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
  • \textit{a} Provide high-speed positive drive locking-element
  • \textit{a} Light-weight rotating components (flight like)
  • \textit{b} Housing design (modular, possibility of windage shrouds)

\textit{a} requirement dropped
\textit{b} 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

Forward Bearing

Speed Reduction Gear Train

Low-Speed Shaft

Aft Bearing

Rotating Feed-Through (RFT)

Input Shaft

Control Clutch (Dry or Wet)

Sprag Clutch

Output Shaft

Gear Modules

Clutch Modules

15,000 rpm

7,500 rpm

15,000 rpm
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
- 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**
- **Clutch Hub**
  - **Release Bearing Ass’y**
  - **Rotating Feed-Through**
- **Output Shaft (DC)**

* Ratio=2:1
* Ratio=1:1

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

- Low-Speed Shaft
- Drive Helical Springs
- Sprag Clutch
- Ratio=2:1
- Ratio=1:1
- Drive Plates
- * Annular Release Piston
- * Rotating Feed-Through
- * Output Shaft (WC)

* 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

RFT Inlets: Lube / Clutch / Lube

Drains (Bleed)

* 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

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

Axis of Rotation

Pressures & Reactions

PC = P1
PC > P1

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

\[ P = \text{Contact Pressure MPa} \]

\[ V = \text{Tangential Sliding Speed (m/s)} \]

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)

Cruise 7,500 rpm (20 m/s)
Hover 15,000 rpm (40 m/s)

Experimental P-V limits by lab test
Actual PV condition of Commercial VESPEL® SP seal rings from survey
Not Recommended

DuPont™ VESPEL® SP
PEEK
PTFE
OK
RFT Isolated Power Loss Experiments

- 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)

**Wet Clutch Configuration**
- Clutch disengaged at 1.38 MPa (200 psi)
- Clutch engaged at 0 MPa (0 psi)

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

<table>
<thead>
<tr>
<th></th>
<th>Dry Clutch</th>
<th>Wet Clutch</th>
</tr>
</thead>
<tbody>
<tr>
<td>Engaged</td>
<td>r^2 = 0.0008;</td>
<td>r^2 = 0.9;</td>
</tr>
<tr>
<td>Disengaged</td>
<td>r^2 = 0.2</td>
<td>r^2 = 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) = (5.8E-07 × Ω + 2.8E-02) × ΔP
Power (hp) = (9.2E-12 × Ω² + 4.5E-07 × Ω) × ΔP

where: Ω is rpm and ΔP is psi

Torque (N-m) = (6.6E-08 × Ω + 3.2E-03) × ΔP
Power (W) = (6.9E-09 × Ω² + 3.4E-04 × Ω) × ΔP

where: Ω is rpm and ΔP 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</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>A (psi)</td>
<td>B (psi)</td>
<td>C (psi)</td>
</tr>
<tr>
<td>Dry Clutch 1:1</td>
<td>80</td>
<td>0</td>
<td>0</td>
</tr>
<tr>
<td>Dry Clutch 2:1</td>
<td>80</td>
<td>375</td>
<td>375</td>
</tr>
<tr>
<td>Wet Clutch 1:1</td>
<td>80</td>
<td>0</td>
<td>80</td>
</tr>
<tr>
<td>Wet Clutch 2:1</td>
<td>80</td>
<td>200</td>
<td>80</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 |

![Graph showing comparison of dry clutch configurations with trend lines for different pressure settings.](chart1.png)

**Wet Clutch Configuration**

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

![Graph showing comparison of wet clutch configurations with trend lines for different pressure settings.](chart2.png)
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?