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USAAVLABS TECHNICAL REPORT 65-31

ENDURANCE TEST OF AN-1 ROLLER GEAR DRIVE

TRW
Engineering Report 6371

August 1965

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

CONTRACT DA 44-177-AMC-30(T)

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DEPARTMENT OF THE ARMY
U. S. ARMY AVIATION MATERIEL LABORATORIES
FORT EUSTIS, VIRGINIA 23604

This report represents a part of a continuing U. S. Army Aviation Materiel Laboratories research program for the investigation of new concepts of high-speed reducers for use as main transmissions in helicopters. The main efforts of this program are directed toward deriving a reduction unit or units, with a reduction ratio significantly higher (40:1 and above) than those of currently used transmissions, which would be more compatible with the high rotational speeds of aircraft turbine engines. With this objective in mind, the roller gear drive investigation was undertaken.

This Command concurs with the contractor's conclusions and recommendations reported herein and has taken the same into consideration in the continuing design and fabrication program.

Task 1M121401D14404
CONTRACT DA 44-177-AMC-30 (T)
USAAVLABS Technical Report 65-31
August 1965

ENDURANCE TEST OF AN-1 ROLLER GEAR DRIVE

TRW
ENGINEERING REPORT 6371

PREPARED BY
TRW Inc.
ACCESSORIES DIVISION
CLEVELAND, OHIO

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

SUMMARY

THIS REPORT PRESENTS A REVIEW OF THE 1,000-HOUR ENDURANCE TEST OF THE ROLLER GEAR POWER TRANSMISSION BY TRW, CLEVELAND, OHIO, FOR USAAML* UNDER CONTRACT DA 44-177-AMC-30 (T). THE PRIMARY PURPOSE OF THIS PROGRAM WAS TO DETERMINE THE FEASIBILITY OF ROLLER GEAR DRIVE ARRANGEMENTS FOR POWER REDUCTIONS IN HELICOPTER TRANSMISSIONS AND TO DETERMINE CRITICAL DESIGN PARAMETERS FOR THE DRIVE.

AN ENDURANCE RUN WAS MADE FOR 1,000 HOURS ON TWO ROLLER GEAR DRIVES MOUNTED IN A BACK-TO-BACK TEST RIG WITH A 200-HORSEPOWER LOAD APPLIED CONTINUOUSLY WITH THE HIGH-SPEED SHAFT AT 28,000 RPM AND THE LOW-SPEED SHAFT AT 600 RPM IN EACH BOX. AN INSTRUMENTATION SYSTEM WAS PROVIDED TO MEASURE LOADS, SPEEDS, OIL FLOWS, AND TEMPERATURES, AND TO MEASURE VIBRATION LEVELS.

THE TEST HAS CONFIRMED THAT THE ROLLER GEAR DRIVE IS FEASIBLE FOR HELICOPTER POWER REDUCTIONS AND HAS THE ABILITY TO ACCEPT HIGH INPUT SPEEDS AND TO ACCOMPLISH HIGH RATIO REDUCTIONS IN ONE PLANE WITH VERY HIGH EFFICIENCY, 98.5 PERCENT OR BETTER, WITH A HIGH DEGREE OF MECHANICAL RELIABILITY.

THE DESCRIPTION OF THE ROLLER GEAR DRIVE, THE TEST RIG, AND THE TEST PERFORMANCE DATA ARE INCLUDED IN THIS REPORT. IN ADDITION, RECOMMENDATIONS FOR ACQUIRING ADDITIONAL INFORMATION TO BETTER DETERMINE THE ADVANTAGES OF THE ROLLER GEAR DRIVE ARE INCLUDED IN THE REPORT.

*Changed to USAAVLABS in June 1965.

FOREWORD

THE U. S. ARMY AVIATION MATERIEL LABORATORIES ENTERED INTO A CONTRACT WITH TRW FOR THE PURPOSE OF CONDUCTING A PARAMETRIC ANALYSIS, PREPARING A DETAIL DESIGN, AND FABRICATING A ROLLER GEAR DRIVE UNDER CONTRACT DA 44-177-AMC-30 (T). THIS REPORT IS A REVIEW OF THE 1,000-HOUR ENDURANCE RUN ON AN EXISTING TRW ROLLER GEAR DRIVE, CONDUCTED AS A PART OF THE DETAIL DESIGN PHASE, TO DETERMINE FEASIBILITY OF THE DRIVE FOR HELICOPTER APPLICATION AND TO DETERMINE CRITICAL DESIGN PARAMETERS OF THE DRIVE.

THE ROLLER GEAR DRIVE AND TEST STAND HAD BEEN DESIGNED AND THE HARDWARE PROCURED BY TRW AS PART OF A COMPANY SPONSORED RESEARCH ACTIVITY TO DEVELOP IMPROVED POWER TRANSMISSIONS PREVIOUS TO THE AWARDING OF THIS TESTING CONTRACT. THE ROLLER GEAR DRIVE HAD CONSIDERABLE TEST TIME ON IT AT THE TIME OF THE CONTRACT COMMITMENT BY USAAML. IT WAS ORIGINALLY DESIGNED FOR 300 HORSEPOWER, 42,000-RPM INPUT, AND 600-RPM OUTPUT.

THE ENTIRE PROGRAM AT TRW WAS CONDUCTED BY DR. A. L. NASVYTIS, WHO IN ADDITION TO BEING PROGRAM MANAGER IS THE INVENTOR AND HOLDS PATENT RIGHTS TO THE ROLLER GEAR DRIVE. THE WORK UNDER THIS CONTRACT WAS DONE IN THE ACCESSORIES DIVISION OF TRW UNDER THE MANAGEMENT OF MR. BEN BARISH; MR. W. M. SHIPITALO WAS THE PROJECT ENGINEER.

THE LIAISON WORK WITH USAAML HAS BEEN CONDUCTED WITH THE FOLLOWING PERSONNEL: MESSRS. HOMER BROWN AND WAYNE HUDGINS, UNDER THE DIRECTION OF MR. NELSON DANIEL .

THE DESIGN OF THIS DRIVE WAS COMPROMISED, BECAUSE INITIALLY A DUAL OUTPUT WITH CONTRAROTATING SHAFTS WHICH PROVIDED FOR A BALANCED TORQUE REACTION ON THE HOUSING WAS ENVISIONED. SUBSEQUENTLY, TEST-RIG LIMITATIONS ELIMINATED THE CAPABILITY OF TESTING A CONTRAROTATING OUTPUT ROLLER DRIVE WITHOUT PROVIDING COSTLY TEST-RIG CHANGES. CONSEQUENTLY, THE ROLLER GEAR DRIVE AS TESTED DID NOT APPROACH AN OPTIMUM DESIGN FOR A SINGLE OUTPUT AND YIELDED A HOUSING WHICH DID NOT HAVE SUFFICIENT TORSIONAL STIFFNESS. THIS REQUIRED ADDITIONAL STIFFENING BARS AND ADJUSTMENT OF THE DRIVE UNDER FULL PRELOAD TO COMPENSATE FOR THE TORSIONAL WINDUP BEFORE ENDURANCE TESTING COULD BEGIN.

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OBJECTIVES

THE BASIC OBJECTIVE OF THIS CONTRACT PROGRAM WAS TO RUN A 1,000-HOUR CONTINUOUS ENDURANCE TEST UPON A TRW HIGH RATIO REDUCTION ROLLER GEAR DRIVE TO DETERMINE THE BASIC TECHNICAL FEASIBILITY OF ROLLER GEAR DRIVE ARRANGEMENTS FOR USE AS THE MAIN POWER REDUCTION IN HELICOPTER TRANSMISSIONS.

INTERMEDIATE OBJECTIVES FOR THE ROLLER GEAR DRIVE WERE AS FOLLOWS:

1. TO MEASURE EFFICIENCY.
2. TO MEASURE GEAR TOOTH LOAD DISTRIBUTION AMONG THE PLANET GEARS.
3. TO MEASURE AND EVALUATE THE RELIABILITY OF THE LUBRICATION SYSTEM.
4. TO EVALUATE THE RELIABILITY OF THE MECHANICAL ARRANGEMENT OF THE COMPONENTS.
5. TO MEASURE AND EVALUATE THE VIBRATION LEVEL.
6. TO EVALUATE SERVICE LIFE CRITERIA AND WEAR RATES.

CONCLUSIONS

THE TRW ROLLER GEAR DRIVE HAS BEEN SUCCESSFULLY OPERATED IN EXCESS OF 1,000 HOURS OF TRANSMISSION TEST TIME. THE FOLLOWING CONCLUSIONS WERE DERIVED FROM THIS EXPERIENCE:

1. THE HIGH RATIO REDUCTION ROLLER GEAR DRIVE IS FEASIBLE FOR MAIN POWER REDUCTION IN HELICOPTER TRANSMISSIONS.

2. THE MEASURED EFFICIENCY FOR A 70:1 AND A 46.6:1 RATIO REDUCTION WAS 98 PERCENT AND 98.5 PERCENT, RESPECTIVELY. THIS EFFICIENCY IS CONSIDERED TO BE BETTER THAN, OR EQUAL TO, ANY OTHER KNOWN REDUCTION UNIT OF SIMILAR REDUCTION RATIOS.

3. THE INSTRUMENTATION USED TO MEASURE GEAR TOOTH LOAD DISTRIBUTION AMONG THE PLANET GEARS WAS NOT SUFFICIENTLY SENSITIVE TO BE ABLE TO PREDICT LOAD DISTRIBUTION WITH SUFFICIENT ACCURACY.

4. PRESSURE JET LUBRICATION WAS DIRECTED TO THE GEAR MESHES WITH THE ROLLER CONTACTS DEPENDING UPON THE RESULTING OIL SPRAY MIST FROM THE GEARS. THIS CONVENTIONAL METHOD OF LUBRICATION PROVIDED SATISFACTORY GEAR AND ROLLER CONTACT SURFACES.

5. SUPPORTING ROLLER GEAR ELEMENTS IN A TOGGLE ARRANGEMENT WITH THREE ALMOST EQUIDISTANT PRELOADED LINE CONTACTS ON THE ROLLER SURFACES HAS MAINTAINED EXCELLENT ALIGNMENT OF THE GEAR TOOTH SURFACES. IN ADDITION, IT IS BELIEVED THAT THE LOADED ROLLER CONTACTS AFFORD A DEGREE OF ELASTIC FLEXIBILITY WHICH CONTRIBUTES TO HAVING MORE UNIFORM LOADING OF THE GEAR CONTACTS.

6. THE VIBRATION LEVEL OF THE ROLLER GEAR DRIVE WAS QUITE LOW, BUT SUFFICIENT DATA WAS UNAVAILABLE TO PROJECT AND COMPARE WITH OTHER DRIVES OPERATING UNDER SIMILAR CONDITIONS.

7. AT THE CONCLUSION OF THE 1,000-HOUR ENDURANCE TEST, ALL ROLLER AND GEAR CONTACTS WERE IN EXCELLENT CONDITION. THE OPERATING STRESS LEVELS DURING THE ENDURANCE RUN WERE NOT HIGH ENOUGH TO MAKE ANY SIGNIFICANT LIFE AND WEAR PREDICTIONS FOR HELICOPTER APPLICATIONS.

RECOMMENDATIONS

IT IS RECOMMENDED THAT A ROLLER GEAR DRIVE BE DESIGNED, BUILT, AND TESTED WHICH MORE CLOSELY RESEMBLES AN ACTUAL HELICOPTER POWER TRANSMISSION APPLICATION. A TEST OF A ROLLER GEAR DRIVE SPECIFICALLY DESIGNED FOR HELICOPTER APPLICATION CAN BE COMPARED AND EVALUATED TO STUDY THE FOLLOWING LISTED DESIRED ADVANTAGES OF IMPROVED HELICOPTER TRANSMISSIONS.

1. IMPROVEMENT IN POWER-TO-WEIGHT RATIO.

2. INCREASED RELIABILITY.

3. ABILITY TO ACCEPT HIGHER INPUT SPEEDS.

4. IMPROVED EFFICIENCY.

5. INCREASED SIMPLICITY.

6. LOWER PRODUCTION COSTS.

THE ROLLER GEAR DRIVE WHICH HAS SUCCESSFULLY COMPLETED AN ENDURANCE RUN AT A 200 HORSEPOWER LOAD SHOULD BE TESTED AT A 300 HORSEPOWER LEVEL. THIS LOAD LEVEL WOULD RESULT IN THE THIRD MESH CARRYING A BENDING STRESS WHICH IS BELIEVED TO BE TYPICAL OF THE BENDING STRESSES ENCOUNTERED IN HELICOPTER TRANSMISSIONS. AN ENDURANCE RUN OF AT LEAST 500 HOURS SHOULD BE MADE AT THIS LOAD LEVEL, AND THEN IF CONDITIONS PERMIT, THE LOAD SHOULD BE INCREASED GRADUALLY AND RUNS MADE FOR SHORTER PERIODS OF TIME UNTIL FAILURE OCCURS.

THE PURPOSE OF THIS TESTING IS TO FULLY UNDERSTAND THE DESIGN LIMITATIONS OF THE ROLLER GEAR DRIVE.

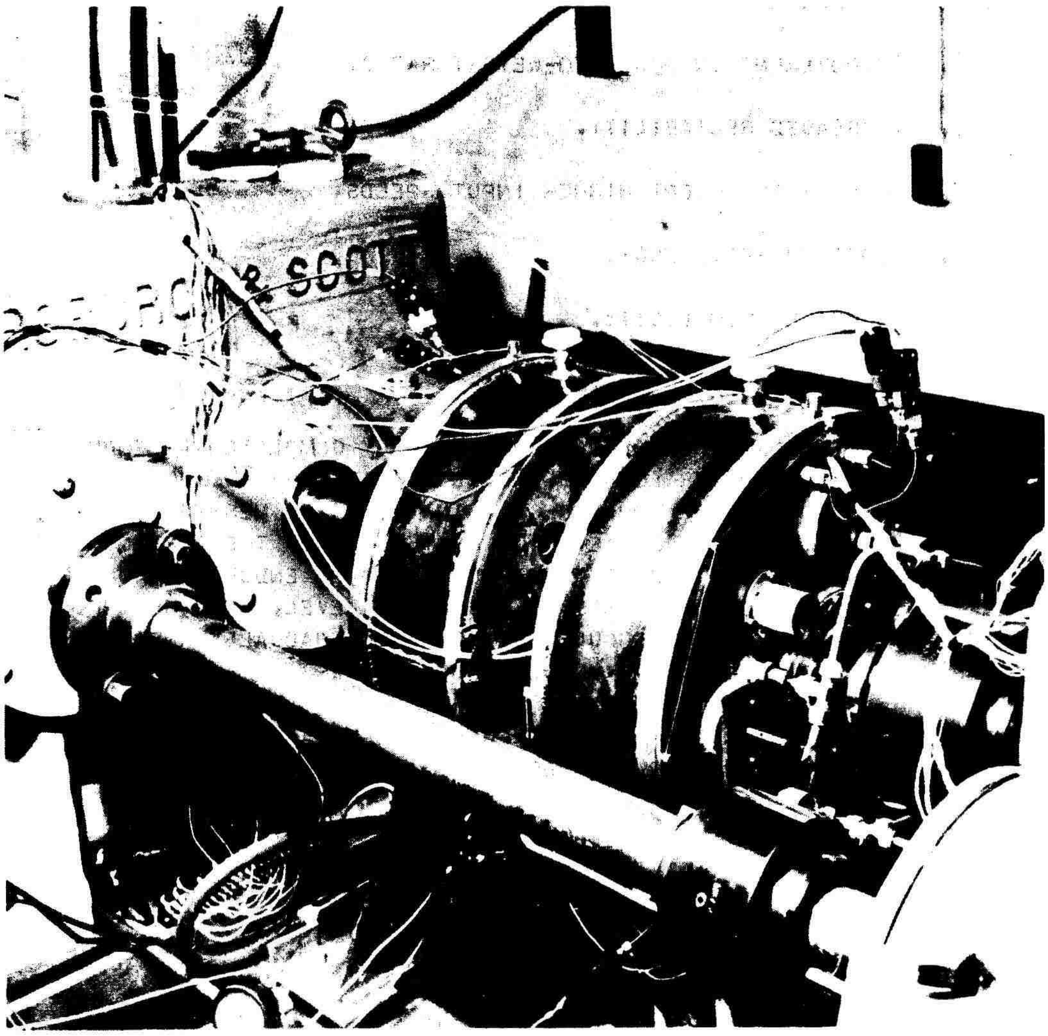

FIGURE 1. ROLLER GEAR DRIVE TEST SETUP

TEST METHOD

THE TEST SYSTEM INCLUDES A TEST STAND, CONTROL ROOM, LUBRICATION AND COOLING SYSTEM, SAFETY SHUTOFF DEVICES, AND A BLACK BOX TO PROVIDE AUTOMATIC UNATTENDED TESTING WITH RECORDING EQUIPMENT. THE EQUIPMENT HAS PERFORMED VERY SATISFACTORILY AND EXTREMELY LONG LIFE ENDURANCE TESTING CAN BE PERFORMED ON THIS TYPE OF RIG WITH CONFIDENCE AND A LOW-DOLLAR COST PER HOUR OF TESTING.

