

Concepts for Multi-Speed Rotorcraft Drive System - Status of Design and Testing at NASA GRC

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ABSTRACT

In several studies and on-going developments for advanced rotorcraft, the need for variable/multi-speed capable rotors has been raised. Speed changes of up to 50% have been proposed for future rotorcraft to improve vehicle performance. A rotor speed change during operation not only requires a rotor that can perform effectively over the operating speed/load range, but also requires a propulsion system possessing these same capabilities. A study was completed investigating possible drive system arrangements that can accommodate up to a 50% speed change. Key drivers were identified from which simplicity and weight were judged as central. This paper presents the current status of two gear train concepts coupled with the first of two clutch types developed and tested thus far with focus on design lessons learned and areas requiring development. Also, a third concept is presented, a dual input planetary differential as leveraged from a simple planetary with fixed carrier.

INTRODUCTION

Rotorcraft propulsion is a critical element of the overall rotorcraft. Unlike fixed wing aircraft, the rotor/propulsion system provides lift and control as well as forward thrust. As a result, the rotorcraft engine/gearbox system must be highly reliable and efficient. Future rotorcraft trends call for more versatile, efficient, and powerful aircraft, all of which challenge state-of-the-art propulsion system technologies. Variable speed rotors have been identified as having a large impact on many critical rotorcraft issues.

Currently, rotor speed can be varied only a small percentage by adjusting the speed of the engine. This is generally limited by engine efficiency and stall margin permitting speed changes limited to approximately 15% of the maximum operating speed (used in current tilt-rotor applications).

The NASA Heavy Lift Study (Ref. 1) had shown that variable speed propulsion is necessary for all aircraft concepts studied. Variable speed propulsion, without loss of

efficiency and torque, is necessary to permit efficient high speed operation with reduced noise. The heavy lift study suggests that increased speed variations of 50% will have a dramatic effect on reducing external noise while increasing rotorcraft performance. Previous NASA variable speed transmission studies concentrated on 15% speed changes (Refs. 2 and 3). To achieve the large speed variation capability for the heavy lift application, advanced variable/multi-speed drive system concepts must be developed.

This paper is a follow on to an earlier concept study that identified viable concepts for both two-speed and variable speed drive transmission configurations (Ref. 4). From this earlier study, the top three concepts were selected for development and testing (Ref 5).

This paper provides 1) an overview of the design for two of the top three concepts selected from the earlier conceptual study summary, as ranked from a comprehensive study of possible concepts that provide up to a 50% speed reduction range, 2) an overview of the follow on design and initial test status of a modular-multiple-configuration scale model test articles currently under development and testing at the NASA Glenn Research Center, 3) a summary of design lessons learned from initial testing, and 4), an overview of a paper-design for the third selected concept (Ref. 4), for possible future development.

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For the purpose of designing scale developmental test hardware to evaluate selected concepts, a 50% speed range was identified as a notional design limit. Speed changes less than the above were thought to be more easily realizable. A relevant maximum input speed of 15,000 rpm, with output speeds of 15,000 rpm (high-range hover mode) and 7,500 rpm (low-range cruise mode) was selected for design.

The test article features a modular design offering the ability of testing two interchangeable simple gear train designs, as well as two clutch designs. The gear trains and clutches share common interfaces, allowing assembly of combinations of either of the two gear trains with either of the two clutch configurations, resulting in the ability to test four different configurations. The modular approach offers a highly flexible baseline test article, with the added flexibility of focused hardware refinement, while allowing for the retention of hardware not requiring refinement. The result is a highly exploitable, yet economically alterable, baseline test article to identify technology barriers, and evaluate design refinements at the concept demonstration level.

The first gear train configuration is the DSI (Dual Star-Idler), a sun-gear input, ring-gear output, simple single-stage planetary with reversing idler planet gears. The second gear train, the OCG (Offset-Compound Gear), is a unique configuration employing an offset cluster gear comprised of both an internal and external gears. The OCG has only two gear meshes. The impetus behind the selection of the above two gear trains is simplicity and minimal parts count.

