Two-Speed Rotorcraft Research Transmission Power-Loss Associated with the Lubrication and Hydraulic Rotating Feed-Through Design Feature

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Topics

• Background – why are we doing this?

• Modular inline concentric two-speed research transmission configuration

• Rotating Feed-Through (RFT) design feature
  • RFT System (Shaft and RFT) in the two-speed transmission
  • Isolated RFT power loss experiment and results
  • Conclusions and future RFT development
Background

• Advances in rotorcraft propulsion systems require increased efficiency, power, and enhanced capabilities

• Studies show that variable/multi-speed rotors are required for:
  • Enhanced capabilities: increased speed, payload, and range
  • Reduction in noise

*Advances require varying rotor speed up to 50%.*

*Present Limitations* ~15% via engine output shaft speed control.
Future Rotorcraft Propulsion System Configuration, Variable/Multi-Speed Gearbox Application

To Mid-Wing Gearbox

V/M-S Gearboxes

Gas Turbine Engines

Combiner Gearbox

Tilt-Axis Gearbox

Reduction Gearbox

Hover Ratio 131.4 : 1 Cruise Flight Ratio 243.6 : 1
Two-Speed Research Transmission Design Requirements

- 250 HP nominal (200 HP facility capacity)
- Inline concentric configuration
- Input Speed 15,000 rpm
- Output Speeds 15,000 rpm (hover), 7,500 rpm (cruise)
- Lubricant: DOD-PRF-85734A, synthetic ester-based oil
- Drive should fail safe to the high-speed (hover) mode
- Employ straight spur gear geometry
- a Provide high-speed positive drive locking-element
- a Light-weight rotating components (flight like)
- b Housing design (modular, possibility of windage shrouds)

\[a \text{ requirement dropped}\]
\[b \text{ not an original requirement}\]
Research Transmission Modules: Gear & Clutch

1:1 Direct Drive (Hover Mode)
- Control Clutch Engaged

2:1 Reduction Drive (Cruise Mode)
- Control Clutch Disengaged
Gear Module 1: Offset-Compound Gear (OCG)

Input Gear

Ring Gear

Input Gear

Ring Gear

Ratio 2:1

Ratio 1:1

OCG Cluster Offset Axis

Cluster Gear (OCG)
Gear Module 2: Dual Star-Idler Planetary (DSI)

Star Gear
Reversing Idler Gear
Sun Gear
Ring Gear
Carrier (Fixed)

Ratio 2:1
Ratio 1:1
Clutch Module: Dry-Clutch (DC)

- Dry-Clutch
- Drive Diaphragm Spring
- Intermediate Shaft
- Sprag Clutch
  - 16 sprag elements
  - 4-lube inlets/drains
- Low-Speed Shaft
- Ratio=2:1
- Ratio=1:1
- Clutch Hub
  - * Release Bearing Ass’y
  - * Rotating Feed-Through
- Output Shaft (DC)

* Unique hardware necessary to meet the inline design requirement
Clutch Module: Wet-Clutch (WC)

- Wet-Clutch
- Low-Speed Shaft
- Drive Helical Springs
- Sprag Clutch
- Drive Plates
- * Annular Release Piston
- * Rotating Feed-Through
- * Output Shaft (WC)

Ratio=2:1

Ratio=1:1

* Unique hardware necessary to meet the inline design requirement
Rotating Feed-Through (RFT) Design Feature

Output Shaft and RFT in the Two-Speed Transmission
Power Loss Experimental Setup
Power Loss Experimental Results
Output Shaft* - Hydraulic & Lubrication Passages
(Wet-Clutch Shown)

Passage C
Bearing Oil Jets
Drive Plates & Oil Jets

Passage B
Release Piston

Passage A
Sprag Clutch At Inner Race

Drains
Orifice (Bleed)
Low-Speed Shaft

A B C
RFT Inlets: Lube / Clutch / Lube

* Unique hardware necessary to meet the inline design requirement
Hydraulic/Lubricant Rotating Feed-Through (RFT*)