1. TEST STAND - FIGURE 1

THE TESTING SYSTEM EMPLOYS THE PRINCIPLE OF BACK-TO-BACK COUPLING OF TWO TEST UNITS IN A CLOSED POWER PATH. THE TWO OUTPUT SHAFTS OF THE TEST UNITS ARE CONNECTED BY FLEXIBLE COUPLINGS TO THE OUTPUT SHAFTS OF RIGHT- AND LEFT-HAND CLOSING GEARBOXES. THE CLOSING GEARBOXES ARE STANDARD INDUSTRIAL TYPE REDUCER UNITS WITH A FIXED RATIO REDUCTION OF 5:1. THE COMMERCIAL GEARBOXES ARE LIMITED TO 300 HORSEPOWER WITH A 3,000 RPM INPUT SHAFT AND A 600 RPM OUTPUT SHAFT. THE INPUT SHAFTS OF THE CLOSING BOXES ARE CONNECTED TO EACH OTHER WITH A TORQUE COUPLING FORMING A CLOSED POWER PATH. THE TORQUE COUPLING CONSISTS OF TWO PLATES WITH HOLES AND SLOTS PERMITTING THE ATTACHMENT OF A TORQUE BAR WITH CALIBRATED WEIGHTS TO APPLY KNOWN TORQUE LOADS TO THE SYSTEM. THIS PRELOADED TORQUE IS APPLIED STATICALLY AND LOCKED WITH BOLTS IN THE FLANGES OF THE TORQUE COUPLING, AFTER WHICH THE TORQUE BAR AND WEIGHTS ARE REMOVED. THE INPUT SHAFT OF ONE CLOSING BOX IS ALSO CONNECTED TO A DRIVING DYNAMOMETER WHICH HAS A SPEED CONTROL THAT PROVIDES VARIABLE SPEEDS AND MEASURES THE TOTAL DELIVERED TORQUE TO THE CLOSED POWER LOOP TRANSMISSION SYSTEM. ALSO, LUBRICATION AND COOLING SYSTEM IS PROVIDED FOR THE TRANSMISSION SYSTEM.

A SCHEMATIC DRAWING OF THE TEST STAND IS SHOWN IN FIGURE 2.

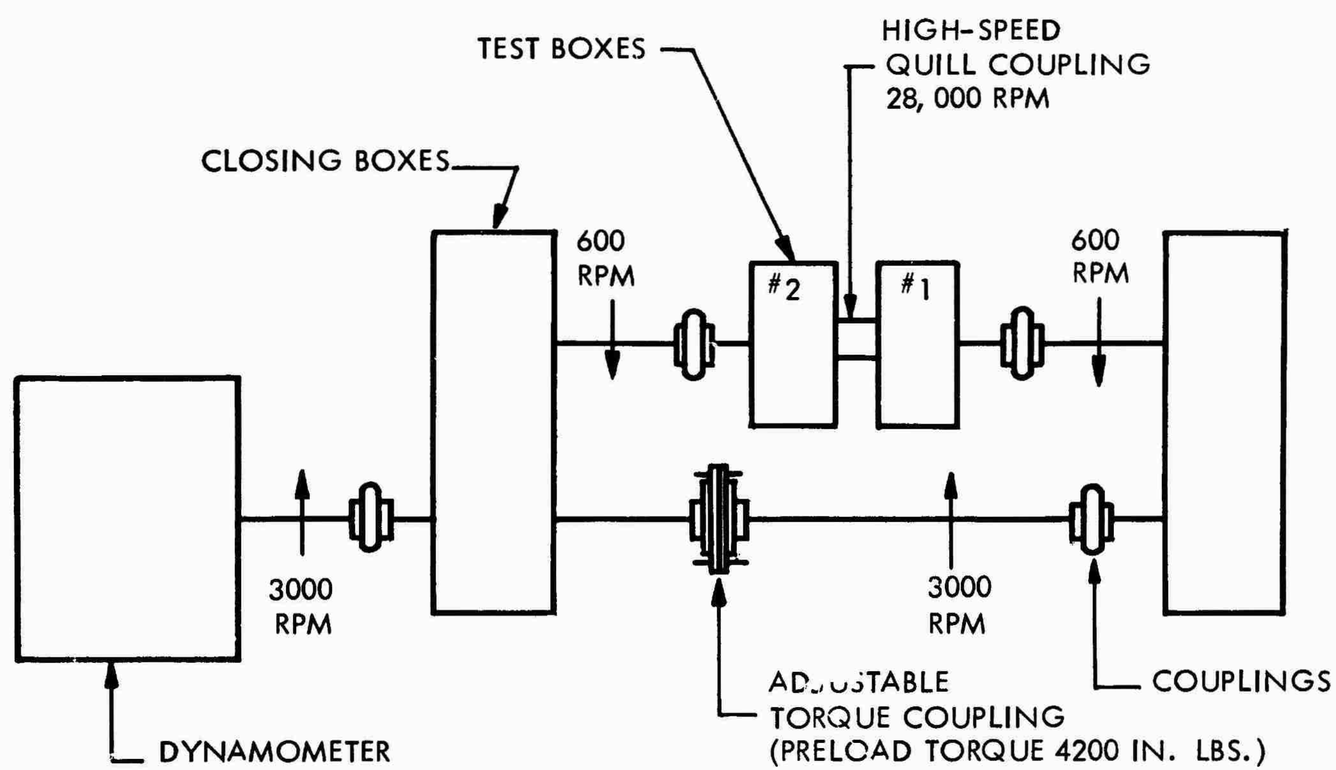

FIGURE 2. TEST RIG SCHEMATIC

2. INSTRUMENTATION SYSTEM

THE INSTRUMENTATION SYSTEM (FIGURE 3) USED FOR TESTING IS DESIGNED TO MONITOR THIRTY-SEVEN OUTPUTS OF TEMPERATURE, PRESSURE, OIL FLOW, SPEED, TORQUE, VIBRATION, CHIP DETECTORS, AND STRAIN GAGE READINGS FOR THE TEST TRANSMISSIONS AND TEST STAND ROTATING EQUIPMENT. THE MONITOR IN TURN WILL EXERCISE SUPERVISORY FUNCTIONS OVER THE SYSTEM AS A WHOLE, SO THAT VISUAL SIGNALS, RECORDING, SHUTDOWN, AND OTHER FUNCTIONS CAN BE AUTOMATICALLY PERFORMED.

THE INSTRUMENTATION SYSTEM CONSISTS OF THE FOLLOWING:

A. TEMPERATURE RECORDING EQUIPMENT, INCLUDING VISUAL INDICATION AND A STRIP CHART PLOTTING RECORDER.

B. TEMPERATURE MONITOR TO SCAN TEMPERATURES AND PROVIDE AUTOMATIC SHUTDOWN FOR EXCESSIVE TEMPERATURES.

C. FLOWMETERS, SENSORS, AND VISUAL INDICATORS FOR DISPLAY OF OIL-FLOW SIGNALS.

D. PRESSURE GAGES FOR DISPLAY OF LUBRICATION OIL PRESSURES AND A MONITOR TO PROVIDE LOW-PRESSURE SHUTDOWN.

E. TACHOMETER GENERATOR AND INDICATOR FOR RECORDING SPEED, AND A MONITORING SYSTEM TO PROVIDE AUTOMATIC SHUTDOWN WHEN THE TEST RIG IS RUNNING OVER OR UNDER PRESELECTED SPEED.

F. ELECTRIC CHIP DETECTORS TO PROVIDE A VISUAL SIGNAL OF FOREIGN MATERIAL IN THE TEST BOXES AND TO PROVIDE AUTOMATIC TEST-RIG SHUTDOWN.

G. STRAIN GAGE EQUIPMENT, INCLUDING VISUAL INDICATORS AND AN OSCILLOSCOPE FOR DISPLAY OF AMPLITUDE SIGNAL.

H. AN ELECTRICAL DYNAMOMETER THAT PROVIDES A TORQUE INPUT READING THAT IS MONITORED TO PROVIDE AUTOMATIC TEST-RIG SHUTDOWN FOR OVER OR UNDER STEADY STATE TORQUE VALUES.

J. VIBRATION METER.

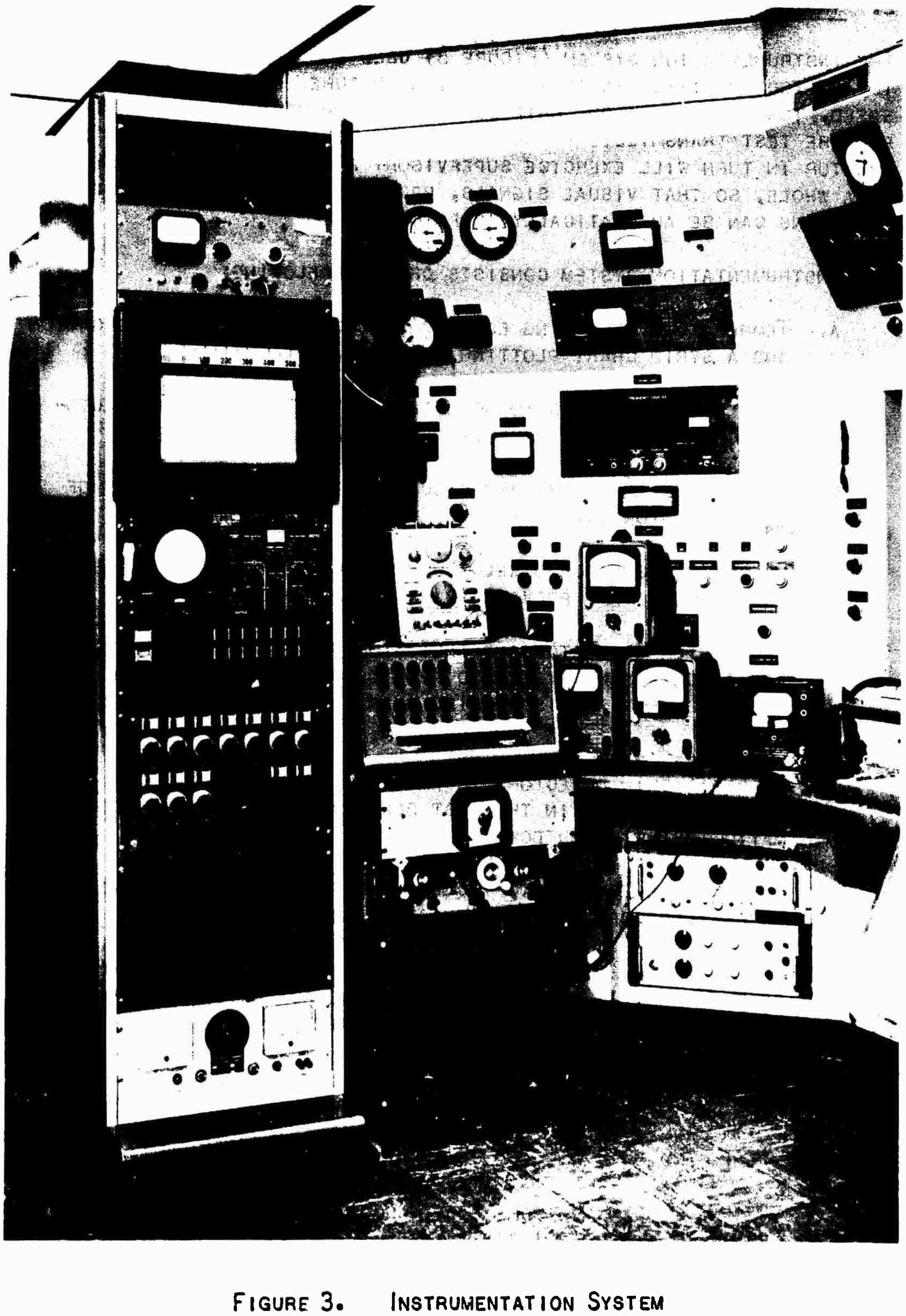

FIGURE 3. INSTRUMENTATION SYSTEM

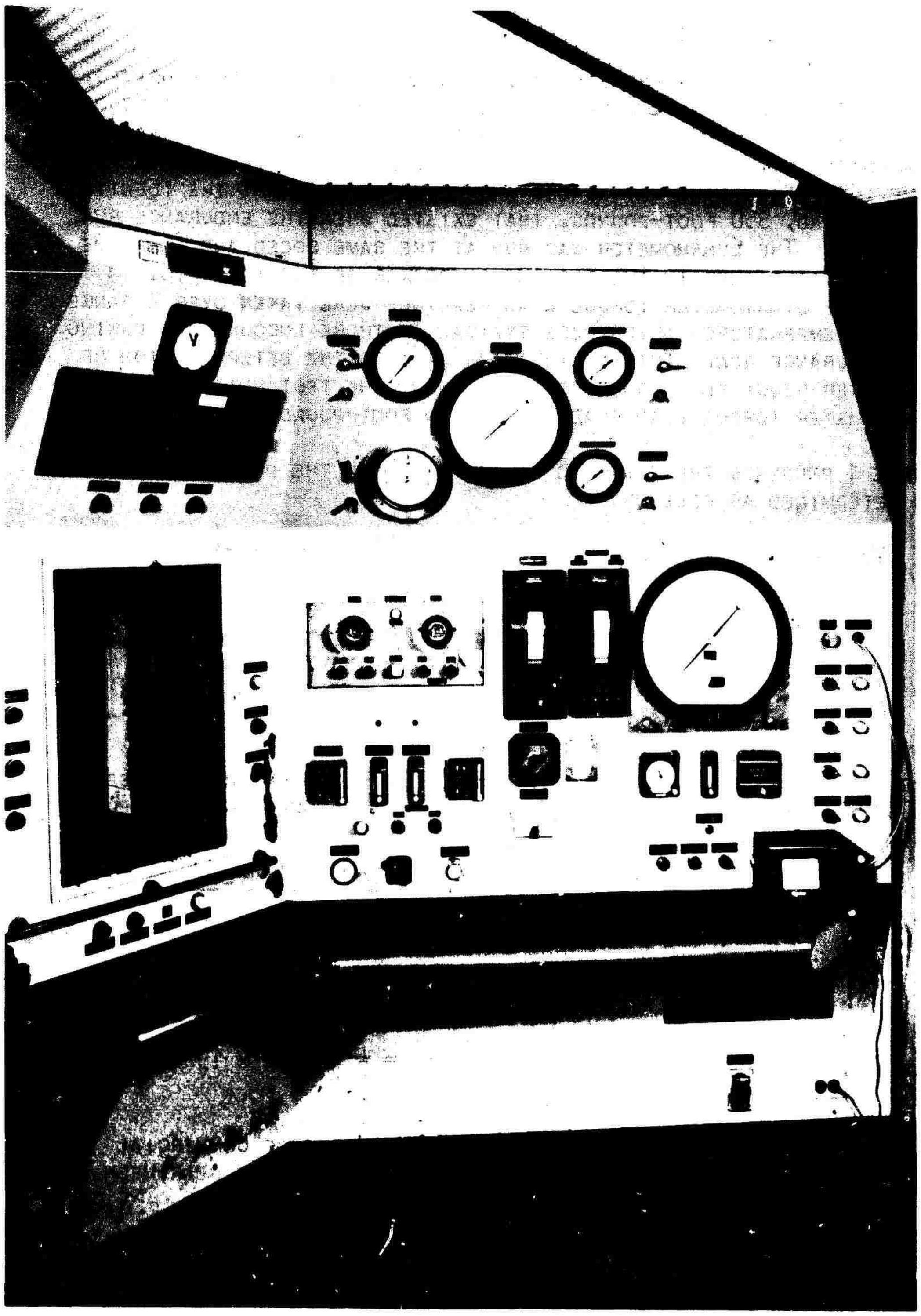

FIGURE 4. TEST CONSOLE

3. TEST-RIG CALIBRATION

AT THE COMPLETION OF THE ENDURANCE RUN ON THE ROLLER GEAR DRIVES, THE TEST UNITS WERE REMOVED FROM THE BACK-TO-BACK RIG AND A COUPLING SHAFT WAS CONNECTED TO THE RIGHT- AND LEFT-HAND COMMERCIAL CLOSING GEARBOXES IN THE SPACE FORMERLY OCCUPIED BY THE TESTED UNITS. THE SAME TORQUE COUPLING WAS LOADED AND ADJUSTED TO CARRY THE IDENTICAL TORQUE LOAD, 350 FOOT-POUNDS, THAT EXISTED WHEN THE ENDURANCE RUN WAS MADE. THE DYNAMOMETER WAS RUN AT THE SAME SPEED AND THE SAME OIL WAS MAINTAINED IN THE CLOSING GEARBOXES OF THE TEST RIG. A SERIES OF DYNAMOMETER TORQUE LOAD READINGS WERE TAKEN OVER A RANGE OF OIL TEMPERATURES WHICH WERE TYPICAL OF THOSE ENCOUNTERED DURING THE ENDURANCE RUN. THIS CALIBRATION PERMITS THE DETERMINATION OF THE POWER REQUIRED TO RUN THE RIG WITHOUT THE TEST UNITS. THE DYNAMOMETER TORQUE LOAD READINGS ARE IN FOOT-POUNDS.

