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POSITIVE-ENGAGEMENT CLUTCH

Charles J. Wirth

Kaman Aerospace Corporation

Prepared for:

Army Air Mobility Research and Development Laboratory

December 1973

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The research described herein was conducted by Kaman Aerospace Corporation under the terms of Contract DAAJ02-72-C-0055. The work was performed under the technical management of Mr. D. P. Lubrano, Technology Applications Division, Eustis Directorate.

V/STOL drive systems must incorporate an overrunning (freewheel) clutch unit so that in the event of engine malfunction, the aircraft can safely autorotate or, in the case of multiengines, proceed on single-engine operation. Current clutches are based on friction operating mechanisms with inherent limitations in capacity and wear. The objective of this program was to evaluate a positive engagement clutch operating at 11,500 rpm and 40,000 in.-lb. maximum continuous torque.

Appropriate technical personnel of this Directorate have reviewed this report and concur with the conclusions contained herein.

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Contract DAAJ02-72-C-0055
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December 1973

POSITIVE-ENGAGEMENT CLUTCH

Final Report

Kaman Engineering Report R-1180

By

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for

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ii

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13. ABSTRACT
Overrunning clutches are universally used in helicopter drive systems to couple the engine and rotor in powered flight while permitting rotor autorotation with the engine stopped.

The clutches in current use depend on friction for power transmission, and suffer from structural inefficiency which imposes a weight penalty and centrifugal influences which restrict operating speed. The positive-engagement overrunning clutch described herein offers the possibility of better structural efficiency and lifting of the overrunning speed limitation. This clutch transmits power through splines, and suffers no detriment to overrunning operation from centrifugal force.

A positive-engagement clutch was designed to meet the operating and space requirements of the HLH helicopter, and a test program was conducted to explore the operating characteristics of the clutch. Feasibility was demonstrated through static torque tests, running engagements and disengagements, and sustained overrunning. Full-scale operational tests are recommended.

92

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SUMMARY

The purpose of this program was to design, fabricate, and test a positive-engagement clutch consistent with the requirements of the HLH main engine overrunning clutch.

A concept was developed using a face spline to connect driving and driven members. A pawl and ratchet system provides proper synchronization and indexing for smooth engagement, and a spiral spline is used as the means of engagement/disengagement actuation.

The test program included static testing to design torque, overrunning tests including a lubrication requirements survey, and operating-speed engagement tests.

The test program successfully demonstrated the ability of the clutch to engage, transmit torque, and disengage, statically and at speeds to 11,000 RPM. Testing also showed that the clutch is capable of long-term overrunning with oil flows down to .5 GPM.

The test program has not explored the durability of the clutch for long-term full power engaged running, or for exposure to high-acceleration-impact engagements. It is recommended that exploration of these operational areas be the subject of future testing.

FOREWORD

The U.S. Army Air Mobility Research and Development Laboratory (USAAMRDL) supervises research programs aimed at advancing helicopter power transmission systems. Overrunning clutches have been identified as a subsystem requiring improvement. The positive-engagement clutch is a type of overrunning clutch with wide application in marine and power generation drive systems. The positive-engagement clutch appears to have great advantages when proportioned for helicopter power transmissions. Accordingly, the positive-engagement clutch has been investigated under USAAMRDL Contract DAAJ02-72-C-0055, Task 1G162207AA7201. Technical direction was provided by Mr. Dominick P. Lubrano, Aerospace Engineer, Eustis Directorate, USAAMRDL.

This report covers some design considerations, analyses, and bench tests of the positive-engagement clutch. The curvic coupling portions of the test clutch were designed and manufactured with the cooperation of Gleason Works, Rochester, New York. The program reported herein was accomplished during the period from 10 April 1972 to 3 June 1973. The work was conducted at Kaman Aerospace Corporation, Bloomfield, Connecticut, under the technical supervision of Mr. Charles J. Wirth, Project Engineer. Overall cognizance of the program was maintained by Mr. Robert Bossler, Jr., Chief of Mechanical Systems Research.

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TABLE OF CONTENTS

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Page

APPENDIXES

LIST OF ILLUSTRATIONS

LIST OF ILLUSTRATIONS (Continued)

LIST OF TABLES

LIST OF SYMBOLS

A	=	addendum
C_p	=	specific heat = .46 Btu/lb F^o
C_F	=	coefficient of friction
D_F	=	pitch dia, face spline - in.
D_S	=	pitch dia., helical spline - in.
F_A	=	axial load generated in clutch - lb
F_T	=	pawl tip force, radial - lb
f_n	=	pawl natural frequency - Hz
f_t	=	ratchet tooth exciting frequency - Hz
g	=	acceleration due to gravity = 386.4 in./sec^2
J	=	polar mass moment of inertia - lb/in.sec^2
K	=	radius of gyration - in.
K_S	=	spring rate - lb-in./rad
L	=	length of engagement of spiral spline - in.
P_S	=	pawl spring force - lb
m	=	spring moment - in.-lb
N	=	number of teeth
N_i	=	input RPM
N_D	=	output RPM
n	=	spring deflection - turns
Q	=	clutch torque - in.-lb
q	=	oil flow - GPM
S	=	lateral travel of synchronizer - in.
t	=	temperature - ^oF

LIST OF SYMBOLS (Continued)

T = time - sec

R = ratchet teeth number

α = pressure angle, face spline - deg

β = helix angle, spiral spline - deg

θ = torsional deflection, input to output - deg

INTRODUCTION

In many drive systems, overrunning clutches are used as an automatic means of transmitting engine power into a drive train when that engine develops power, while permitting engine decoupling at times when engine power ceases and the driven member continues to run. Overrunning clutches are universally used in helicopters to permit autorotation with a dead engine or to permit an engine shutdown in a multiengine installation.

The most common types of overrunning clutches in current use are sprag or ramp roller clutches, which depend on friction to transmit torque between the input and output members. While these clutches are entirely satisfactory in many applications at moderate power levels, their inherent structural inefficiency severely penalizes the high torque applications in size and weight demanded. Their use at high speed is also limited, since centrifugal effects in the sprags and rollers interfere with proper release and accelerate wear.

A type of overrunning clutch is in present use in very heavy machine and marine drive applications, based on the principle of carrying the drive load through engaged spline teeth. Synchronization required to produce engagement at matched speed is provided by a ratchet and pawl system, while a spiral spline provides the transfer motion needed to engage the splines. While these clutches in their present form are completely outsized and unsuitable for aircraft use, the basic principles are validly applicable to overrunning clutches of any size, and they offer advantages which appear to be particularly attractive in the realm of high-powered aircraft drive systems.

A positive drive clutch, wherein the teeth on one member engage mating teeth on the other, transmits torque directly through surface compression, bending and shear. Since there is no dependence on friction, or the huge normal loads required to develop that friction, structural efficiency is greatly improved, and size and weight savings can be realized. This benefit increases as torque levels increase.

By elimination of a friction-dependent engagement system, concern over centrifugal modification of that friction is eliminated, along with the need to remove the heat generated by that friction. This benefit becomes more important with increasing operating speeds.

Sprag and ramp roller clutches depend upon shallow-angle cams which develop huge radial forces and which permit very small radial motions. This characteristic demands very stiff, close-tolerance races and severely limits allowable wear. By elimination of small wedging angle cams, required stiffness, manufacturing tolerances, and wear allowance are all significantly liberalized. This results in fabrication benefits, extends useful life, and improves repairability of the positive-engagement clutch.

This program demonstrated the feasibility of creating an overrunning clutch of the positive-engagement type specifically suited to an aircraft application. This experimental clutch was competitive in weight and size to the sprag type clutch developed for this same application. Further, testing of the experimental model of the positive-engagement clutch demonstrated satisfactory operation and the potential for a very long life without dependence on the large oil flow demanded by comparable sprag and ramp roller clutches.

The space and performance requirements for the HLH main engine overrunning clutch form the basis for the design and test criteria used in this program.

This report describes the original clutch concept, the design approach used, and the test program to which the clutch was subjected.

DESCRIPTION OF OPERATION

Figure 1 illustrates operation of the clutch, which is pictured as being disengaged. In the position shown, the output is free to rotate in the overrunning direction while the input remains stationary. The face splines clear each other, and the ratchet ramps slide freely over the pawls.

If the input is rotated in a clockwise direction relative to the output, the pawls will be brought into contact with the ratchet teeth, holding the synchronizer rotationally fixed to the output member with the face splines disengaged but poised in the proper engagement position. Further rotation of the input member clockwise has the same effect as turning a screw inside a nut; the synchronizer is moved axially into engagement with the output member by the action of its spiral splined connection to the input.

The clutch will transmit torque in this condition, once the face spline is fully engaged, with the torque being passed from the input to the synchronizer through the spiral spline, and torque passing from synchronizer to output member through the face spline.

The thrust component derived in the spiral spline holds the face spline in intimate contact as long as direction of torque is positive (input moment clockwise, output drag moment counterclockwise). On reversal of this torque direction, the spiral spline thrust reverses and the synchronizer face spline is immediately propelled out of engagement.

Two features of the clutch concept intended to significantly enhance its ability to operate at high speed and power levels are worthy of mention:

- As shown in Figure 2, the face spline tooth alignment at the instant of contact is on the load bearing side, and occurs when the depth of tooth engagement is only one-third to one-half of final engagement. The slight rotation that results as the splines fully engage backs the pawls away from the ratchet to assure that no driving torque is imposed on them.

- At any appreciable overrunning speed, the pawls become hydrostatically supported by the oil annulus inside the ratchet. This eliminates metal-to-metal wear of the pawls and ratchet.

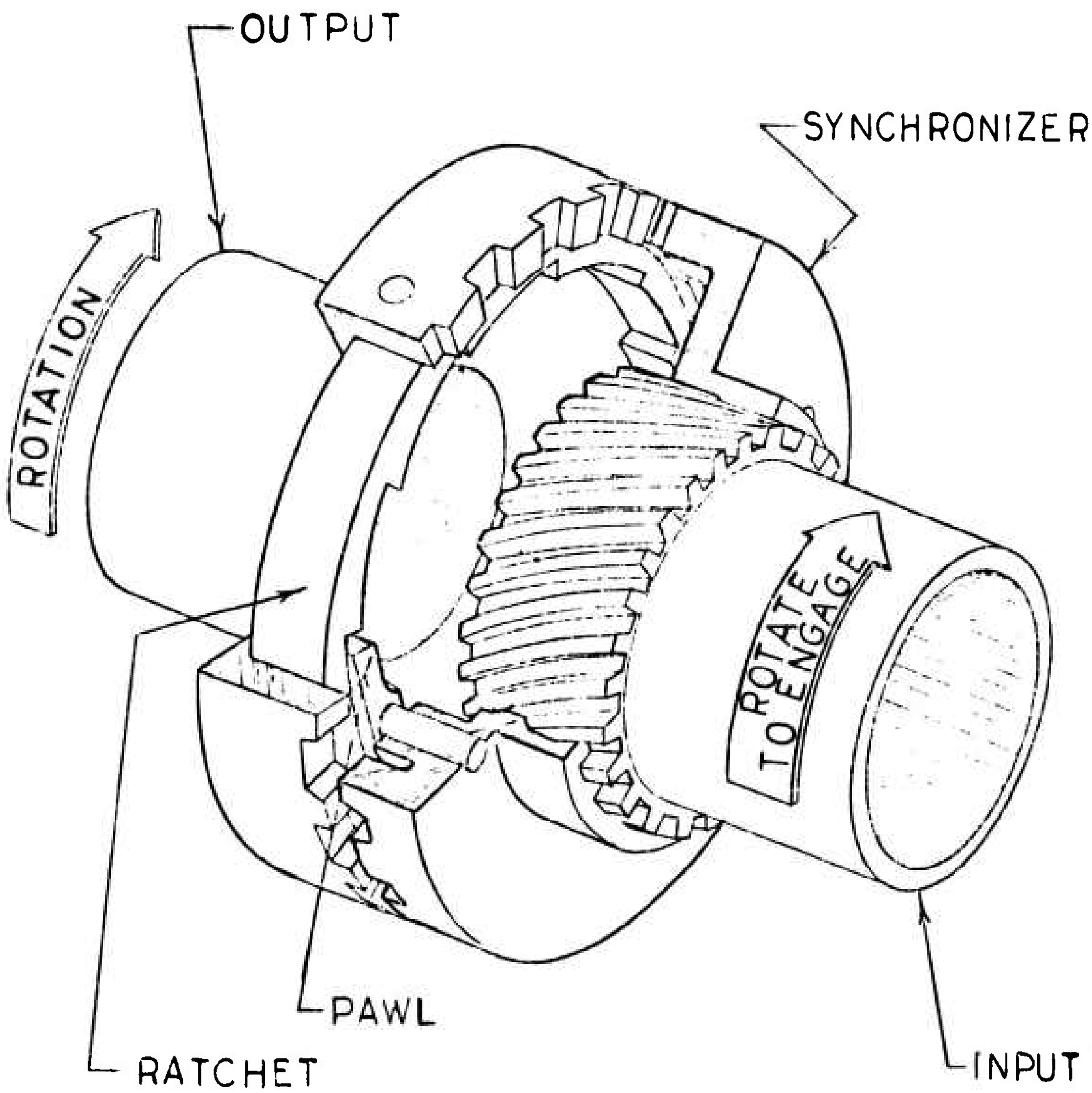


Figure 1. Functional Diagram of Clutch.

PERFORMANCE REQUIREMENTS

The specific requirements chosen for this feasibility evaluation are the operating requirements, available space, and interface arrangements of the HLH main engine overrunning clutch.

Performance requirements are as follows:

Operating interval is 2000 hours
Maximum continuous torque is 45,550 inch-pounds
Peak torque (limit) is 80,000 inch-pounds
Ultimate torque is 120,000 inch-pounds
Operating speed is 11,500 RPM
Lubricant is MIL-L-7808 or MIL-L-23699
Oil inlet temperature is 195°F
Oil inlet pressure not to exceed 100 psig

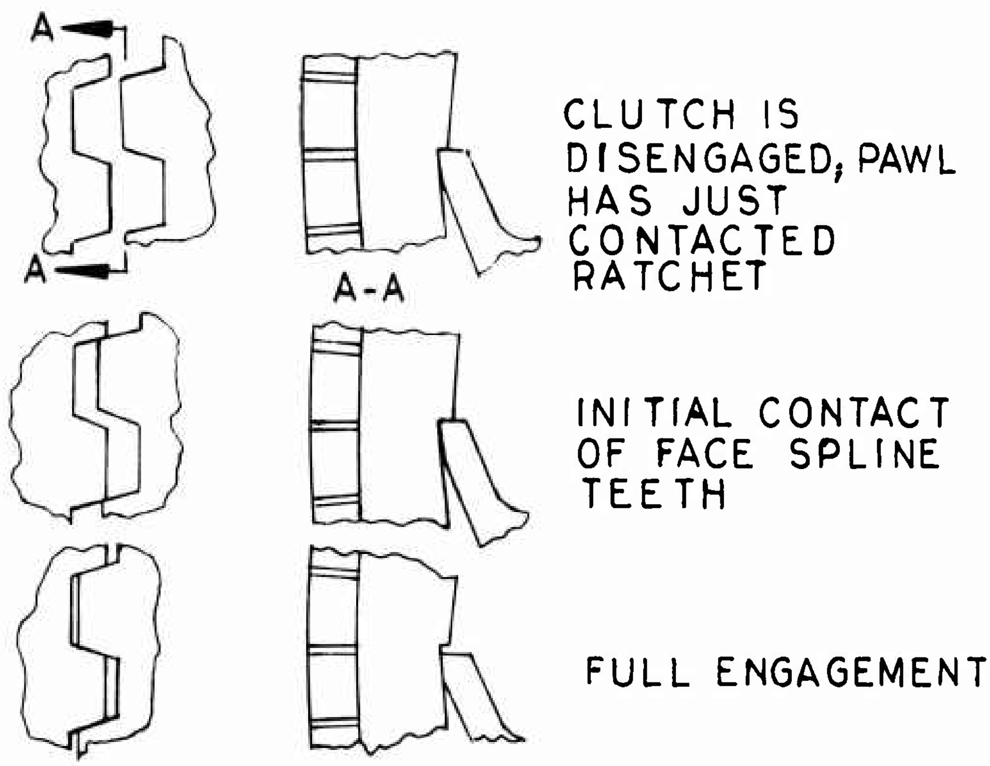


Figure 2. Engagement Sequence.

HLH CLUTCH - GENERAL ARRANGEMENT

Figure 3 shows a general arrangement of the clutch designed to meet these requirements, superimposed on a view of the surrounding HLH parts (shown in phantom). The clutch assembly is designed for installation as a complete bench-assembled unit. The input member (1) supports and retains the output member (9) on a set of bearings. All induced loads are completely balanced within the clutch, with no thrust or side reactions being fed to the mounting bearings as a result of transmitted torque.

TABLE I. IDENTIFICATION OF CLUTCH PARTS

NO.	QTY/CLUTCH	PART NAME
1	1	Input Shaft
2		Spiral Spline Between Input Shaft and Synchronizer
3	1	Pawl Carrier
4	6	Pawl
5	6	Pawl Return Spring
6	1	Ratchet
7	1	Synchronizer
8		Face Spline Between Synchronizer and Output Shaft
9	1	Output Shaft
10	1	Oil Retainer
11		Ratchet Tooth

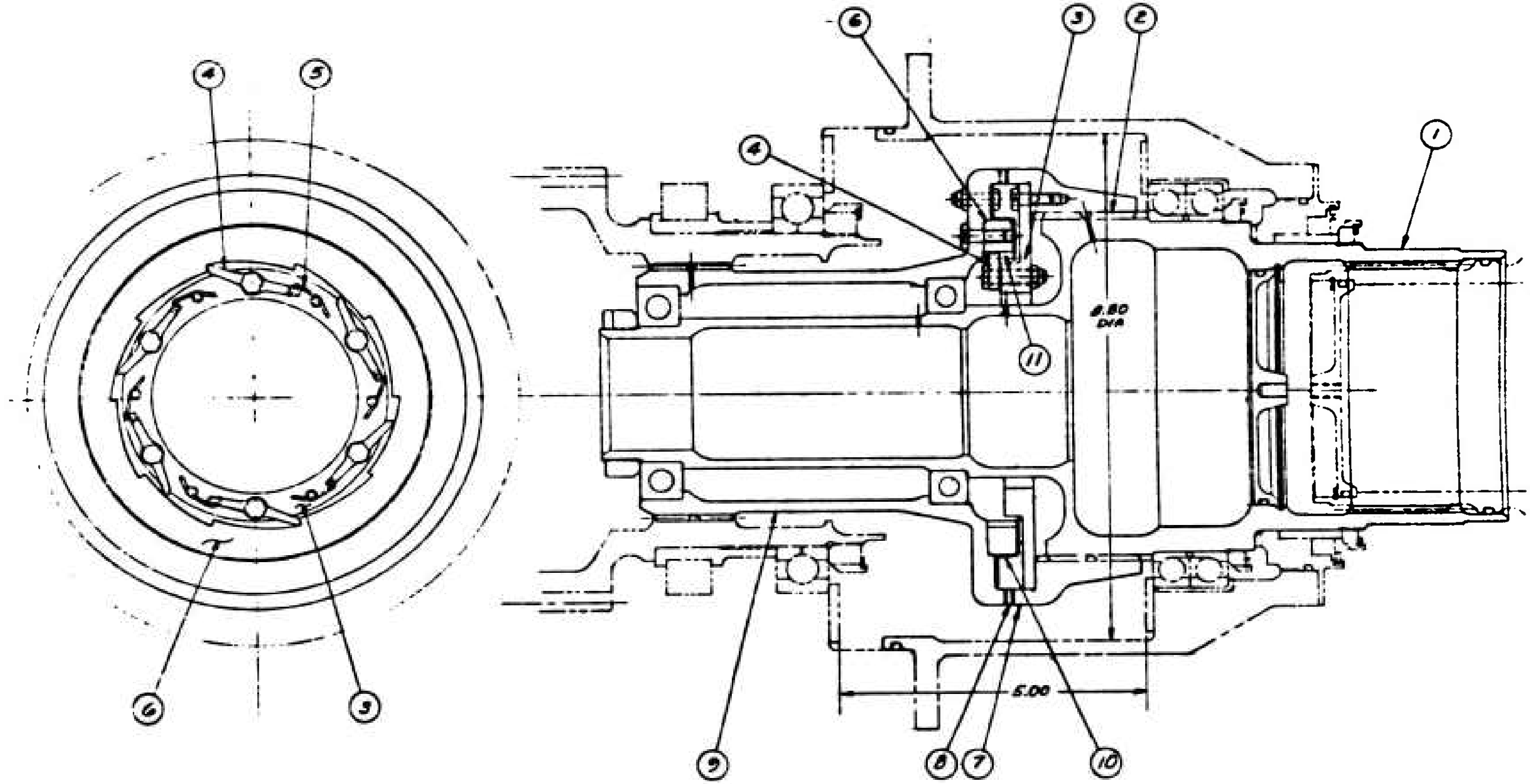


Figure 3. Positive-Engagement Clutch in HLH Drive System.