The first clutch option is a single-plate dry clutch which is an automotive small diameter high performance clutch. The second clutch is a new design multi-plate wet-clutch. Both the wet and dry clutches employ a mechanical force driving element for the 1:1 ratio hover mode, and a hydraulic clutch release and sprag drive for the 2:1 ratio cruise mode.

The hardware described in this paper is intended for use as development and test hardware and is not intended to replicate flight hardware. Consideration of light weight design yielded to that of a flexible and easily accessible, easily modifiable design which can be exploited in the test cell environment, although some limited aspects of the design are light-weight, such as the gear trains.

This design overview includes a review of design requirements, an examination of detail cross sections of the four modular test configurations, insight into the design evolution and philosophy, as well as pointing out limitations and technical challenges of each.

The above configurations have been manufactured and are in the process of being assembled into four test article hardware configurations for testing at NASA GRC to validate operational capability of the concepts.

TEST ARTICLE DESCRIPTION

Application

The subject fractional scale drive is a development model for an inline drive configuration intended to be an add-on unit, or directly integrated into the overall design of a propulsion drive system. Figure 1 shows a representative tilt rotor drive system schematic and the subject speed changer located between the engines and main combiner gearbox where power transfer is at high-speed and low-torque.

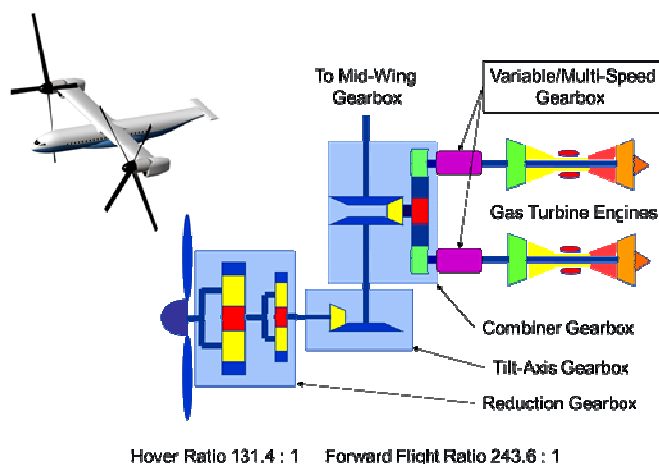


Figure 1. Propulsion Drive Schematic.

Concept Identification and Selection

An earlier study was completed that identified a large number of possible configurations (Ref. 4) of both multi and variable speed drives. The study included a broad number of possible configurations found in the open literature and also a number of original concepts. A set of criterion was established to permit down select from the above concepts identifying the top three potential candidates for concept development and testing (Ref. 5). The primary metric for selection was simplicity, seen as translating to light weight and robustness. Three concepts were selected meeting the above. Based on available funding, two of the concepts were designed and fabricated for testing. A third remains undeveloped, although it is now possible to advance this concept as a portion of the design basis is part of one of the above concepts. An overview of an updated design basis is given at the end of the paper. This paper focuses on the design and manufacturing of these test articles, on areas identified for future development, and lessons learned.

Design Requirements

Test article design requirements are summarized in Table 1 and a representative aircraft mission cycle is defined in Table 2.

Table 1. Test Article Design Requirements.

250 HP nominal (200 HP facility capacity)
Input Speed 15,000 rpm
Output Speeds 15,000 rpm (hover), 7,500 rpm (cruise)
Employ straight spur gear geometry (budget consideration)
Drive should fail safe to high-speed (hover) mode
^a Provide high-speed positive drive element
Lubricant: DOD-PRF-85734A, synthetic ester-based oil
40C 104F 23.0 cSt
100C 212F 4.90-5.40 cSt
-54C -65F pour point
Inline configuration (input-output shafts)
^b Light-weight rotating components (flight like)
^c Housing design (modular, possibility of windage shrouds)

^a requirement dropped due to complexity and budget
^b requirement dropped due to scope and budget
^c not an original requirement

Table 2. Duty Cycle - High Speed & Low Speed Ranges.