- Ring Seals
- 15,000 rpm Max

* Unique hardware necessary to meet the inline design requirement
**RFT Example Single Passage Pressures, Speeds, Velocities**

- **O-ring, Viton (Static)**
- **Ring Seal, Polyimide (Dynamic)**
- **Clearance**

**Pressures & Reactions**

- $P_1 \rightleftharpoons P_1$
- $P_0 \rightleftharpoons P_0$
- $P_C \rightleftharpoons P_1$
- $P_C \rightleftharpoons P_1$

**Axis of Rotation**

- **P1 – RFT Passage Pressure**
- **P0 – Atmospheric Pressure - or - Adjacent Passage Pressure**

**Stator**

- 0 rpm

**Rotor**

- 15,000 rpm Maximum Shaft Speed
- 1,571 in/s (40 m/s) Ring Seal Side Surface Velocity

(Ref: 2 inch nominal diameter)

**Output Shaft Not Shown**
RFT Seal Pressure and Speed Operating Points

Field Survey of Automatic Transmission Ring Seal P-V Conditions

P = Contact Pressure MPa

V = Tangential Sliding Speed (m/s)

RTF Ring Seal Conditions at Cruise and Hover:
- Dry-Clutch control (initial* & modified** design)
- Wet-Clutch control (0-200 psi)

Ref. 9: Graphic Basis (0 - 20 m/s); RFT operation (20 - 40 m/s)
RFT Isolated Power Loss Experiments

- Blank Flange
- Output Shaft
- Duplex Bearing
- Oil In, Passages A, B, & C
- Passage A Sprag Orifice
- Passages B & C Orifice or Plug (To Simulate Each Clutch)
- TorqDisc* & Drive Motor

Notes:
* - Torque Is Measured
- Power Is Calculated
- Bearing Tare is Measured
- Test Speed < 8,000 rpm
RFT Experiment Ring Seal Torque Drag Vs. Speed

Dry Clutch Configuration
Clutch disengaged at 2.59 MPa (375 psi)
Clutch engaged at 0 MPa (0 psi)

Experimental data linear trend line correlation coefficients, $r^2$

<table>
<thead>
<tr>
<th>Clutch Configuration</th>
<th>Engaged $r^2$</th>
<th>Disengaged $r^2$</th>
</tr>
</thead>
<tbody>
<tr>
<td>Dry Clutch</td>
<td>0.0008</td>
<td>0.2</td>
</tr>
<tr>
<td>Wet Clutch</td>
<td>0.9</td>
<td>0.2</td>
</tr>
</tbody>
</table>

Note: Torques shown above is for the RFT ring seal drag less the duplex bearing torque measured separately.

Note: Speed range limited due to rotor dynamic response of experiment setup.
RFT Power Loss Trend Line Equations

Dry-Clutch disengaged (cruise)
| 80 psi (sprag) | 375 psi (clutch) | 375 psi (clutch) |
Torque (in-lb) = 1.4E-4 × Ω + 20
Power (hp) = 2.4E-9 × Ω² + 3.2E-4 × Ω
Torque (N-m) = 1.6E-5 × Ω + 2.3
Power (Watts) = 1.7E-6 × Ω² + 2.4E-1 × Ω

Dry-Clutch engaged (hover)
| 80 psi (sprag) | 0 psi (clutch) | 0 psi (clutch) |
Torque (in-lb) = 1.2E-5 × Ω + 6.5
Power (hp) = 1.9E-10 × Ω² + 1.0E-4 × Ω
Torque (N-m) = 1.4E-6 × Ω + 0.73
Power (Watts) = 1.4E-7 × Ω² + 7.7E-2 × Ω

Wet-Clutch disengaged (cruise)
| 80 psi (sprag) | 200 psi (clutch) | 80 psi (bearing lube) |
Torque (in-lb) = 2.4E-04 × Ω + 14
Power (hp) = 3.9E-09 × Ω² + 2.2E-04 × Ω
Torque (N-m) = 2.8E-05 × Ω + 1.5
Power (Watts) = 2.9E-06 × Ω² + 0.16 × Ω

