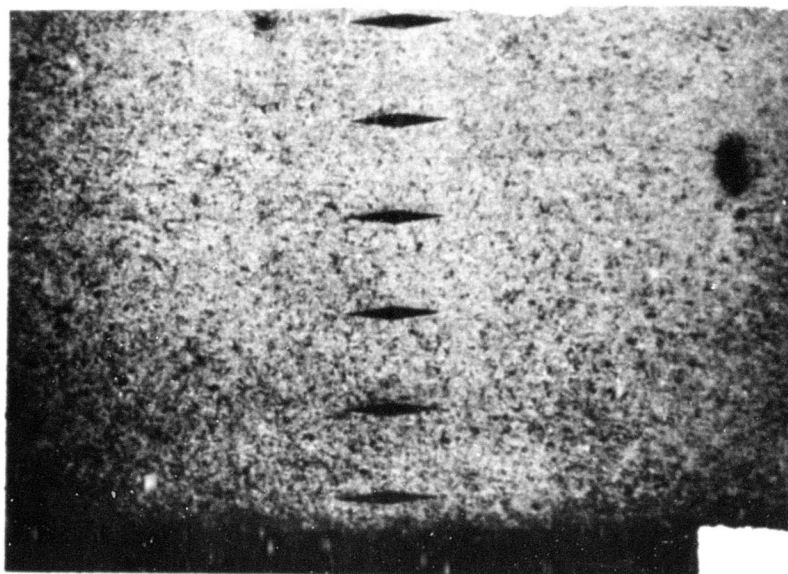


Figure 51. Worn surface 100X.

The results of this survey may be found in Figure 52. Hardness measurements taken from the unworn surface show a hardness of Rockwell C 60 to 62 approximately 0.002 inch below the surface. At a distance of 0.025 inch below the surface, the hardness was Rockwell C 56. The case hardness dropped below Rockwell C 50 at approximately 0.047 inch. Core hardness measured on the Rockwell A scale and converted to Rockwell C was 32 to 34.