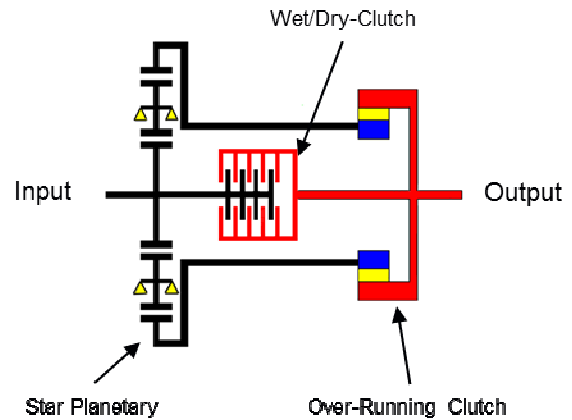
Operation	time (minutes)	notes
Taxi	4	
Ground check	1	(60% power)
Climb to cruise	25-30	
Convert (shift)	2	(TBD)
Cruise	180	
Transfer altitude	20	
Final approach	5	
Vertical landing	1	
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High Speed	60	→ ~25% (TBD)
Low Speed	180	→ ~75% (TBD)

(Adapted from Ref. 6.)

Concept Schematic

An initial impetus for a simple light weight configuration was based on the notional schematic shown in Figure 2. In this schematic, a couple of observations can be made for comparison to the designs presented later. These include the simplicity of the gear trains and the relative size of the main clutch and the sprag over-running clutch. The above will be shown as unrealistically simplistic and poor in both relative

scale and proportion when design requirements are applied to design actual hardware.



High Speed: W/D Clutch Engaged, Over Running Clutch Free Wheel
Low Speed: W/D Clutch Disengaged, Over-Running Clutch Driving

Figure 2. Initial Concept Schematic.

Two-Speed Drive Rig Modules & Configurations

Modules. The test article design is comprised of four modules consisting of two interchangeable gear trains and two interchangeable clutch systems in a single housing structure as shown schematically in Figure 3 below.

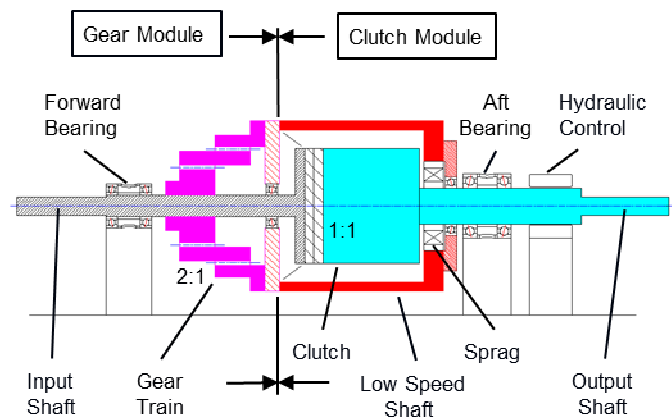


Figure 3. Modules Schematic.

In the following sections, each of the four modules are briefly overviewed below for familiarization of the overall test rig configuration. Later in the paper modules are discussed in detail at the individual component and function level that comprises each individual module, as well comparisons between the complementary modules, (i.e.,

comparisons between gear trains modules and clutch system modules respectively).

Both gear train modules that follow are interchangeable. High-speed 1:1 is at the central shaft aft flange and the low-speed ~2:1 reduction output is at the ring gear output spider hub flange.

(OCG) Offset Compound Gear Drive. The OCG consists of an input gear having external teeth, an output ring gear having internal teeth, and a cluster gear that has internal teeth on the input end and external teeth on the output end, mating with the above, and running on a centerline which is offset from the main machine axis. The OCG gear train has two meshes and four basic bearings, two at the cluster gear and two at the input shaft. The OCG is shown in Figure 4.