TABLE 1 PRESENTS THE TEST-RIG CALIBRATION AND THE POWER REQUIRED IS DETERMINED AS FOLLOWS:

$$\text{Total Power Required} = \frac{\text{RPM} \times \text{Dyno Torque (in-lbs)}}{63{,}000}$$

$$\text{Total Power Required} = \frac{3000 \times 11.50 \times 12}{63{,}000} = 6.57 \text{ HP (at } 120^{\circ}\text{F oil temp.)} \quad (1)$$

THE PRELOAD TORQUE ON THE CLOSING BOX IS APPLIED ON THE HIGH-SPEED 3000-RPM SHAFT. ACCORDINGLY, THE TRANSMITTED POWER LOAD PER BOX IS DETERMINED AS FOLLOWS:

$$\text{Transmitted Power} = \frac{\text{RPM} \times \text{Preload Torque (in-lbs)}}{63{,}000} \quad (2)$$

$$\text{Transmitted Power} = \frac{3000 \times 350 \times 12}{63{,}000} = 200 \text{ HP}$$

$$\text{Average percentage loss per gearbox} = \frac{6.57}{2 \times 200} \times 100 = 1.64\% \quad (3)$$

THIS LOSS CAN BE CONSIDERED TO BE REPRESENTATIVE OF COMMERCIAL GEARBOXES AT CORRESPONDING LOADS AND SPEEDS WHERE ONE GEAR MESH AND SPLASH LUBRICATION IS INVOLVED.

TABLE 1. TEST-RIG CALIBRATION

SPEED RPM	TORQUE FT-LBS	TEMP. ^{0}F	HP LOAD	PRELOAD FT-LBS	HP REQD	TIME
3000	12.5	100	200	350	7.14	0940
	12.35	102	200	350	7.06	
	12.20	104	200	350	6.97	
	12.05	106	200	350	6.88	
	12.0	108	200	350	6.86	
	11.9	110	200	350	6.80	
	11.8	112	200	350	6.74	
	11.7	114	200	350	6.68	
	11.6	116	200	350	6.63	
	11.45	118	200	350	6.54	
	11.50	120	200	350	6.57	
	11.40	122	200	350	6.52	
	11.30	124	200	350	6.46	
	11.30	126	200	350	6.46	
	11.30	128	200	350	6.46	
2998	11.25	130	200	350	6.43	01100

TEST HARDWARE

1. DESCRIPTION

AS SHOWN IN FIGURE 5, THE DRIVE EMPLOYS A PLANETARY ARRANGEMENT OF GEARS AND ROLLERS, CONSISTING OF AN INPUT SUN ROLLER GEAR AND TWO CONCENTRIC CLUSTERS OF STATIONARY ROLLER GEAR PLANETS, AND AN OUTPUT SUN GEAR WHICH IS COAXIAL WITH THE INPUT SUN GEAR. THE SPEED REDUCTION OF THE DRIVE IS ACCOMPLISHED BY TWO ROWS OF STEPPED PLANET GEARS AND THE CONTACT OF THE DRIVING PINION IN THE SECOND ROW PLANET GEAR WITH THE OUTPUT SUN GEAR. THE SPEED REDUCTION OF THIS ROLLER DRIVE IS EQUIVALENT TO ANY STANDARD THREE-MESH GEAR SYSTEM.

THE SUN ROLLER GEAR , A, IS A HOLLOW CYLINDRICAL ONE-PIECE ELEMENT WITH AN INTERNAL, CENTRALLY LOCATED, INPUT SPLINE TO MINIMIZE EFFECTS OF TORSIONAL WINDUP ACROSS THE FACE OF THE GEAR TEETH. THE SUN GEAR TEETH ARE INTEGRALLY CUT AND FINISH GROUND ON BOTH ENDS OF THE CYLINDER. THEN, THE ROLLER ELEMENTS ARE PRESS FITTED TO THE GEAR CYLINDER AT BOTH ENDS AND THE ROLLING SURFACES ARE FINISH GROUND AND LAPPED TO CONFORM WITH THE PITCH DIAMETER OF THE SUN GEAR AND TO BE CONCENTRIC WITH THE GEAR. FLANGES ARE PROVIDED ON THE TWO ROLLERS TO FURNISH AXIAL RETENTION OF THE SUN ROLLER GEAR ELEMENT. THE FLANGES ARE SLIGHTLY TAPERED TO PROVIDE A MINIMUM CONTACTING SURFACE AS NEAR THE ROLLING SURFACES AS POSSIBLE. FIGURE 6 DESCRIBES THE SUN ROLLER GEAR ELEMENT.

THERE ARE FOUR STEPPED, EQUALLY SPACED, PLANET ROLLER GEARS SURROUNDING THE INPUT SUN ROLLER GEAR IN THE FIRST ROW. EACH OF THESE FIRST ROW ROLLER GEARS CONSIST OF THREE GEAR ELEMENTS; TWO LARGER GEARS, X_1, AT THE END AND A SMALLER GEAR, Y_1, AT THE CENTER. INITIALLY THE SMALLER GEAR, Y_1, IS GROUND TO FINAL SIZE, AFTER WHICH THE TWO LARGER GEARS ARE PRESSED AND PINNED TO SHAFT EXTENSIONS FROM THE SMALLER GEAR, Y_1. THE LARGER GEARS, X_1, ARE THEN GROUND TO MAINTAIN A TOOTH INDEX RELATIONSHIP WITH THE Y_1 GEAR OF PLUS OR MINUS .0001 INCH AND AT THE SAME TIME THE TWO ROLLER SURFACES ON BOTH SIDES OF THE Y_1 GEAR ARE FINISH GROUND AND LAPPED TO CONFORM PRECISELY WITH THE Y_1 GEAR. FINALLY, THE TWO LARGER ROLLERS ARE PRESSED IN PLACE ON THE ENDS OF THE TWO LARGER GEARS, X_1, AND GROUND AND LAPPED TO CONFORM. THIS ASSEMBLY CONSTITUTES THE DESCRIPTION OF THE X_1Y_1 ROLLER GEAR ELEMENT, AS SHOWN IN FIGURE 7. THE TWO LARGE GEARS, X_1, ALONG WITH THEIR CORRESPONDING MATCHING ROLLING SURFACES ARE IN CONTACT WITH THE ROLLING AND GEAR SURFACES OF THE SUN ROLLER, A, AS SHOWN IN FIGURE 5, PROVIDING FOUR EQUALLY SPACED LINE CONTACTS.

SURROUNDING THE FIRST ROW ROLLER GEARS, FOUR STEPPED PLANET ROLLER GEARS ARE EQUALLY SPACED (FIGURE 10). THE SECOND ROW PLANET GEARS HAVE SPHERICAL ROLLER BEARINGS ON THE ENDS OF THEIR SHAFTS WHICH ARE MOUNTED IN BEARING LINKS. THESE ARE BOLTED AND LOCKED TO THE HOUSING OF THE DRIVE TO PROVIDE A SUSPENSION FOR THE ROLLER GEAR DRIVE ELEMENTS AND TO TAKE THE TORQUE REACTION. EACH OF THE SECOND ROW ROLLER GEARS ESSENTIALLY CONSIST OF TWO GEAR ELEMENTS; ONE LARGER GEAR, X_2, AT ONE END, AND A SMALLER GEAR, Y_2, AT THE OTHER END. BOTH OF THESE GEARS ARE CUT AND FINISH GROUND INTEGRALLY WITH THE SHAFT WITH AN INDEX TOOTH RELATIONSHIP HELD ON A LARGE GEAR TOOTH, X_2, TO A TOOTH ON THE SMALL GEAR, Y_2, WITHIN PLUS OR MINUS .0001 INCH. ROLLERS ARE PRESSED IN PLACE ON BOTH SIDES OF THE LARGER GEAR, X_2, AND GROUND AND LAPPED TO CONFORM PRECISELY WITH THE GEAR X_2. AS SHOWN IN FIGURE 5, EACH LARGE GEAR, X_2, AND ITS CORRESPONDING MATCHED ROLLERS IS IN CONTACT WITH TWO GEARS, Y_1, AND THEIR CORRESPONDING MATCHED ROLLERS. GEOMETRY IS PREDETERMINED TO PRODUCE A TOGGLE ANGLE OF 27^{o} FOR THIS DRIVE, RESULTING IN A SUBTENDED ANGLE OF 36^{o} FOR THE GEAR X_2 TO MAKE CONTACT WITH THE TWO Y_1 GEARS. THIS ANGULAR RELATIONSHIP REQUIRES THAT THE NUMBER OF TEETH IN GEAR X_2 BE A WHOLE NUMBER WHEN MULTIPLIED BY THE RATIO $36/360 = 1/10$. THE NUMBER OF TEETH IN X_2 GEAR IS 130. AS SHOWN IN FIGURE 5, THE RESULTING SUBTENDED ANGLE ON GEAR Y_1, AS A RESULT OF THE TWO X_2 ROLLER GEAR CONTACTS WITH IT, IS 126^{o}, REQUIRING THAT THE NUMBER OF TEETH IN Y_1 BE A WHOLE NUMBER WHEN MULTIPLIED BY THE RATIO 126/360. THE NUMBER OF TEETH IN Y_1 GEAR IS 20. SMALL PINION GEAR, Y_2, HAS NO MATCHING ROLLERS BECAUSE THE FOUR Y_2 GEARS MAKE A CONVENTIONAL GEAR MESH IN A SECOND PLANE DIVORCED FROM THE PLANE OF ROLLER GEARS, A, X_1Y_1, AND X_2 ELEMENTS. THERE ARE FOUR EQUALLY SPACED Y_2 PINIONS UPON WHICH THERE ARE NO TOOTH NUMBER REQUIREMENTS SURROUNDING THE OUTPUT C GEAR. THE NUMBER OF TEETH IN GEAR C (FIGURE 9) MUST BE DIVISIBLE BY FOUR, AND THE NUMBER OF TEETH IN GEAR C IS 104. GEAR C FLOATS WITHIN THE Y_2 CONTACTS AND IS TRAPPED AXIALLY WITH LOOSE FITTING BRONZE THRUST WASHERS TO THE STATIONARY HOUSING. GEAR C HAS AN INTERNAL LOOSE FITTING SPLINE INTO WHICH THE OUTPUT SHAFT OF THE DRIVE IS INSERTED. THIS OUTPUT SHAFT IS CONVENTIONALLY SUPPORTED ON TWO STANDARD BALL BEARINGS.

SINCE ROLLER GEARS X_2Y_2 ARE CLAMPED TO THE HOUSING BY THE BEARING LINK ASSEMBLIES, AXIAL LOCATION FOR THE FIRST ROW GEAR ELEMENTS AND SUN GEAR ARE PROVIDED BY THE FLANGES OF THE ROLLER CONTACTS. IN ADDITION, THE TOGGLE ARRANGEMENT OF ROLLER GEARS, AS DESCRIBED, RESULTS IN A X_1Y_1 ROLLER GEAR ELEMENT WHICH IS SUPPORTED BY THREE ALMOST EQUALLY SPACED ROLLER CONTACTS. THESE CONTACTS MAINTAIN THE ROLLER GEAR ELEMENTS IN A VERY PRECISE ALIGNMENT, WHICH RESULTS IN EXCELLENT GEAR CONTACT. NO HIGH-SPEED ANTIFRICTION BEARINGS

WITH THEIR RESULTING LOOSENESS AND LOCATION TOLERANCES ARE REQUIRED, WHICH FREQUENTLY RESULTS IN A SKEWING OF GEAR ELEMENTS IN A MESH SYSTEM WITH VERY UNSATISFACTORY GEAR TOOTH CONTACTS.

THE HOUSING OF THE DRIVE CONSISTS OF TWO STEEL END PLATES CONNECTED TO EACH OTHER BY FOUR POSTS ARRANGED TO FORM A SQUARE PATTERN. ON EACH OF THE SIDES OF THE RESULTING SQUARE, THE TWO BEARING LINK ASSEMBLIES ARE FASTENED FOR EACH X_2Y_2 ROLLER GEAR. EACH BEARING LINK HAS AN ATTACHED LOAD CALIBRATED STRAIN GAGE. ONE END OF THE BEARING LINK IS TIGHTENED SECURELY TO THE HOUSING CONSTITUTING THE FIXED END OF A BEAM, WHILE THE OTHER FREE END OF THE LINK ASSEMBLY HAS A SLOTTED HOLE INTO WHICH A FIXED STUD PROJECTS FROM THE HOUSING. AS A NUT IS TIGHTENED ON THIS STUD, THE ROLLERS OF GEAR X_2 PRELOAD ALL THE OTHER ROLLERS. THE PRELOAD IS APPLIED AT EACH OF THE EIGHT FREE ENDS IN A PROGRAMMED STEP LOAD FASHION. IT SHOULD BE QUITE APPARENT THAT FOR THE TWO-PLANE DESIGN ARRANGEMENT LOAD APPLICATION WAS RATHER DIFFICULT BECAUSE OF NONSYMMETRY. BECAUSE OF THIS, THE NECESSARY PRELOAD WAS DETERMINED BY EXPERIMENTATION AND WAS A SUFFICIENT LOAD TO MATCH THE APPLIED TEST SYSTEM PRELOAD TORQUE TO MAINTAIN THE PRECALCULATED CENTER DISTANCE RELATIONSHIP FOR THE X_2Y_2 ROLLER GEAR ELEMENTS. THIS RELATIONSHIP WAS EASILY MEASURED ON THE SHAFT EXTENSIONS OF THE FOUR X_2Y_2 ROLLER GEARS.

IN ADDITION, THE HOUSING CONTAINS FOUR OTHER BOLTED POSTS CONNECTING THE TWO END PLATES. THESE POSTS WERE INCLUDED FOR ADDITIONAL TORSIONAL HOUSING RIGIDITY. IT BECAME EVIDENT IN EARLY TESTING OF THIS ROLLER GEAR DRIVE THAT FINAL ADJUSTMENT OF THE DRIVE COULD ONLY BE MADE ON THE TEST STAND. THIS FINAL ADJUSTMENT IS MADE WITH FULL APPLIED TORQUE LOAD AND A RED LEAD MARKING COMPOUND APPLIED TO THE OUTPUT GEAR MESHES BETWEEN GEAR Y_2 AND SUN OUTPUT C. THESE FOUR BOLTED POSTS CARRY ADJUSTMENT SCREWS WHICH CAN BE USED TO MOVE LOOSENED BEARING LINKS AGAINST THE REACTION GEAR LOAD UNTIL A PERFECT GEAR MESH IS OBSERVED. THE ADJUSTMENT IS MADE WITH THE COVER OF THE HOUSING REMOVED, THEREBY NOT CONTRIBUTING TO THE TORSIONAL STIFFNESS OF THE HOUSING. THE DESIGN OF THIS ROLLER GEAR WAS ORIGINALLY PLANNED FOR AN APPLICATION WHICH REQUIRES CONTRAROTATING SHAFTS, RESULTING IN A BALANCED TORQUE REACTION ON THE HOUSING. SUBSEQUENT CONSIDERATIONS ELIMINATED THE CONTRAROTATING REQUIREMENT BUT THE DESIGN OF THE HOUSING REMAINED UNCHANGED AND THE SHAFT EXTENSIONS FROM ROLLER GEAR X_2Y_2 TO CARRY THE DRIVE PINIONS FOR AN INTERNAL RING GEAR MESH ARE STILL EVIDENT.