Since this design concept is unique, with no similar or preceding design to follow as a guide, it was necessary to develop a rational design approach. Refinements in this design approach will evolve with subsequent new designs. The approach that was used is believed to be valid and adequate.

DESIGN AND ANALYSIS

The sequence of examination followed in development of design definition falls into three broad categories:

- o Load-path analysis - design to maintain proper engagement, to transmit torque, and to withstand induced internal loads.
- o Engagement-system analysis - design the pawl system for positive engagement at synchronous speed, and long life at overrunning speed.
- o Critical-speed analysis - design to assure that critical speed of the clutch package is always higher than 115% operating speed.

Basic clutch parameters, established for convenience of fit, ease of manufacture, and based on preliminary analysis, are as follows:

1. Pawls to be arranged in 3 pairs, 6 total.
2. Ratchet to have 10 teeth, giving 30 different pawl and ratchet engagement positions.
3. Face spline to be curvic type, 30 teeth to match the 30 ratchet positions.
4. Pawl clearance with clutch engaged = .010 in. minimum.
5. Pitch diameter of spiral spline = 5.900 in.

LOAD-PATH ANALYSIS

Face Spline Design

Mating face spline halves on the output member and synchronizer make up the basic driving connection between input and output halves of the clutch.

During the engagement process, tooth contact must be avoided until one member has sufficiently penetrated the other to provide adequate load-bearing area of the contacting surfaces (see Figure 2). In the fully engaged position, the synchronizer must have rotated back far enough to pull the pawl tips back from the ratchet teeth.

Strength of the face spline must be adequate to carry the normal load without galling and the ultimate load without failure.

Pressure angle of the face spline teeth must exceed 10°, to avoid sticking in engagement, and yet the angle must not be so large that excessive separating forces are generated.

The face spline chosen for use in this clutch is the curvic type, introduced by the Gleason Works. In this type of spline, precise fit, interchangeability, and good tooth strength can be achieved at reasonable cost. Design guidance is offered, both in handbook form and in the form of expert analytical review of a proposed spline design, by Gleason.

4340 steel, heat treated to 150,000 psi, was chosen as the material to be used in the load-path members. This material is readily available, is tough, and has good dimensional stability. A variety of surface treatments are available to enhance wear resistance and reduce fretting tendencies. In this initial design, the chosen surface treatment for the splined areas was silver plating to inhibit fretting.

Using the design procedure given in the Gleason design handbook*, face spline diameter and tooth proportions appropriate to continuous torque of 45,550 inch-pounds were derived. The face spline definition thus derived is as follows:

Spline outside diameter	- 7.56 inches
diametral pitch	- 3.970 inches
whole depth	- .222 inch
addendum	- .098 inch
dedendum	- .123 inch
tip-to-root clearance	- .025 inch

Derivation of the face spline pressure angle was based on a need to keep the angle small to minimize separating force, while providing sufficient tooth slope to assure proper engagement and pawl load relief in the presence of wear and adverse tolerances.

*"Curvic Coupling Design", Copyright 1964, Gleason Works, Rochester, N.Y.

It was assumed to be desirable that tooth contact on engagement not occur with less than one-third of tooth overlap. This dictates that in the remaining two-thirds tooth penetration, tolerances, wear, and desired pawl setback be accommodated.

2/3 of tooth overlap = 2/3 (2) (.098) = .130

Total tolerance - pawl length ($\pm$ .002)	.004
Total tolerance - ratchet tooth position ($\pm$ .002)	.004
Total tolerance - pawl to spline tooth	.004
Total tolerance - ratchet tooth to spline tooth	.003
Pawl & ratchet wear -	.008
Minimum pawl setback -	.010
Total	.033

$$\mathrm{Tan}\ \alpha = \frac{.033}{.130} = .250$$

Face spline pressure angle - 14.2° = 15°

The face spline data thus derived was submitted to the Gleason Works for their review, comment, and tooling recommendations. Their response indicates that the face spline should perform satisfactorily (see Appendix I).

The separating force generated by the spline teeth, and the force which must be overcome to drive the teeth into engagement while carrying torque, can be calculated by assuming a coefficient of friction between the spline teeth. For this analysis, a coefficient of .05 was used.

$$\text{Axial force required to engage} = \frac{Q}{(.5)D_F}\ \mathrm{Tan}\,\alpha + \frac{Q\ C_F}{.5\ D_F\ \mathrm{Cos}\,\alpha}$$

$$\text{Axial force required to maintain engagement} = \frac{Q}{(.5)D_F}\ \mathrm{Tan}\,\alpha \quad \frac{Q\ C_F}{.5\ D_F\ \mathrm{Cos}\,\alpha}$$

where Q = Clutch torque

D_F = Diameter of face spline = 7.56 in.

α = Face spline pressure angle = 15^{o}

C_F = Coefficient of friction; assume C_F = .05

Axial force to engage = .085 Q

Axial force to maintain engagement = .057 Q

In practice, it is necessary to provide an excess of axial preload beyond the minimum required for engagement so that good clamp-up of the face spline is maintained. This clamping effect brings more teeth into engagement and minimizes the working which may occur at tooth contact surfaces in response to cyclic torque variations.

As recommended by Gleason Works, a clamping force to torque ratio of .18 lb/lb-in.was used (see Appendix I). This is approximately double the axial force required to bring the teeth into engagement, and it assures that full tooth contact will be maintained under all driving conditions.

Spiral Spline

The spiral spline between the input member and synchronizer transmits driving torque and at the same time generates the axial force required to hold the face spline members in intimate engagement.

The spline diameter, tooth size, and length of engagement must be chosen to carry the clutch torque adequately, and the helix angle must be chosen to achieve the proper proportion between axial load and torque.

A standard SAE spline tooth form, fillet root side fit, of moderate pitch was tentatively chosen for the spiral spline, with availability of tooling to be considered in the final choice.

Tentative spline parameters chosen for convenience of arrangement, which will be checked for adequacy, are:

D_S = Spline pitch diameter = 5.900 in.

Diametral pitch = 10/20

Number of teeth = 59

Pressure angle = 30°

Minimum engaged length = 1.6 in.

The axial force requirement determines the helix angle to be used in the spiral spline as follows:

$$\text{Axial Force} = (\text{Tooth Load})(\text{Tan}\,\beta) - (\text{Tooth Load})(C_F)(\text{Cos } 30^{\circ})$$

where Tooth Load = $\dfrac{Q}{1/2\ D_S}$

β = Helix angle

Q = Clutch torque

C_F = Coefficient of friction - .05

$$\text{Axial Force} = \frac{Q}{1/2\ D_S}(\text{Tan}\,\beta) - \frac{Q}{1/2\ D_S}(.05)(\text{Cos } 30^{\circ})$$

Letting the axial force = .18Q as recommended by Gleason and solving for β , we get

$$\beta = \text{Tan}^{-1}\ \frac{.18Q(1/2\ D_S)}{Q} + .05(\text{Cos } 30^{\circ}) = \text{Tan}^{-1}\ .565$$

$$= 29^{\circ}\ 28'$$

Tooling was available to produce a helix angle of 29° 52' 8", so this was used.

Surface stress in the spline, based on projected area and on the assumption of 25% tooth contact, is found as follows:

$$\text{Surface Stress} = (\text{Tooth Load})\frac{1}{.25(N)(L)(2A)}$$

N = Number of spline teeth = 59

L = Length of engagement = 1.6 in.

A = Addendum = .1 in.

$$\text{Surface stress} = \frac{Q}{2.59 \times .25 \times 59 \times 1.6 \times .1} = .164Q$$

Surface stress at continuous load	=	7452 psi
Surface stress limit load	=	13,120 psi
Surface stress ultimate load	=	19,680 psi

Experience has shown that surface stresses of this magnitude are not excessive, and the spline is judged to be adequate.

Thrust Path

The axial load exerted by the spiral spline must be reacted, either within the clutch itself or by the input and output support bearings in the HLH transmission. Since the load is high enough to severely tax those bearings, this clutch is designed to completely react the axial load internally.

As Figure 4 indicates, input and output members are supported by a duplex bearing pair (A & B) which is preloaded by an 80-pound spring to eliminate radial play and to enhance smooth running.

When the clutch engages and axial forces far in excess of the spring preload are generated, the input (which is slip fitted into the bearings) moves to the right until the thrust collar "C" contacts the outer race of bearing A. All thrust loads then follow the path indicated in the sketch, with none of the load taken through the bearings.

An added feature of the bearing mounting scheme is the removal of preload from bearing "B" in the engaged condition. This increases the radial clearance in the bearing so that it does not fight the powerful tendency of the face spline to center the members on its own axis, which may deviate slightly from the bearing center (by virtue of manufacturing tolerances).

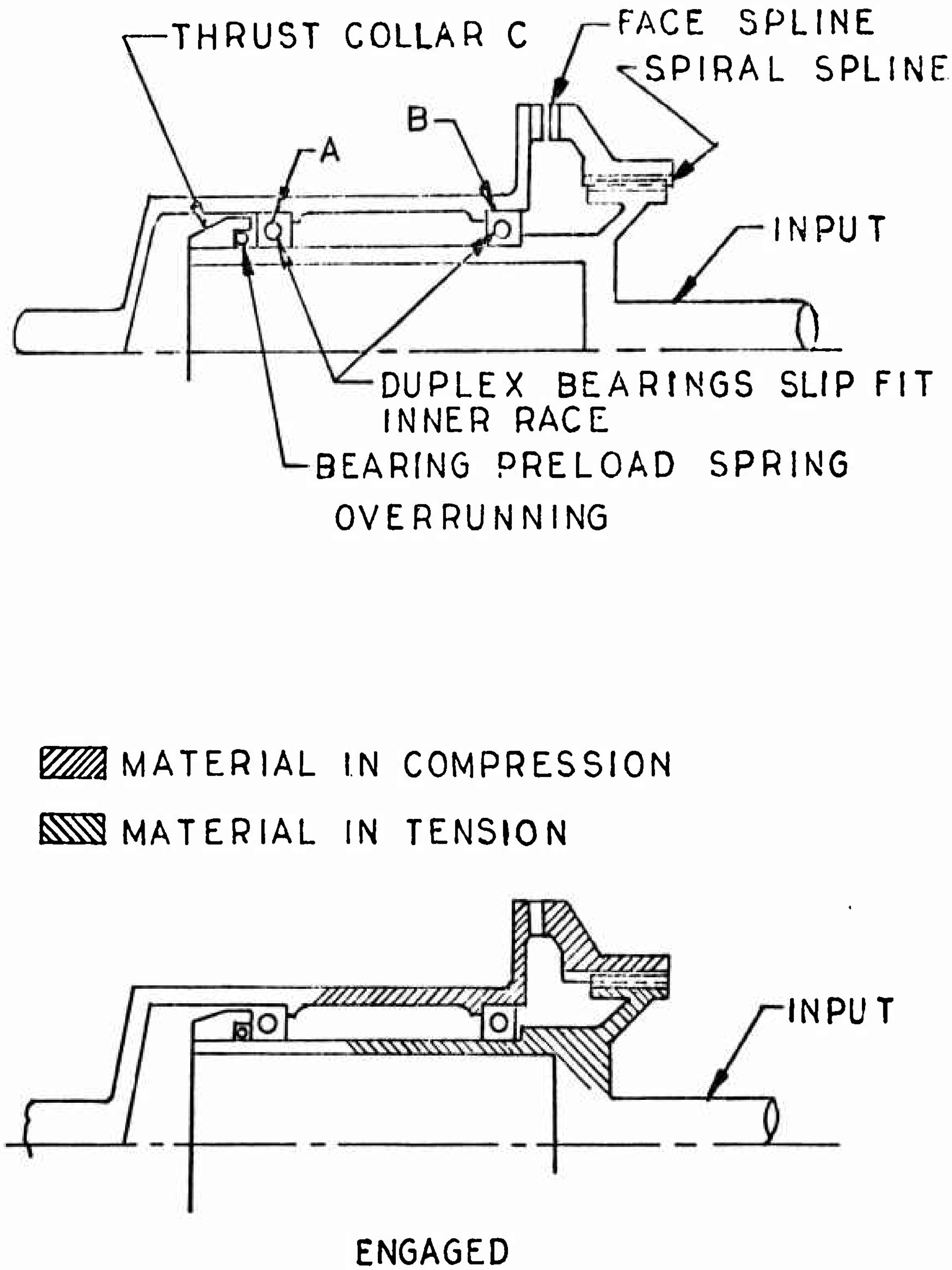


Figure 4. Thrust Path in Clutch.

ENGAGEMENT SYSTEM ANALYSIS

Pawl System

The pawl system of the positive engagement clutch provides the means for sensing the proper instant to initiate engagement and for operating the synchronizer to complete the engagement. It is essential that the engagement operation commence as soon as input speed has overtaken output speed and the face spline teeth are in alignment.

In the subject clutch, positive pawl system operation is assured by prevention of random pawl bounce during the time interval immediately preceding engagement.

The second element in assurance of reliable pawl system operation involves design to minimize wear and fatigue of the pawl-ratchet system in long-term overrunning. By promoting pawl hydroplaning at all continuous overrunning conditions, metal-to-metal wear between pawls and ratchet teeth can be avoided and return spring cyclic motion minimized.

Figure 5 is a plot of pawl action versus differential input to output speed. By assuming uniform acceleration of the input from 0 to full speed in 5 seconds, a scale of time to go to synchronous speed can be added to the abscissa; and by knowing the number of teeth in the ratchet, ratchet tooth exciting frequency can be added to the ordinate.

As the plot indicates, pawl system behavior passes through three distinct phases as differential speed changes:

- Pawl hydroplaning, wherein differential speed between the pawls and the oil annulus carried in the ratchet is sufficient to support the pawl hydrodynamically.

- Resonant behavior, where the speed differential will no longer support hydroplaning, so that the pawls become propelled by the ratchet teeth. When the ratchet pulses occur at a frequency which the pawls are unable to follow, then the pawls will tend to bounce at their natural frequency.

- Coherent pawl and ratchet interaction occurs when the click frequency becomes less than the pawl natural frequency and each pawl falls into each ratchet pocket as it comes by.

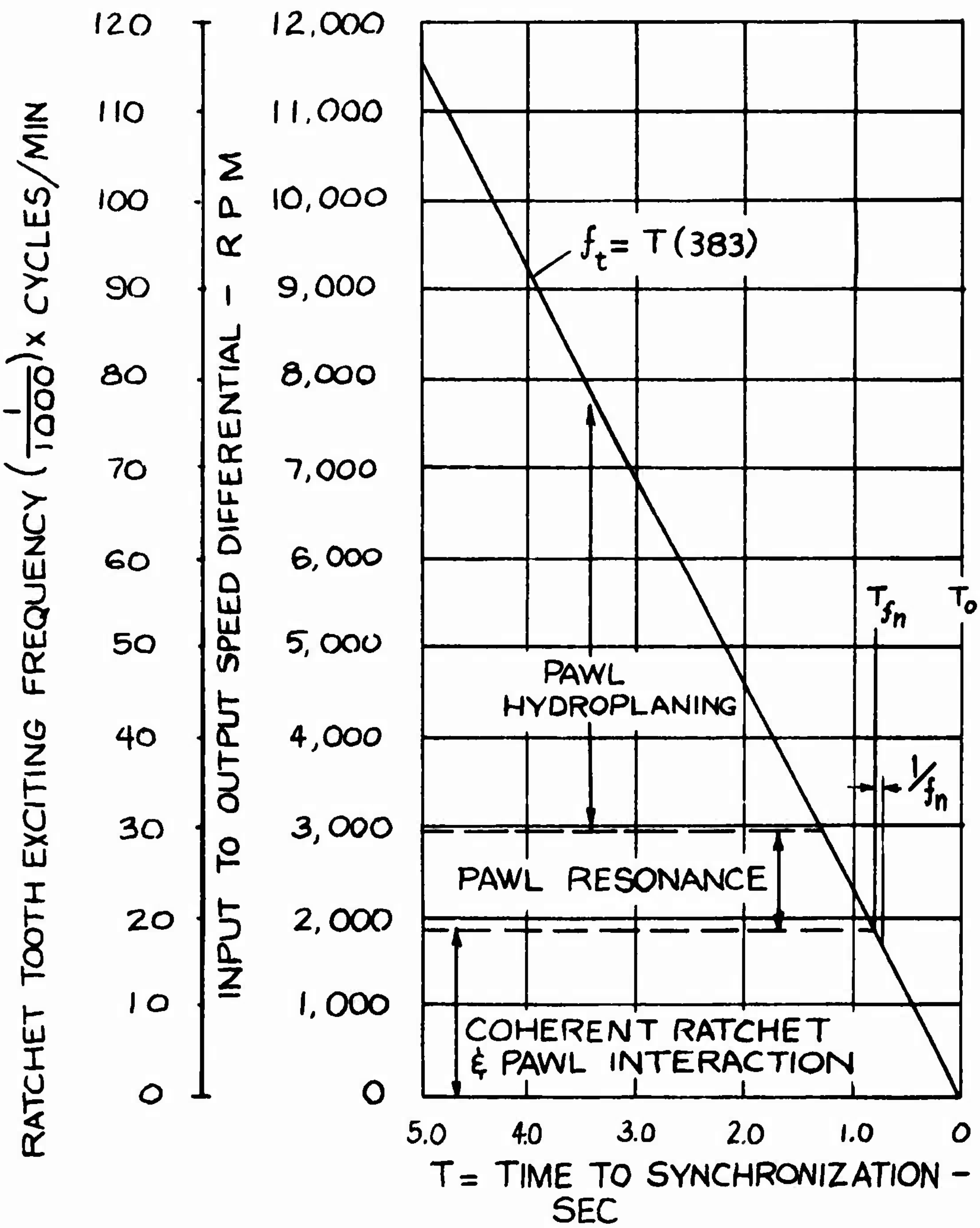


Figure 5. Differential Speed and Ratchet Tooth Exciting Frequency Vs Time.

The main thrust of pawl system design is to exercise control over the extent of these zones to promote reliable operation and long life.

<u>Determination of Ability of Pawl to Engage Ratchet in the Available Time</u>

In the HLH drive system, it is possible for the speed differential between input and output to change rapidly as an engine accelerates up to engagement speed. It is assumed that this acceleration rate can be as high as 0 to 11,500 RPM in 5 seconds.