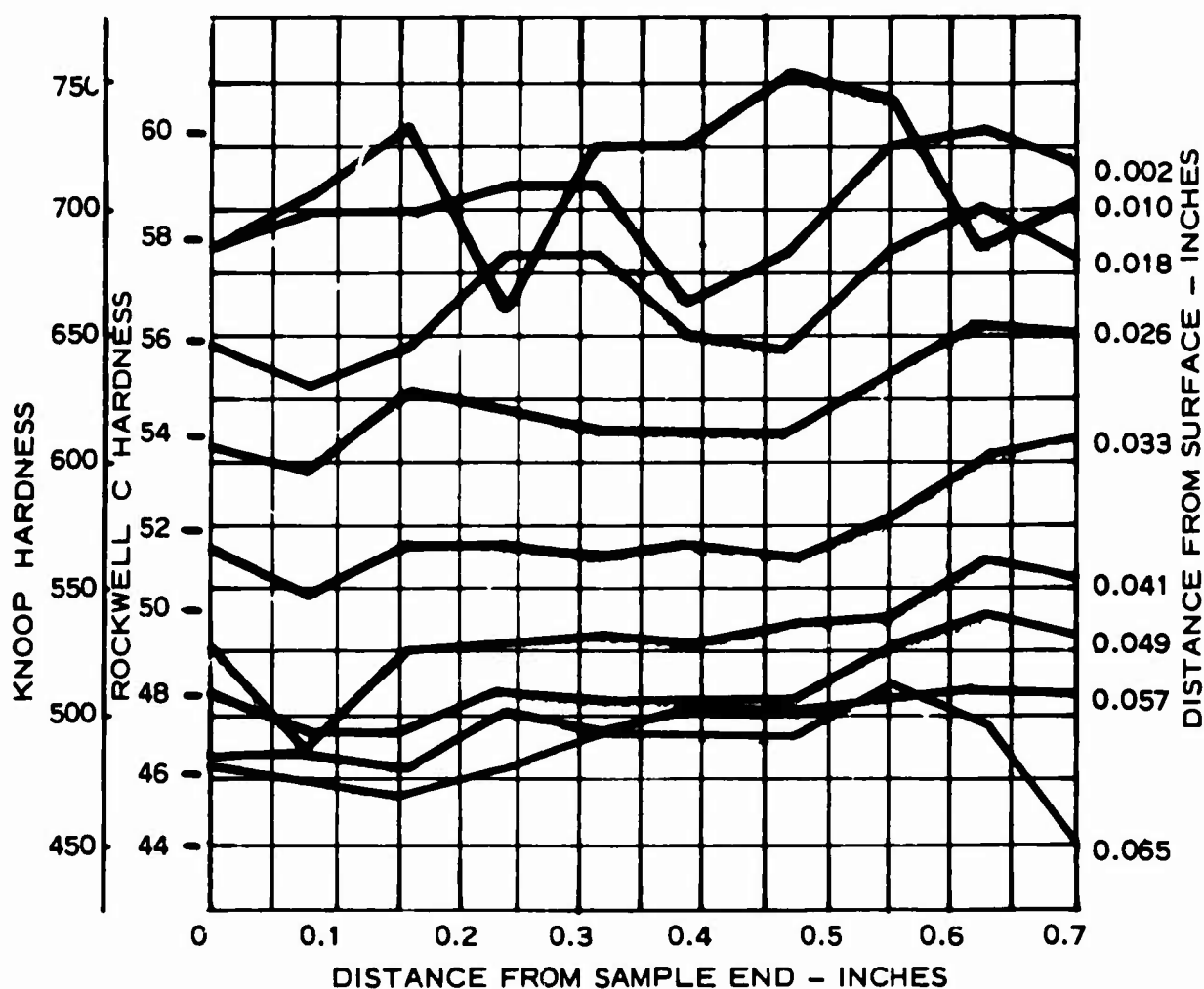


Figure 52. Hardness survey of crankshaft from transmission S/N 102. Material was SAE 4820 carburized casting with core hardness of 32 - 34 Rc.

Photomicrographs, Figures 53 and 54, were taken of the area surrounding an oil hole which was added subsequent to carburizing and tempering. The hole was lined with white particles which were assumed to be ferrite. These particles were not observed in the first 0.015 inch in from the surface. This ferritic condition in the hole would be expected; however, it is not thought that this would have any detrimental effects.

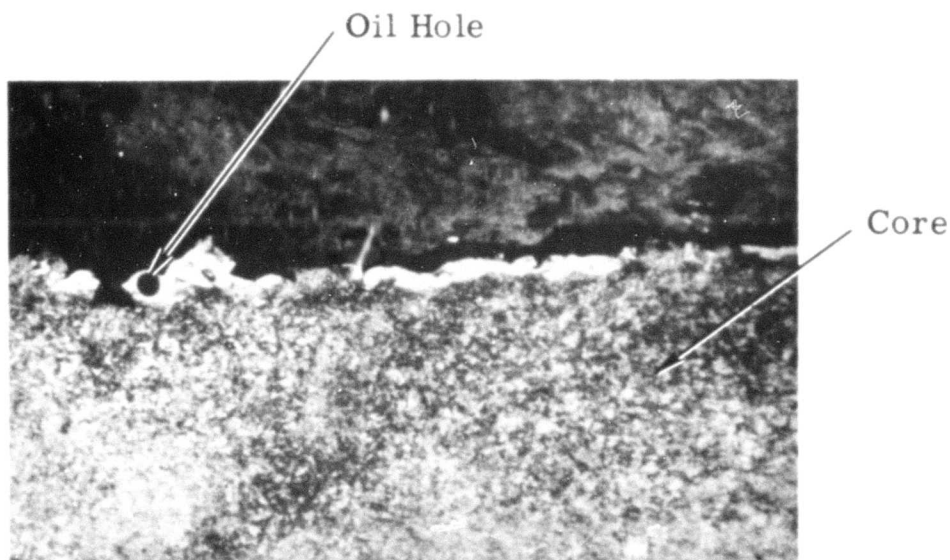


Figure 53. Oil hole and core 100X.

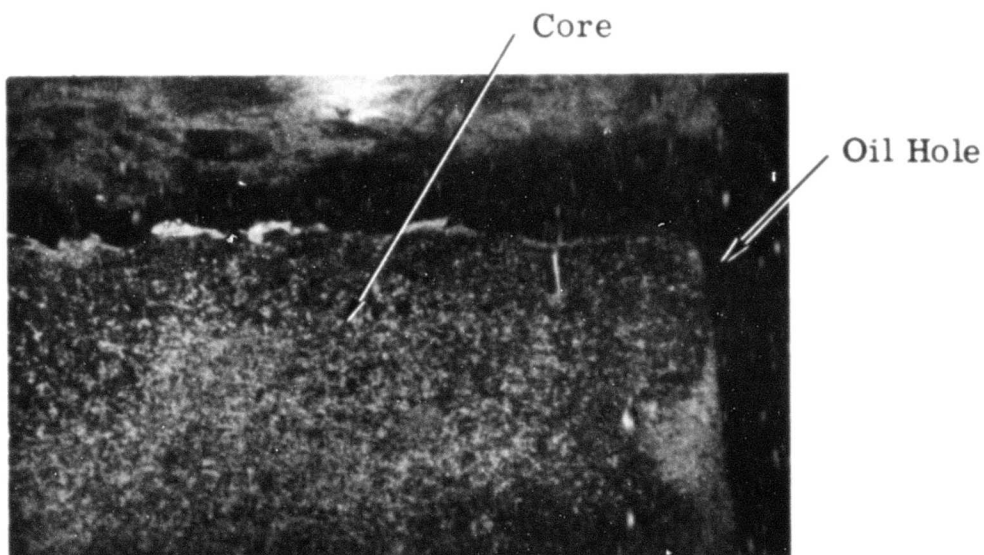


Figure 54. Oil hole and core 100X.

The crankshaft was X-rayed and found to be free of defects. The material was chemically analyzed and found to be within the acceptable limits for SAE 4820 steel.