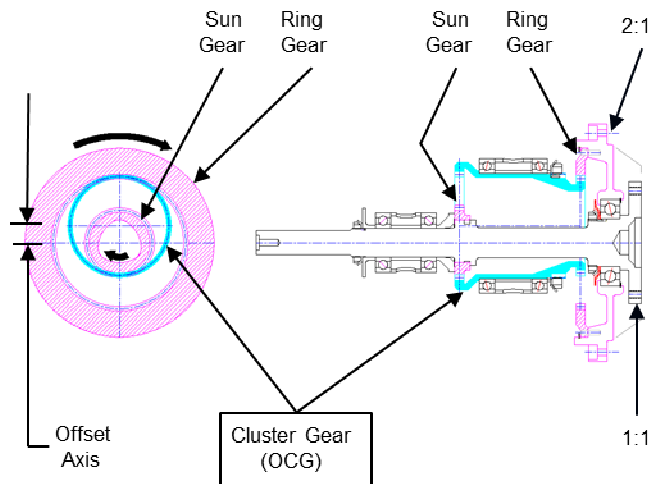


Figure 4. OCG Gear Train Module.

(DSI) Dual Star-Idler Planetary Gear Drive. Like the OCG, the DSI planetary configuration consists of an input gear with external teeth, a ring gear with internal teeth, and the OCG cluster gear is replaced by a set of three star gear pairs, where the first meshing gear is a planet gear and the second meshing planet gear serves as an idler, reversing the direction of rotation of the ring gear, such that both the input gear and ring gears rotate in the same direction as shown in Figure 5. Note that the need for reverse rotation idler gears was a major oversight of the simple planetary in the baseline schematic (refer to Figure 2) in which the input and output would rotate in opposite directions as shown in that figure.

For both gear trains, the input gears are coupled to their respective input shafts using a three-lobe polygon drive geometry and the ring gear is attached to, and drives the low-speed shaft.

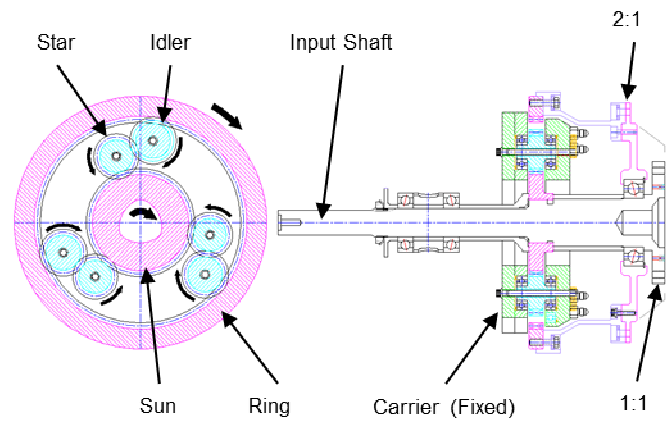


Figure 5. DSI Planetary Gear Train Module.

Dry-Clutch. The dry-clutch is a commercially available small diameter high-performance automotive clutch with 15,000 rpm capability. It was selected as a baseline clutch due to prior experience of having been employed in another unrelated test rig application and thought to be the quickest to integrate into the overall design. A number of design changes implemented to meet the current design requirements are described later. This initial baseline clutch was intended primarily to get the test article and facility operational and enables tuning of the clutch control system. The dry-clutch arrangement is shown below in Figure 6.

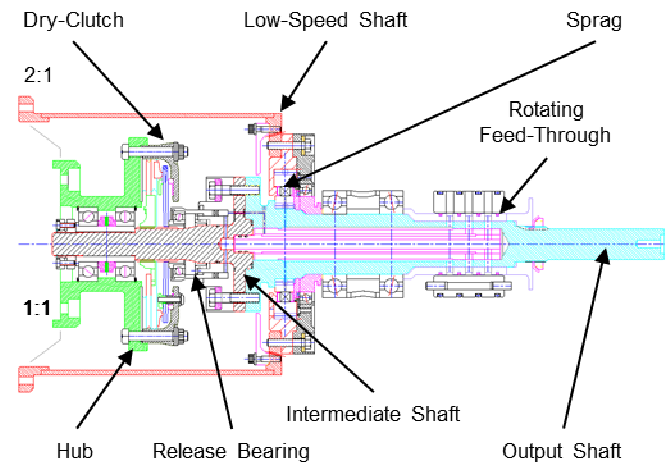


Figure 6. Dry-Clutch Module.