The plot given in Figure 5 shows that the input to output speed differential diminishes as the instant of synchronization (T_o) approaches. The rate of change of differential speed is

$$\frac{11{,}500 \text{ RPM}}{5 \quad \text{sec}} = 2300 \frac{\text{RPM}}{\text{sec}}$$

Converting to rev/sec^2, we get $\frac{2300}{60} = 38.3$ rev/sec^2.

Since there are 10 teeth in the ratchet, the rate of change of ratchet tooth exciting frequency = (10)(38.3)Hz.

And $\quad f_t = 383\ T \qquad (1)$

where $\quad f_t$ = Ratchet tooth exciting frequency

$\quad T$ = Time remaining to synchronization

Working back from the instant of synchronization (T_o), we can say that the last resonant pawl cycle must be completed at T_o or sooner. It can also be seen that the last opportunity for the start of that resonant cycle occurred when the ratchet tooth exciting frequency just equaled the pawl natural frequency ($T = T_{f_n}$).

The period of time for one resonant cycle = $\frac{1}{f_n}$

where $\quad f_n$ = Pawl natural frequency

From (1), the time preceding T_o for f_t to equal f_n is

$$T_{f_n} = \frac{f_n}{383}$$

As a minimum let $\frac{1}{f_n} = \frac{f_n}{383}$.

Therefore, $f_n = \sqrt{383} = 19.57$ Hz

For a rotationally vibrating system,

$$f_n = \frac{1}{2\pi}\sqrt{\frac{K_s}{J}}$$

where K_s = Spring rate of pawl return spring, lb-in./rad

J = Pawl polar mass moment of inertia = $\frac{(W)}{g}K^2$ lb-in. sec^2

W = Pawl weight, lb

K = Pawl radius of gyration, in.

g = Acceleration of gravity = 386.4 in./sec^2

From Appendix II, we find that pawl weight is approximately .0159 lb, $J = 7.795 \times 10^{-6}$ lb-in. sec^2.

if f_n = 19.57 Hz,

$$19.57 = \frac{1}{2\pi}\sqrt{\frac{K_s}{7.795 \times 10^{-6}}}$$

K_s = .113 lb-in./rad or .712 lb-in./turn

Note that this is a minimum; any spring rate exceeding this value is satisfactory.

Other considerations in the design of the pawl return spring, such as available space, installation tolerances, desirable stress levels, and desirable pawl preload, all lead to a practical range of spring rate of from 4 to 6 inch-pounds per turn. Pawl natural frequencies of 50 Hz are therefore assured.

Determination of Pawl Hydroplaning

It is desired that the pawls be supported entirely by hydrodynamic planing for all continuous overrunning conditions. Hydrodynamic planing requires that the pawl skating on the oil film develop a lift moment greater than opposing moments. The pawl moment which must be overcome by the hydrodynamic moment consists of the return spring torque and the centrifugally magnified pawl nose heaviness. Expected values for these moments are:

Spring torque = .5 to 1.0 lb-in.

Centrifugal moment = .5 lb-in. at 6000 RPM

Total moment to be reacted by planing = 1.0 to 1.5 lb-in.

Analytical determination of pawl planing torque is a difficult, uncertain process, with little background data available on the scale of clutch parts. An analysis was made using methods and data presented by Hoerner*, pages 11-20 and 7-18, and the results of that analysis are given in Figure 6. Analysis indicated that half-speed overrunning (with the input at 6000 RPM and the output at 11,500 RPM) generated the lowest hydrodynamic pawl lift, and this condition is represented by Figure 6. Testing has shown, however, that the analysis has underestimated planing ability by several hundred percent.

While Figure 6 indicates marginal planing with .19 wide pawls at 5500 RPM differential, testing shows planing down to 2500 RPM differential.

CLUTCH CRITICAL SPEED

Assurance was required that the subject clutch have a critical speed well above its operating speed of 11,500 RPM. Appendix III contains a summary of that analysis, the results of which are presented here.

Critical speed as calculated - clutch disengaged	=	56,200 RPM
Critical speed assuming bearing spring rate 1/2 calculated	=	49,000 RPM

No critical speed problem was encountered during testing, nor should any be encountered in operation with this clutch.

*Sighard Hoerner, "Fluid Dynamics Drag", 1958.

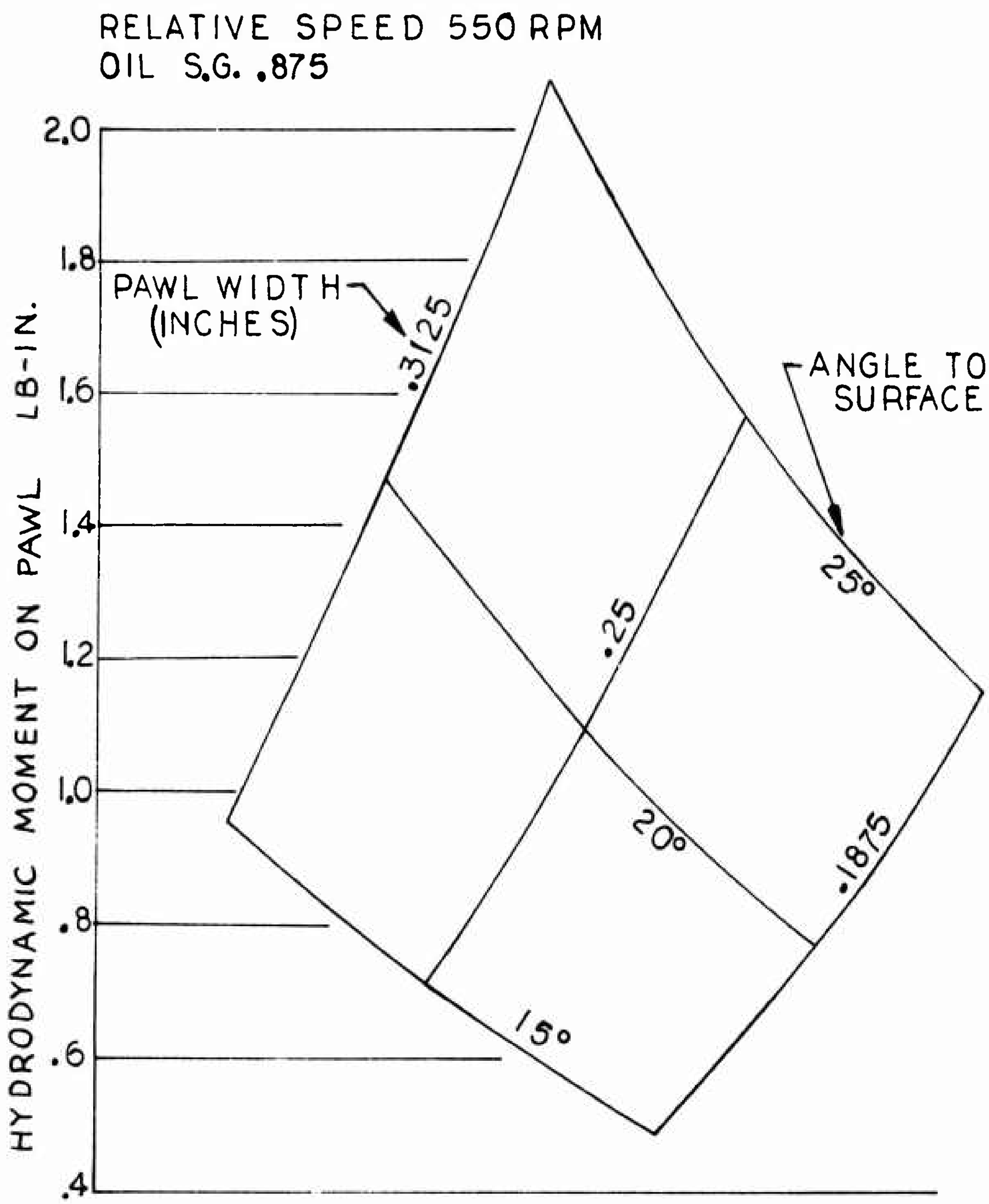


Figure 6. Hydrodynamic Moment About Pivot.

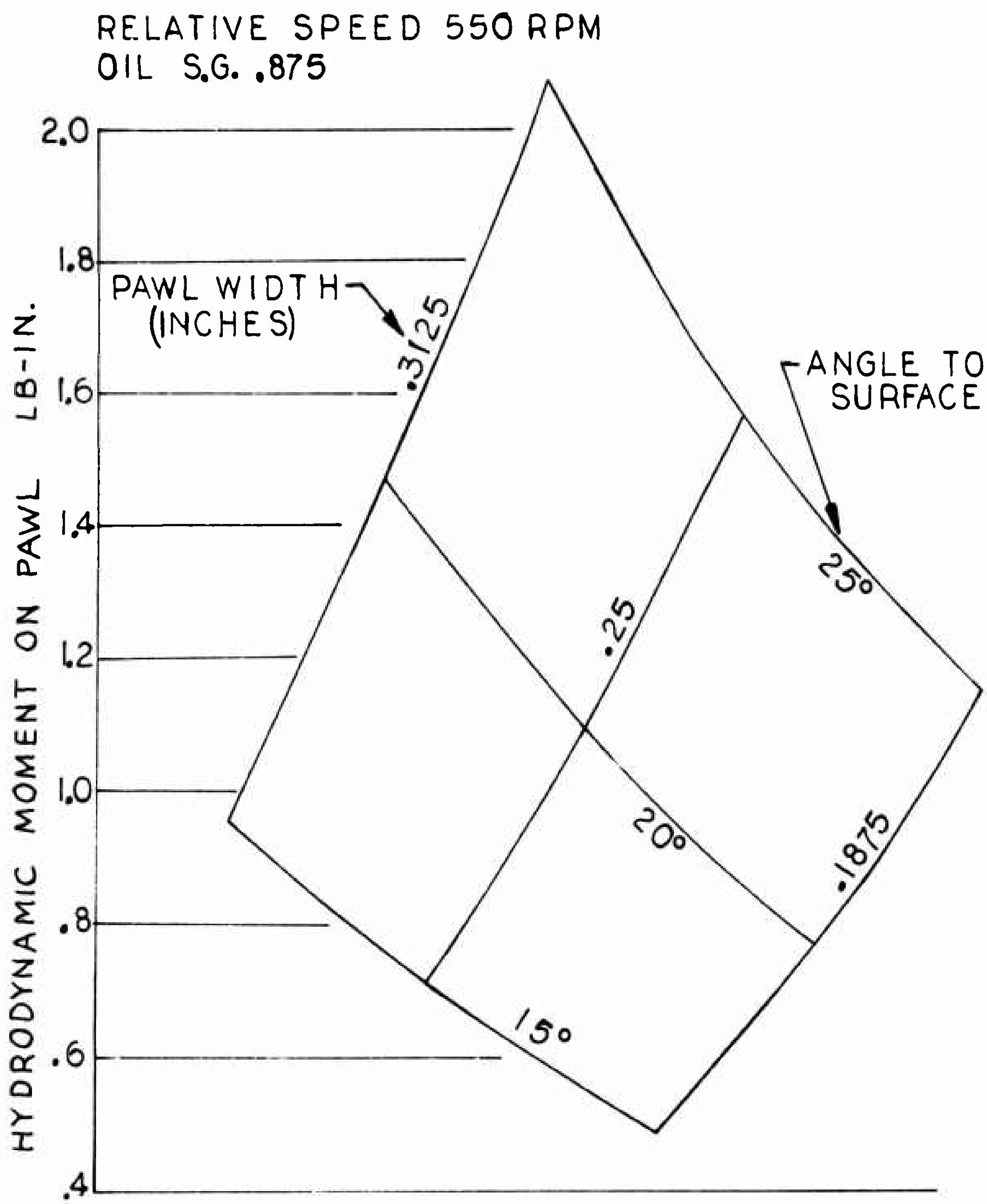


Figure 6. Hydrodynamic Moment About Pivot.

TEST PROGRAM

INTRODUCTION

In order to investigate the feasibility of the positive engagement clutch and to evaluate its applicability to the HLH, a clutch was built, a test plan was developed, and tests were conducted. The characteristics which this program intended to explore were the ability to engage, transmit torque, disengage, and overrun; and the heat generation, lubrication requirements, and durability in the overrunning mode. Testing included both static and dynamic tests, and observations of deflections, temperatures, vibrations, noise, speed, and wear.

TEST SPECIMEN

Figure 7 shows an assembly drawing of the test specimen clutch and its test housing. This clutch is similar to the proposed HLH clutch shown on Figure 3 in all significant aspects of design, material, heat treatment, and method of manufacture. The design and analysis referred to in the preceding section were applied to this test specimen. Outside of the clutch itself, input and output adapter shafts have been added to the clutch to facilitate design of the drive system used in the dynamic test.

TABLE II. TEST SPECIMEN MATERIALS LIST

PART NO.	PART NAME	MATERIAL	HEAT TREAT
SK-22-9	Input Shaft	4340 STL	150,000 psi min.
SK-22-13	Adapter-Helical Spline	4340 STL	150,000 psi min.
SK-22-12	Synchronizer	4340 STL	150,000 psi min.
SK-22-15	Output Shaft	4340 STL	150,000 psi min.
SK-22-18	Thrust Collar	4130 STL	Condition N
SK-22-16	Ratchet	4340 STL	Salt Bath Nitride per AMS 2755
SK-22-22	Pawl	4340 STL	Salt Bath Nitride per AMS 2755
SK-22-24	Pawl Return Spring	Music Wire	
SK-22-23	Bushing-Pawl Pivot	4340 STL	Salt Bath Nitride per AMS 2755
SK-22-14	Carrier	4130 STL	180,000 psi

TABLE III. CLUTCH GEOMETRY

Face Spline	-	7.56-in. 0. diameter, 30 teeth, 15° PA Curvic
Helical Spline	-	5.900-in. pitch diameter, 59 teeth, 10/20 pitch, 30° Helix
Ratchet	-	10 teeth
No. of Pawls	-	6 (3 pairs)

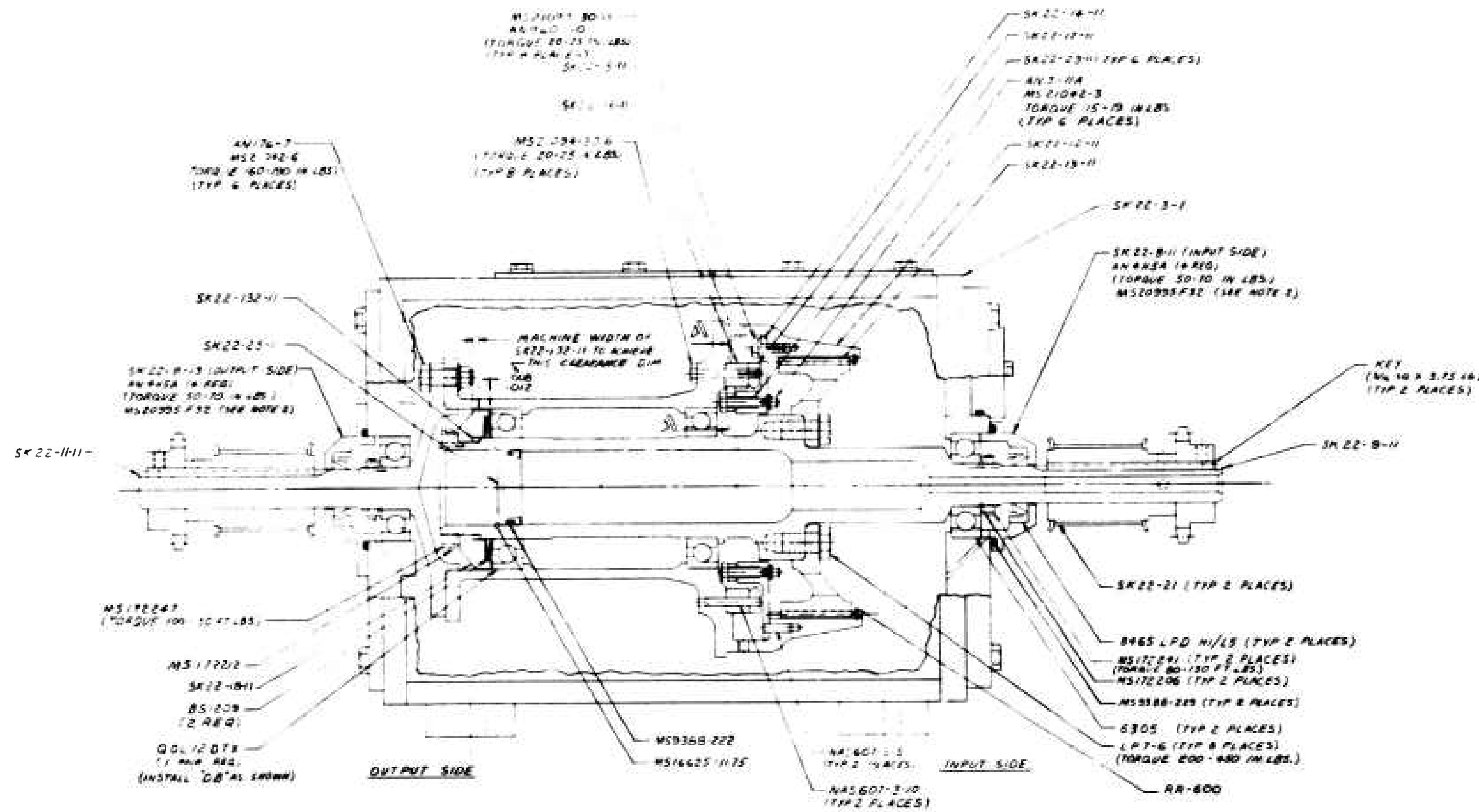


Figure 7. Assembly - Test Specimen Clutch.

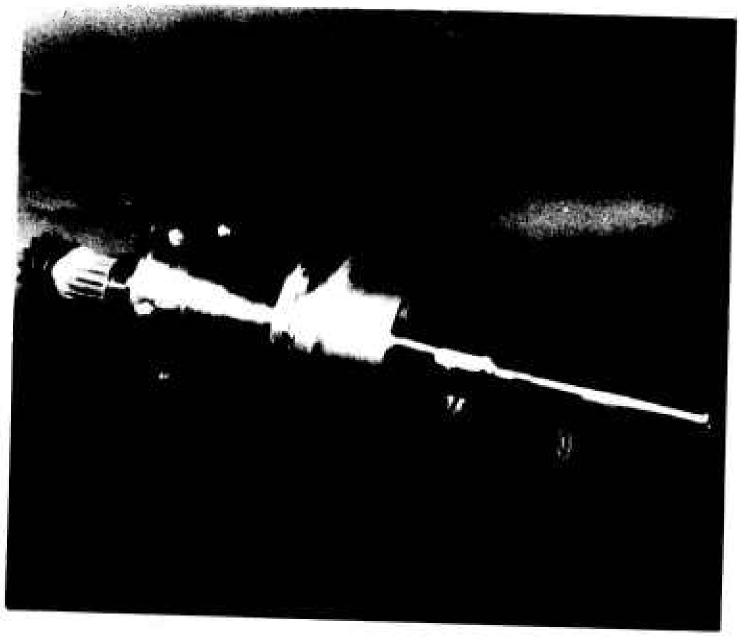


Figure 8. Test Clutch Shown Engaged.