Some of this wear is attributed to the oxidation effect of MIL-L-7808D oil with the bronze cages which in turn could cause roller skewing. The wear particles tend to form a sludge in the oil which creates a lapping compound. Most of the sludge shown in Figure 55 is magnetic in nature. The magnetic material probably came from the fixed pin rollers since they showed an appreciable amount of wear. The lapping or grinding action is shown by the tract wear (Figure 56, page 63).



Figure 55. Various drain plugs from the lubrication system of transmission S/N 102 showing wear particles after load testing.

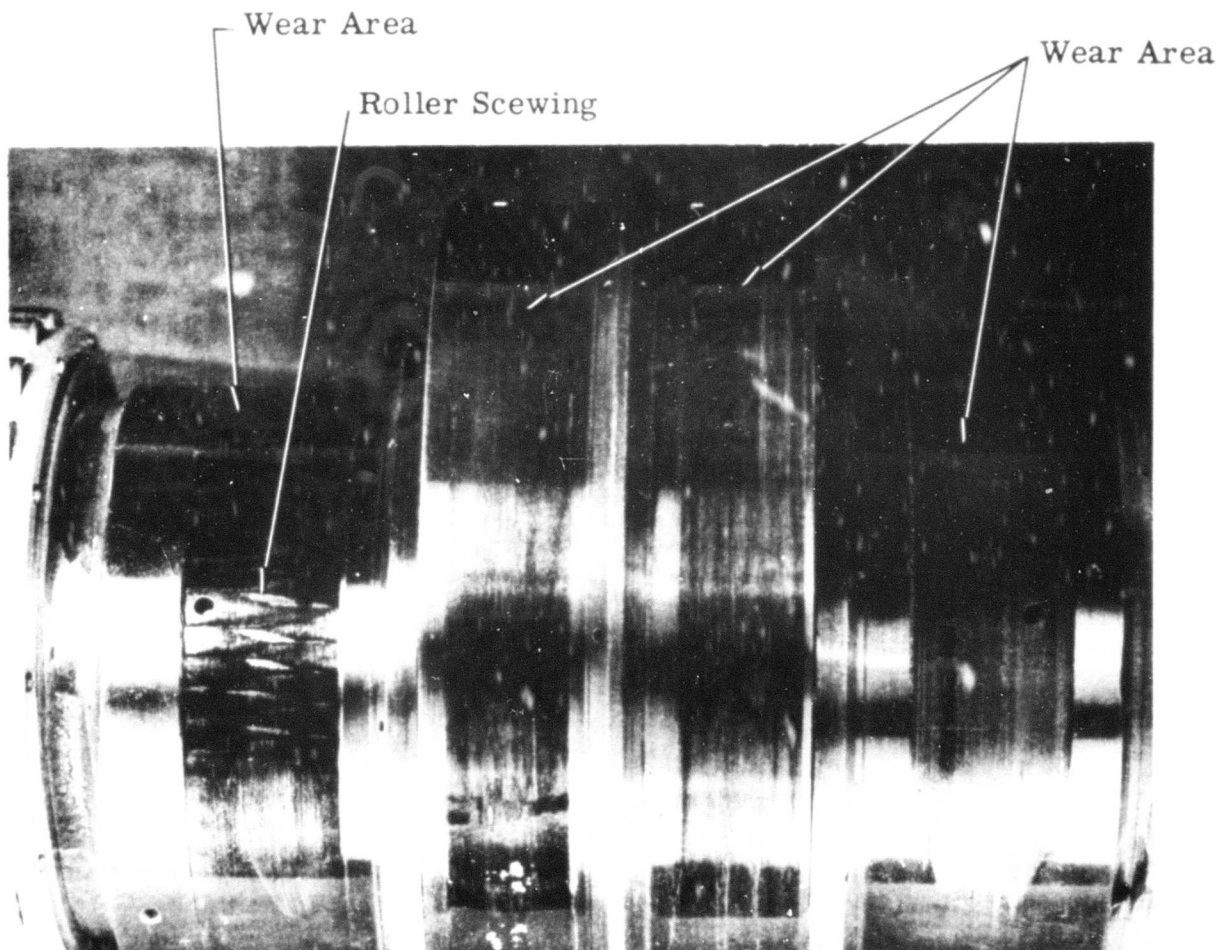


Figure 56. Crankshaft from transmission S/N 102.

LUBRICATION CALCULATIONS FOR TRANSMISSION S/N 102

The quantity of oil needed for the transmission increases with temperature, load, speed, and bearing size. The optimum quantity required for a particular application can be determined only by experimentation. As a general guide the following formula was used in establishing oil quantity, bearing in mind that the more extreme the conditions, the more oil is required.

<u>Method of Application</u>	<u>Oil Quantity</u>	<u>Load Conditions</u>
Jet and Spray	(ML/Minute) 5D* to 15D	Moderate to Heavy

*where D = bore in millimeters

TABLE 2
THEORETICAL OIL SUPPLY

Bearing (No.)	Quantity	Oil Supply (ml/minute)
XLS 2-7/8	1	365
RXLS 2-7/8	1	365
XLS 2-3/4	1	350
1313AR-RA	2	1200
1021AR-RA	2	830
SKF 7215 BY/G	2	750
MR 1917	1	425
MS 305-901	48	5300
6144C97	80	7200
KD55CP	1	700
HJ263520	1	205
J108	12	960
HJ142216	12	1320
NTA 1018	12	<u>960</u>
TOTAL		20,930

Conversion: 3784 ML = 1 gallon

$$\text{Minimum Flow} = \frac{20930}{3784} = 5.5 \text{ g.p.m.}$$

$$\text{Maximum Flow} = 3 \times 5.5 = 16.5 \text{ g.p.m.}$$

Somewhere in between these two values would be the optimum flow for the transmission. An educated guess would be 7 g.p.m. The actual oil flow used was 10 g.p.m.