Wet-Clutch. The wet clutch is a NASA design based loosely on friction/steel drive elements of traditional automotive automatic transmission design and uses automotive wet friction/steel drive plates. A significant difference in the design compared to common automotive applications is that instead of hydraulic force application to provide the

clamping and power transfer, and mechanical spring release of the drive plates, the opposite mode of operation is used. That is, mechanical springs provide clamping forces to transfer power through the drive plates and hydraulic pressure is used to release the drive plates. The above meets a basic design requirement to fail safe in the high-speed 1:1 direct drive, clutch engaged, hover mode. This baseline development configuration uses some other commercial automotive transmission parts to reduce costs. The wet-clutch design arrangement is shown in Figure 7.

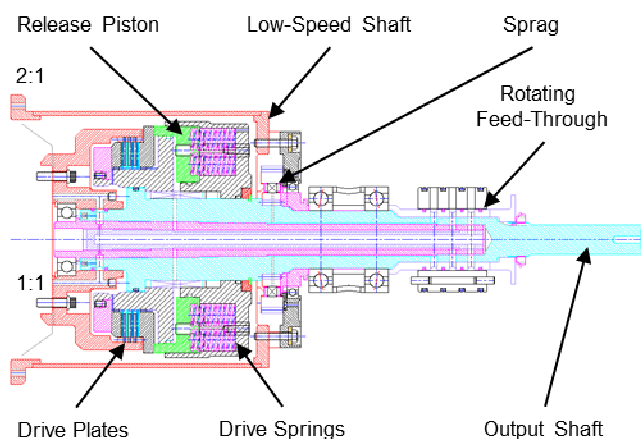


Figure 7. Wet-Clutch Module.

Balance Plan for Modular Design. The multi-configuration modular design approach, as well as any given overall configuration assembly, often requires special balance considerations. Each configuration has some inherent geometry that warrants special considerations as well as those considerations necessary due to the placement of modular interfaces. For each overall assembly there are multiple sub-assemblies that spin at different speeds. Some sub-assemblies spin on different centerlines from the primary machine axis. The modular design approach further complicates the design by introducing parting planes, required for module interchangeability, within some of the obvious balance subassemblies. The parting faces between modules which occur within an obvious balance subassembly required careful consideration of where unbalance corrections are made. This is necessary to permit direct module change out without rebalance of the overall assembly. Taking into consideration the module interfaces and obvious balance subassemblies, a typical overall rotating assembly requires up to six balance subassemblies. Six detail parts or sub-assemblies were required for the example overall rotating assembly are as shown in Figure 8. Thus far the approach to balancing the subassemblies with module interchangeability has been very successful.

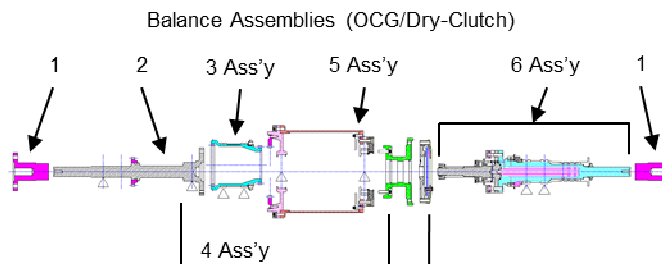


Figure 8. Configuration Balance Assembly Breakdown.

Housing - Base Arrangement: Stations, Module Interfaces, and Bearing Locations. A basic similarity exists between the top two concepts, a gear train coupled to a clutch. It was further observed that significantly more technology and experience could be gained with lower cost if the above similarity was exploited. Two of the top three concepts could be realized, with the third potentially realizable, if a modular design approach was undertaken. It was therefore decided to create a modular test bed with interchangeable gear trains and interchangeable clutches. The above has enabled additional test configurations over that of a single test article initially envisioned.

To accomplish the above, module interfaces were established that included a number of fixed stations for bearings and mating flanges. It was further decided to forego a more flight-like housing configuration and design a single common unit to permit more of the budget to be dedicated to rotating components than to multiple housings. A common housing approach was also deemed more practical for test rig to permit integration into the facility, addition of instrumentation, and maximum accessibility. In addition, the rotating parts are generally applicable to future design whereas the housing is application specific and best left to later development when design requirements for a specific application are in play. The most probable specific future application is a mid-power level combined engine and transmission test article although funding for such hardware development and testing is not presently available.