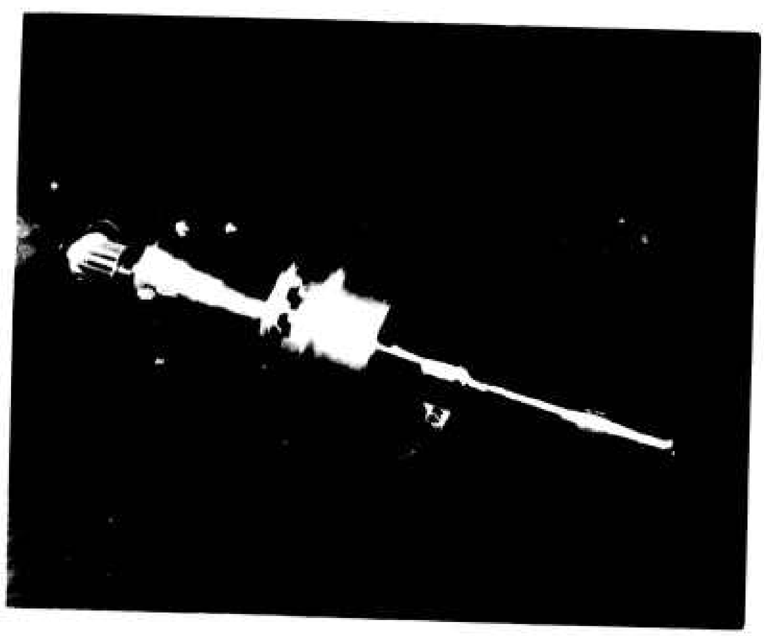


Figure 9. Test Clutch Shown Disengaged.

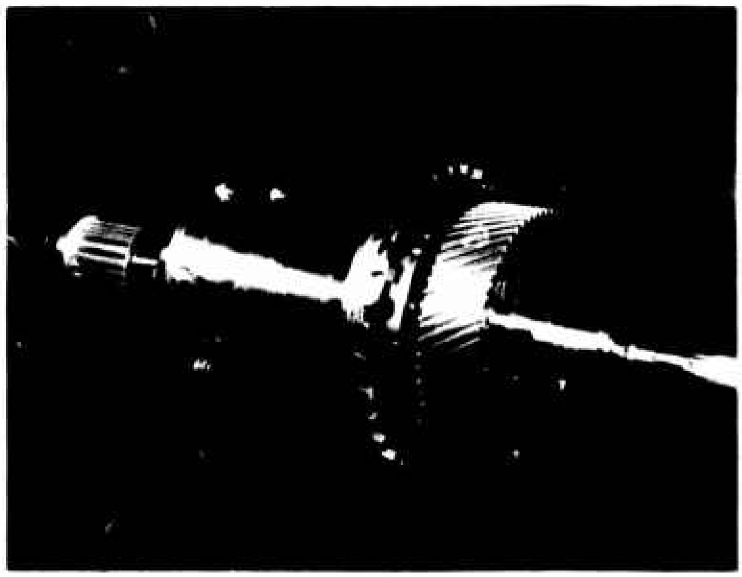


Figure 10. Clutch Assembly
Showing Spiral Spline.

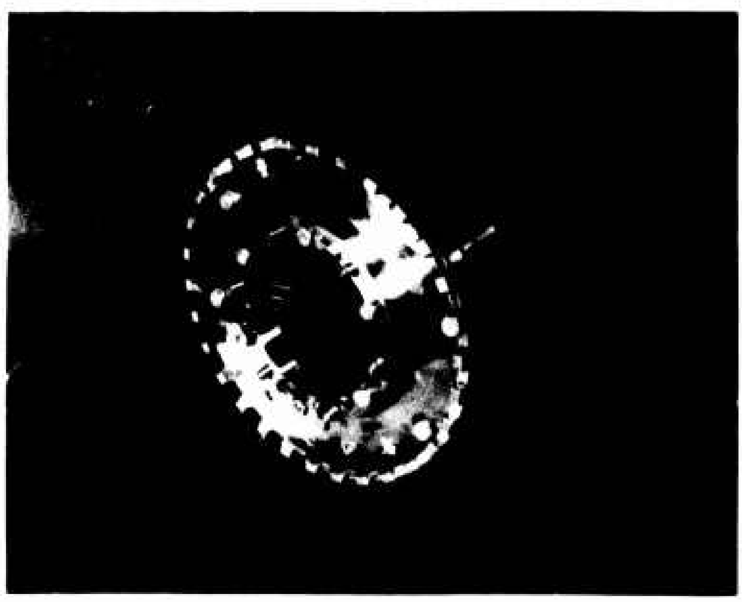


Figure 11. Synchronizer Subassembly.

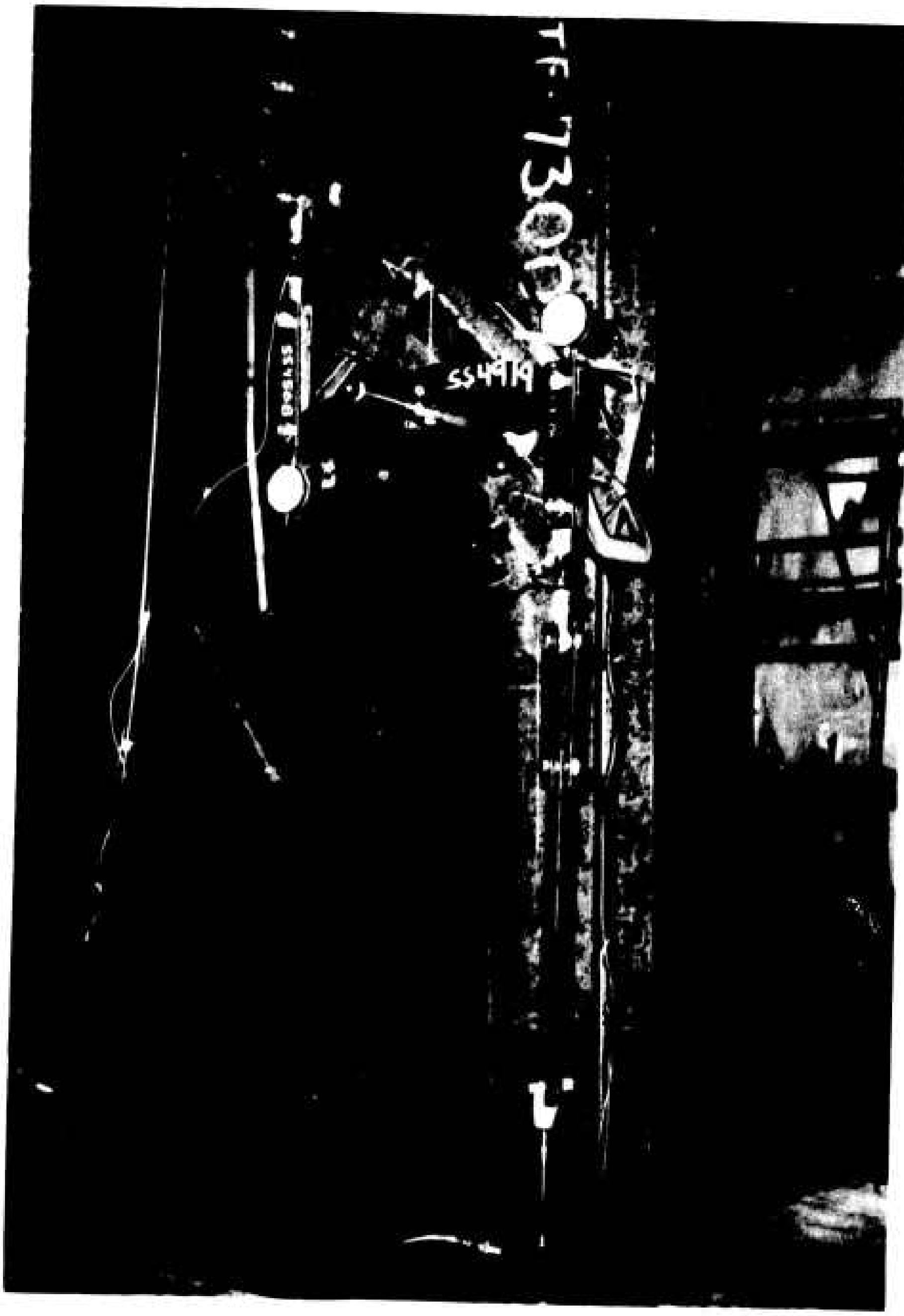


Figure 12. Clutch Installed in Static Test Fixture.

STATIC TEST

This first test to which the clutch was exposed investigated the ability of the clutch to engage, transmit torque and disengage. Effort required to actuate the clutch and deflections under load were measured.

Test Rig

Figure 12 shows the clutch installed in the static test fixture with the loading beam in place and load cylinders connected.

In this test setup, the output member was bolted directly to massive structure. The loading beam was rigidly fastened to the input member, and hydraulic cylinders were coupled to the ends of the loading beam by calibrated load links.

Large loads were applied by pressurizing the two hydraulic cylinders simultaneously, providing a pure couple. For application of the small actuating loads, a torque wrench was applied to a hexagonal fitting in the center of the load beam.

Load was monitored by torque wrench readings or by readout from the strain-gaged cylinder links. Clutch rotation was measured by the dial indicators on the load beam, axial displacement by measuring the gap between the load beam and synchronizer.

Test Plan

The test plan called for four load cycles, starting with the clutch disengaged, proceeding through engagement, load application in 10,000 inch-pound increments to 40,000 inch-pounds, relief of the load, and disengagement. Loads and deflections were monitored for all steps.

Test Results

Results of the testing showed that torque required to engage the clutch ranged from 40 to 80 inch-pounds, if engaged very slowly, while 20 inch-pounds would actuate the engagement if completed in a continuous motion. Disengagement torque was 10 inch-pounds, with no sign that the clutch tended to be locked in engagement after application of full load. Engagements were tried in all ratchet positions, with no significant differences noted. There was no tendency to bind.

The results of the load cycles are shown on Figure 13. Runs 2, 3 and 4 show good repetition; but it is evident that Run 1 took up some slack in the test rig, accounting for the fact that the

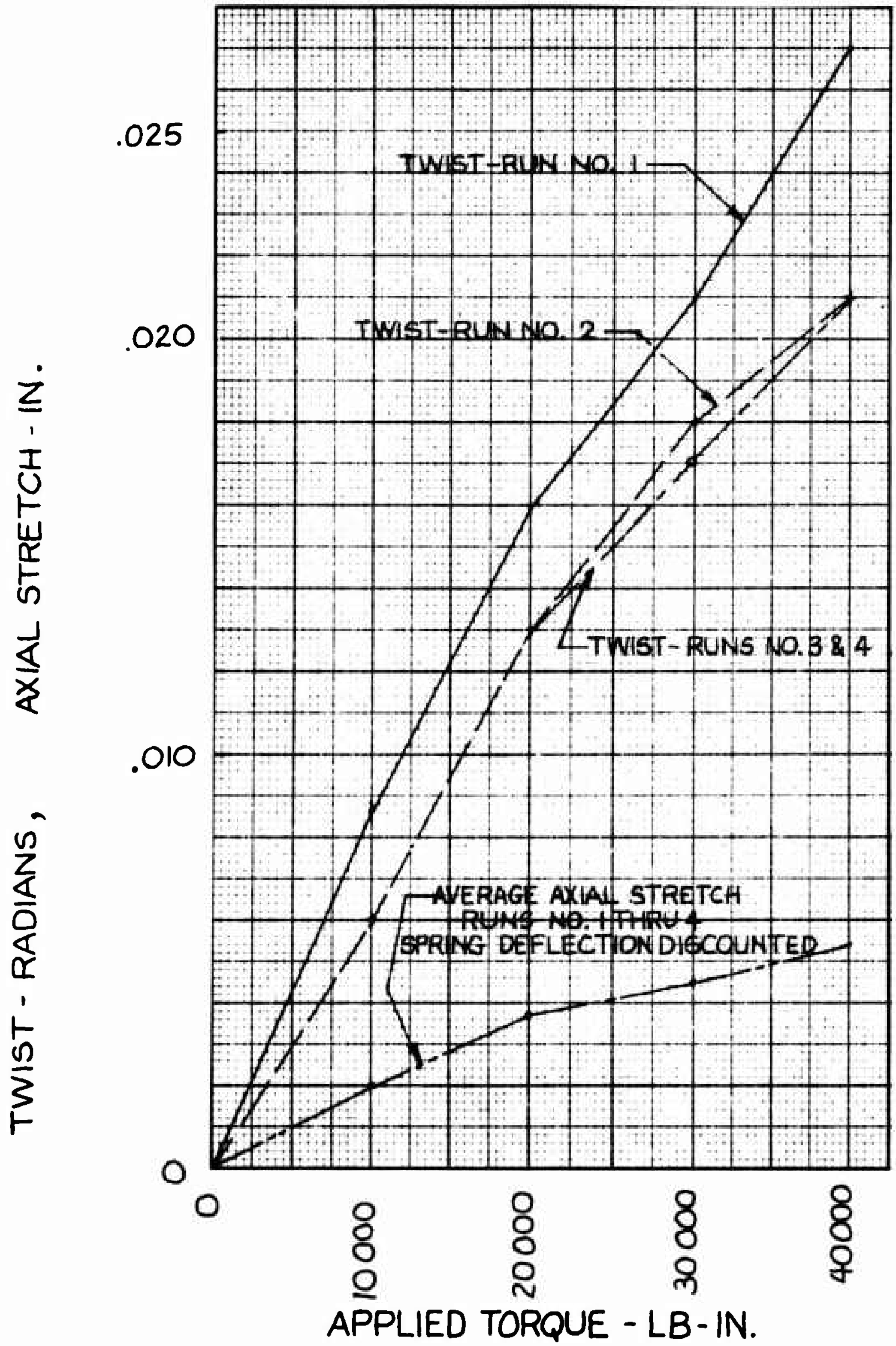


Figure 13. Clutch Deflections Versus Torque.

plot of Run No. 1 deflections is displaced from the remaining plots.

Clutch torsional deflection is shown to be fairly linear, with a slight tendency to stiffen up after 20,000 inch-pounds, probably as more spline teeth are made to share the load. Torsional deflection at design torque of 40,000 inch-pounds is 1.2^{o}.

As the clutch is engaged, and load applied, there is an initial axial offset of .010 inch, due to compression of the bearing preload springs until thrust is taken by the internal thrust collar. Figure 13 reflects the net elastic deflection of the primary structural members of the clutch. Axial deflection of the clutch is shown to be .006 inch at design torque.

An effort was made to investigate load sharing of the pawls by bluing the tips and examining for rub-off after engagement, but this method was not considered reliable. As an alternative test method, one pawl was removed from a pair, and engagements were attempted. It was found that attempted single pawl engagements could not be made with a smooth slow torque application, as internal binding led to a "stick-slip" type action. Since the double pawl engagements were entirely free of binding, it was judged that both pawls were sharing the load.

Discussion

The static test program demonstrated that the positive engagement clutch will engage, maintain engagement when subjected to operating torque, and disengage without binding, as torque is reversed.

In performing these operations, the clutch did not exhibit any tendencies which would be detrimental to use as an overrunning clutch in the HLH drive system.

Torsional stiffness of the clutch shows that deflection at design torque is only 1.2^{o}, which is less than half the deflection which would be expected with a sprag or ramp roller clutch. This could be an important factor in avoidance of torsional natural frequencies in the drive train.

The loads required for clutch actuation are repeatable, and very small in terms of engine torque (less than .25%). This indicates that engagements and disengagements should be smooth and very responsive.

The face spline pressure angle and spiral spline helix angle are considered to be satisfactory.

LUBRICATION REQUIREMENTS TESTS

Introduction

This series of tests was intended to explore the lubrication needs of the positive engagement clutch, particularly during overrunning.

As in other overrunning clutches, a supply of oil is required to carry away heat of friction and to sustain an oil film separating the overrunning parts and their raceway. Excessive oil flow can be detrimental because of additional heat generated by oil churning, while insufficient oil flow can greatly accelerate part wear. Determination of a flow range which avoids those extremes is essential to the proper integration of this clutch into a power transmission system.

In the course of design, the proper oil flow for the subject clutch was estimated to be 1.5 gallons per minute for either MIL-L-7808 or MIL-L-23699 oil. The test scheme was to operate throughout the overrunning speed spectrum at that flow, and at 50% above and below that flow. The heat generated during those runs was to be monitored, and through observation of vibration, noise and wear, pawl operation was to be assessed.

Test Rig

For this, and other dynamic tests, a rig was built with the capability of spinning input and output at independent, variable speeds. The test rig is shown in Figure 14.

Input and output are each driven by a 5-HP varidrive, coupled to the clutch by a flat cog belt system with 3:1 step up in speed.

The clutch test housing can be seen mounted on the table between the varidrives.

Lubricating oil supply (MIL-L-23699) is fed into the center of the input shaft through a standpipe. The oil supply tank is equipped with a thermostatically controlled immersion heater so that inlet oil temperature can be raised to 195° F, valving is furnished to control oil flow, and a flowmeter is provided to measure that flow.

Other instrumentation is provided to monitor the following:

Inlet oil temperature
Outlet oil temperature

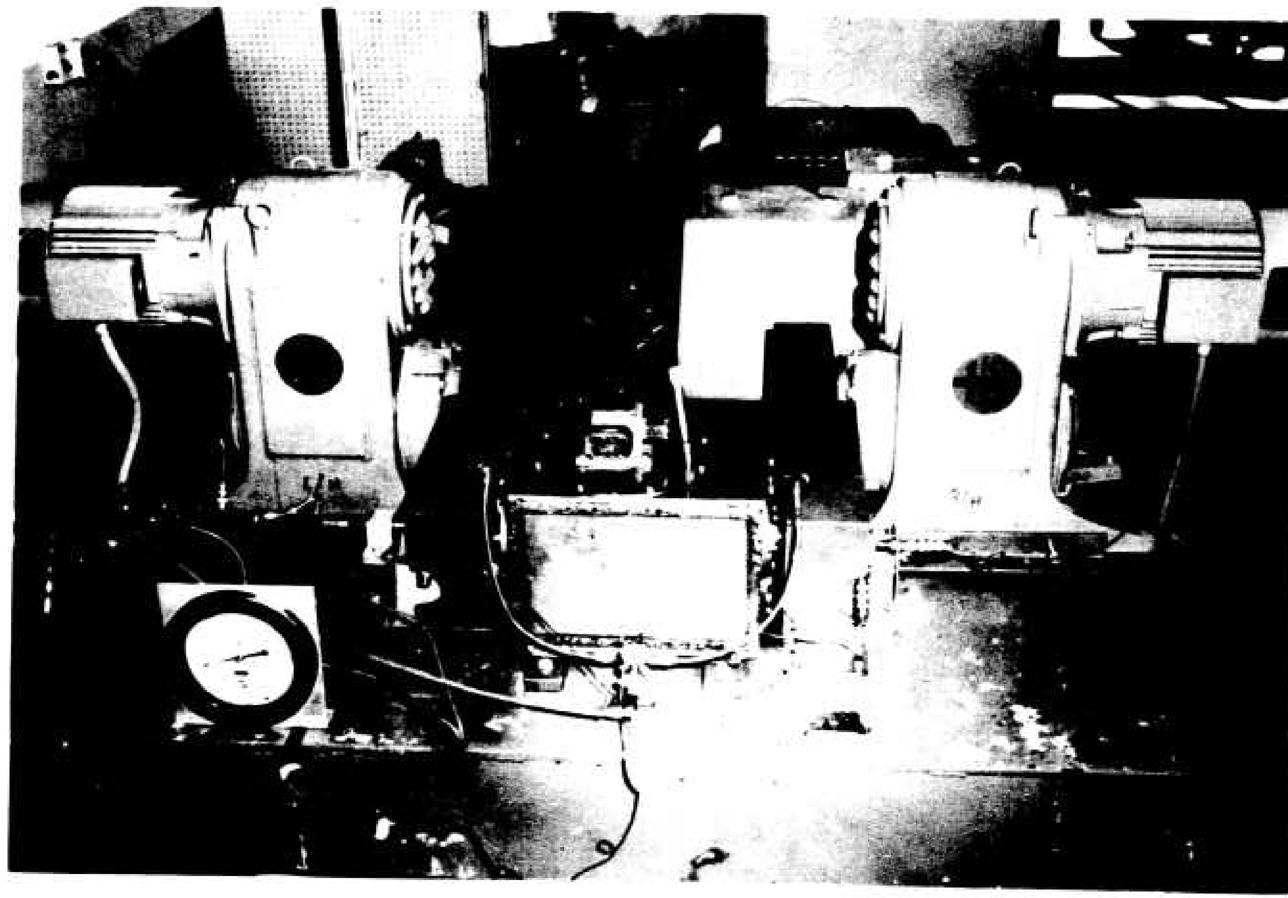


Figure 14. Dynamic Test Facility.