For speeds in excess of 600,000 DN (D = bore in mm, N = RPM), it becomes necessary to use great care in selecting and applying ball or roller bearings. The higher the speed, the more sensitive the bearing is to the adverse effects of misalignment, poor lubrication, improper fitting practice, poor heat dissipation, and contamination. In the present design, the crank bearings fall into this class

Bearing	DN		
1313AR-RA	120 mm x 6000	=	720,000
1021AR-RA	83 mm x 6000	=	500,000

In examining the bearings from transmission S/N 101, there were indications that more oil would be required and directed into the start of the loading zone. This was accomplished by drilling two oil holes (.093 inch diameter) radially spaced in the unloaded under each crank bearing (opposite the throw). In determining orifice size, standard tube selection data and tube size charts were used for rough approximations.

TABLE 3
THEORETICAL ORIFICE SIZE

Bearing (No.)	Flow (g.p.m.)	Orifice (in.)
XLS 2-7/8	.0965	.030
RXLS 2-7/8	.0965	.030
XLS 2-3/4	.0925	.030
1313 AR-RA	.95	.085
1021 AR-RA	.66	.068
MR 1917C	.113	.030
MS 305-90	.115	.030
6144C97	.0715	.030
KD 55CP	.184	.038

Crankshaft for Transmission S/N 102

Inlet Area

$$2 \text{ key ways} \quad .375 \times .187 \times 2 \quad = \quad .070 \times 2 \quad = \quad \underline{\underline{.140 \text{ in.}^2}}$$

Orifice Areas (Outlets)

$$\text{Crank bearings} \quad 8 \quad .093 \text{ dia} \quad .0069 \times 8 \quad = \quad .0562 \text{ in.}^2$$

$$\text{Spacers} \quad 4 \quad .0625 \text{ dia} \quad .0031 \times 4 \quad = \quad .0125 \text{ in.}^2$$

$$\text{Bearing Spacers} \quad 24 \quad .055 \text{ dia.} \quad .00235 \times 24 \quad = \quad \underline{\underline{.0064 \text{ in.}^2}}$$

$$\text{TOTAL ORIFICE AREA} \quad = \quad .1350 \text{ in.}^2$$

Therefore, areas balance out.

Reverter Shafts for Transmission S/N 102

Inlet Area

$$\text{One } 1/2 \text{ inch NPT} - \text{Housing} \quad = \quad .1964 \text{ in.}^2$$

Orifice Areas (outlets)

Reverter Cage

$$12 - 3/16 \text{ holes} \quad .0276 \times 12 \quad = \quad .3312 \text{ in.}^2$$

$$8 - 3/8 \text{ holes} \quad .1105 \times 8 \quad = \quad .8840 \text{ in.}^2$$

Therefore no restrictions for inlet oil.

Adapter Seal

$$8 - 3/16 \text{ holes} \quad .0276 \times 8 \quad = \quad .2208 \text{ in.}^2$$

Therefore no restrictions for inlet oil.

Reverter Shafts

$$4 - .0625 \text{ holes} \quad .0031 \times 4 \quad = \quad .0124 \text{ in.}^2$$

$$12 \text{ Shafts} \quad .0124 \times 12 \quad = \quad .1488 \text{ in.}^2$$

Crank Bearing - (Outer Spider)

$$4 - .0625 \text{ holes} \quad .0031 \times 4 = .0124 \text{ in.}^2$$

$$\text{Total Area} = .1488 + .0124 = \underline{\underline{.1612 \text{ in.}^2}}$$

Therefore areas balance out.

Output Spider

$$8 - 1/8 \text{ holes} \quad .0123 \times 8 = .0984 \text{ in.}^2$$

$$8 - 1/8 \text{ holes} \quad .0123 \times 8 = \underline{\underline{.0984 \text{ in.}^2}}$$

$$\underline{\underline{.1968 \text{ in.}^2}}$$

Fixed Shafts for Transmission S/N 102

Inlet Area

$$\text{One } 1/2 \text{ inch NPT} \quad \text{Housing} = .1964 \text{ in.}^2$$

Fixed Shafts

$$4 - .055 \text{ holes} \quad .00235 \times 4 = .00940 \text{ in.}^2$$

$$20 \text{ Shafts} \quad 2 \times .00940 = \underline{\underline{.188 \text{ in.}^2}}$$

Therefore areas balance out.

Crankshaft and Reverter System for Transmission S/N 102

Inlet Area - Mast

$$\text{One } 1/8 \text{ inch NPT} \quad \text{Housing} = .0123 \text{ in.}^2$$

Adapter Seal

$$8 - 3/16 \text{ holes} \quad .0276 \times 8 = .2208 \text{ in.}^2$$

Therefore, no restrictions for inlet oil.