The basic test article housing consists of a precision ground base plate with multiple bearing supports that are horizontally split on the center of rotation to form a lower saddle and upper cap and other integral features to accommodate the rotating assembly. The base and supports are shown in Figure 9. The base plate has longitudinal and transverse rails both above and below adding to the overall stiffness.

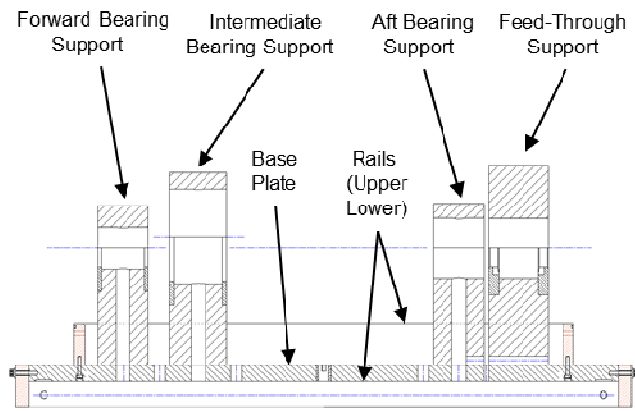


Figure 9. Housing and Supports.

The base is enclosed with simple flat plates. This type of housing/bearing support arrangement is loosely analogous to an axial-split housing where the entire rotating assembly is assembled outside of the housing and installed within the split housing. For the test bed housing, the above configuration is also deemed the most practical to provide ready access to the rotating assembly and also allow for the possibility of adding windage reduction shielding later in testing. The bearing supports, consisting of a base and cap, split on the centerline, were partially machined, then located and pinned on the baseplate, and align bored. The above was machined in a single set-up as an assembly to assure that the centerline of each support was in line with the others, and also assuring axial position of the stations. Within the single set-up, the intermediate bearing support which differs between the gear modules, were switched out and finish machined to maintain centerlines and stations between all of the modules, and assure interchangeability at assembly. All machining was completed during initial manufacture so that configuration changes in the test cell would preclude the need to perform additional machining. The side and top plates are not intended to contribute to the overall housing stiffness but provide for containment of oil and possible parts failure.

Due to the difference in length of the basic OCG and the DSI gear trains, the length of the DSI gear train was increased to match the OCG by increasing the input shaft length and moving the DSI planetary carrier forward. In addition, it was necessary that a different support design be created for each gear train. The input shaft bearing support and the output shaft bearing support, as well as a rotating feed-through, were positioned to common fixed stations. Due to the configuration being a test bed, the overall length of each module was lengthened somewhat to provide some buffer space in the event of a need for a significant revision to any of the modules or components contained within. A generous envelope was provided for the sprag over-running clutch area providing space for future design revision.

Axial position of the overall rotating assembly is controlled at the gear train as shown in Figure 10. Position of the overall rotating assembly is constrained at the input shaft duplex bearing set for both gear trains. For the OCG, the cluster gear position is controlled by the cluster duplex bearing set. For the DSI, the carrier frame is keyed to the support which in turn controls planet gear axial position and also provides anti-rotation reaction point. The output shaft duplex bearings are free to float axially for thermal expansion without increasing or decreasing bearing thrust load (i.e., change axial preloads). The hydraulic rotating feed-through has no bearings and is non-contacting other than ring seals.

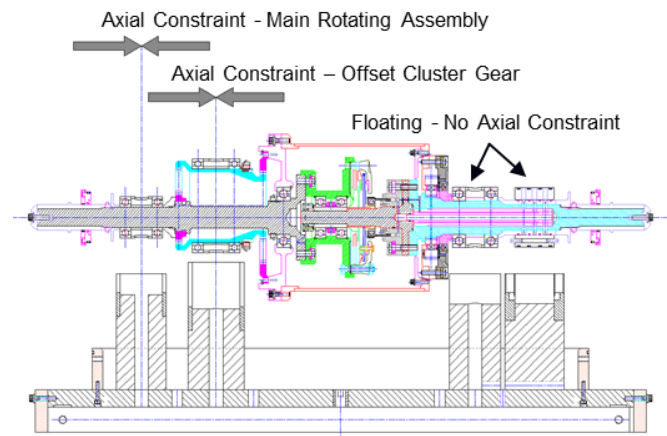


Figure 10. Axial Constraints.