Wet-Clutch engaged (hover)
| 80 psi (sprag) | 0 psi (clutch) | 80 psi (bearing lube) |
Torque (in-lb) = 4.7E-04 × Ω + 4.1
Power (hp) = 7.4E-09 × Ω² + 6.5E-05 × Ω
Torque (N-m) = 5.3E-05 × Ω + 0.46
Power (Watts) = 5.5E-06 × Ω² + 4.9E-02 × Ω

Where Ω is shaft speed in rpm

Note: RFT torque and power loss shown above is less duplex bearing torque and power loss.
Generalized Ring Seal Power Loss Equations

Torque (in-lb) = \( (5.8 \times 10^{-7} \times \Omega + 2.8 \times 10^{-2}) \times \Delta P \)
Power (hp) = \( (9.2 \times 10^{-12} \times \Omega^2 + 4.5 \times 10^{-7} \times \Omega) \times \Delta P \)  
where: \( \Omega \) is rpm and \( \Delta P \) is psi

Torque (N-m) = \( (6.6 \times 10^{-8} \times \Omega + 3.2 \times 10^{-3}) \times \Delta P \)
Power (W) = \( (6.9 \times 10^{-9} \times \Omega^2 + 3.4 \times 10^{-4} \times \Omega) \times \Delta P \)  
where: \( \Omega \) is rpm and \( \Delta P \) is MPa

Comparison of Power Loss from Experimental Data Trend Line Equations with Power Loss Estimates from the Generalized Ring Seal Power Loss Equation

<table>
<thead>
<tr>
<th>Clutch Drive Ratio</th>
<th>RFT Passage Pressure</th>
<th>Summed Seal ΔP Pressure Differentials (psi)</th>
<th>% Error 1,000 rpm</th>
<th>% Error 7,500 rpm</th>
</tr>
</thead>
<tbody>
<tr>
<td>Dry Clutch 1:1</td>
<td>A (psi) 80</td>
<td>B (psi) 0</td>
<td>C (psi) 0</td>
<td>160</td>
</tr>
<tr>
<td>Dry Clutch 2:1</td>
<td>A (psi) 80</td>
<td>B (psi) 375</td>
<td>C (psi) 375</td>
<td>750</td>
</tr>
<tr>
<td>Wet Clutch 1:1</td>
<td>A (psi) 80</td>
<td>B (psi) 0</td>
<td>C (psi) 80</td>
<td>320</td>
</tr>
<tr>
<td>Wet Clutch 2:1</td>
<td>A (psi) 80</td>
<td>B (psi) 200</td>
<td>C (psi) 80</td>
<td>400</td>
</tr>
</tbody>
</table>
Estimating RFT Power Loss

Comparison of Experimental RFT Power Loss Data Linear Trend Lines versus the Generalized Ring Seal Power Loss Equation

Dry Clutch Configuration

| 80 psi (sprag) | Clutch psi | Clutch psi |

Wet Clutch Configuration

| 80 psi (sprag) | Clutch psi | 80 psi |

![Graphs showing comparison of power loss data for dry and wet clutch configurations.](image-url)
RFT Conclusions & Future Considerations

Conclusions

• The RFT power loss at ~80 psid is low and is a reasonable option to provide lubrication internal to a rotating system provided that seals are not required to be leak free.

• The RFT power loss does not scale with system power, but does increase when designs require larger shaft diameters, higher speeds, or higher pressure.

• The RFT and total transmission power loss can be minimized by designing any components supplied through the RFT with the lowest required pressures necessary for proper function.

• The polyimide ring seals performed well for the experimental time accumulated.

• All experimental data and results are valid only for polyimide ring seal materials.

Future Considerations

• Test all ring seal materials under consideration as friction coefficients vary considerably.

• The RFT design used standard ring seals and installation geometry. Future design should consider thermal expansion with respect to operating temperatures.

• The RFT rotor geometry should be optimized:
  Outside diameter: Increase seal sliding contact area.
  Groove width: Increased width ensures pressure is applied radially outward.
Questions?