Temperature of carrier bearings
Vibration amplitude of clutch housing in three axes
Oil pressure

Test Plan

The test plan closely followed the usual procedure for evaluating lubrication requirements for other clutches. The clutch was to be overrun through the full-speed spectrum at design oil flow to find the point of maximum heat generation. The clutch would then be run at that speed at flows 50% above and below the design flow to determine the effect of flow variation on heat generated. Observations were also to be maintained on vibration, noise, and wear, as possible indicators of detrimental effects due to the oil flow variations.

Clutch drag at the various oil flows was also to be measured and recorded.

In order to separate the pawl system heat, noise and vibration effects from those generated in the clutch support bearings or in the rig, the tests were to be run first with no pawl system installed, then repeated with the full clutch assembly.

Test Results

In conducting the planned tests, it was found that heat generation attributed to clutch operation was slight. The absence of significant heat generation diminishes the importance of identifying the exact relationships between heating and variables such as oil flow or speed. The raw test data is given in Appendix IV, and reveals temperature rises which are typically 7° to $12^{\circ}F$. Attempts to plot most of this information resulted in erratic curves. Several important conclusions can be drawn, however, from the data:

- Heat generation during overrunning of the positive engagement clutch is slight.
- Heat generation is not significantly responsive to oil flow variations in the .75 to 2.25 GPM range.
- Heat generation is not significantly responsive to overrunning speed variations.

Measurement of clutch drag revealed a linear relationship between overrunning RPM and drag. The values were fairly small, reaching a peak of under 20 inch-pounds.

Power absorption represented by this drag amounts to about 3.0 HP.

Discussion

Evidence gathered in this test indicates that the heat removal criterion which establishes the oil flow requirements for sprag and roller clutches is not an appropriate concern in the positive engagement clutch. The lowest flow tested, .75 GPM, showed no indication of producing a trend to higher heat generation.

PAWL SYSTEM EVALUATION

Introduction

During the overrunning which was done as part of the lubrication requirements test, there was evidence that the pawl system was not operating in a manner conducive to long life. There were no indications that pawl hydroplaning occurred at any speed, and a number of failures were encountered in the pawl return spring system.

The methods of observation available in the lubrication requirements test arrangement were not useful for study of pawl system operation, or disclosure of problems, since the pawl system was completely surrounded by other clutch parts and hidden from view.

This test series was undertaken to permit study of pawl system operation, identification of problems, and development of corrections. It was intended, through this test program, to refine the pawl system so that operation in accordance with design intent could be realized.

Discussion of Failures

In the course of lubrication requirements testing, several episodes of failures had occurred involving the pawl return springs. The failure pattern appeared to involve nearly coincident breakage of several springs or spring attachments through high stress, low cycle fatigue. It was suspected that some unanticipated action of the pawl or spring was generated under particular overrunning conditions, leading to these failures. Identification of those particular conditions, or even the exact time of failure, could not be deduced from the observations made during the lubrication requirements testing.

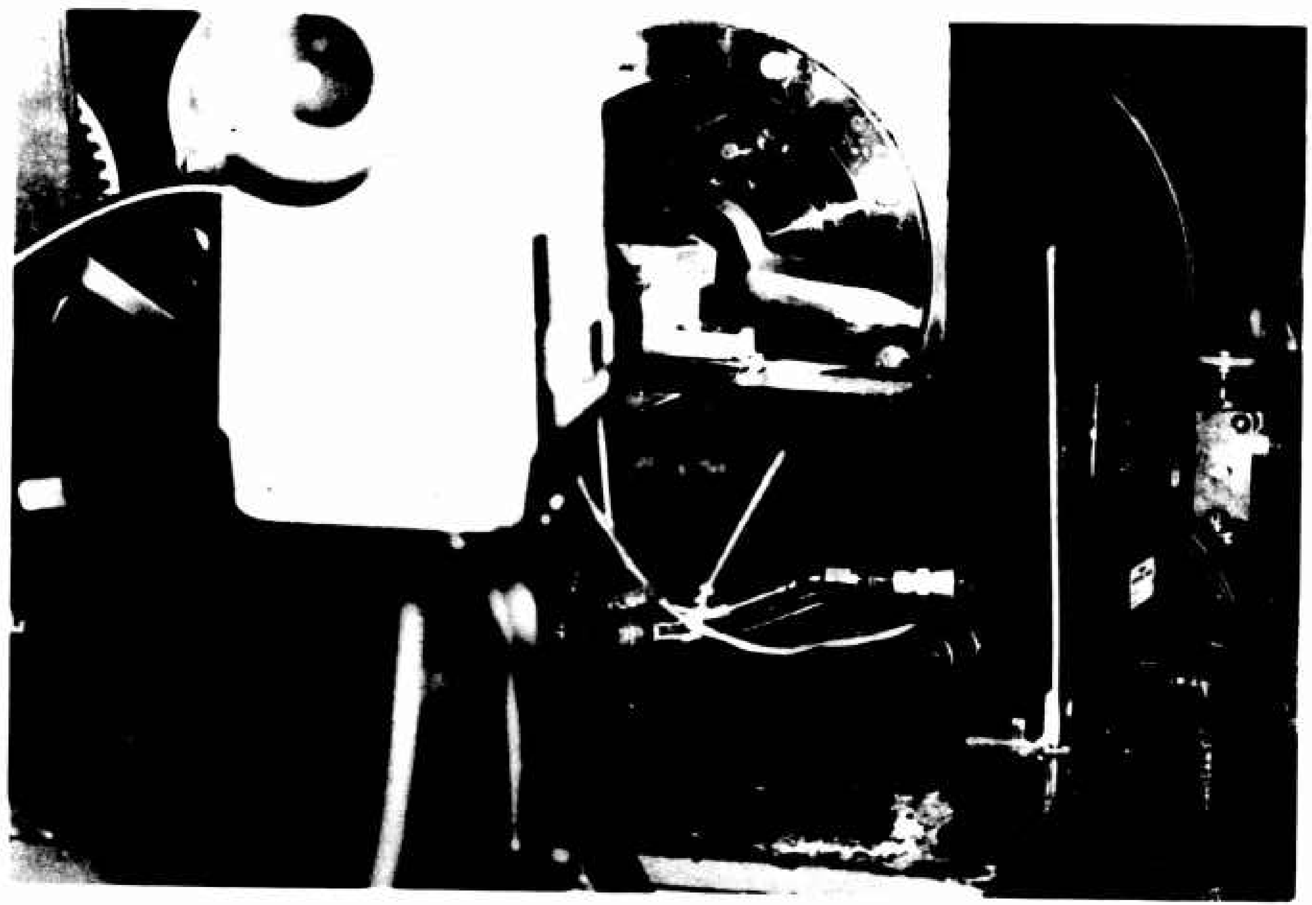


Figure 15. Pawl System Evaluation Test Arrangement.

Test Rig

Figure 15 shows the clutch test rig as set up for the pawl system evaluation tests. In this test arrangement, a single pawl and return spring are mounted to a fixed support in such a way that they can be readily viewed as the output member and ratchet are overrun at any selected speed. A video-tape recording system was incorporated into the test setup to supplement direct visual observation. All events were recorded on tape for subsequent replay at reduced or stopped speed.

The lubrication system remained unchanged from the previous (lubrication requirements) test, allowing control and measurement of oil flow and temperature.

Stroboscopic illumination was used throughout the test, so that pawl system operation could be closely observed.

This test arrangement provided a means to gain direct, intimate knowledge of the pawl system action. In the full clutch assembly tests, the pawl system was completely enclosed, and only indirect methods of observation were feasible.

Test Plan

The plan for this portion of the test program was to study the behavior of the pawl system throughout the full overrunning speed range. Through this study, the unanticipated, failure generating behavior could be identified, and corrective action evaluated.

Specifically, the plan was to operate the pawl system which had been subject to failure in the lubrication requirements test through the full overrunning speed range. By means of direct visual observation of the stroboscopically illuminated pawl system, and aided by video tape records of pawl operation, it was intended to study the conditions which had contributed to the previous failures. Particular attention was to be directed to the spring motion, pawl motion, and interaction between the pawl and ratchet. Alertness to extraneous spring motion and erratic or violent pawl action was to be maintained.

The next planned test phase was the evaluation of the effect of changes which could readily be made. Changes slated for trial were to include spring and spring support variations, pawl stop cushions and ratchet profile variation. The same methods of observation were to be used, and operation through the full overrunning speed spectrum surveyed.

A final evaluation was to be based on a pawl system incorporating the revisions deemed necessary and desirable on the basis of the first two test phases. Pawl system function throughout the overrunning speed range was to be examined and compared to the type of operation intended by design. To be acceptable, pawl hydroplaning would be required as the mode of pawl operation for all overrunning speeds in excess of 6000 RPM. Determination of the lubricating flow range to support acceptable pawl operation was a goal of this test phase.

Test Results

The initial test in the pawl system evaluation series called for operation with the pawl, spring, and ratchet which was run in the lubrication test series. The pawl return springs in this configuration had been subject to very early failure in previous testing, and observation of the operation leading to these failures was desired.

The plan for this test called for running the output (and ratchet) in increasing 1000 RPM increments, starting at 2000 RPM which was the rig minimum speed.

A considerable amount of extraneous spring motion was observed in this running, in the form of lateral movement of the spring tail in the pawl slot and squirming of the body of the spring on its arbor (see Figure 16).

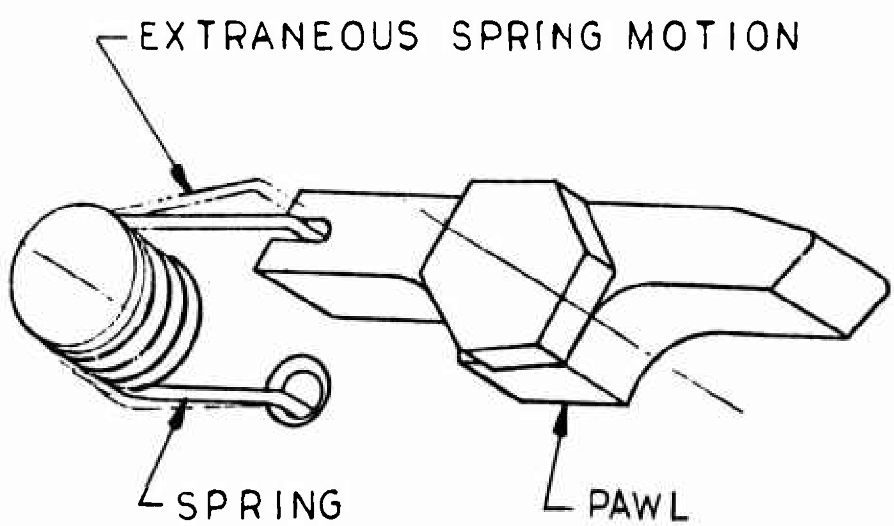


Figure 16. Extraneous Spring Motion.

This spring squirming increased with overrunning speed, and while increasing from 7000 to 8000 RPM, the spring lateral deflection allowed the spring tail to disengage from the pawl.

It is felt that the unwanted, extraneous spring motion which was observed increased cyclic stresses within the spring-wire severely. This higher stress level is the probable reason failures previously encountered in this spring.

The corrective measure taken to reduce this unwanted spring working was the addition of a thin, .500-inch-diameter washer under the .375-inch arbor flange. This provided more lateral support for the spring tail and a better endwise fit for the spring body.

This configuration was run and observed at 1000 RPM increasing increments of ratchet speed, with the spring motion found to be much better, and the squirming eliminated. As speed was increased, the pawl motion became increasingly violent, with no indication of support through hydroplaning. A travel stop is provided for the pawl in the form of a .190-inch-diameter roll pin, and the pawl rebounded against this pin quite hard. At 8000 RPM, the stop pin broke off.

It was evident from the observations made to this point in the test program that the pawl motions being achieved, particularly at overrunning speeds over 2500 to 3000 RPM, were not conducive to long life of the pawl system components.

It was considered vital to pawl system survival that hydroplaning action be established at least by 5500 RPM, since long term overrunning operation at that differential is viewed as probable.

It was thought that a probable reason for the failure to hydroplane was that the ratchet tooth peaks were protruding through the surface of the oil annulus inside the ratchet. The pawls striking these tooth peaks would be flipped outwardly, against the stops, and then rebound back against the ratchet. It can be visualized that this type of motion could build up to increasing violence with successive cycles.

A ratchet with revised profile was chosen as the subject for the next test. Figure 17 compares the original ratchet tooth profile with this revised one.

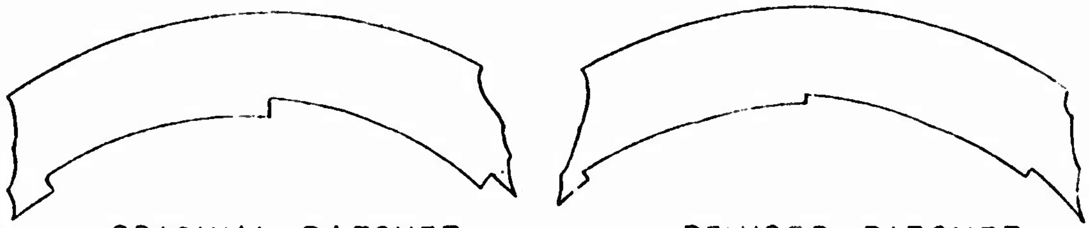


Figure 17. Comparison of Original and Revised Ratchet Profiles.

This configuration was tested, as were the previous ones, in 1000 RPM increasing increments of ratchet speeds.

The pawl was found to run with significantly less violence at speeds to 6000 RPM, and at all speeds above 6000 RPM hydroplaning action took over. During hydroplaning, the pawl motion stopped completely as the pawl rode smoothly on the surface of the oil film within the ratchet. It was clear that pawl system wear and distress would be reduced practically to the point of nonexistence in this mode of operation.

Testing next focused on pawl system durability in the non-planing mode. Since observation had indicated an increase in pawl motion up to transition to hydroplaning, this same configuration was continuously overrun at 5000 RPM. Thirty-five minutes of running were accomplished before the stop pin again broke.

Interim Conclusions

On the basis of pawl system observations up to this point, a number of conclusions were formed:

- The spring distress encountered in the lubrication tests was probably due to extraneous spring motion.

Stabilization of the spring to eliminate squirming and design to preclude lateral bending are required to assure reliable spring performance.

- Pawl planing is an achievable overrunning mode, offering operation free from fatigue and wear-producing pawl motion. If the differential overrunning speed required for planing could be reduced below the smallest continuous differential speed expected in normal transmission system operation, the result would be almost unlimited overrunning endurance.

Final Pawl System Revision

Based on the previous test observations and conclusions, a final pawl system configuration was assembled and tested.

Figure 18 shows this final pawl system in contrast to the system which ran in the lubrication tests. The improvements are as follows:

- The spring was designed with straight tangs rather than bent, a closer fit was provided between the spring and arbor, and the spring action was reversed so that increasing load tended to wind up the spring rather than unwind it. The spring preload was also reduced, since this new spring would be centrifugally augmented rather than diminished, which was previously the case.

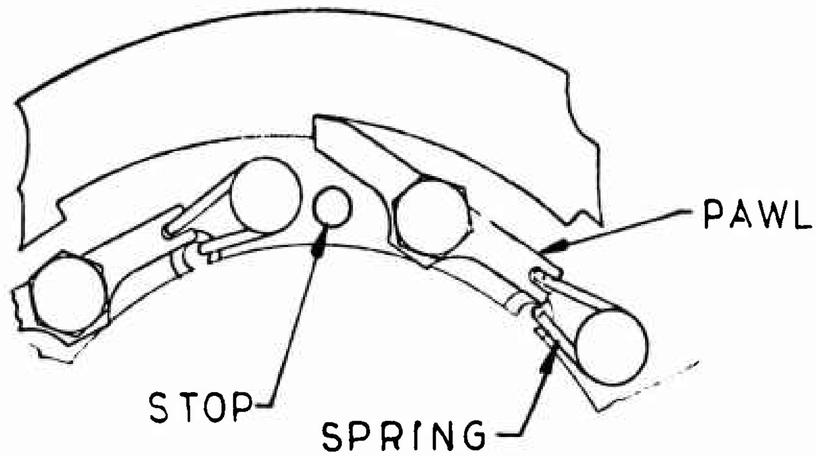
- The ratchet with revised profile which was used in the previous test was assembled in this configuration.

This configuration displayed a smooth pawl action with considerably less violent bouncing than any previous configuration.

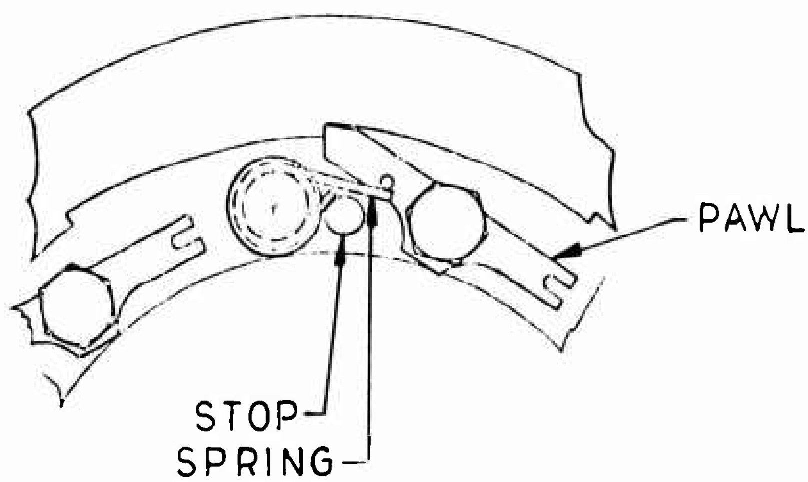
A careful survey was made of pawl cycling rate versus overrunning RPM, with the finding that the pawl responded to each ratchet tooth up to an overrunning speed of 1850 RPM (pawl frequency of 308 Hz).

Pawl motion above 1850 RPM was not precisely repetitive, but was not violent, with only occasional moves against the stop pin.

Hydroplaning of the pawl commenced at 3000 RPM overrunning speed, at an oil flow of .25 GPM, and persisted at all higher speeds.



PAWL SYSTEM USED IN LUBRICATION REQUIREMENT TEST



FINAL PAWL SYSTEM

Figure 18. Comparison of Pawl Systems.

Both visual and audible clues to the degree of violence of pawl motion indicated a significant reduction in comparison to the previous configuration. The clutch overran quietly, and contact with the stop pin was infrequent and light.

Since the maximum activity level of the pawl appeared to take place at 1850 RPM (the point where pawl natural frequency and excitation frequency coincide), it was decided to overrun for 2 hours at this speed with an oil flow of .75 GPM. It should be noted that this condition would only occur as a transient condition in normal operation.

Following the 2-hour run, the parts were examined and exhibited only superficial wear of the pawl and ratchet.

Conclusions

The pawl system performance as revealed in this single-pawl test setup was adequate and in accord with design intent.

A question of transferability of these results to a full six-pawl clutch required that this system be evaluated in a complete clutch assembly, with both input and output rotating.