A light cylindrical cover surrounds the roller gear elements and is supported by both ends of the cover plates. Basically, this cover contains the oil, with one of the end plates providing for a structural support of the unit in the test stand.

All gears in the drive were made from standard AMS-6260 gear steel, Rc 60 hardness and finish ground for a surface roughness of 32 microinches. Roller elements were made from standard 52100 bearing steel, Rc 60 hardness and finish ground and lapped for a surface roughness of 8 microinches. The connecting high-speed quill shaft, with splines at both ends to permit back-to-back testing of the two roller gear units, was made from 4340 steel hardened to Rc 30. The splines of this quill shaft were the only area showing obvious distress at the end of the 1000-hour endurance test.

No attempt was made to produce a lightweight aircraft type design for this original roller gear drive and all the housing elements were made from low-carbon boiler-plate steel, welded or bolted together to form the structure.

The total reduction of this drive is:

$$\frac{X_1}{A} \times \frac{X_2}{Y_1} \times \frac{C}{Y_2}$$

A = 24 teeth

X_1 = 48 teeth

Y_1 = 20 teeth

X_2 = 130 teeth

Y_2 = 29 teeth

C = 104 teeth

$$\text{Reduction Ratio} = \frac{48}{24} \times \frac{130}{20} \times \frac{104}{29} = 46.62 \qquad (4)$$

TRW HAS PREPARED A REPORT (NUMBER 64-29)* PUBLISHED BY USATRECOM WHICH IS A PARAMETRIC STUDY OF ROLLER GEAR DRIVES. THIS REPORT PRESENTS THE MATHEMATICAL CAPABILITIES AND LIMITATIONS OF POSSIBLE GEAR TOOTH COMBINATIONS AND ARRANGEMENTS TO ACCOMPLISH A REQUIRED POWER REDUCTION.

THE APPROXIMATE DIMENSIONS OF THE ROLLER DRIVE ARE 18.50 INCHES IN DIAMETER X 7.0 INCHES IN LENGTH. FIGURES 10 AND 11 PORTRAY THE ASSEMBLY OF THE DRIVE AND ALL THE BASIC ELEMENTS.

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* "PARAMETRIC STUDY ON THE ROLLER GEAR REDUCTION DRIVE," USATRECOM REPORT NO. 64-29, U. S. ARMY TRANSPORTATION RESEARCH COMMAND, FORT EUSTIS, VIRGINIA, MAY 1965.

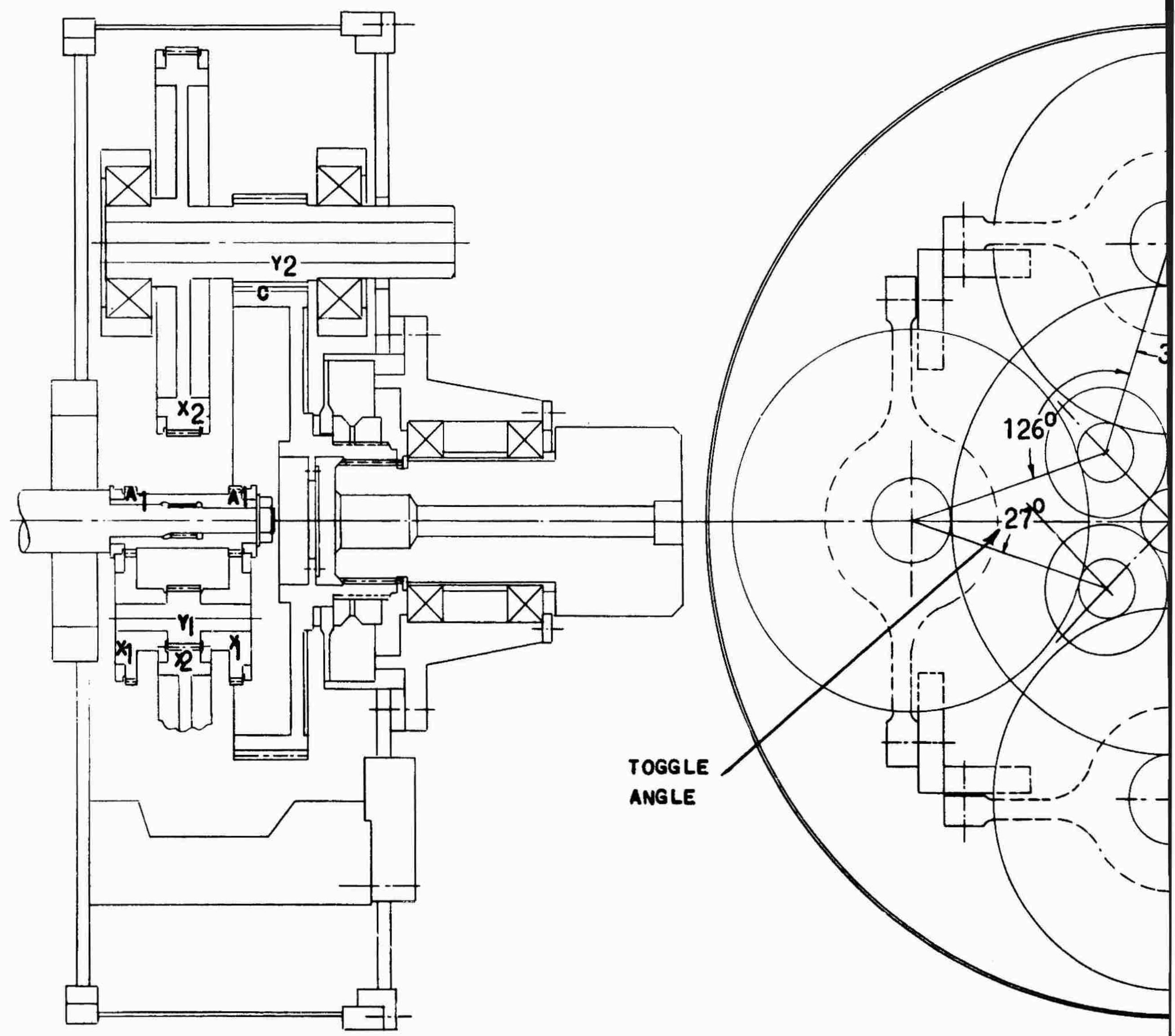
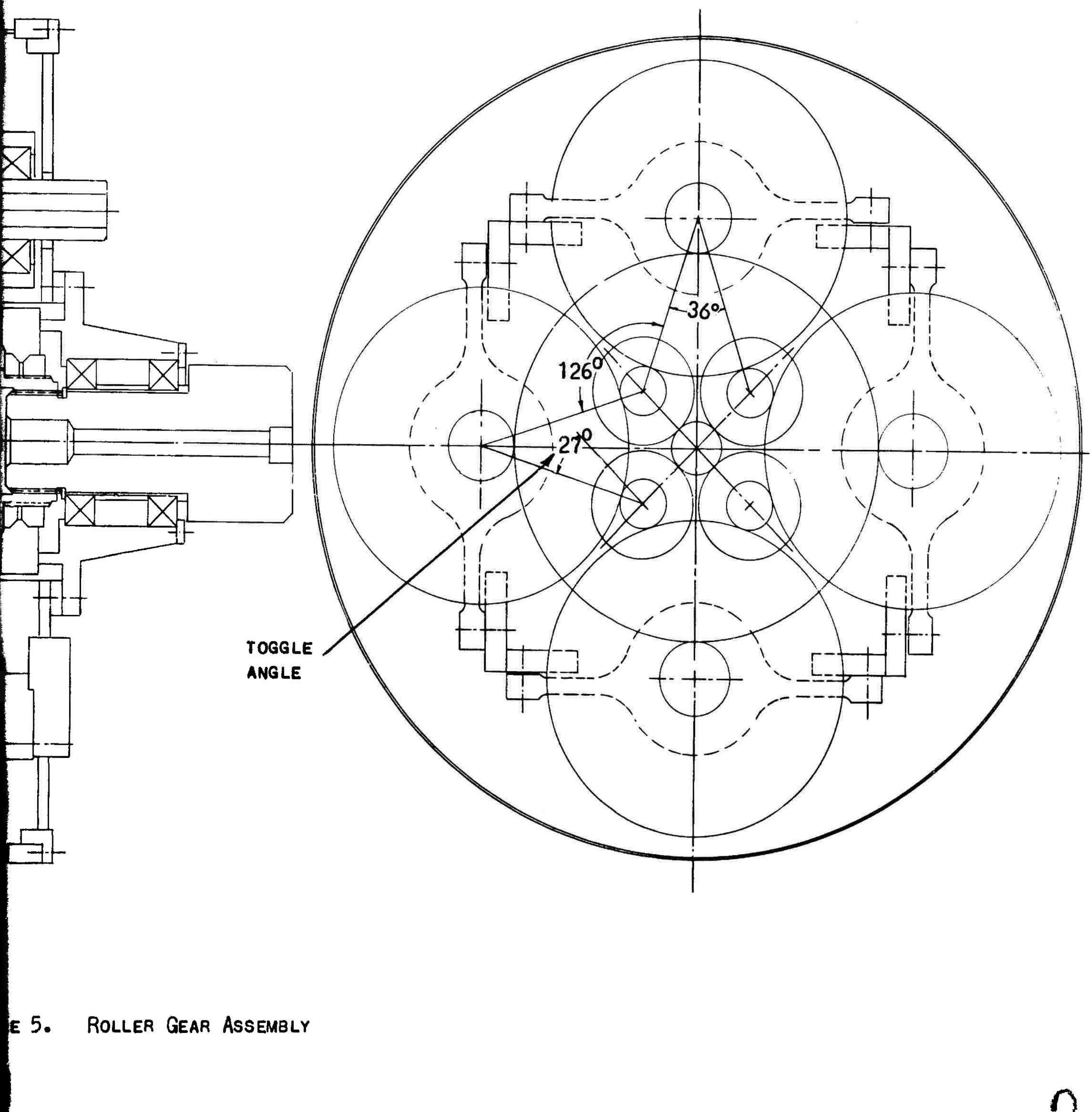