Revised Inlet Oil Condition, S/N 102

	<u>Area</u>
(1) 1/2 inch NPT	.19635 in. ²
(2) 1/2 inch NPT	.19635 in. ²
(3) 1/8 inch NPT	<u>.01228 in.²</u>
Manifold for the three inlet lines	<u>.30498 in.²</u>
1 inch NPT	<u>.7854 in.²</u>

Therefore, no restrictions for the inlet.

Revised Outlet Oil Condition, S/N 102

3/4 inch NPT	.44179
1 inch NPT	<u>.7854</u>
	1.22719 in. ²
Manifold for the two outlet lines	
1-1/2 inch NPT	<u>1.767 in.²</u>

Therefore, no restrictions for the outlet.

Figure 57, page 69, is a viscosity versus temperature graph for three current aircraft and missile lubricants. Two of the lubricants were tested in the transmission, namely, MIL-L-6086 and MIL-L-7808. Lubricant MIL-L-6086 was considered the better oil for this particular application.

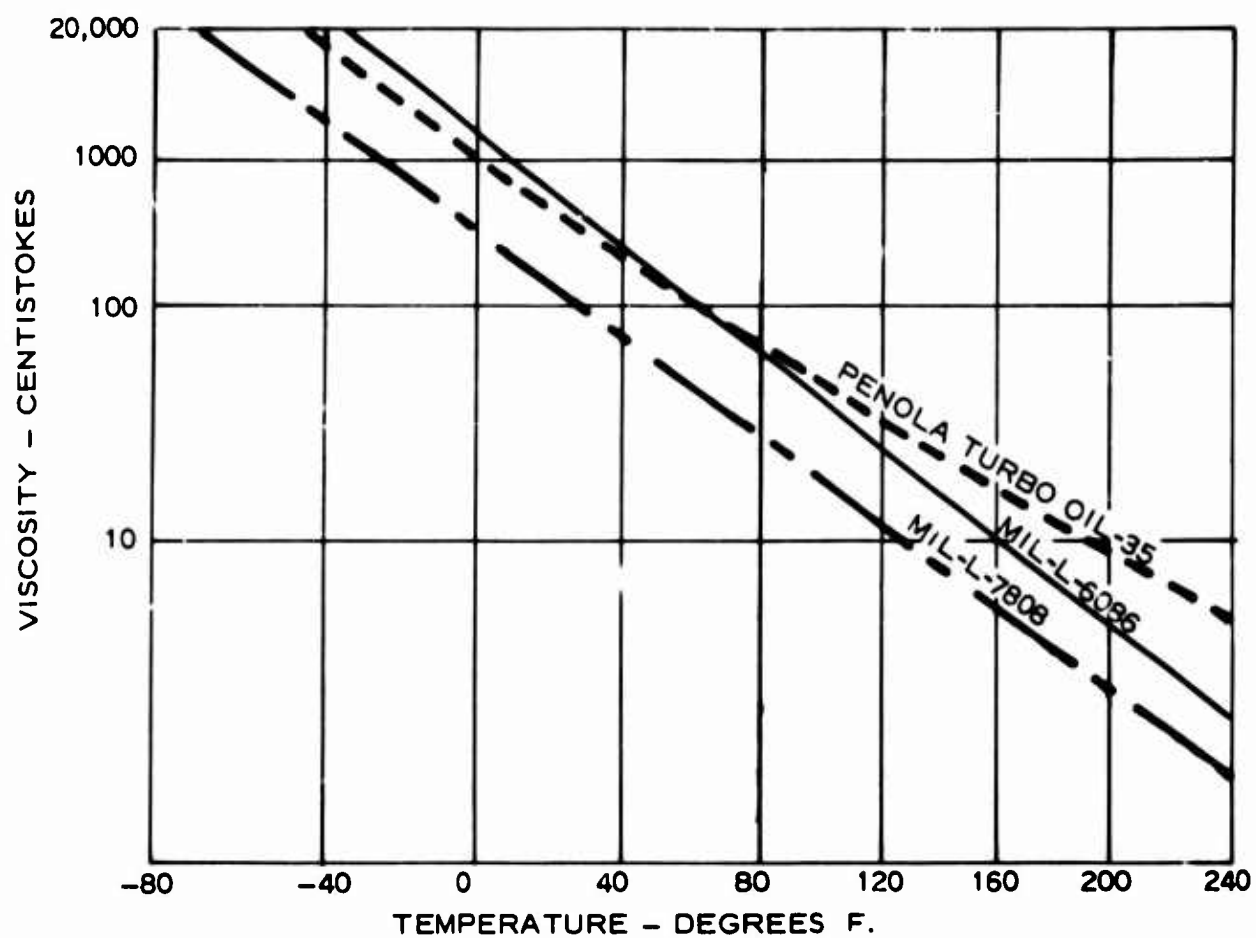


Figure 57. Viscosity versus temperature graph for three transmission lubricants.