OVERRUNNING ENDURANCE TEST

Introduction

Testing to this point had proven the ability of the positive engagement clutch to engage, carry torque and disengage. Lubrication tests revealed that oil flow requirements were modest, and at least at flows down to .75 GPM there were no indications that lubrication was marginal. Difficulties within the pawl system (spring failures, inability to hydroplane) had been corrected by the system adjustments derived in the pawl system evaluation. An important question remaining concerned the ability of the positive engagement clutch to overrun and engage as required throughout the operational speed range.

Overrunning demonstrations would be required in addition to those accomplished in the pawl system evaluation, because several operational factors could not be explored in that test.

- o The ability to maintain an oil annulus while planing six pawls would probably differ from the single-pawl situation.

- o Dynamic engagements, the primary task for this clutch, remained untried.

Test Rig

The test setup used in the lubrication tests was again used for this overrunning endurance test (see Figure 14). Instrumentation included 3 axis vibration pickups, pulse type input and output revolution counters, inlet and outlet oil thermocouples, and lubricating oil flowmeter.

Test Specimen

The full clutch as shown in Figure 7 was assembled, incorporating the ratchet and pawl system design which was used in the final pawl system evaluation test.

Test Plan

In an attempt to simulate actual operation, the test plan called for overrunning the output at high speed with the input stationary for a continuous 2-hour period. The clutch was then to be inspected, reassembled and overrun with the input at 6000 RPM for 2 hours. Following this overrunning, engagements and disengagements would be made at 2000 RPM increasing speed increments.

In the course of running this complete clutch, a test was also planned to measure the effect of oil flow variation on the hydroplaning transition point.

Test Results

The full clutch assembly was found to operate with the pawls hydroplaning at differential speeds ranging from 2500 RPM to 4000 RPM, depending on lubricating oil flow. A curve of the point of onset of hydroplaning is shown on Figure 19.

If pawl hydroplaning is used as a basis for establishment of lubricating oil flow, then a nominal flow in the .75 to 1.0 GPM range is seen as a good choice. Adequate margin thereby remains for a 50% diminution of flow with no detriment expected. Accordingly, the subsequent tests of overrunning and engagements were made with lubrication flow adjusted to 1 GPM.

The first 2-hour overrun was made with the input held stationary and the output running at 10,700 RPM. Inlet and outlet oil temperatures are plotted in Figure 20, showing a differential of about 25° F. It should be noted that the total

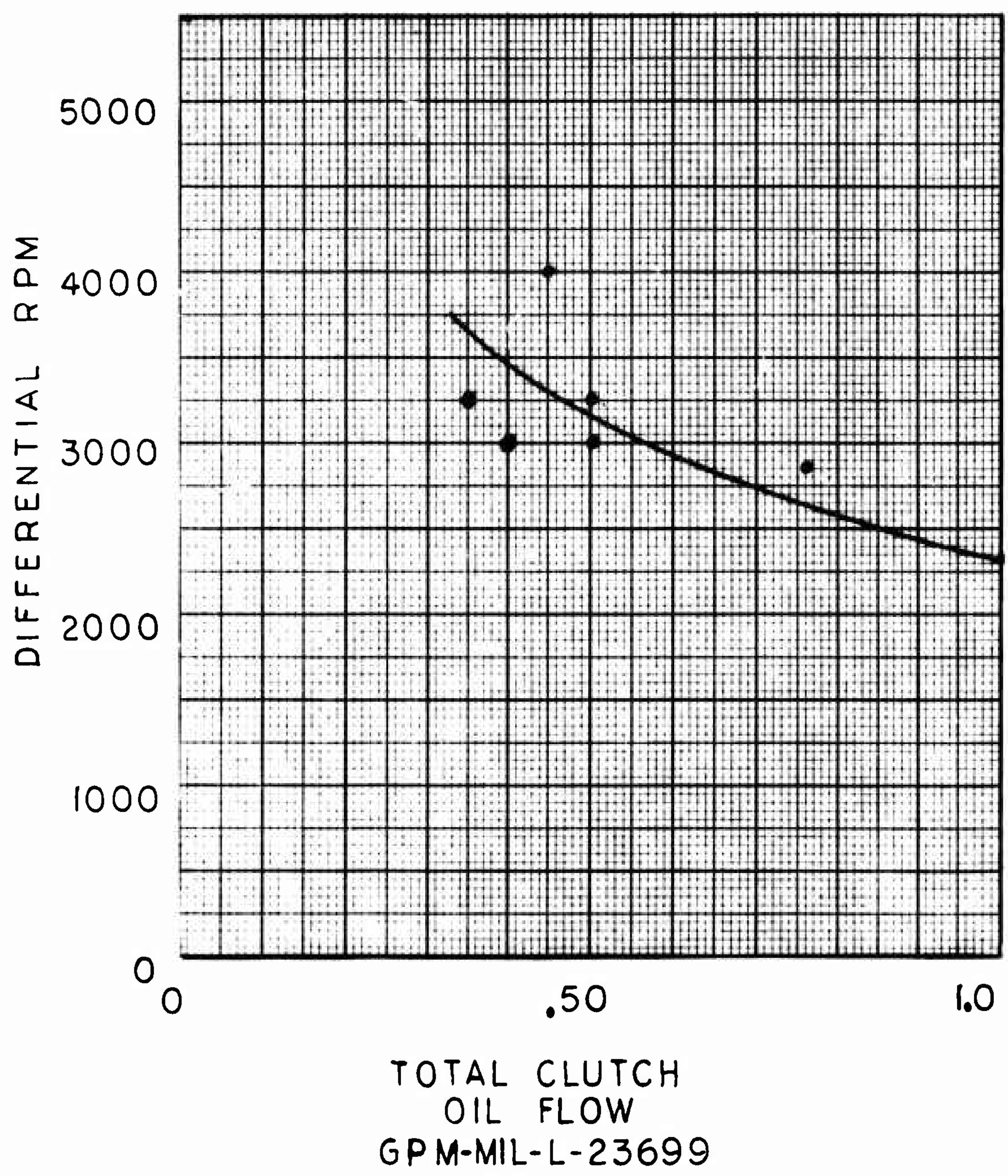


Figure 19. Onset of Pawl Planing
Vs Lubricating Oil Flow.

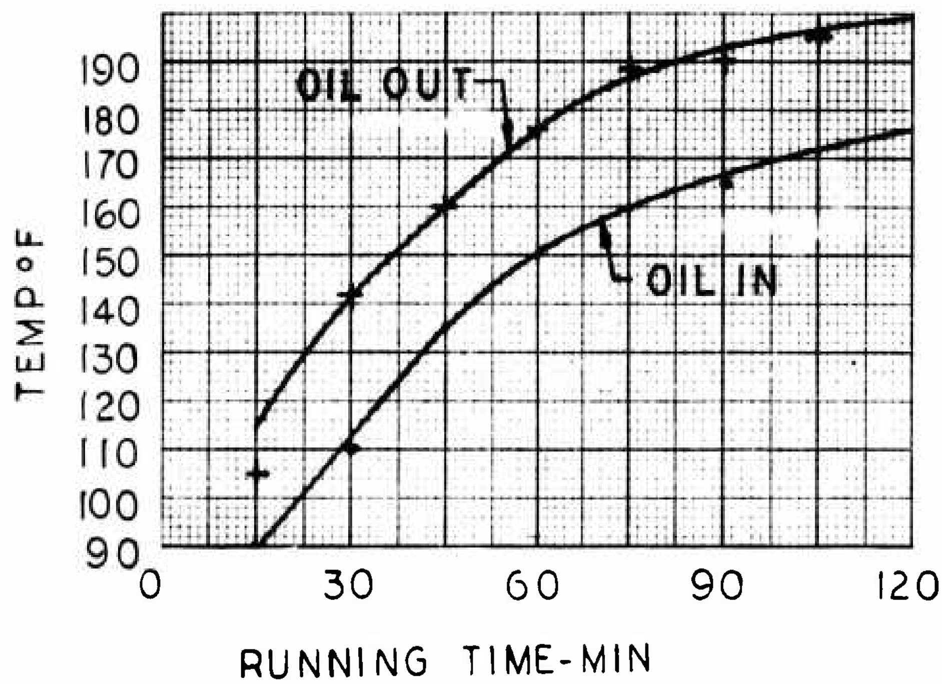


Figure 20. Oil Temp. Increase - Full Overrun (Output - 10,500 RPM; Input - 0 RPM).

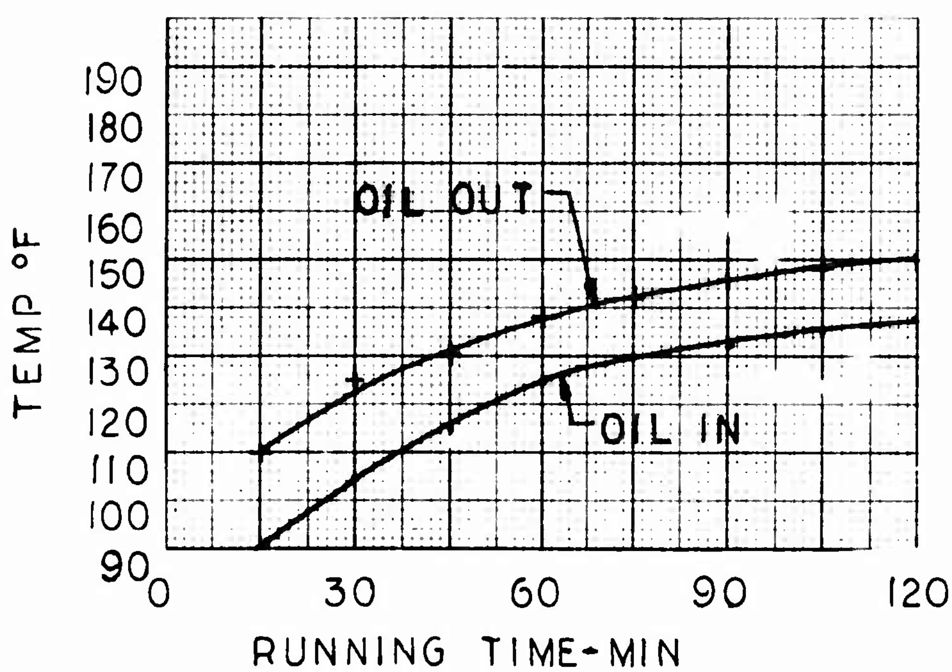


Figure 21. Oil Temp. Increase - Half-Speed Overrun (Output - 10,500 RPM: Input - 6,000 RPM).

oil system duty included supply for pawl hydroplaning, and lubricating two overrunning clutch bearings, two rig bearings, and rig seals.

Disassembly of the clutch after this run disclosed no appreciable wear of the pawls and ratchet, and no indication of distress in any clutch part.

The clutch was reassembled and overrun with the output at 10,700 RPM and the input at 6000 RPM. Oil flow was again held at 1 GPM, and temperatures were noted every 15 minutes. Figure 21, which is a plot of the temperature differential between inlet and outlet, shows this differential to be about 15° F.

RUNNING ENGAGEMENTS

Without change to the test setup, a series of clutch engagements and disengagements was made, in accordance with the following procedure:

> With the input stationary, oil flow at 1 GPM, output speed was adjusted to the desired engagement speed. The input shaft was then energized at its minimum speed and the input speed rapidly increased to produce an engagement. Traces of input and output speed were made as engagement was approached through the engagement event. Input speed was then readjusted to its minimum and shut down.

The sequence was repeated, producing engagements at 2000, 4000, 6000, 8000, 10,000 and 10,700 RPM. The engagements and disengagements were positive, smooth, and without harmful incident. These engagements are plotted on Figures 22 through 27.

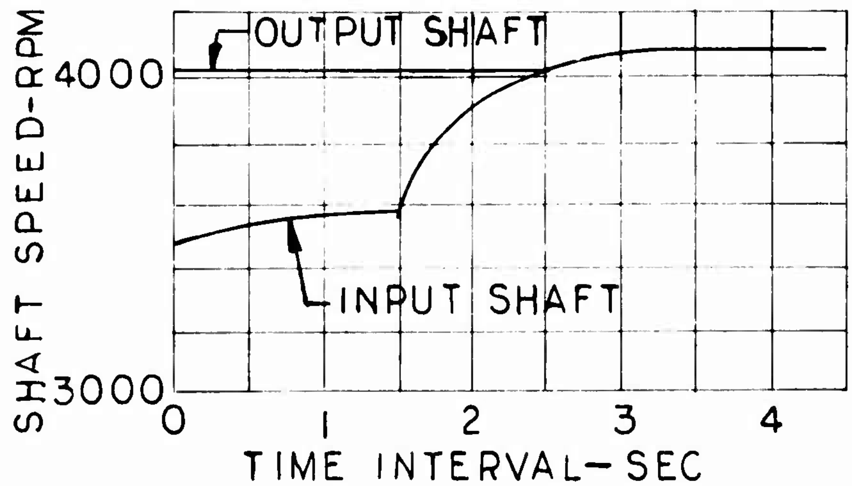


Figure 22. Clutch Engagement at 4020 RPM.

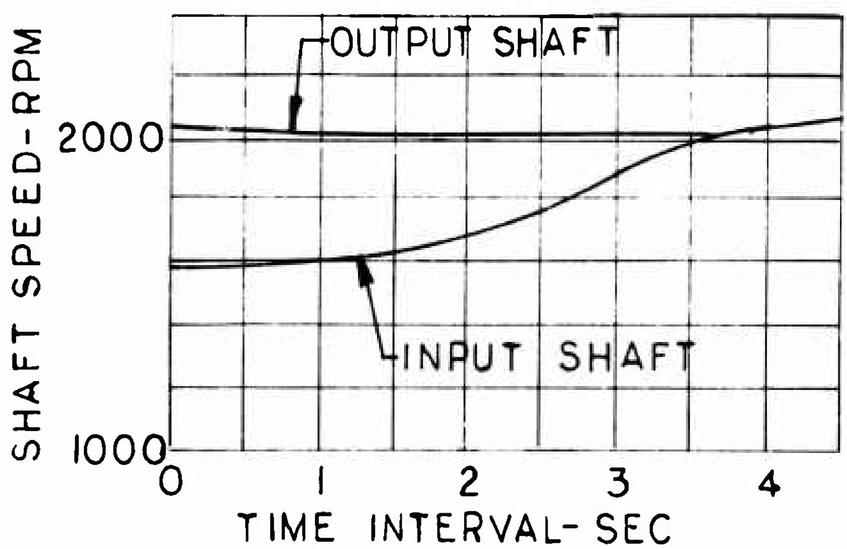


Figure 23. Clutch Engagement at 2010 RPM.

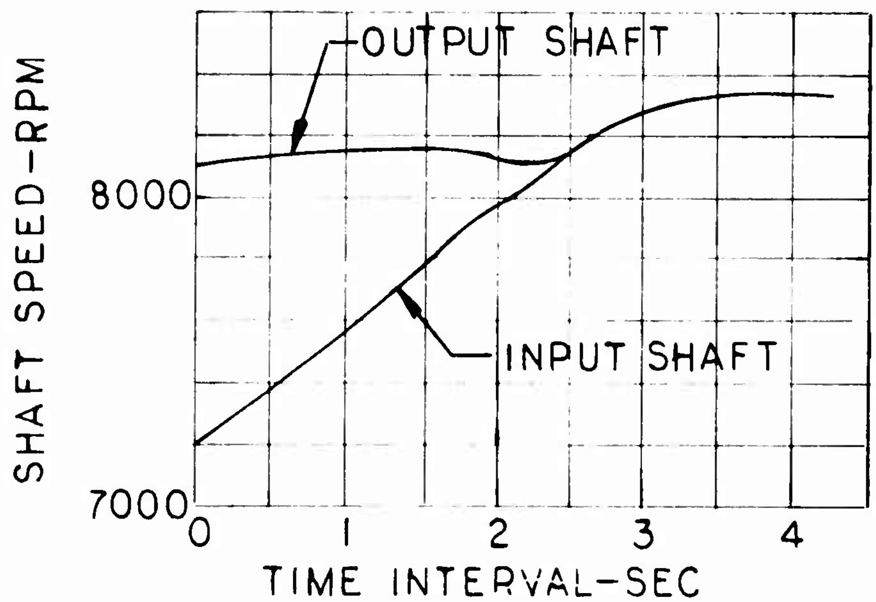


Figure 24. Clutch Engagements at 8160 RPM.

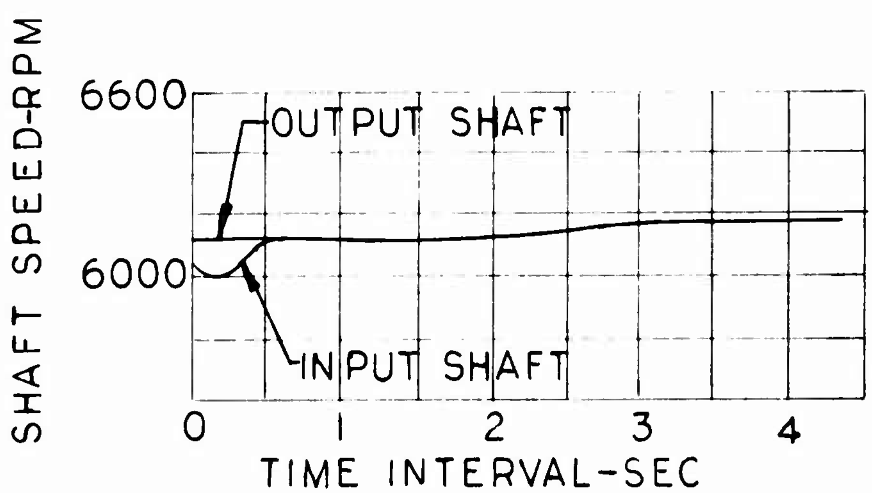


Figure 25, Clutch Engagement at 6120 RPM.

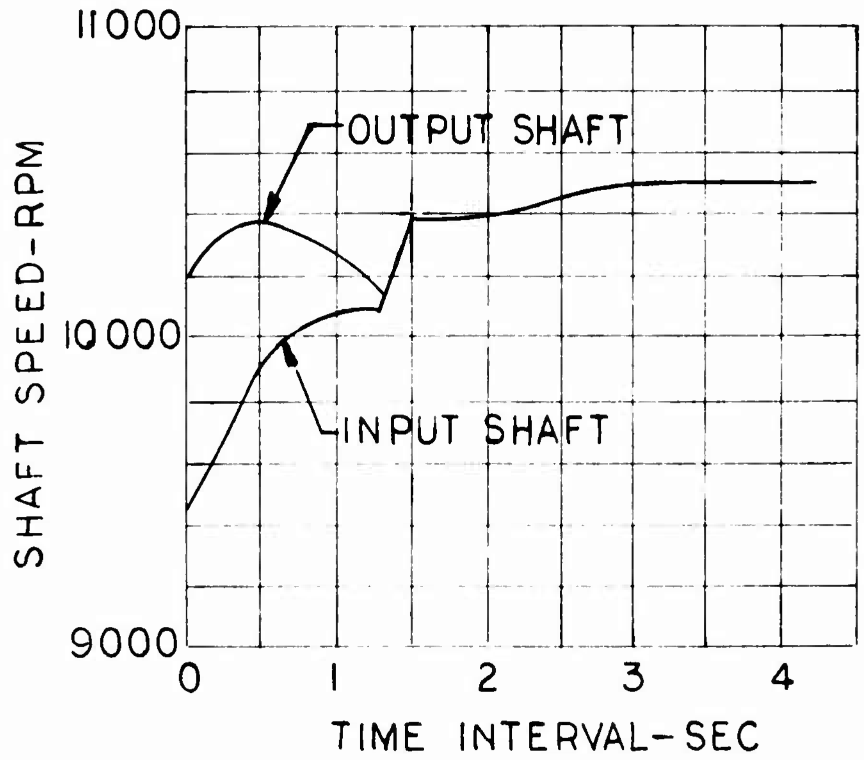


Figure 26. Clutch Engagement at 10,100 RPM.