FIGURE 5. ROLLER GEAR ASSEMBLY



E 5. ROLLER GEAR ASSEMBLY

2

FIGURE 6. INPUT SUN ROLLER GEAR, A

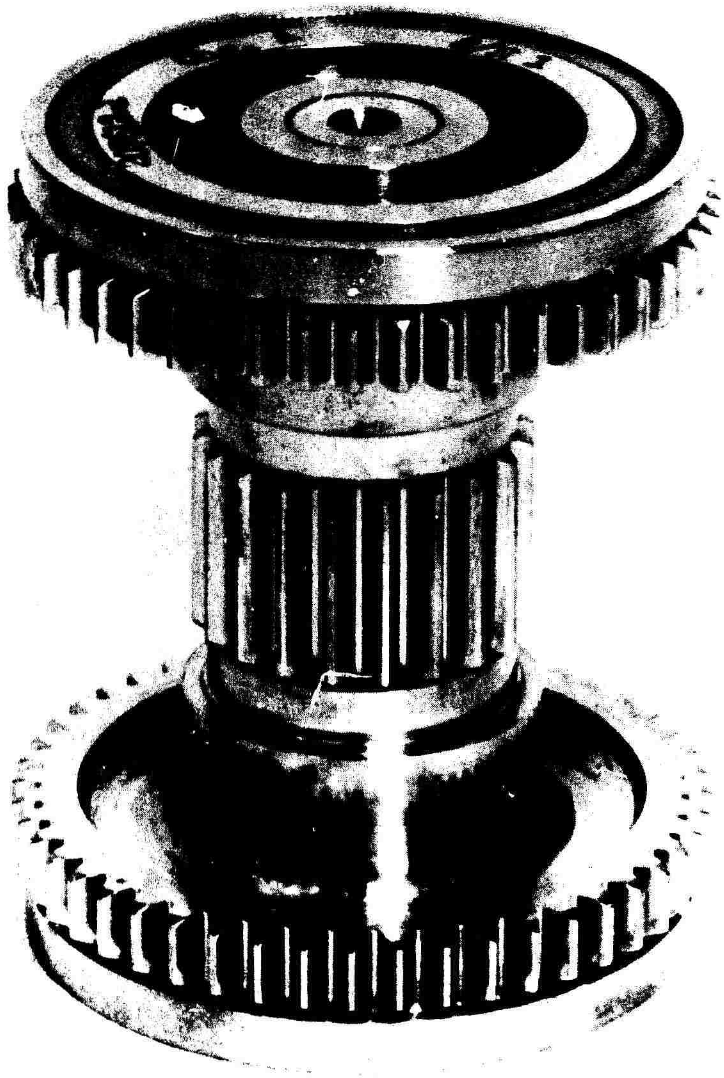


FIGURE 7. FIRST ROW ROLLER GEAR, X_1Y_1

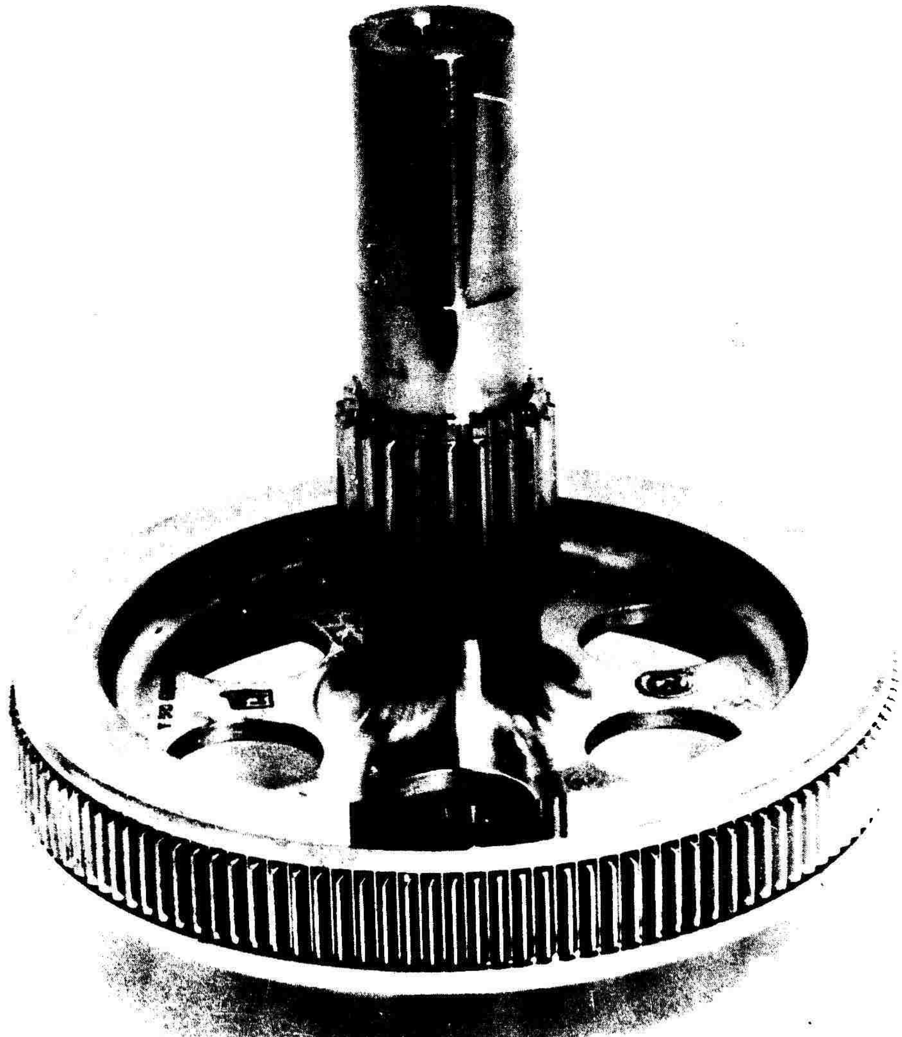


Figure 8. Second Row Roller Gear, X_2Y_2

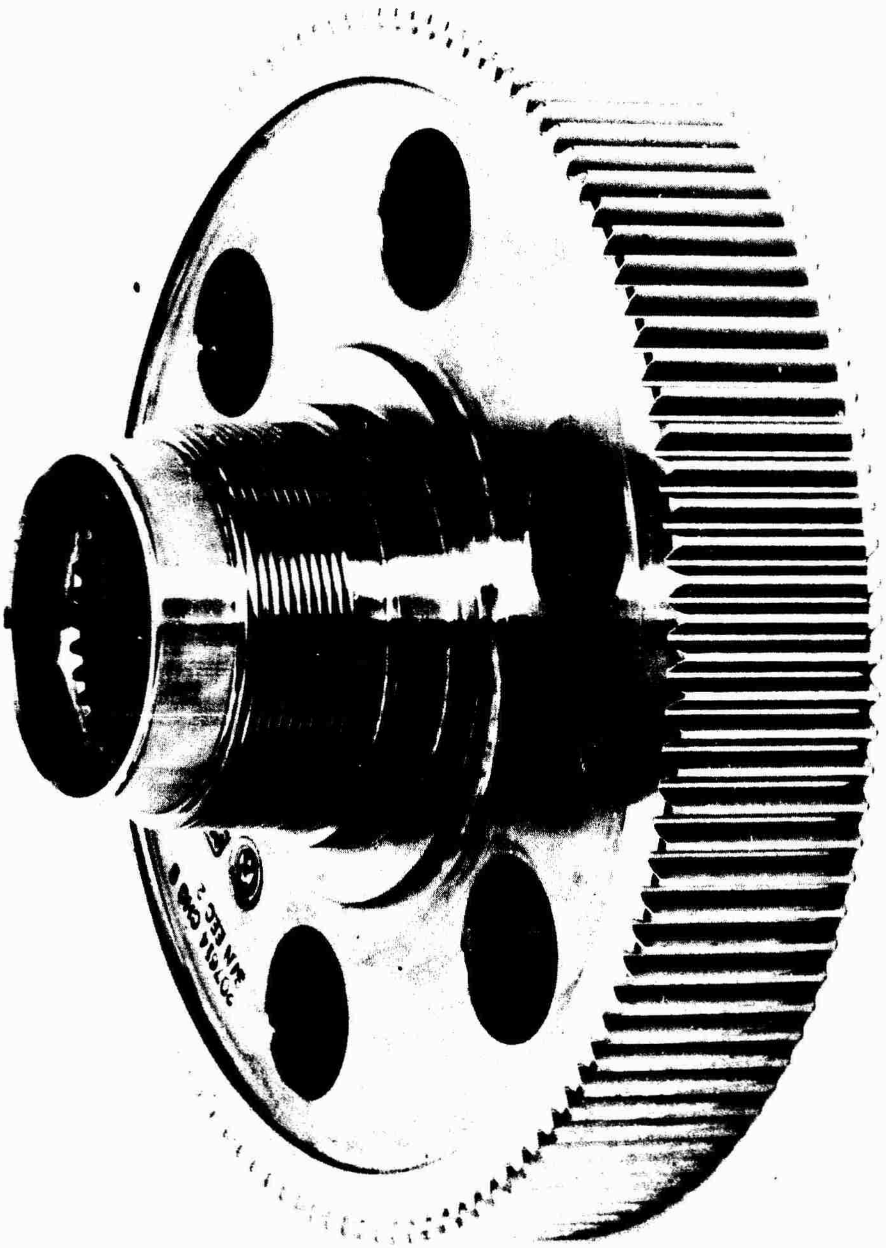


FIGURE 9. OUTPUT SUN GEAR

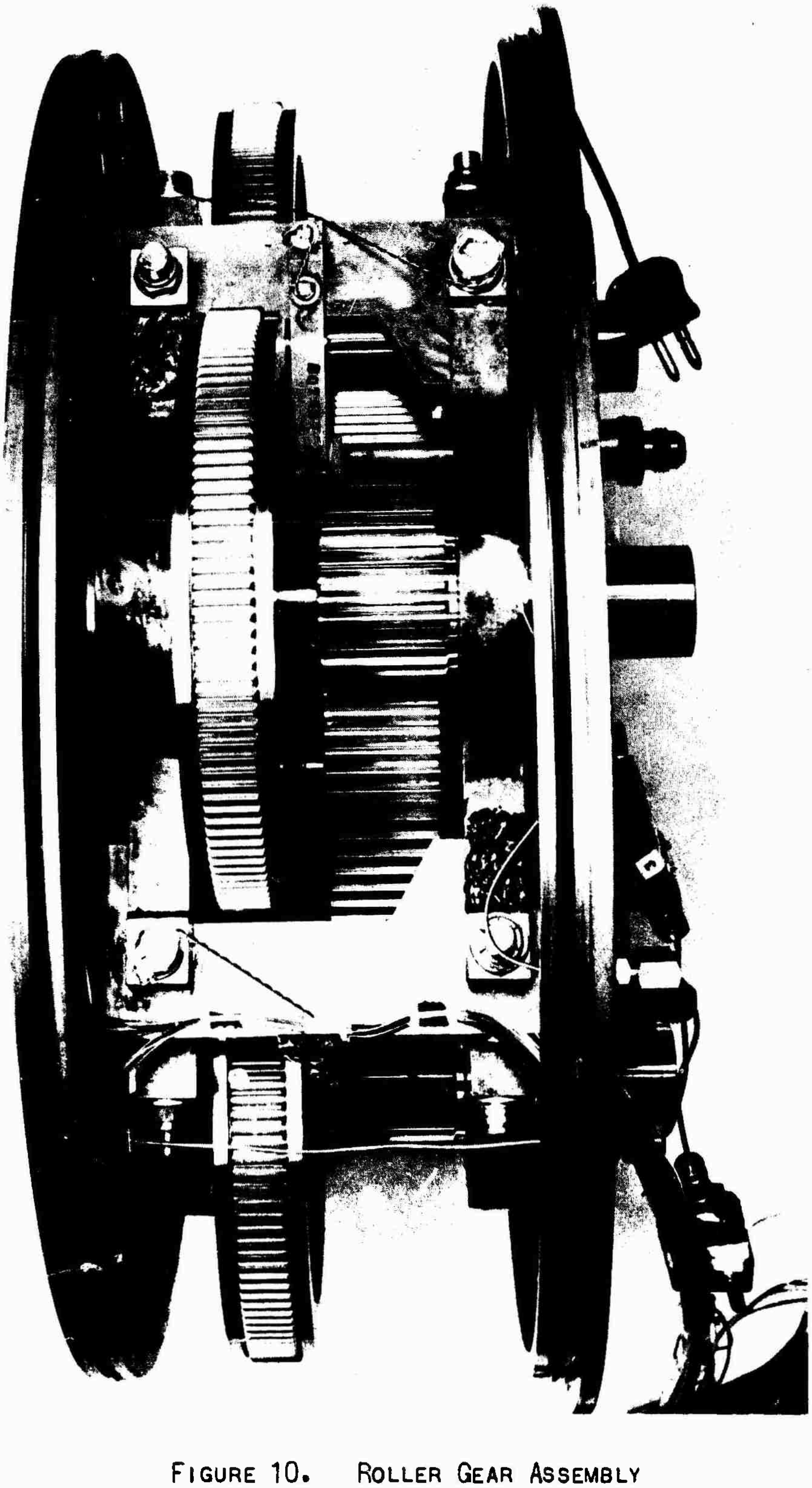


FIGURE 10. ROLLER GEAR ASSEMBLY

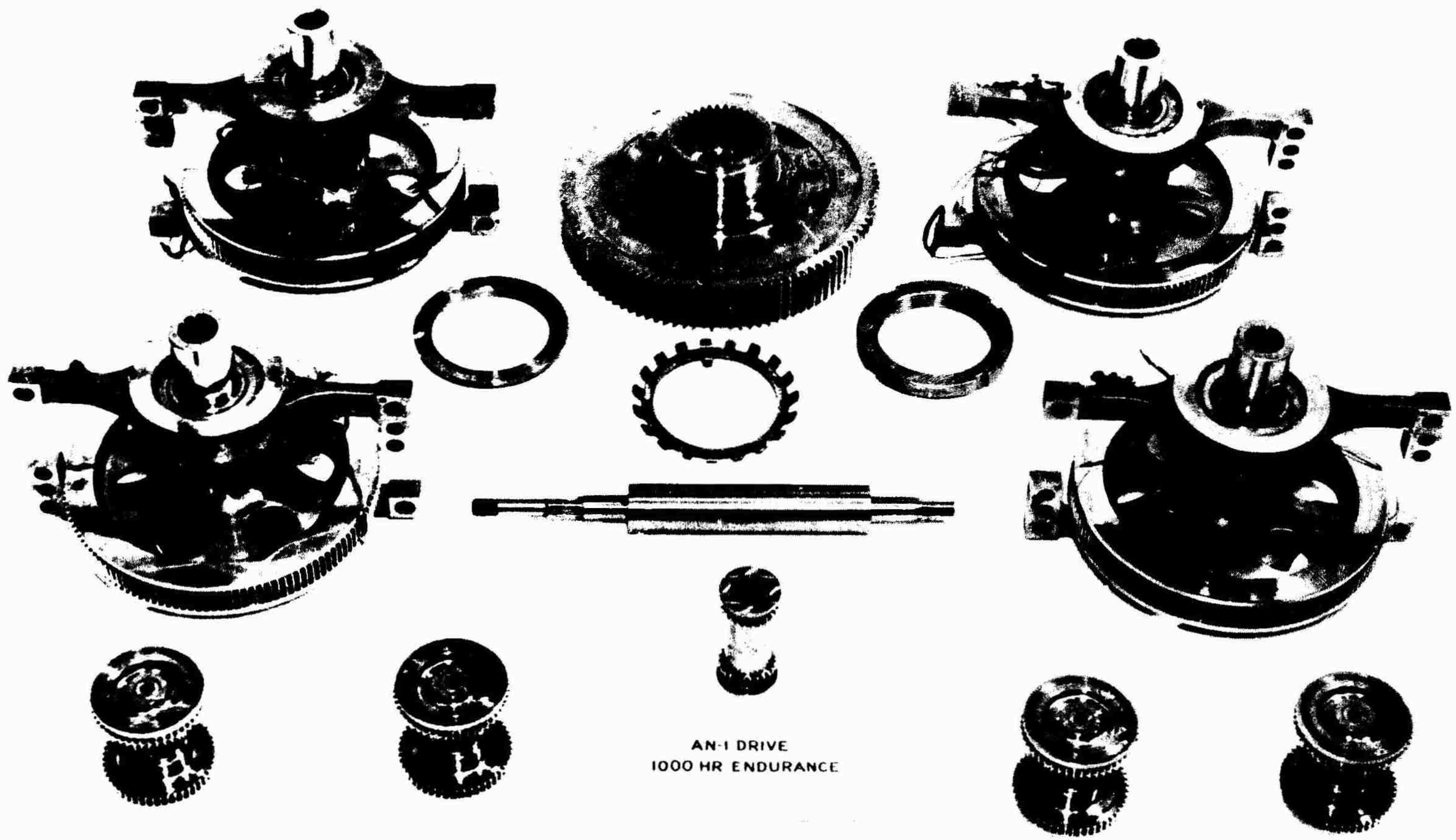


FIGURE 11. ROLLER GEAR ELEMENTS

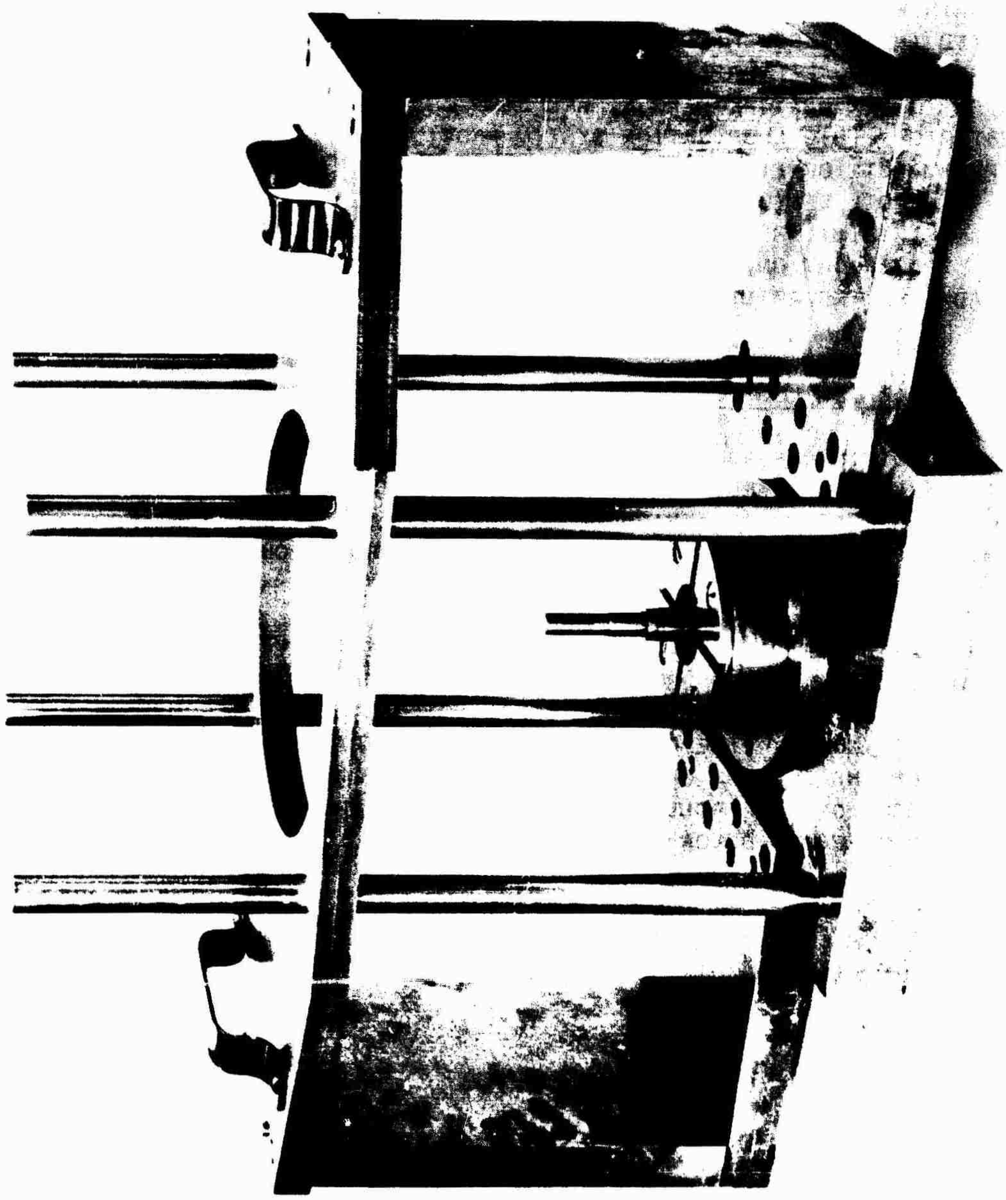


Figure 12. Assembly Fixture

2. LUBRICATION

EVERY GEAR MESH IN THE ROLLER GEAR DRIVE WAS LUBRICATED BY A PRESSURE JET, GENERALLY ORIENTED IN THE DIRECTION OF THE MESH, AND WHERE POSSIBLE ON THE OUTGOING SIDE OF THE MESH. THE ROLLER CONTACTS WERE LUBRICATED ONLY FROM THE ENSUING OIL SPRAY OF THE GEAR MESHES. A GRAVITY DRAIN WAS USED FOR EACH TEST UNIT WITH ELECTRIC CHIP DETECTORS MOUNTED IN AN ELBOW FITTING FROM EACH TEST BOX. A PUMP, PRESSURE REGULATOR, FILTER (10 MICRON), FLOWMETERS, PRESSURE SHUTDOWN SWITCHES, HEAT EXCHANGER, AND SUMP ARE UTILIZED ON THE BASIC TEST RIG. THIS LUBRICATION SYSTEM PROVIDED APPROXIMATELY 1.5 GALLONS PER MINUTE OF FILTERED OIL AT A PRESSURE OF 30 PSI TO EACH TEST BOX. THE OIL SELECTED FOR THE TEST IS A COMMERCIALLY AVAILABLE OIL AND IS SIMILAR IN MANY CHARACTERISTICS TO OILS MEETING THE MIL-L-7808 SPECIFICATION, BUT IT IS HIGHER IN VISCOSITY AND HAS A HIGHER LOAD-CARRYING ABILITY.