FAILURE ANALYSIS FOR TRANSMISSION S/N 102

Crankshaft:

1. The crank bearing roller path wear is not considered a Hertz stress condition.
2. The roller bearing paths show skewing and edge loading of the rollers. The tracks also show that a grinding or lapping action took place.
3. The lapping compound is formed with the oil and wear particles, such as the cage material of the roller bearings and the fixed pin wear (Figure 42, page 50).
4. The roller bearing paths show very slight sliding of the rollers.
5. For 360 degrees of travel, the bearing roller track shows roller "tilt and cocking".
6. The "washboard" effect comes from the same roller hitting in the same place for each revolution of the crankshaft.
7. The bearing roller skid marks show a single impact effect.
8. The bearing rollers show evidence of "locking up".
9. The nature of the wear particles removed from the crankshaft suggests that a solid forging would be better than the present casting.
10. The bouncing and skidding of the bearing rollers is considered a secondary effect. The primary effect is the roller tilt and roller lock.
11. There is a phase lag due to the dragging of the rollers into the load zone.
12. Staggering the two oil holes in the bearing roller track area is undesirable. One oil hole in the center of the track, with special emphasis on having the hole honed in order to avoid any sharp edges for stress risers, would be more desirable.
13. The center of the bearing roller path load zone is 50 degrees from the horizontal. Therefore, the combined resultant load might be 90 degrees from this location.
14. The Coriolis force effect can be neglected. There is no evidence of this effect being present.

Cam Plates:

15. The cam plate crank roller bearing raceways show edge loading of the rollers which is more severe with the two inner cam plates.
16. The cam plate guide plates show lack of crank bearing roller contact. This would indicate too much roller float clearance. The drawing specifications of 0.003 to 0.004 inch should be reduced to 0.001 inch.
17. There is little or no contact in the middle of the crank bearing roller path for the two inner cam plates.
18. The cam plate guide plates rely on the step guide pins for axial location. This should be changed by fastening the guide plates to the cam plates with screws.
19. A taper of 0.001 inch from the vertical should be added to the guide plates. This will allow more surface contact area for the crank bearing rollers and help prevent roller "skew".
20. An oil passage should be added to allow the oil to leave the outer crank bearing raceway. This could be done by adding a V groove by the step pins in the guide plates.
21. The reason the outer crank bearings raceways are in such good condition is because the rollers continuously change relative position. The cam plates rotate at output speed while the crankshaft rotates at input speed.

Crank Bearing Cages and Rollers:

22. The rollers show a tendency to move toward the input side of the transmission.
23. There is somewhat excessive cage-pocket wear. Possibly this wear could be reduced by silver plating.
24. The calculated cage unbalance is about 30 g. Bronze cages can take about 300 g so the present cages are considered adequate strength-wise.
25. The major loading in the cages was in the pockets. The wear indicates the rollers were "rocking". The ends of the pocket show wear.
26. The outer cam plate crank bearing rollers show less tilt action than the inner.

27. The outer cam plate crank bearings indicate adequate guiding, but the inner do not.
28. The centrifugal force (possibly 300 g) on each roller created the mechanism for the lapping process as noted on the crankshaft. The lapping action was very gradual. The bronze dust and steel dust mixture increased with time which in turn intensified the crankshaft wear.
29. The "skewing skidding" action of the rollers can be reduced by testing incremental clearance changes such as 0.003, 0.0015, and 0.0005, to 0.0008 inch.
30. Other cage material might be investigated such as "ARMALON". The "ARMALON" offers a resiliancy over the brass cages. If "ARMALON" is used, the cage riding clearance should be 0.025 to 0.030 inch.

Fix Pin Rollers:

31. The scuffing and skidding of the rollers and cam plate lobes does not indicate a Hertz stress problem.
32. The scuffing action comes from the inability of the roller to overcome the frictional drag.
33. The inertia forces can be reduced on the roller by reducing the weight in half. If this is done, the scuffing should be eliminated.
34. As a method of checking whether item 33 is correct, the transmission should not be operated under the no-load condition. The loading speed should be gradually increased to half speed. Then the rollers should be inspected to see if they roll or skid. If no skidding is present, then the speed should be increased to design speed.

Conclusions of Failure Analysis for Transmission S/N 102:

1. The crank bearing solution of reducing radial clearances may lead to turbine engine manufacturing techniques where outer raceways are "egg shaped" and the rollers are preloaded.
2. Roller skidding under no-load conditions might become a very difficult problem to solve and could involve some fancy, exotic methods for solution.

3. As a matter of precaution, it would be advisable to plate all bronze parts when used with MIL-L-7808D oil.
4. For the best grain structure, it would be well to make the next crankshaft out of a solid forging, possibly SAE 9310 steel.
5. During the testing phase, the transmission should not be operated under no-load conditions until all of the test loads have been completed satisfactorily. After the test loading problems have been solved, then the no-load testing should be tackled.
6. The inertia loading on the fixed pin rollers should be reduced by 50 percent.