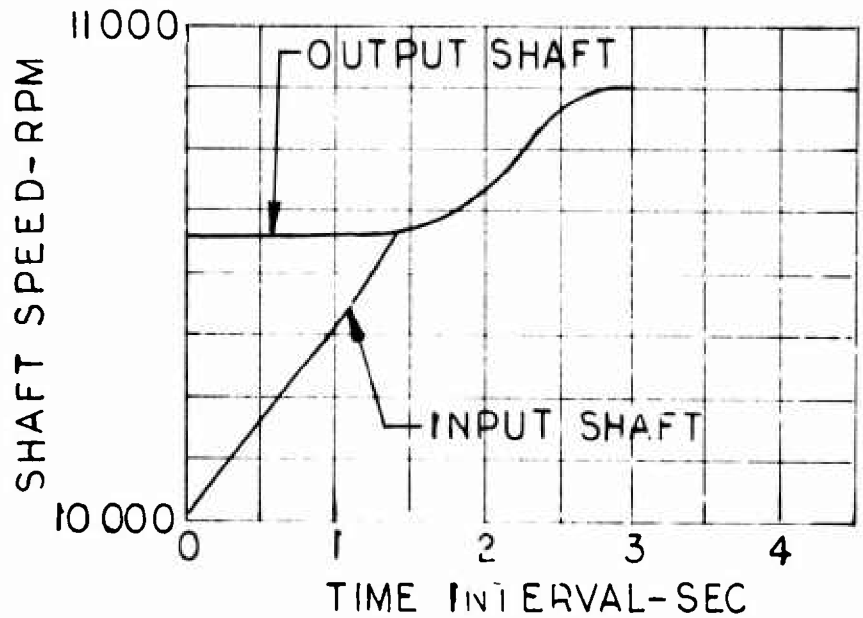


Figure 27. Clutch Engagement at 10,920 RPM.

ANALYSIS OF HARDWARE AFTER TEST

Following the tests the clutch was disassembled and detail parts were examined. Very little wear was observed in the pawl and ratchet system, and no wear was detected in the remaining parts, including bearings, face spline and helical spline.

The internal ratchet surfaces showed no more than light polishing in the track of the overrunning pawls, with the pawls themselves exhibiting somewhat heavier surface polishing. A single pawl, typical of all six, was cross sectioned and subjected to microscopic study for determination of wear.

Figure 28A is a microphotograph of an unworn section of the pawl surface, while Figure 28B shows a similar view of the worn portion of the same pawl. Material of this part is 4340 steel with a core hardness of Rc 32, case hardened (Tuftrided) by a salt bath nitride treatment. Figure 28A reveals a white surface layer which is .00016 to .00024 inch thick, which is typical of the condition expected as a result of the heat treatment. Micro hardness checks reveal hardened material below the white layer to a depth of .005 to .006 inch, with a gradual decrease to core hardness beneath this level. This case-hardened area is visible in the microphotograph as a layer of finer grained material.

The microphotograph of the worn area (Figure 28B) shows remaining vestiges of the white layer, so that the material removed through wear can be estimated as less than .0002 inch. This wear amounts to approximately 3% of the depth of relatively hard case material which would continue to support operation with little increase in wear rate.

All clutch parts remained in good condition following the test program, with no evidence of material distress and only slight wear of the pawls. Since the test experience leading to this wear consisted of 4 hours of hydroplaning overrunning together with approximately 1/2 hour of nonhydroplaning overrunning, a prediction of overrunning life capability on the order of 150 hours can be made. This is recognized as a somewhat tenuous prediction, but in view of the readily available design alternatives for wear enhancement which could be employed if needed, it is regarded as completely feasible.

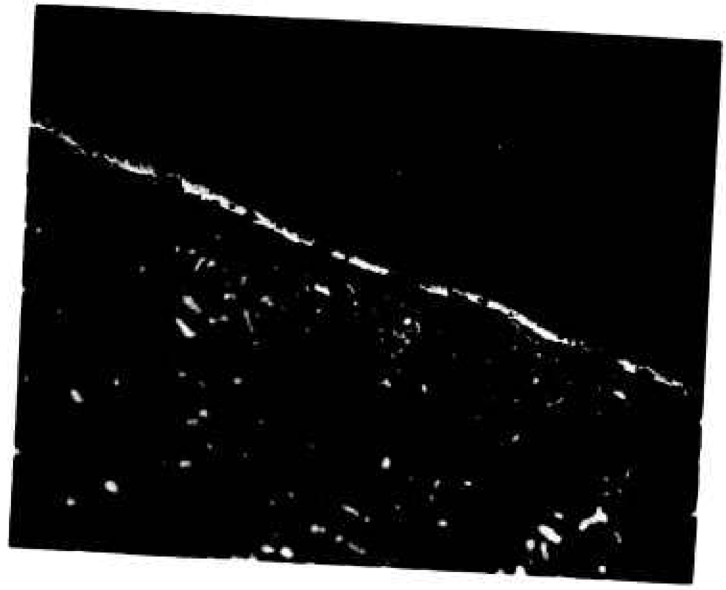


Figure 28A. Microphotograph of Unworn Surface of Pawl. (500X)

Figure 28B. Microphotograph of Worn Surface of Pawl. (500X)

CONCLUSIONS

Static testing of the positive-engagement clutch demonstrated the following:

1. The clutch engaged smoothly and properly with no more than 80 inch-pounds of torque required.

2. The engaged clutch transmitted 40,000 inch-pounds of torque with no distress to any clutch part and no tendency to disengage. Torsional deflection was 1.2°, and axial deflection was .006 inch.

3. Disengagement was accomplished with the application of 10 inch-pounds reverse torque, with no binding resulting from the preceding application of full design torque.

Lubrication requirements testing showed that heat generated by the clutch while overrunning was slight, and that cooling requirements would not be the criteria to use in determining the optimum oil flow. This testing also revealed weaknesses in the pawl system design, as evidenced by pawl return spring failures.

Pawl system evaluation tests showed that:

1. With relatively small adjustments to the original pawl system design, overrunning modes leading to reliable operation and long life could be achieved.

2. Pawl hydroplaning is a remarkable overrunning mode, wherein the pawl is immobile. There is no metal-to-metal wear and no cyclic loading of the return spring.

3. Pawl motion below hydroplaning speed is not distressful; and at differential speeds of less than 1850 RPM, the pawl motion faithfully responds to each ratchet tooth.

The overrunning endurance test of the complete clutch demonstrated that:

1. Overrunning at both full and half differential speed produced very low wear rates, indicating the feasibility of overrunning life of 150 hours or greater.

2. Lubricating oil flow of 1 GPM was satisfactory, with satisfactory operation demonstrated over the range from .50 to 2.0 GPM.

The engagement tests showed that the clutch does engage when input speed coincides with output speed, and the clutch disengages when the input is slowed; these operations are accomplished smoothly and without incident.

RECOMMENDATIONS

The reported program has successfully met the initial goal of designing, fabricating, and testing a positive engagement clutch to HLH scale. Engagement, torque transmission, disengagement, and overrunning have all been demonstrated. Some aspects of operation which the clutch would face in actual use have not been probed, however: long-time driving endurance with fluctuating torque loads such as occur in helicopter drive trains and exposure to high-impact engagements. The high-impact engagement represents an emergency situation in which the clutch must survive engagement at maximum engine acceleration.

It is recommended that the existing clutch, which remains in good, operable condition, be submitted to an extended program which would explore both of the aforementioned operational aspects, and then proceed to an ultimate torque test.

The endurance test could be run in the static torque test rig through incorporation of a high-frequency cyclic load source (shaker). In this way, various cyclic torques could be superimposed on a schedule of steady driving torques for extended periods of time.

A dynamic rig modification could be made to provide momentum, spring rate, and acceleration equivalent to the HLH engine input system, so that high impact engagement effects can be probed.

Testing to ultimate torque can be accomplished in the present static torque test rig.

In view of the potential benefits which would follow from development of the positive engagement clutch, it is urged that the above recommendations be implemented.

APPENDIX I
DIMENSIONAL DEFINITION OF FACE SPLINE

The CURVIC Coupling Dimension Sheet No. 563.168 gives the set-up instructions and finished part dimensions which Gleason Works recommends for the positive-engagement clutch face splines. These recommendations were adhered to in fabrication of the subject test clutch.

Spline stresses calculated by Gleason Works for loads of 85,000 lb-in. and 120,000 lb-in. are as given below. These calculated stresses assume that the clamping force is 1.5 times the separating force.

Torque Load	85,000 lb-in.	120,000 lb-in.
Shear Stress (psi). . . .	8,100	11,420
Surface Stress (psi). . .	20,600	29,000
Equivalent Surface Stress (psi)	36,050	50,800
Recommended Clamping Force (lb)	15,450	21,800

Gleason Works recommended allowable limits for 150,000 psi ultimate tensile strength steel are 15,000 psi shear and 40,000 psi surface stress for applications involving both bending and torsion. The recommended limit for the equivalent surface stress is 145,000 psi. The calculated stresses are well within these recommended limits.

TABLE IV. FACE SPLINE DIMENSIONS

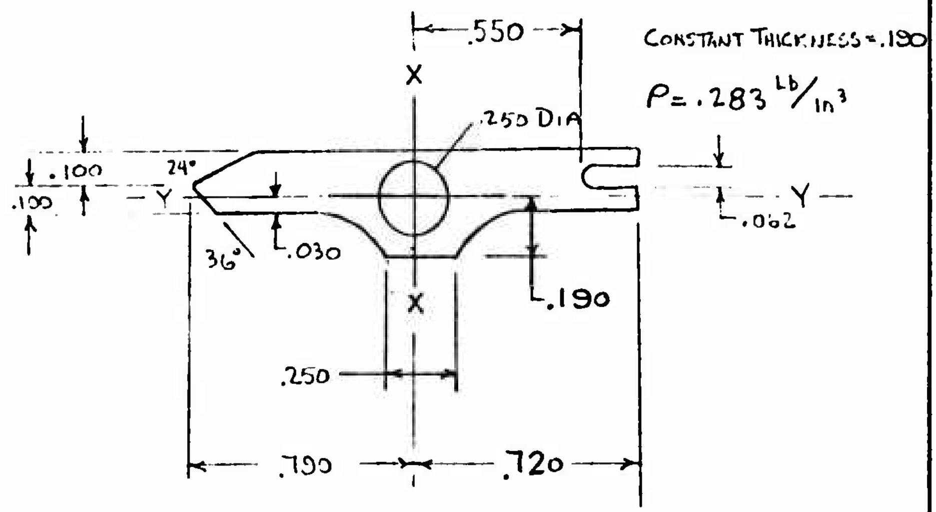
Dimensional Sheet No. 563.168 Form A 6/05/72

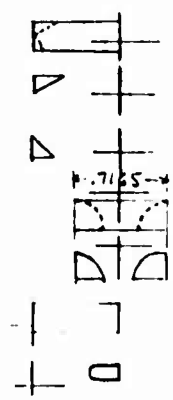
Number of Teeth	30	Outside Diameter	7.557"
Diametral Pitch	3.970	Outside Circular Thickness	.398"
Total Face Width	.250"	Face Angle of Blank	90°
Pressure Angle	15°	Root Angle	90°
Circular Pitch	.791"	Gable Angle Theoretical	$0^{\circ}20'$
Whole Depth	.222"	Gable Height Theoretical	.001"
Clearance	.025"	Backlash	.000"
Chamfer Depth	.023"	Design Desired	Std
Addendum	.098"	Final Design	Std
Dedendum	.123"	Tooth Proportions	Givn
		Manufacturing Method	Grnd

	Concave	Convex
Point Diameter - Wheel	11.186"	11.318"
Actual Point Width	.228"	.232"
Mutilation Point Width	.293"	.299"
Outer Slot Width	.327"	.332"
Limit Slot Width	.327"	.306"
Limit Slot Width Cone Dist	3.779"	3.529"
Inner Slot Width	.301"	.306"
End Angle	$0^{\circ}20'$	$0^{\circ}20'$
Dresser Block Angle	$32^{\circ}0'$	$32^{\circ}0'$
Pressure Angle - Wheel	$15^{\circ}0'$	$15^{\circ}0'$
Clearance Pressure Angle	$17^{\circ}0'$	$17^{\circ}0'$
Maximum Radius - Interference	.065"	.065"
Edge Radius - Wheel	.065"	.065"
Radius to Point of Tangency Std	3.654"	
Layout Radius - Wheel	5.626"	5.626"
Layout Radial - Wheel	6.708"	6.708"
Separating Force Factor K1	.0733	
Shear Stress Factor K2	.0954	
Surface Stress Factor K3	.2422	
Bolt Load Stress Factor K4	3.3020	

APPENDIX II
ANALYSIS OF PAWL SYSTEM

PAWL WEIGHT & BALANCE





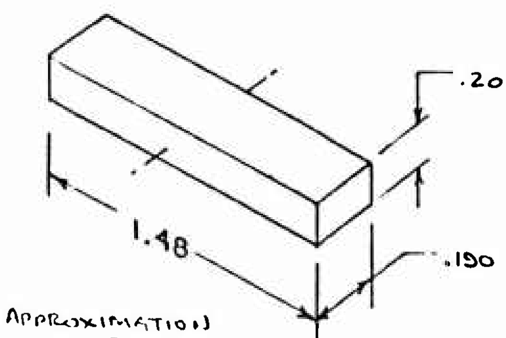
WT +/−	VOL	Wt	X	Y	Mx	My
+	.03002	.008496	.395	.070	.00336	.000595
−	.00214	.0006	.715	.136	.000432	.0000823
−	.00069	.00019	.766	.003	.000149	.000000586
+	.02178	.00616	0	.110	0	.000678
−	.01657	.00469	0	.083	0	.000391
+	.02736	.00774	.36	.070	.00279	.00054
−	.00164	.00046	.6525	.081	.00030	.0000375
SUMMATION		.0165 LB			.000289 (NOSE HEAVY)	.000728

REVISED:

PAWL INERTIA:

$W = .0165$ LB

$K = \sqrt{I/A}$



APPROXIMATION OF PAWL

$I = \frac{bd^3}{12} = \frac{(.20)(1.48)^3}{12} = .054$

$A = 1.48 \times .20 = .296$

$\therefore \; K = \sqrt{\frac{.054}{.296}} = .427$

$J = \left(\frac{W}{g}\right) K^2 = \frac{.0165}{386.4}(.182)$

$= .00000\,7795$

PAWL NATURAL FREQUENCY:

$$f_n = \frac{1}{2\pi}\sqrt{\frac{K_s}{J}}$$

K_s = SPRING RATE OF PAWL RETURN SPRING

$= \frac{\text{TORQUE (LB INS)}}{\text{DEFLECTION (RADIANS)}}$

$\frac{M}{T} = 3.38 \frac{\text{LB INS}}{\text{TURN}}$, OR $3.38 \frac{\text{LB INS}}{2\pi \text{ RAD}}$.

$K_s = \frac{3.38}{2\pi} = .538$ LB INS/RAD

$$f_M = \frac{1}{2\pi}\sqrt{\frac{.538}{.0000078}} = 41.80 \text{ Hz}$$

(OK, MIN. ACCEPTABLE = 20 Hz)

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PAWL RETURN SPRING

$d = .039''$ = WIRE DIA
$d^3 = .0000593$, $d^4 = .0000023$
$D = .393''$ = MEAN DIA
$N = 4.6$ = NO. OF COILS

$$\frac{M}{T} = \frac{E\ d^4}{10.2\ D\ N} = \frac{\text{INCH LBS}}{\text{FULL TURN DEFL.}}$$

$$\frac{M}{T} = \frac{30 \times 10^6 \times 2.3 \times 10^{-6}}{10.2 \cdot .426 \cdot 4.7} = 3.38\ {}^{\text{IN LBS}}/_{\text{TURN}}$$

BY LAYOUT, WORKING DEFLECTION = .111 TURNS - .144 TURNS

$M = .375$ TO $.486\ {}^{\text{LB}}/_{\text{INS}}$

$$\text{BENDING STRESS} = \frac{32\ M}{\pi\ d^3} = \frac{32 \times .486}{\pi \times 5.93 \times 10^{-5}} = \underline{83,480\ \text{psi}}$$

REVISED:

APPENDIX III
CRITICAL SPEED ANALYSIS

SUMMARY

A cursory analysis of the Kaman designed HLH clutch (P-2087) indicates that its critical whirl speed is well above its proposed operating speed of 20,000 RPM.

DISCUSSION

A preliminary model of a HLH clutch is in the design stage and it was desired to ascertain if this design configuration would operate in the region of the critical whirl speed.

An analysis was performed utilizing an existing digital computer program (ZGIC). The clutch assembly was simulated by a series of 15 lumped masses. The structural inertia (EI) and mass moment of inertia (polar) were calculated for use in the program. Additional calculations were performed to determine the spring rate of the support bearing. The effect of spring rate and structural inertia on the whirl frequency was ascertained for various combinations. The results are listed in Table V, and it can be noted that the calculated speeds are well above the operating frequency of 20,000 RPM.

Table VI contains the physical characteristics calculated for the clutch assembly and are included for reference.

TABLE V. CRITICAL WHIRL SPEEDS FOR HLH CLUTCH

CASE	SPRING RATE AT SUPPORTS	EI DISTRIBUTION	CRITICAL SPEED (RPM)
1	Infinite	Calculated	66,270
2	Calculated	Calculated	56,200
3	½ Calculated	Calculated	49,000
4	Twice Calculated	Calculated	60,900
5	Calculated	½ Calculated	43,080
6	Calculated	½ Calculated Between Supports	53,700

Operating RPM: 20,000

Calculated Spring Rate of Bearing: 5.4×10^6 lb/in.

See Table VI for Calculated EI Distribution.