3. BASIC LOAD AND STRESS CALCULATIONS (REFERENCE FIGURE 2)

THE DRIVE OUTPUT TORQUE IS COMMON TO THE 600 RPM OUTPUT SHAFT OF THE CLOSING TEST-RIG GEARBOXES. THE CLOSING BOXES HAVE A BUILT-IN RATIO OF 5:1, RESULTING IN A HIGH-SPEED SHAFT OF 3000 RPM. THE PREDETERMINED PRELOAD TORQUE IS APPLIED TO THE TORQUE COUPLING ON THIS 3000-RPM SHAFT.

THE ROLLER GEAR DRIVE PRELOAD, DETERMINED EXPERIMENTALLY ON THE TEST RIG AS BEING THE LOAD WHICH PREVENTED SEPARATION OF THE ROLLER CONTACTS WITH 350 FOOT-POUNDS OF TORQUE COUPLING LOAD, VARIED FROM 800 TO 900 MICROINCHES. THE CALIBRATION OF THE STRAIN GAGES RESULTS IN A PRELOAD ON THE ROLLERS OF APPROXIMATELY 600 POUNDS. THIS PRELOAD ROLLER FORCE ON X_2Y_2 ROLLER GEAR HAS BEEN DETERMINED TO PRODUCE CONTACT ROLLER STRESSES BELOW 250,000 PSI.

Table 2. Basic Gear Characteristics

Gear		A	X_1	Y_1	X_2	Y_2	C
No. of Teeth	Early Test	24	48	20	130	21	112
	Endurance Test	24	48	20	130	29	104
	Overload Test	24	48	20	130	29	104
RPM	Early Test	42,000	21,000	21,000	3,270	3,270	600
	Endurance Test	28,000	14,000	14,000	2,154	2,154	600
	Overload Test	28,000	14,000	14,000	2,154	2,154	600
Tangential Load	Early Test	187#		205#		1800#	
	Endurance Test	187#		205#		1290#	
	Overload Test	281#		307#		1940#	
Face Width	Early Test	.544		.703		1.75	
	Endurance Test	.544		.703		1.50	
	Overload Test	.544		.703		1.50	
Bending Stress	Early Test	22,800 psi		17,900 psi		42,000 psi	
	Endurance Test	22,800 psi		17,900 psi		34,000 psi	
	Overload Test	34,200 psi		26,850 psi		51,000 psi	
Hertz Comp. Stress	Early Test	123,500 psi		101,000 psi		167,000 psi	
	Endurance Test	123,500 psi		101,000 psi		126,000 psi	
	Overload Test	151,164 psi		123,600 psi		154,200 psi	

TEST RESULTS

1. EARLY TEST

PRIOR TO THE BEGINNING OF THE ENDURANCE TESTING OF THE TRW ROLLER DRIVE TO MEET CONTRACTUAL REQUIREMENTS, TRW HAD ACCUMULATED APPROXIMATELY 60 HOURS OF TEST TIME. THIS TESTING WAS DONE ON A BOX WHICH HAD A BUILT-IN RATIO OF 70:1. THE BASIC DIFFERENCE IN THIS DRIVE FROM ALL OTHER DRIVES SUBSEQUENTLY TESTED WAS THAT IT HAD A SMALLER PINION, Y_2, IN MESH WITH A LARGER OUTPUT SUN GEAR, C. THE FIRST AND SECOND STAGE ROLLER GEARS HAVE SEEN CONTINUED USAGE IN ALL THE TESTING PERFORMED TO DATE. WHEN MOUNTED IN THE TEST RIG, THE APPLIED PRELOAD TORQUE REACHED 6300 INCH-POUNDS FOR A 300-HORSEPOWER LOAD WHEN THE INPUT SHAFT ROTATED AT 42,000 RPM. THE HIGH-SPEED SHAFT OF THE CLOSING BOXES, UPON WHICH THE TORQUE COUPLING IS MOUNTED, ROTATED AT 3000 RPM. DURING THE EARLY TESTING, MEASUREMENTS WERE MADE AT VARIOUS SPEEDS AND LOAD LEVELS AND RECORDED ON A PERFORMANCE CHART (FIGURE 16). THE DESCRIPTION OF THE LOADING CONDITIONS AND GEAR STRESSES FOR THIS TESTING APPEARS IN THE SECTION OF THIS REPORT ENTITLED "BASIC STRESS CALCULATIONS" AND IS IDENTIFIED AS THE "EARLY TEST". IN EVALUATING THE DATA, IT MUST BE REMEMBERED THAT THE HARDWARE AS ASSEMBLED WAS FAR FROM IDEAL IN THAT THE MESHES WERE INADVERTENTLY MISALIGNED SINCE THE OUTPUT SHAFT ASSEMBLIES, WHICH COUPLE THE DRIVES TO THE CLOSING BOXES, DID NOT EXIST AT THIS TIME. ALSO, THERE WERE NO TORSIONAL STIFFENING BARS INCORPORATED INTO THE ROLLER GEAR HOUSINGS. IN SPITE OF THESE SERIOUS DEFICIENCIES, THE UNITS PERFORMED WITH A MEASURED 2-PERCENT LOSS AT DESIGN LOAD.

AT THE END OF 60 HOURS OF TESTING, SEVERAL Y_2 PINION GEARS WERE FAILED AND THE MOST CONVENIENT WAY TO REPAIR THE DRIVE AT THAT TIME WAS TO PRESS LARGER Y_2 PINION GEARS OVER THE FAILED PINIONS AND USE SMALLER OUTPUT GEARS, C, SINCE THE CENTER DISTANCE REMAINED CONSTANT. THIS RESULTED IN A LOWERING OF THE GEAR RATIO TO 46:6. THE FAILURE IN THE Y_2 PINION GEARS APPEARED TO BE AN EXCELLENT EXAMPLE OF CASE CRUSHING, AND SINCE WE WERE UNDER THE REQUIRED EFFECTIVE CASE DEPTH, WE ATTRIBUTED THE FAILURE TO SUCH A CAUSE AND OVERLOOKED THE PRIME CAUSE, WHICH WAS REALLY THE MISALIGNMENT OF THE Y_2 GEARS UNDER RUNNING LOAD.

BEFORE STARTING THE ENDURANCE TEST WITH THE REBUILT HARDWARE FOR THE 46:6 REDUCTION RATIO, WE DETERMINED THAT THE PRELOAD TORQUE WOULD HAVE TO BE LOWERED TO 4200 INCH-POUNDS ON THE 3000-RPM CLOSING BOX SHAFT. THIS PRELOAD TORQUE IS REQUIRED IN ORDER TO MAINTAIN THE SAME GEAR STRESSES FOR THE FIRST TWO MESHES THAT EXISTED IN THE "EARLY TEST".

As indicated in the section of this report entitled "Basic Load and Stress Calculations," this effectively results in a drive horsepower of 200. Consequently, it was decided to begin testing at a preload torque coupling shaft speed of 4500 RPM, which would have resulted in a horsepower of 300 and would have been equal to that of the "Early Test". Before running at this higher speed, an attempt was made to recalibrate the basic rig without the product boxes in the test loop. A serious vibration appeared at a speed of 3800 RPM, and there was pronounced knocking at 4500 RPM in the commercial closing boxes. The closing boxes are basically designed to run at 1750 RPM and they were extended considerably when running at a speed of 3000 RPM. Consequently, it was decided to start the endurance run with a preload torque of 4200 inch-pounds applied on the closing box shaft which would operate at an input speed of 3500 RPM.

2. ENDURANCE TEST

The endurance test started April 24, 1964, and ended on October 15, 1964, with a total accumulated time of 1117.8 hours. The loading and gear stresses for this endurance run are exactly as shown in the section entitled "Basic Load and Stress Calculations," with the one exception of the fact that the preload torque of 4200 inch-pounds was applied to the input shaft of the closing boxes which was driven at a speed of 3500 RPM. This speed was maintained at 3500 RPM for the first 106 hours of endurance testing and the drive horsepower under these conditions was equal to 200 x 3500/3000 = 233 horsepower

On May 1, 1964, a failure was discovered in test box number 2 after 106 hours of testing. This failure was quite significant because several very important changes were made to the roller drives at that time. The failure occurred in only one X_2Y_2 roller gear and it was quite obvious that overloading of the pinion tooth Y_2 was such as to cause the tooth breakage, as shown in Figure 13. In fact, at that time it was quite evident that the unit had random misalignment on all the output pinion meshes.

A COMPLETE MAGNAFLUX INSPECTION WAS MADE ON THE ROLLER GEAR ELEMENTS AND THE ONLY OTHER VISIBLE DAMAGE WAS THE LOOSE ROLLERS ON X_1Y_1 ROLLER GEARS. THE LOOSENING OF THE ROLLERS WAS TO BE EXPECTED BECAUSE THE INTERFERENCE FIT WAS NOT SUFFICIENT IN THE ORIGINAL DESIGN. AFTER REASSEMBLY OF THE TWO UNITS WITH TIGHTER FITTING ROLLERS ON X_1Y_1 ROLLER GEAR, AND HAVING REPLACED THE BROKEN X_2Y_2 ROLLER GEAR, IT WAS DECIDED THAT THE MESH OF THE OUTPUT GEARS WOULD BE OBSERVED WITH RED LEAD COMPOUND AND WHILE UNDER FULL PRELOAD TORQUE. AFTER EXHAUSTIVE ATTEMPTS TO PROVIDE FOR PROPER ADJUSTMENT OF TOOTH CONTACT UNDER PRELOAD CONDITIONS HAD PROVED FUTILE, SEVERAL IMPOSSIBLE CONDITIONS WERE MADE APPARENT. FIRST, THERE WAS NOT ENOUGH TORSIONAL RIGIDITY IN THE ROLLER GEAR HOUSING; SECOND, THE OUTPUT SUN GEAR WAS ACTING AS ONE-HALF OF A FLEXIBLE GEAR TYPE COUPLING; AND THIRD, AN ASSEMBLY FIXTURE WAS NEEDED TO SPEED UP ASSEMBLY PROCEDURES.

TESTING WAS RESUMED ON AUGUST 21, 1964, AFTER THE AFOREMENTIONED PROBLEMS HAD BEEN CAREFULLY STUDIED. SPEED LIMITATIONS ON COMMERCIAL CLOSING GEARBOXES REQUIRED THE TORQUE COUPLING SHAFT SPEED TO BE REDUCED FROM 3500 RPM TO 3000 RPM WHILE MAINTAINING THE PRELOAD TORQUE OF 4200 INCH-POUNDS, RESULTED IN AN EFFECTIVE 200 HORSEPOWER TEST. THE SPEED REMAINED AT THIS LEVEL FOR THE REMAINDER OF THE ENDURANCE TEST. DUE TO THIS MODIFICATION TO THE TEST PROGRAM, THE 1,000-HOUR ENDURANCE TEST IS CONSIDERED TO HAVE STARTED WITH THE TEST CONDUCTED ON AND SUBSEQUENT TO AUGUST 21, 1964. AN ASSEMBLY FIXTURE AND A SET OF FOUR TORSIONAL STIFFENING BARS WERE PROCURED FOR EACH TEST BOX, AND AN OUTPUT SHAFT ASSEMBLY IN EACH BOX PERMITTED COUPLING WITH THE CLOSING BOXES, WHICH IN NO WAY INTERFERED WITH THE PROPER MESHING OF THE FOUR DRIVE PINION GEARS, Y_2, WITH THE OUTPUT SUN GEAR, C. THE MESH PATTERN WAS STUDIED CAREFULLY UNDER FULL PRELOAD TORQUE.

ON AUGUST 23, 1964, AFTER HAVING RUN FOR 11.5 HOURS TO ACCUMULATE 117.4 HOURS, AN AUTOMATIC SHUTDOWN OCCURRED BECAUSE OF AN OVER-TEMPERATURE CONDITION ON THE INPUT SUN ROLLER OF TEST BOX NUMBER 2. TEARDOWN OF THE UNIT DISCLOSED A DISCONCERTING FAILURE ON THE FIRST ROW ROLLER GEAR, X_1Y_1, AS SHOWN IN FIGURE 14. ONE SHAFT OF Y_1 GEAR, WHICH EXTENDS IN BOTH DIRECTIONS AND UPON WHICH THE X_1 ROLLER GEARS ARE PRESS FITTED, HAD FAILED, BUT APPARENTLY CONTINUED TO OPERATE FOR A CONSIDERABLE NUMBER OF HOURS BEFORE THE OVER TEMPERATURE SIGNAL ON THE INPUT SUN GEAR PROVIDED AUTOMATIC RIG SHUTDOWN. THE ASSEMBLY OF THE ROLLER GEAR IS SUCH THAT A FAILURE OF THIS TYPE MAINTAINED ALL ELEMENTS IN THEIR TRAPPED OR PREASSEMBLED POSITION. BECAUSE OF THE RESULTING RUBBING OF THE FRACTURED SURFACES IT WAS DIFFICULT TO ISOLATE THE CAUSE OF THIS FAILURE. THERE WAS AMPLE EVIDENCE TO INDICATE THAT THE Y_1 SHAFT HAD EXPERIENCED TEARING AND GOUGING OF ITS SURFACE WHEN THE ROLLER GEARS WERE INITIALLY PRESSED UPON THIS SHAFT BY THE GEAR VENDOR, AS SHOWN IN FIGURE 15.

ON AUGUST 23, 1964, THE TESTING OF THE UNITS WAS BEGUN AFTER REPLACING THE FAILED GEAR OF TEST UNIT NUMBER 2, AND CAREFULLY ADJUSTING THE ASSEMBLIES UNDER LOAD, AS PREVIOUSLY DESCRIBED. TESTING CONTINUED WITHOUT ANY SIGNIFICANT PROBLEMS UNTIL SEPTEMBER 9, 1964, WHEN AFTER 331 HOURS OF ACCUMULATED TESTING, THE RIG EXPERIENCED AN AUTOMATIC SHUTDOWN BECAUSE OF AN OVER TEMPERATURE SIGNAL ON THE INPUT SUN ROLLER OF TEST BOX NUMBER 1. TEARDOWN DISCLOSED A FAILURE ON A FIRST ROW ROLLER GEAR, X_1Y_1, ALMOST EXACTLY AS HAD OCCURRED PREVIOUSLY ON THE Y_1 SHAFT. THIS FAILURE OCCURRED IN THE SHARP FILLET AREA WHERE APPARENTLY THE CHAMFER, IN THE X_1 GEAR WHICH IS PRESSED ONTO THE SHAFT, WAS NOT SUFFICIENT TO ENSURE CLEARANCE. THIS CONDITION RESULTED IN AN EXTREME STRESS RISER, AND IS BELIEVED TO BE THE CAUSE OF FAILURE. BOTH OF THE PRECEDING FAILURES COULD HAVE BEEN PREVENTED BY A MORE THOROUGH INSPECTION AND QUALITY CONTROL WHICH WAS NOT USED FOR THE PRESS-FITTED ASSEMBLIES. IN ADDITION, THE TEARDOWN DISCLOSED EXTREME FRETTING ON THE QUILL-SHAFT COUPLING BETWEEN THE TWO TEST UNITS, AND BECAUSE OF AN OBVIOUS TORSIONAL CRACK IN THE SPLINE TOOTH AREA THE QUILL COUPLING WAS ALSO REPLACED. AT THIS TIME, INSULATION WAS PLACED AROUND BOTH TEST BOXES TO ELIMINATE TEMPERATURE DIFFERENTIALS AND THEIR RESULTING EFFECT UPON THE PRELOAD FORCES OF THE ROLLER CONTACTS. IN ADDITION, IT WAS FELT THAT A CORRELATION COULD BE DETERMINED BETWEEN THE MEASURED MECHANICAL LOSSES OF THE TEST BOXES AND THE CALCULATED THERMAL OIL LOSSES.