BIBLIOGRAPHY

BOOKS

Brown, V., Epicyclic Gearing, Kinematics and Torque Distribution,
The Machinery Publishing Co., Ltd. Yellow Back Series No. 33

Bruhn, E. F., Analysis and Design of Airplane Structures,
Tri-State Offset Company, Cincinnati 2, Ohio - 1949

Buckingham, Earle, Spur Gears,
McGraw-Hill Book Company, New York, N.Y. - 1928

Hardy, H. W., Planetary Gearing (Design and Efficiency),
The Machinery Publishing Co., Ltd. Yellow Book Series No. 37

Roark, Raymond J., Formulas for Stress and Strain,
McGraw-Hill Book Company, New York, N.Y., 3rd Edition - 1954

Shaw, Milton and Macks, Fred, Analysis and Lubrication of Bearings,
McGraw-Hill Book Company, New York, N.Y. - 1949

MAGAZINE ARTICLES

Bonin, L.S., "The Planetary Friction Drive",
Product Engineering, October 1959

McKee, L. M., "Ultra High Speed Ball Bearings",
Missile Design & Development, August 1960

Poppings, R., "The Efficiency of Planetary Gear Trains",
The Engineers' Digest, December 1950

Robinson, George A. and Thomson, Robert F. "What Metallurgical
Engineering Can Do for Management",
Metals Progress, September 1962

PAMPHLETS

"Brayco 880 Conojet Lubricating Oil, Gas Turbine, Aircraft MIL-L-7808D",
Bray Oil Company, June 1961

"Bearing Failures and Their Causes",
SKF Industries, Inc. 1953

"Torrington Bearings Catalog 562",
The Torrington Company 1962

"Ball Bearing Troubles",
Marlin-Rockwell Corporation, 1939

PAPERS

- Boyd, Alexander, "A Research Programme on Bearing Materials for Nuclear Reactors", The Canadian Mining & Metallurgical Bulletin, August 1961
- Brass, E. A., "Two Stage Planetary Arrangements for the 15:1 Turboprop Reduction Gear", ASME Paper 60-SA-1, June 1961
- Gross, M. R. and Rawlings, H. M., "Propulsion Gear Materials: Investigation of Case Carburizing Steel and Practices", U.S.N. Engineering Experiment Station Report 910016, December 1958
- Jakobson, Bengt, "Torque Distribution, Power Flow and Zero Output Conditions of Epicyclic Gear Trains", Report No. 10 from the Institute of Machine Elements, Chalmers University of Technology Gothenburg, Sweden, 1960
- White, R. H., "Needle Bearings in Epicyclic Gearing for the Earth Moving and Agricultural Tractor Industries", S.A.E. Paper 476 A, 1962
- White, R. H., "Rolling Element Bearings in Passenger Car Automatic Transmissions", S.A.E. Paper 359F - 1961

APPENDIX

Historical Background

During Phase II of the contract, monthly design meetings were held for the purpose of reviewing progress. Listed below are the highlights.

MARCH 1961 - The basic module design was roughed out based on the general configuration and specifications of a typical helicopter transmission. The design rating was 280 HP at 6,000 r.p.m. input with an overall reduction ratio of 19:1. The maximum design load for the mast was 7,980 pounds vertical and 1,400 pounds horizontal.

Mr. Loren Braren of West Germany, an inventor of industrial cycloidal cam reducers, expressed his interest in exchanging information on the subject.

APRIL 1961 - A feature article on the design of cycloidal cam transmissions was published in the August 18, 1961 issue of "Design News".

Patent applications were initiated for three versions:

Docket No. 4168	Roller Transmission
Docket No. 4169	Right Angle Cam Ring Transmission
Docket No. 4170	Cycloidal Cam Transmission

Preliminary design commenced for the 840 HP version. Three arrangements were submitted to USATRECOM for their review. The 280 HP preliminary design module was also submitted to USATRECOM.

MAY 1961 - The "cloverleaf design" for the 840 version was adopted and detailed. Envelope size, weight and cost analysis were completed. Bearing loads were determined for the 280 HP module with the design based on B₁₀ life of 1000 hours. Other design factors also included were the reaction pin cage arrangement, cam plates, output shaft bearings, crankshaft studies, and reverter eccentric assemblies. Cost analysis of the design factors dictated that various trade-off studies be made in order to finalize the 280 HP transmission. The analysis was accomplished and detail drawings were prepared for the 280 HP module.

JUNE 1961 - A test procedure was formulated for the 280 HP module which included run-in and 50 hours of load testing.

Trade-off studies were completed and submitted to USATRECOM for their review.

Study No. 1	Segmented Crankshaft
Study No. 2	Reverter Pin
Study No. 3	Future Design Incorporating Studies 1 and 2

Design specifications were completed for both the 280 and 840 HP transmissions.

JULY 1961 - Phase I Report was completed and submitted to USA-TRECOM for approval, TREC Technical Report 61-38, Cycloidal Cam Transmission, Revision June 1961.

NOVEMBER 1961 - Special tooling was ordered for the contour grinding of the cam plates. Manufacturing was started on various components. Oil seals and bearings were ordered. Housing patterns were completed and castings poured. Crankshaft pattern was completed and castings poured. Cam plate forgings were completed and ready for rough machining.

DECEMBER 1961 - All hardware items were ordered. Housings crankshaft, cam plates were rough machined with two cam plates expedited for heat treatment. Test stand tooling drawings were completed.