TABLE VI. CALCULATED VALUES OF MASS, EI AND MASS MOMENT OF INERTIA

GENERAL INFLUENCE COEF
'20/66

HLH CLUTCH CHECK OF WHIRL FREQ   CALCULATED SPRING RATE 5

'UT DATA

STA. INS	EI	K LB/IN	C IN-LB/RAD	M SLUGS	I SLUG IN S2
0.0	1.7265D 06	0.0	0.0	8.7800D-04	4.2[illegible]-05
2.500	1.7265D 06	0.0	0.0	1.3700D-03	7.[illegible]-05
* 4.375	2.6616D 06	5.4000D 06	0.0	3.7600D-04	2.2300D-05
5.100	4.0816D 08	0.0	0.0	1.6910D-03	1.2443D-03
5.500	3.4096D 09	0.0	0.0	8.7100D-03	1.3193D-02
6.220	7.8400D 08	0.0	0.0	5.1900D-03	1.7460D-03
6.900	5.3300D 08	0.0	0.0	5.0600D-03	1.7460D-03
8.920	4.3100D 08	0.0	0.0	1.1720D-02	2.4500D-03
10.950	6.9100D 08	0.0	0.0	8.5780D-03	3.0077D-03
11.650	9.2400D 09	0.0	0.0	1.5078D-02	4.1179D-02
12.240	5.7470D 09	0.0	0.0	1.0672D-02	7.1210D-03
13.220	4.6000D 09	0.0	0.0	3.5162D-02	3.0141D-02
15.370	2.6700D 07	0.0	0.0	4.2130D-03	6.5800D-04
* 17.120	2.2160D 06	5.4000D 06	0.0	3.0560D-04	1.4100D-05
19.000	1.1480D 06	0.0	0.0	8.3200D-04	4.0000D-05

* DENOTES ℄ OF SUPPORT BEARING

+ MATERIAL STEEL E = 30×10^6

APPENDIX IV
RAW TEST DATA

PMIC TEST RIG CHECK

1. OIL FLOW VS PRESS:

PRESS	FLOW @ 170°F	FLOW @ 200°F
	GPM	GPM
20	1.35	1.30
30	1.65	1.57
40	1.88	1.86
50	2.15	2.11
60	2.33	2.32
70	2.50	2.49
80	2.65	2.55

2. BASE LINE VIBRATION CHECK:

RPM	VERT.	HOR.	FORE & AFT
1400	.0001	.0000	.0002
2000	.00015		.00025
3000	.00022		.0008
4000	.00025		.00075
5000	.00036		.0003
6000	.00043		.00058
7000	.00068		.00036
8000	.0016		.0012
9000	.0009		.0014
10000	.0007		.0010
11000	.00055		.0005
12000	.00045	.0000	.00055

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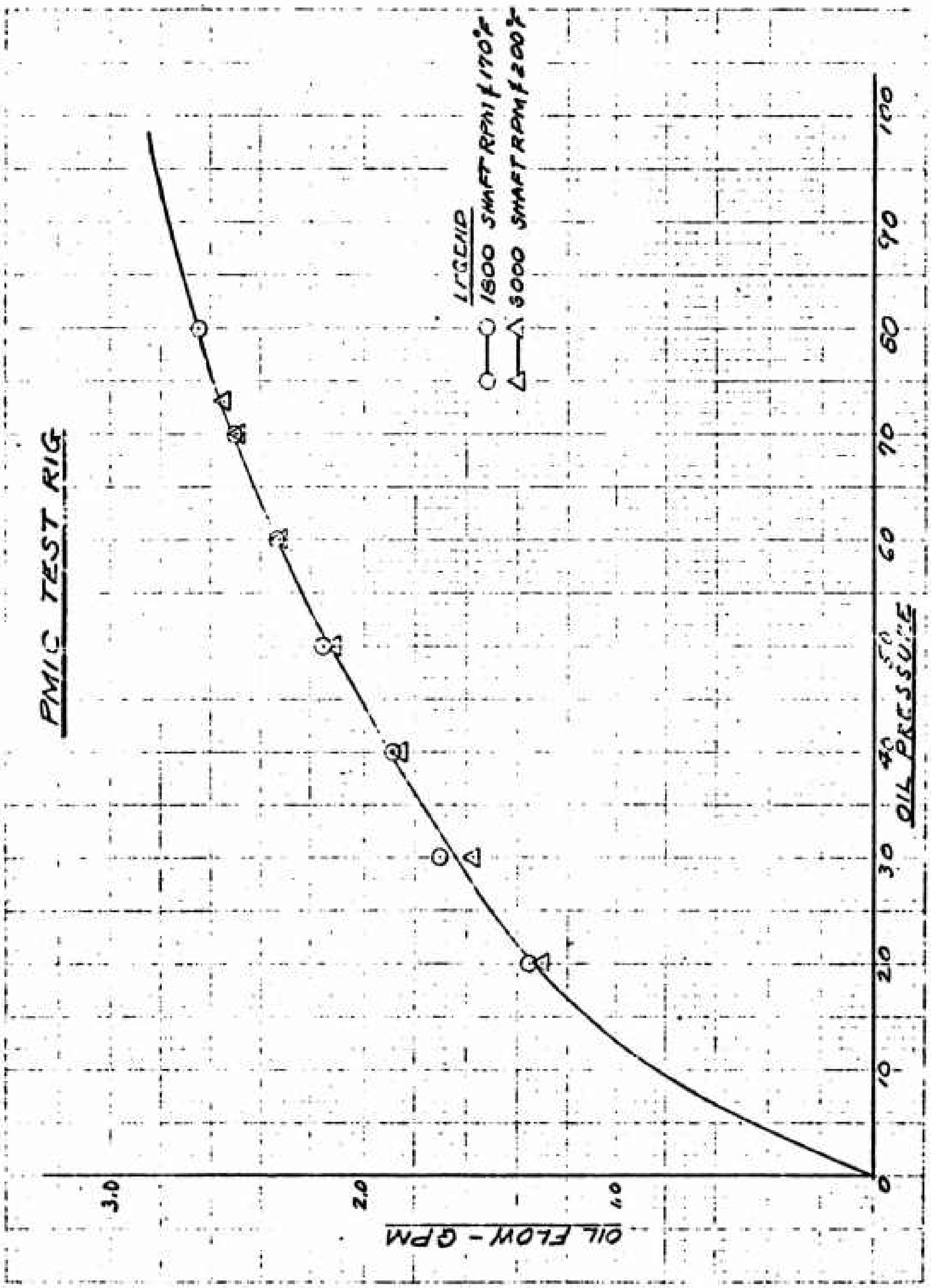
ENGINEERING GENERAL REPORT K65 REV. 4/70



PMIC TEST RIG
LEGEND
1800 SHAFT RPM / 170°F
3000 SHAFT RPM / 200°F
OIL FLOW - GPM
3.0
2.0
1.0
0
10
20
30
40
50
60
70
80
90
100
OIL PRESSURE

POSITIVE ENGAGEMENT CLUTCH
STATIC TEST DATA

SHT 1

RUN NO STRETCH RUN

GAP A	TORQUE TO ENGAGE	APPLIED TORQUE LOAD	DIAL INDICATOR # 1	DIAL INDICATOR # 2	TORQUE TO DISENGAGE	TOOTH CLICK
.2630		—	.880	.880		
.2640		2500	.860	.862		
.2669	60:/80	10 000	.791	.791	10 LB IN	
.2685	LB IN	20 000	.575	.545		
.2702		30 000	.268	.420		
.2720		40 000	.231	.305		
		0	.461	.534		

RUN NO 1

GAP A	TORQUE TO ENGAGE	APPLIED TORQUE LOAD	DIAL INDICATOR # 1	DIAL INDICATOR # 2	TORQUE TO DISENGAGE	TOOTH CLICK	SOLDER (TOOTH SPACE)
.2490		—	.559	.643			.0484
.2608		10 000	.451	.535			
.2639		20 000	.359	.445			
.2639		30 000	.305	.391			
.2670	40 LB IN	40 000	.230	.320	10 LB IN		.0336
.2650		0	.476	.567			

POSITIVE ENGAGEMENT CLUTCH
STATIC TEST DATA

SHT 2

RUN No 2

GAP A	TORQUE TO ENGAGE	APPLIED TORQUE LOAD	DIAL INDICATOR #1	DIAL INDICATOR #2	TORQUE TO DISENGAGE	TOOTH CLICK	SOLDER (TOOTH SPACE)
.2670		0	.480	.580			.0490
.2696		10 000	.411	.511			
.2710		20 000	.320	.424			
.2716		30 000	.266	.372			
.2698	40 LB IN	40 000	.230	.331	10 LB IN		
.2664		0	.473	.573			

RUN No 3

GAP A	TORQUE TO ENGAGE	APPLIED TORQUE LOAD	DIAL INDICATOR #1	DIAL INDICATOR #2	TORQUE TO DISENGAGE	TOOTH CLICK	SOLDER (TOOTH SPACE)
.2666		—	.460	.560			.046
.2675		10 000	.392	.495			
.2690		20 000	.305	.407			
.2706		30 000	.251	.355			
.2706	70 LB IN	40 000	.203	.312	10 LB IN		.033
.2700		0	.450	.550			

POSITIVE ENGAGEMENT CLUTCH

STATIC TEST DATA

SHT	3
RUN No	4

GAP A	TORQUE TO ENGAGE	APPLIED TORQUE LOAD	DIAL INDICATOR # 1	DIAL INDICATOR # 2	TORQUE TO DISENGAGE	TOOTH CLICK	([illegible] TOOTH SPACE)
.2658		0	.470	.580			.046
.2660		10 000	.398	.515			
.2670		20 000	.310	.431			
.2680	70 LB IN	30 000	.254	.378	.10 LB IN		
.2708		40 000	.212	.337			.0326
.2650		0	.450	.561			

RUN No	

GAP A	TORQUE TO ENGAGE	APPLIED TORQUE LOAD	DIAL INDICATOR # 1	DIAL INDICATOR # 2	TORQUE TO DISENGAGE	TOOTH CLICK

NO SYNCHRONIZER OR PAWLS

POSITIVE ENGAGEMENT CLUTCH

TEST NO C-461

TEST DIR. E. ROEHL	OPERATOR	ROOM TEMP 75°	DATE 1/24/73	SHT. 1

TIME	OIL TEMP IN	OIL TEMP OUT	BRG. TEMP INPUT	BRG. TEMP OUTPUT	OIL FLOW	INPUT RPM	OUTPUT RPM	VIBRATION VERT	VIBRATION HOR	VIBRATION FORE & AFT	OIL PRESS
1045	160	160	165	175	57 / 2.25	2000	10 500	.001	.0008	.0.02	58
1050	165	177	170	179	52.3	4000	"	.001	.001	.0002	55
1055	168	169	168	180	52.3	6000	"	.001	.001	.0002	60
1100	174	174	175	180	53.0	8000	"	.0018	.0009	.0001	60
1105	175	175	178	186	50.0	10000	"	.0014	.0009	.0006	60
1110	175	185	181	185	53.5	10500	"	.0014	.0008	.001	58
1323	150	158	155	162	35.5 / 1.5	2000	"	.0015	.001	.0008	32
1329	152	162	155	168	36.5	4000	"	.0015	.001	.0008	30
1335	154	165	160	168	37.8	6000	"	.0013	.001	.0008	28
1340	155	165	162	171	36.2	8000		.0020	.001	.0008	30
1345	155	168	165	173	37.0	10000	"	.0015	.001	.0015	28
1350	156	168	168	175	37.3	10500	"	.0015	.001	.0015	28
1405	160	165	155	160	17.7 / .75	2000	"	.0010	.0007	.0005	11
1410	159	168	165	173	18.0	4000	"	.0009	.0009	.0003	8
1415	158	167	164	172	16.8	6000	"	.001	.0007	.0003	6
1420	155	163	161	168	16.6	8000	"	.0018	.0009	.0002	6
1425	153	160	160	166	16.1	10000	"	.0013	.0013	.0003	7
1430	151	158	158	166	17.0	10500	"	.001	.001	.0003	10

NO SYNCHRONIZER OR PAWLS

POSITIVE ENGAGEMENT CLUTCH

TEST NO C-461

TEST ENG. SPURR					OPERATOR BEDFORD			ROOM TEMP		DATE 1-26-73	SHT. 2
TIME	OIL TEMP		BRG. TEMP		OIL FLOW	INPUT RPM	OUTPUT RPM	VIBRATION			OIL PRESS
	IN	OUT	INPUT	OUTPUT				VERT	HOR	FORE & AFT	
1230	155	150	—	—	.75	0/10550	10,500				13.0
1355	158	155	—	—	2.25	0/8750	10,500				81.0
	ACCELERATION RUNS - INPUT RELEASED & ALLOWED TO ACCELERATE IN RESPONSE TO INTERNAL DRAG. TIME HISTORY OF INPUT RPM WAS RECORDED.										
	MEASUREMENT OF DRAG REQ'D TO HOLD INPUT STATIONARY										
	160°F	13.5 IN LBS			1.5	0	10 500				35
	160	14.0 " "			2.25	0	10 500				62
	155	8.5 " "			.75	0	10 500				12

COMPLETE CLUTCH POSITIVE ENGAGEMENT CLUTCH TEST № C-461

TEST DIS. SPURR				OPERATOR HASHAN			ROOM TEMP 71°F	DATE 1-31-73	SHT. 3	
TIME	OIL TEMP		BRG. TEMP		OIL FLOW	INPUT RPM	OUTPUT RPM	VIBRATION			OIL PRESS
	IN	OUT	INPUT	OUTPUT				VERT	HOR	FORE & AFT	
1252											
1302	202	210	200	215	2.5	0	10480	.0013	.0018	.0001	53
1308	192	203	200	187	2.5	2016	10 704	.0013	.0014	.0001	53
1313	197	203	198	192	2.5	4004	10 672	.0013	.0015	.0001	51
1318	193	195	198	107	2.5	6004	10 506	.0013	.0015	.0001	57
1323	205	212	210	215	2.5	8016	10 524	.002	.0018	.001	58
1328	203	215	212	212	2.5	10 008	10 032	.0014	.0021	.001	58
1330	BLEW CIRCUIT BREAKER ON #1 VARI[illegible]:										
	DISASSEMBLY REVEALED SPRING TABS BROKEN ON PAWLS										

COMPLETE CLUTCH POSITIVE ENGAGEMENT CLUTCH TEST № C-461

TEST ENG. SPURR			OPERATOR BEDFORD					ROOM TEMP 70°F		DATE 2-2-73	SHT. 4
TIME	OIL TEMP		BRG. TEMP		OIL FLOW	INPUT RPM	OUTPUT RPM	VIBRATION			OIL PRESS
	IN	OUT	INPUT	OUTPUT				VERT	HOR	FORE & AFT	
1020	195	195	145	155	2.25	0	10 400	.0012	.0012	.0001	48.0
1027	208	210	192	200	2.25	2000	10 520	.0012	.0012	.0001	46.0
1033	204	209	205	205	2.25	4000	10 500	.0012	.0012	.0001	42.0
1037	203	208	203	207	2.25	6000	10 500	.0012	.001	.0001	42.0
1039		SHUT	DOWN								
1103	198	204	187	193	2.25	2000	10 500	.0011	.0009	.0008	44
1108	197	204	188	193	2.25	2000	10 500	.0011	.0009	.0008	45
		SHUT	DOWN								
1223											
1228	202	204			2.25	2000	10500				45
1233	202	211	205	206	2.25	2000	10500	.0011	.0009	.0009	45
1238	202	210			2.00	2000	10500	.0011	.0009	.0009	32
1243	199	209	204	204	2.00	2000	10500	.0011	.0009	.0007	32
1248	197	207	199	199	1.50	2000	10 500	.0011	.0009	.0009	22
1253	195	205	198	200	1.50	2000	10 500	.0011	.0009	.0009	22
1258	193	203	193	193	1.20	2000	10 500	.0011	.0007	.0007	13.5
1303	191	199	192	192	1.20	2000	10 500	.0011	.0009	.0009	13.5
1308	192	198	192	192	1.20	2000	10 500	.0011	.0009	.0009	14.0
1313	190	195	192	193	.75	2000	10 500	.0011	.0009	.0009	7.0
1318	200	203	194	199	.75	2000	10 500	.0011	.0009	.0009	7.0
		SHUT	DOWN								
			NO ERRATIC NOISE OBSERVED								

[illegible] MEASUREMENTS — POSITIVE ENGAGEMENT CLUTCH — COMPLETE CLUTCH — TEST No. C-461

TEST DWG. SPURR — OPERATOR BEDFORD — ROOM TEMP 72 °F — DATE 2-5-73 — SHT. 5

TIME	OIL TEMP IN	OIL TEMP OUT	BRG. TEMP INPUT	BRG. TEMP OUTPUT	OIL FLOW	INPUT RPM	OUTPUT RPM	VIBRATION VERT	VIBRATION HOR	VIBRATION FORE & AFT	OIL PRESS
		TORQUE TO HOLD INPUT									
0930	190 °F	3	IN LB		2.25	0	2000				
		6	" "		"	0	4000				
		8	" "		"	0	6000				
		18	" "		"	0	10300				
0937	STOP										
0952	190 °F	2.5	IN LB		1.75	0	2000				
		5.5	" "		"	0	4000				
		9.0	" "		"	0	6000				
		11.5	" "		"	0	8000				
		12.5	" "		"	0	10500				
0957	STOP	-	OIL SHUT DOWN,		CLUTCH WOULD NOT OVERRUN.			FOUND	5 PAWL	SPRINGS	BROKEN

POSITIVE ENGAGEMENT CLUTCH

TEST № C-461

TEST ENG. SPURR			OPERATOR BEDFORD				ROOM TEMP		DATE 2-20	SHT. 6
TIME	OIL TEMP		BRG. TEMP		OIL FLOW	INPUT RPM	OUTPUT RPM	VIBRATION			OIL PRESS
	IN	OUT	~~INPUT~~	~~OUTPUT~~				VERT	HOR	FORE & AFT	
			CLUTCH WITH NEW SPRINGS (SK 23-26) & PAWLS (SK 23-25)								
			INPUT DRAG								
1057			6.5	IN LB	1.5	0	1464				
			6.5	" "	1.5	0	2000				
			8.0	" "	1.5	0	4000				
			14.0	" "	1.5	0	6000				
			18.5	" "	1.5	0	8000				
			15.5	" "	1.5	0	10,500				
1100											
	TEAR DOWN REVEALED 5 BROKEN SPRINGS										

SINGLE PAWL SETUP POSITIVE ENGAGEMENT CLUTCH TEST No C-461

TEST ENG. SPURR | OPERATOR BEDFORD | ROOM TEMP | DATE 5-2-73 | SHT. 7

TIME	OIL TEMP IN	OIL TEMP OUT	BRG. TEMP INPUT	BRG. TEMP OUTPUT	OIL FLOW	INPUT RPM	OUTPUT RPM	VIBRATION VERT	VIBRATION HOR	VIBRATION FORE & AFT	OIL PRESS
1004		DETERMINE PAWL PLANING.				VS OIL	FLOW	(PLANING JUST BEGINS)			
					.225	0	1833				
					.420	0	1750				
					.900	0	1500				
						0	1250	NO PLANING			
1010	START ENDURANCE				.4	0	1250				
1045	STOP TEST - SHEARED STOP PIN										

FULL CLUTCH.

POSITIVE ENGAGEMENT CLUTCH

TEST № C-461

TEST ENG. SPURR | OPERATOR WASHBURN | ROOM TEMP | DATE 5-30-73 | SHT. 8

TIME	OIL TEMP IN	OIL TEMP OUT	BRG. TEMP INPUT	BRG. TEMP OUTPUT	OIL FLOW	INPUT RPM	OUTPUT RPM	VIBRATION VERT	VIBRATION HOR	VIBRATION FORE & AFT	OIL PRESS
			ATTEMPT 2HR RUN - FULL OVERRUN								
12:15					1.5	0	2200	ONSET OF PAWL RATCHETING			20
12:27					1.5	0	10700				20
12:40	88°F	115°F	95°F	130°F	1.0	0	10700				13
13:00	110	143	135	150	1.0	0	10700				13
13:15	135	160	155	164	"	"	"				12
13:30	150	175	170	177	"	"	"				11
13:45	160	188	182	188	"	"	"				10
14:00	165	190	190	190	"	"	"				10
14:15	173	196	196	196	"	"	"				10
		RUN COMPLETE - O.K. NORMAL SHUTDOWN									

FULL CLUTCH

POSITIVE ENGAGEMENT CLUTCH

TEST № C-461

TEST DWG. SPURR					OPERATOR WASHBURN			ROOM TEMP			DATE 6.1.72	SHT. 9
TIME	OIL TEMP		BRG. TEMP		OIL FLOW	INPUT RPM	OUTPUT RPM	VIBRATION			OIL PRESS	
	IN	OUT	INPUT	OUTPUT				VERT	HOR	FORE & AFT		
			ATTEMPT 2 HR RUN - HALF SPEED OVERRUN									
8:45	95°F	85°F	95°F	110°F	1.0	6000	10 500				13	
9:00	90°F	113	112	130	"	"	"				"	
9:15	105	125	125	140	"	"	"				"	
9:30	115	130	130	145	"	"	"				"	
9:45	125	138	138	152	"	"	"				"	
10:00	130	143	144	155	"	"	"				"	
10:15	133	145	148	157	"	"	"				"	
10:30	136	147	150	130	"	"	"				"	
10:45	139	150	150	162	"	"	"				"	
		NORMAL SHUT DOWN - ALL O.K.										
CHECK ONSET OF PLANING												
	107°F				1.0	0	2324				13	
	"				.9	0	2848				9	
	"				.5	0	3500				5.5	
	"				.5	0	3000				5.5	
	"				.42	0	4000				5.0	
	190°F				.41	0	3000				4.5	
	"				.3		3500				3.5	