ON SEPTEMBER 12, 1964, THE UNITS WERE PLACED BACK ON TEST AND REMAINED ON TEST CONTINUOUSLY UNTIL OCTOBER 15, 1964, HAVING ACCUMULATED 1117.8 HOURS OF TESTING. AT THIS TIME, TEST BOX NUMBER 2 ACCUMULATED 1000 HOURS OF CONTINUOUS TEST WHILE TEST BOX NUMBER 1 HAD A TOTAL OF 786.8 CONTINUOUS HOURS WITHOUT ANY EVIDENCE OF MALFUNCTION. TEARDOWN AT THIS TIME DISCLOSED THAT ALL ROLLER AND GEAR SURFACES WERE IN EXCELLENT CONDITION AND CAPABLE OF EXTENDED AND PROLONGED SERVICE AT HIGHER LOADS. THE ONLY ELEMENT WHICH SHOWED CONCERN WAS THE CONNECTING QUILL SHAFT. THERE WAS CONSIDERABLE WEAR AND FRETTING ON THE SPLINE TEETH. THE CALCULATED COMPRESSIVE STRESS ON THE SPLINE TEETH IS 5,600 PSI WHEN OPERATING AT 28,000 RPM, WITH AN INDICATED MISALIGNMENT OF .001 INCH PER INCH OF LENGTH. FOR OPERATING CONDITIONS SUCH AS THESE, AND WITH NO MEANS PROVIDED IN THIS ROLLER GEAR DRIVE FOR OIL LUBRICATION ON THE SPLINE TEETH, THIS SERVICE MET EXPECTATIONS AND CAN BE IMPROVED IN A NEW DESIGN.

THE DATA SHOWN IN **TABLE 3 AND TABLE 4** REPRESENTS A SUMMARY OF THE TEST RESULTS FOR A TOTAL OF 236 HOURS. THIS PERIOD IS PRESENTED AS BEING TYPICAL OF THE DATA RECORDED AND COVERS THE PERIOD FROM 862 HOURS TO 1098 HOURS OF ACCUMULATED TESTING.

From the data shown in Tables 3 and 4, the following information is determined:

Average oil temperature rise, box number 1 = 25^0

Average oil temperature rise, box number 2 = 20.68^0

Average oil flow, box number 1 = 1.545 GPM

Average oil flow, box number 2 = 1.557 GPM

Average dynamometer torque = 22.256 foot-pounds (at 3000 RPM)

Average operating oil temperature in closing boxes = 120^0 F

The measured dynamometer torque for the test rig alone without test boxes = 11.50 foot-pounds (at 3000 RPM)

Total system required =

$$\frac{3000 \times 22.256 \times 12}{63{,}000} = 12.718 \text{ HP} \qquad (5)$$

Test rig required =

$$\frac{3000 \times 11.50 \times 12}{63{,}000} = 6.571 \text{ HP} \qquad (6)$$

Total required by two test boxes =

$$12.718 - 6.571 = 6.147 \text{ HP}$$

Load in test boxes =

$$\frac{4200 \times 3000}{63{,}000} = 200 \text{ HP}$$

Average percentage loss per test box =

$$\frac{6.147}{2 \times 200} \times 100 = 1.536\% \qquad (7)$$

THE SPECIFIC HEAT FOR THE OIL IS $C_p = .47$ AND THE SPECIFIC GRAVITY IS .91. THE FORMULA USED TO DETERMINE THERMAL LOSSES BY MEASURING OIL FLOWS AND TEMPERATURE RISES IS:

$$Q = \frac{M \; C_P \qquad T}{42.4} \quad , \qquad (8)$$

WHERE Q = HORSEPOWER

M = MASS FLOW OF OIL (LB.)

C_p = SPECIFIC HEAT OF OIL

T = TEMPERATURE RISE IN OIL, ^{O}F.

USING A SPECIFIC GRAVITY OF .91, THE WEIGHT OF THE OIL HAS BEEN DETERMINED TO BE 7.59 POUNDS PER GALLON.

$$\text{FOR BOX NUMBER 1, } Q = \frac{1.545 \times 7.59 \times .47 \times 25}{42.4} = 3.25 \text{ HP} \qquad (9)$$

$$\text{FOR BOX NUMBER 2, } Q = \frac{1.557 \times 7.59 \times .47 \times 20.68}{42.4} = 2.71 \text{ HP} \qquad (10)$$

$$\text{TOTAL HEAT LOSS} = 3.25 + 2.71 = 5.96 \text{ HP}$$

CONSIDERING THAT THE RADIATED LOSSES FROM THE INSULATED TEST BOXES HAVE BEEN IGNORED SINCE THEY ARE BELIEVED TO BE SMALL, EXCELLENT CORRELATION BETWEEN THE MEASURED MECHANICAL LOSSES AND THE THERMAL LOSSES FOR THE TEST BOXES EXISTS.

THE REMAINING TEMPERATURES IN TABLES 3 AND 4 ARE SELF-EXPLANATORY. THE SUN ROLLER TEMPERATURES ARE TAKEN BY A THERMOCOUPLE, MOUNTED AS CLOSE AS POSSIBLE TO THE HIGH-SPEED INPUT SUN ROLLER WHICH ROTATES AT 28,000 RPM. THE BEARING LINK TEMPERATURES INDICATE THAT THE ROLLER GEAR ASSEMBLY HAS A UNIFORM TEMPERATURE THROUGHOUT THE ROLLER GEAR ASSEMBLY AND THE HOUSING SKIN TEMPERATURES ARE ABOUT AS TO BE EXPECTED, CONSIDERING ALL OTHER TEMPERATURES.

THE STRAIN GAGE READINGS REPRESENT DEPARTURE LOADS FROM THE PRESET LOAD WHICH EXISTS AT THE BEGINNING OF TEST. AT THE BEGINNING OF TEST, THE STRAIN INDICATOR READINGS ARE ALL SET FOR A ZERO READING. IT IS EVIDENT THAT THERE IS A STABILITY TO THE INDIVIDUAL RECORDED STRAIN READINGS BUT TRW BELIEVES THAT THE ABSOLUTE VALUES OF THESE READINGS CANNOT BE USED RELIABLY TO PREDICT GEAR MESH LOAD SHARING WHILE OPERATING, BECAUSE THE CONNECTING WIRES TO THE STRAIN GAGES ARE ALL OF VARYING LENGTHS, TEMPERATURE EFFECTS INFLUENCE THE STRAIN READINGS AND NO ALLOWANCE HAS BEEN DETERMINED FOR THESE STRAIN READINGS BECAUSE OF THIS TEMPERATURE EFFECT.

THE VIBRATION READINGS RECORDED ARE NOT CONSIDERED SIGNIFICANT BECAUSE THE DISPLACEMENT READINGS WERE TAKEN ON THE CLOSING BOXES AND THE READING IN G WAS TAKEN ON TEST BOX NUMBER 1, WHICH HAD AN EXTREMELY LIGHT AND FLEXIBLE SUPPORT. IN ANY CASE, THERE IS NO DATA UPON WHICH TO MAKE A COMPARISON.

3. OVERLOAD TEST

AT THE COMPLETION OF THE ENDURANCE TEST AS REQUIRED BY CONTRACTUAL ARRANGEMENTS, TRW, CONSIDERING THE EXCELLENT CONDITION OF THE ROLLER GEAR DRIVE ELEMENTS, ELECTED TO CONTINUE TESTING. THIS OVERLOAD TEST IS ESSENTIALLY A 50-PERCENT INCREASE IN TORQUE LOADING, RESULTING IN A 50-PERCENT INCREASE IN GEAR STRESSES AND EFFECTIVE HORSEPOWER OUTPUT. THE CONDITIONS FOR LOADING AND THE RESULTING STRESSES ARE AS OUTLINED IN THE SECTION ENTITLED "BASIC LOAD AND STRESS CALCULATIONS". A TOTAL OF 188 HOURS HAS BEEN ACCUMULATED AT THIS INCREASED LOAD LEVEL. AFTER 110 HOURS OF OVERLOAD TESTING, THE CONNECTING HIGH-SPEED SPLINED QUILL SHAFT BROKE. THE QUILL FAILURE WAS DEFINITELY A TORSIONAL TWIST TYPE FAILURE WITH THE CRACK ORIGINATING FROM THE HIGHLY STRESSED SPLINE ROOT DIAMETER WHERE THERE IS A SEVERE CASE OF FRETTING. THE CALCULATED SHEAR STRESS IN THIS SHAFT IS 37,500 PSI OPERATING AT 28,000 RPM WITH THIS SERIOUS FRETTING CONDITION, AND FAILURES ARE TO BE EXPECTED. DURING THE TEST, MEASUREMENTS SIMILAR TO THOSE TAKEN DURING THE ENDURANCE RUN WERE MADE. CALCULATIONS HAVE INDICATED THAT THE EFFICIENCY OF THE ROLLER DRIVES HAS IMPROVED FROM THE 98.5 PERCENT MEASURED DURING THE ENDURANCE RUN TO 98.9 PERCENT.

THE FAILED QUILL SHAFT CONNECTOR WAS REPLACED WITH A PREVIOUSLY USED AND WORN QUILL SHAFT WHICH PERMITTED TESTING FOR AN ADDITIONAL 78 HOURS. AT THIS TIME, A SIMILAR TORSIONAL FAILURE OCCURRED TO THE QUILL CONNECTOR SHAFT. THE MATERIAL USED FOR THE QUILL SHAFTS WAS 4340 STEEL WITH A HARDNESS MEASURED TO BE Rc 35. A NEW QUILL SHAFT MADE FROM 9310 STEEL WITH A HARDNESS IN THE SPLINE TOOTH AREA OF Rc 60 IS BEING PROCURED ALONG WITH A REQUIREMENT FOR SHOT PEENING IN THE SPLINE AREA. FURTHER TESTING IS CONTEMPLATED TO YIELD AT LEAST 500 HOURS OF TIME AT THIS INCREASED LOAD LEVEL.

TEARDOWN INSPECTION OF THE ROLLER DRIVE AT THE END OF 188 HOURS OF OVERLOAD TESTING DISCLOSED THAT ALL GEARS AND ROLLERS WERE IN GOOD CONDITION EXCEPT THAT A TOOTH WAS MISSING IN THE SUN GEAR, A, OF BOX NUMBER 1. METALLURGICAL INSPECTION INDICATED THAT THE FAILURE WAS DUE TO FATIGUE. THE SUN GEAR HAD RUN A CONSIDERABLE LENGTH OF TIME WITH THE BROKEN TOOTH. IN THIS CONDITION, THERE WAS NO MEASURABLE EFFECT TO THE GEARBOX PERFORMANCE. THE TEETH OF THIS GEAR EXPERIENCED OVER TEN LOAD CYCLES DURING THE PERIOD OF TESTING. AN EXPLANATION FOR THIS FAILURE WILL BE DEPENDENT UPON FURTHER TESTING AND A THOROUGH METALLURGICAL EXAMINATION OF THE BROKEN GEAR.

AN-1 TESTING (REFERENCE FIGURE 2)

EARLY TEST

$$\text{PRELOAD TORQUE} = 6300 \text{ INCH-POUNDS AT } 3000 \text{ RPM}$$

$$\text{DRIVE RATIO} = 70$$

$$\text{DRIVE OUTPUT SPEED} = \frac{3000}{5} = 600 \text{ RPM}$$

$$\text{DRIVE INPUT SPEED} = 600 \times 70 = 42{,}000 \text{ RPM}$$

$$\text{DRIVE OUTPUT TORQUE} = 6300 \times 5 = 31{,}500 \text{ INCH-LBS}$$

$$\text{DRIVE INPUT TORQUE} = \frac{31{,}500}{70} = 450 \text{ INCH-LBS}$$

$$\text{DRIVE HP} = \frac{42{,}000 \times 450}{63{,}000} = 300 \text{ HP} \qquad (11)$$

ENDURANCE TEST (1000 HOURS) (MODIFIED TEST CONDITIONS)

$$\text{PRELOAD TORQUE} = 4200 \text{ INCH-POUNDS AT } 3000 \text{ RPM}$$

$$\text{DRIVE RATIO} = 46.62$$

$$\text{DRIVE OUTPUT SPEED} = \frac{3000}{5} = 600 \text{ RPM}$$

$$\text{DRIVE INPUT SPEED} = 600\ (46.62) = 28{,}000 \text{ RPM}$$

$$\text{DRIVE OUTPUT TORQUE} = 4200 \times 5 = 21{,}000 \text{ INCH-LBS}$$

$$\text{DRIVE INPUT TORQUE} = \frac{21{,}000}{46.62} = 450 \text{ INCH-LBS}$$

$$\text{DRIVE HP} = \frac{28{,}000 \times 450}{63{,}000} = 200 \text{ HP} \qquad (12)$$

OVERLOAD TEST (50-PERCENT INCREASE IN GEAR STRESSES)

$$\text{PRELOAD TORQUE} = 6300 \text{ INCH-POUNDS AT } 3000 \text{ RPM}$$

$$\text{DRIVE RATIO} = 46.62$$

$$\text{DRIVE OUTPUT SPEED} = \frac{3000}{5} = 600 \text{ RPM}$$

$$\text{DRIVE INPUT SPEED} = 600\ (46.62) = 28{,}000 \text{ RPM}$$

$$\text{DRIVE OUTPUT TORQUE} = 6300 \times 5 = 31{,}500 \text{ INCH-LBS}$$

$$\text{DRIVE INPUT TORQUE} = \frac{31{,}500}{46.62} = 675 \text{ INCH-LBS}$$

$$\text{DRIVE HP} = \frac{28{,}000 \times 675}{63{,}000} = 300 \text{ HP} \qquad (13)$$