JANUARY 1962 - The run-in test stand was released for manufacturing, 60 percent of the transmission items were completed. The two cam plates were dimensionally checked for possible heat treat distortion. Other special tooling was completed for the grinding of the cam plates.

FEBRUARY 1962 - A photograph inventory was started to cover the main manufacturing methods of the transmissions.

MARCH 1962 - An optical verlay template, machined scribed, was completed based on the X and Y coordinates of the cam contour. The overlay was used for checking the accuracy of the cam contour of each cam plate. Balancing of the crankshafts was completed.

APRIL 1962 - The reverter and crankshaft bearings were received. The assembly of one transmission was completed.

MAY 1962 - Preliminary no-load run-in tests were completed for one transmission. Lubrication modifications were required as a result of the testing. Formal load testing was finalized and the test stand was released for fabrication. Colored movies were taken showing the assembly and run-in procedures.

JUNE 1962 - Four teardown inspections and evaluations of S/N 101 were made in preparation for the load testing. The transmission (S/N 101) was re-assembled and re-checked for operation. The second transmission (S/N 102) was modified to include the design changes made on the first transmission.

JULY 1962 - The first transmission was delivered to the vendor, for their use in setting up the load testing program. Drawing changes were brought up to the current transmission configuration.

AUGUST 1962 - Check-out runs and load testing was completed on the first transmission S/N 101. An industrial oil was tried in the unit but the temperatures were excessive for the design speed operation. An oil change was made to aircraft type MIL-L-5086 Light which lowered the operating temperatures sufficiently for the top design speed. Heat losses, mechanical losses and efficiencies were calculated along with calibration curves for the various components used in the loading test set-up. Load testing was witnessed by USATRECOM personnel. During the high load testing of S/N 101, excessive wear was noted by the chips found in the circulating oil. After 20 hours of load testing, the unit was removed from the test stand and a teardown inspection was completed.

SEPTEMBER AND OCTOBER 1962 - Detailed analysis was conducted to determine the nature of the failure and the relatively low efficiencies. Reasons for the failure and the low efficiencies were determined and recommendations for redesign were submitted to USA-TRECOM. A bearing consultant was called in to aid in the analysis, namely, Mr. Tom Barish of Cleveland, Ohio.

Data reduction from the load runs were completed including graphs for calibration and performance. This information was submitted to USATRECOM. It was apparent from the data that more accurate methods of monitoring the input and output torque would be necessary in order to determine the efficiency of the transmission. It also was apparent that better methods would be required in order to monitor wear of the transmission.

NOVEMBER AND DECEMBER 1962 - All testing and redesign was held in abeyance for USATRECOM. Torrington submitted their cam follower bearing design to replace the present Rollway eccentric reverter bearings. Byron Jackson submitted their proposal on testing. This consisted of using two load pumps of different capacity instead of one in order to cover the entire loading and speed spectrum desired. Western Gear in-house research was initiated and completed for the β_D torque transducers. One transducer was designed to cover the input torque range of 0 to 3000 inch-pounds and another was designed to cover the output torque range of 0 to 60,000 inch-pounds.

JANUARY 1963 - Amended contract received. Redesign of transmission S/N 102 commenced, incorporating the stiffened reverter cage system and the cam follower bearings.

FEBRUARY AND MARCH 1963 - Drawings were released and manufacturing commenced on the redesign. Bearing and hardware items were ordered. Drawings were released for the torque transducers. Details were completed for modifying Norma Hoffman RXLS 2-3/4 roller bearing to include an outer race shoulder.

APRIL AND MAY 1963 - Manufacturing was completed and transmission S/N 102 was assembled, also the torque transducers. Run-in and checkout runs were completed. Lubrication flow checks were completed. Optimum flow conditions were determined. A teardown inspection was made in preparation for the load testing. The torque transducers were calibrated for both static and dynamic conditions. Patent applications, Dockets 4168, 4169 and 4170 were rejected by the patent office on the basis of the wording in the claims. Further action will be held in abeyance.

JUNE 1963 - The second transmission (S/N 102) was delivered to the vendor for their use in setting up the load testing program. Checkout and load runs were made covering the load spectrum desired. Heat losses, mechanical losses and efficiencies were determined. Load testing was witnessed by USATRECOM personnel. By monitoring the magnetic plugs for wear particles inserted in the lubrication system, all testing was stopped after 17-1/2 hours had been accumulated. The transmission was removed from the test stand and a teardown inspection was made.

JULY 1963 - Inspection revealed excessive wear of the crankshaft and the fixed pin rollers. Detailed analysis was conducted, culminating into a report which was submitted to USATRECOM. Data reduction revealed a considerable improvement in efficiency.

AUGUST 1963 - Crank bearing recommendations were being reviewed by MRC, Rollway and Torrington. Torrington was also reviewing the fixed pin rollers and have submitted their design of a future fixed pin roller. MRC and Rollway have submitted preliminary suggestions for the crank bearings but not their final configurations.

SEPTEMBER 1963 - Phase II and III report was prepared in rough draft form for USATRECOM for their review and comments.