FIGURE 13. FAILED PINION SECTION
OF OUTER PLANETARY GEAR

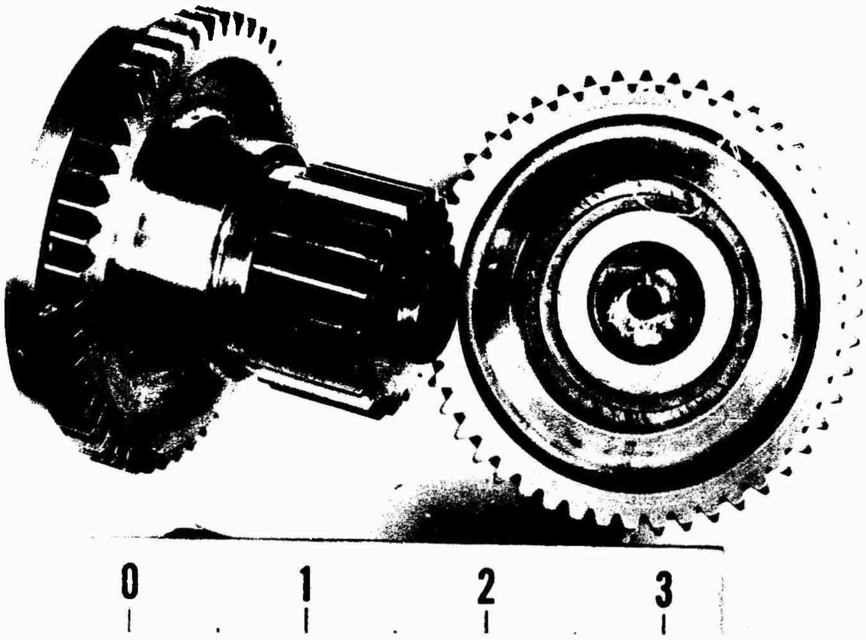


FIGURE 14. INNER PLANETARY GEAR ASSEMBLY - SHOWING FAILURE ON PINION SHAFT

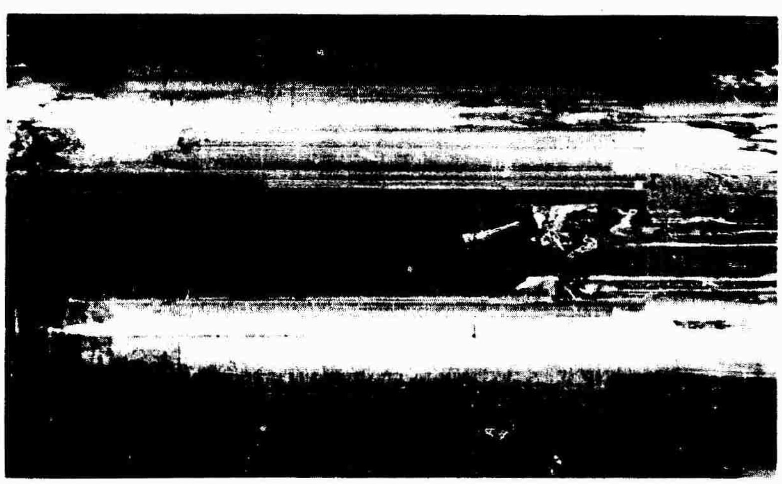


FIGURE 15. BROKEN PINION SHAFT - SHOWING SURFACE DAMAGE CAUSED BY DISASSEMBLY AND/OR ASSEMBLY

PERCENT POWER LOSS RUNNING AT A CONSTANT SPEED
AND AT CONSTANT TORQUE

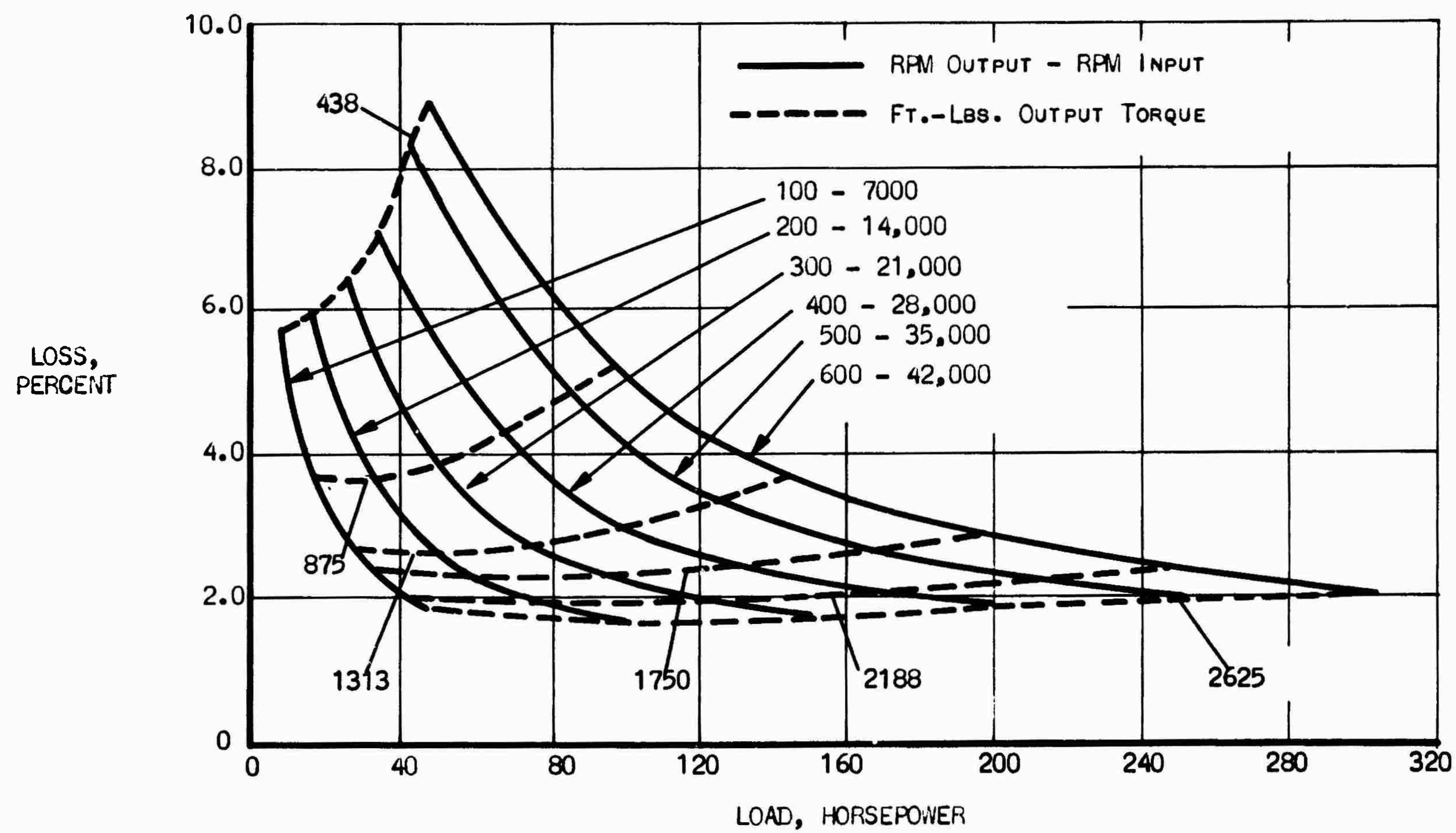


FIGURE 16. PERFORMANCE CHART
70:1 RATIO ROLLER GEAR DRIVE

Table 3
Tabulated Test Data

Roller Gear Drive
Data Sheet

Date 10/4/64

Gearbox No. 1, Part No. 207626-2 Serial No. 1

Gearbox No. 2, Part No. 207626-1 Serial No. 2

Date			10/5		10/6			10/7			10/8			10/9
Time		1600	0900	1610	0800	1315	1615	0815	1300	1600	0800	1330	1630	0800
Applied Load	Ft/Lbs	350	350	350	350	350	350	350	350	350	350	350	350	350
Dynamometer Speed	RPM	3000	3000	3000	3007	3000	3000	3000	3000	3000	3000	3000	3006	3000
Torque	Ft/Lbs	22.3	22.3	22.5	22.3	22.2	22.1	22.3	22.4	22.4	22.3	22.3	22.3	22.25
Oil Pressure, Box #1	PSI	28.0	28.0	28.0	27.8	27.0	27.2	27.0	27.5	27.5	27.2	27.0	27.5	26.7
Oil Pressure, Box #2	PSI	27.6	27.5	27.5	27.3	26.5	26.8	27.8	27.0	27.0	26.8	26.5	27.0	26.5
Oil Flow, Box #1	GPM	1.57	1.57	1.50	1.57	1.48	1.53	1.58	1.47	1.50	1.44	1.41	1.47	1.58
Oil Flow, Box #2	GPM	1.57	1.58	1.54	1.56	1.46	1.52	1.58	1.50	1.52	1.46	1.45	1.50	1.58
Ambient Temperature	°F	96	92	93	95	90	91	94	103	98	94	96	97	96
Commercial Temp., Box #1	°F	120	118	120	121	118	119	118	119	119	120	119	120	120
Commercial Temp., Box #2	°F	122	119	121	121	119	120	118	120	121	121	120	120	120
Oil Inlet Temp., Box #1	°F	160	160	160	160	161	150	155	163	159	162	158	166	166
Oil Inlet Temp., Box #2	°F	160	160	160	160	161	150	155	163	159	162	158	166	166
Exit Oil Temp., Box #1	°F	180	180	180	185	178	180	186	178	178	177	178	180	190
Exit Oil Temp., Box #2	°F	179	178	179	181	171	174	178	174	177	173	172	176	184
Sun Roller Temp., Box #1	°F	196	197	196	196	195	191	194	198	195	197	195	198	200
Sun Roller Temp., Box #2	°F	196	196	196	197	195	191	194	198	195	197	195	198	200
Yoke Temp., Box #1	°F	189	188	188	190	193	194	196	190	188	190	192	188	193
Yoke Temp., Box #2	°F	189	188	188	190	191	194	196	190	188	190	192	188	193
Skin Temp., Box #1	°F	171	171	171	168	171	171	171	172	171	170	171	168	170
Skin Temp., Box #2	°F	171	171	171	168	171	172	171	172	171	170	171	168	170
Vibration, Box #1	G's													
Vibration, Box #2	G's													
Vibration, Horizontal	G's	6.9	5.8	5.7	5.7	5.7	6.0	6.5	5.6	5.6	5.5	5.5	5.5	5.9
Strain Gage 1	u/in	-30	-40	-40	-30	-50	-50	-50	-35	-30	-40	-40	-30	-30
Strain Gage 2	u/in	-120	-130	-125	-120	-130	-135	-130	-120	-120	-120	-120	-120	-125
Strain Gage 3	u/in	100	85	90	95	95	95	95	105	100	100	110	100	90
Strain Gage 4	u/in	-5	-25	-20	-20	-35	-40	-40	-20	-15	-15	-20	-10	-20
Strain Gage 5	u/in	-45	-50	-40	-50	-45	-50	-65	-40	-40	-45	-40	-40	-50
Strain Gage 6	u/in	-10	-30	-20	-15	-40	-45	-50	-20	-10	-25	-25	-10	-25
Strain Gage 7	u/in	-140	-145	-135	-150	-125	-130	-120	-120	-135	-120	-120	-135	-140
Strain Gage 8	u/in	45	30	35	35	30	25	25	40	40	40	45	-55	30
Strain Gage 9	u/in	0	20	0	-30	10	5	10	0	-5	-20	0	-10	-15
Strain Gage 10	u/in	-60	-70	-70	-90	-75	-70	-75	-60	-80	-65	-30	-65	-80
Strain Gage 11	u/in	20	20	30	0	10	10	35	15	0	25	15	20	0
Strain Gage 12	u/in	5	5	0	-25	-10	-10	-20	-15	-20	0	0	5	-15
Strain Gage 13	u/in	40	40	40	10	30	30	20	25	20	35	35	40	20
Strain Gage 14	u/in	-45	-30	-25	-35	-35	-35	-40	-35	-50	-25	-25	-25	-50
Strain Gage 15	u/in	60	80	80	30	75	70	60	65	45	70	75	85	50
Strain Gage 16	u/in	200	225	225	165	200	200	185	200	185	210	200	225	180
Vibration, Commercial, Box #1	Mils	.35	.38	.40	.40	.35	.40	.40	.40	.40	.37	.35	.40	.35
Vibration, Commercial, Box #2	Mils	.69	.67	.67	.70	.65	.70	.65	.68	.70	.65	.62	.70	.68
Time on Test	Hrs.	862.2	879.2	886.4	902.3	907.6	910.5	926.0	931.0	934.0	950.0	955.0	958.0	974.0

Table 4
Tabulated Test Data

ROLLER GEAR DRIVE
DATA SHEET

Date 10/9/64

Gearbox No. 1, Part No. 207626-2 Serial No. 1

Gearbox No. 2, Part No. 207626-1 Serial No. 2

Date		10/9		10/10	10/12			10/13			10/14			
Time		1310	1600	1100	0815	1300	1600	0810	1300	1600	0745	1010	1035	1300
Applied Load	Ft/Lbs	350	350	350	350	350	350	350	350	350	350		350	350
Dynamometer Speed	RPM	3000	3000	3000	3000	3000	3000	3000	3000	3005	3000		3000	3000
Torque	Ft/Lbs	22.0	•22.2	22.3	22.4	22.3	22.25	21.7	22.2	22.2	22.1		22.7	22.1
Oil Pressure, Box #1	PSI	27.0	27.5	29.0	26.0	28.2	28.5	28.5	28.6	28.7	26.8		26.8	26.7
Oil Pressure, Box #2	PSI	26.8	27.0	23.5	27.5	27.8	28.0	28.0	28.0	28.0	26.3		26.4	26.3
Oil Flow, Box #1	GPM	1.46	1.63	1.65	1.55	1.62	1.63	1.60	1.62	1.65	1.53		1.53	1.49
Oil Flow, Box #2	GPM	1.50	1.64	1.64	1.55	1.62	1.63	1.60	1.62	1.65	1.55		1.55	1.55
Ambient Temp.	°F	97	99	84	82	96	97	100	99	100	94		97	97
Commercial Temp., Box #1	°F	120	120	115	114	119	119	120	120	120	119		108	118
Commercial Temp., Box #2	°F	120	122	115	114	120	120	121	121	121	119		108	119
Oil Inlet Temp., Box #1	°F	161	150	166	165	162	160	162	160	159	160		158	161
Oil Inlet Temp., Box #2	°F	161	150	166	165	162	160	162	160	159	160		158	161
Exit Oil Temp., Box #1	°F	182	190	185	190	192	190	191	192	191	191		191	194
Exit Oil Temp., Box #2	°F	178	186	184	184	186	187	186	195	186	185		184	184
Sun Roller Temp., Box #1	°F	197	201	201	199	200	200	200	198	199	200		200	200
Sun Roller Temp., Box #2	°F	197	201	201	198	200	200	200	198	199	200		198	200
Yoke Temp., Box #1	°F	188	194	193	198	197	198	198	198	199	197		197	197
Yoke Temp., Box #2	°F	189	194	193	198	199	198	197	198	199	197		197	197
Skin Temp., Box #1	°F	169	171	173	176	177	176	178	176	177	176		176	176
Skin Temp., Box #2	°F	170	172	173	177	177	177	178	177	177	176		176	176
Vibration, Box #1	G's													
Vibration, Box #2	G's													
Vibration, Horizontal	G's	5.4	5.3	5.6	5.5	5.7	5.8	5.7	5.8	5.8	5.8	5.8		5.7
Strain Gage 1	u/in	- 25	- 40	- 60	- 40	- 40	- 40	- 35	- 35	- 35	- 45			- 40
Strain Gage 2	u/in	-120	-125	-130	-130	-135	-135	-130	-130	-120	-135			-130
Strain Gage 3	u/in	100	110	90	75	90	95	100	100	100	80			- 40
Strain Gage 4	u/in	- 10	- 25	- 30	- 25	- 25	- 20	- 20	- 15	- 15	- 35			- 20
Strain Gage 5	u/in	- 40	- 50	- 50	- 50	- 40	- 40	- 40	- 40	- 35	- 55			- 40
Strain Gage 6	u/in	- 10	- 30	- 40	- 30	- 25	- 25	- 15	- 20	- 10	- 35			- 25
Strain Gage 7	u/in	-130	-120	-130	-140	-140	-140	-135	-135	-135	-140			-140
Strain Gage 8	u/in	45	40	30	30	35	40	40	45	50	30			40
Strain Gage 9	u/in	- 5	0	0	25	20	15	0	20	20	- 10			0
Strain Gage 10	u/in	- 70	- 40	- 70	- 35	- 25	- 75	- 55	- 50	- 40	- 60			- 75
Strain Gage 11	u/in	20	40	60	55	60	10	0	25	35	25			20
Strain Gage 12	u/in	0	- 15	0	10	0	0	- 20	0	0	10			0
Strain Gage 13	u/in	30	25	35	50	30	30	20	30	35	45			35
Strain Gage 14	u/in	- 35	- 40	- 30	- 35	- 45	- 40	- 50	- 40	- 40	- 30			- 40
Strain Gage 15	u/in	75	60	75	75	70	70	60	70	70	80			85
Strain Gage 16	u/in	210	190	200	225	210	215	200	210	220	225			220
Vibration, Commercial, Box #1	Mils	.40	.38	.32	.35	.40	.40	.35	.39	.39	.42		.38	.40
Vibration, Commercial, Box #2	Mils	.70	.70	.59	.59	.72	.72	.69	.70	.71	.70		.75	.73
Time on Test	Hrs.	979	982	1001	1046	1051	1054	1070	1075	1078	1094	1095		1098

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13. ABSTRACT This report presents a review of the 1,000-hour endurance test of the roller gear power transmission by TRW, Cleveland, Ohio, for USAAVLABS under contract DA 44-177-AMC-30 (T). The primary purpose of this program was to determine the feasibility of roller gear drive arrangements for power reductions in helicopter transmissions and to determine critical design parameters for the drive.

An endurance run was made for 1,000 hours on two roller gear drives mounted in a back-to-back test rig with a 200-horsepower load applied continuously with the high-speed shaft at 28,000 RPM and the low-speed shaft at 600 RPM in each box. An instrumentation system was provided to measure loads, speeds, oil flows, and temperatures, and to measure vibration levels.

The test has confirmed that the roller gear drive is feasible for helicopter power reductions and has the ability to accept high input speeds and to accomplish high ratio reductions in one plane with very high efficiency, 98.5 percent or better, with a high degree of mechanical reliability.

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