The rotating subsystem is fully assembled and lowered into the bearing support saddles using a special fixture. The dual purpose fixture serves as both a structural lifting sling and a precision lowering/lifting jacking frame to lower the rotating assembly in the bearing and seal support saddles. There are multiple simultaneous alignments required while lowering the rotating assembly into the supports (e.g., DSI carrier and OCG cluster assemblies, rotating feed through anti rotation pin alignment, and a delicate o-ring seal engagement).

The modular base, top, side and end wall configuration allows for easy modification to secure and pass through instrumentation and any unforeseen needs without impacting the primary base and bearing support structure. Most lubrication supply entry points are located integral in the main base. Some entry points for the rotating feed-through and main duplex bearings are located integral in the aft end wall. Openings for drain sumps are integral machined in the base, and the sumps are attached from below. End walls are split at the shaft centerline for removal. Sealing at the end walls is done using commercial non-contacting labyrinth seals supplemented with splash shields and shaft oil slingers.

Drive Configurations and Test Status. Assembly and overall testing has been completed for the first two configurations, Configuration 1: OCG / Dry-Clutch, and Configuration 2: DSI / Dry-Clutch. Parts for the wet-clutch used in Configuration 3 are still being manufactured.

The rotating assemblies for the above three test configurations are shown below for general comparison in Figures 11, 12, and 13 respectively.

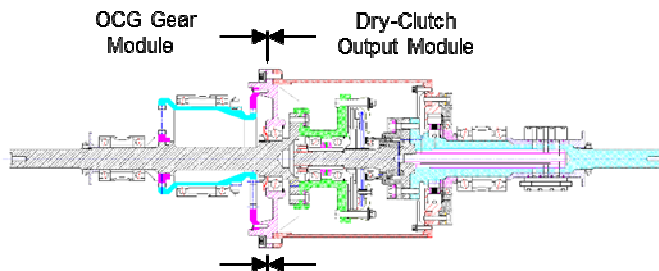


Figure 11. Configuration 1: OCG/Dry-Clutch

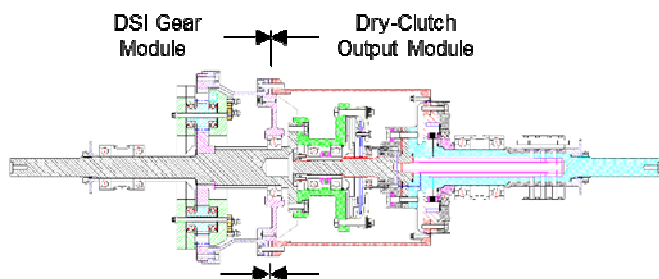


Figure 12. Configuration 2: DSI Planetary/Dry-Clutch

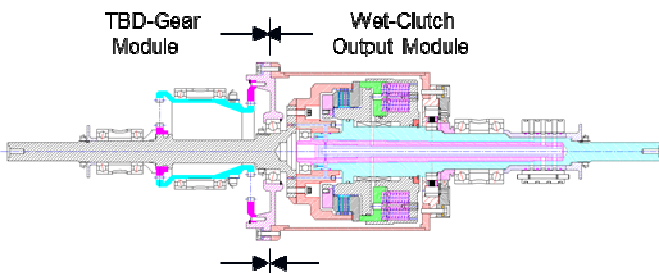


Figure 13. Configuration 3: Wet-Clutch & Gear? TBD.

Components – Detailed Description of Design Features

Dry-Clutch. The dry-clutch utilized in the test article is not foreseen as a candidate clutch configuration for the end application, but as a baseline component used to facilitate test rig operation in the new facility. The above was the quickest path to gain preliminary experience in shifting and varying the period of engagement and disengagement that controls the smoothness of the transition. It was decided to pursue the dry-clutch to get the test article and facility up and running with the available gear modules and follow up

with the wet-clutch design in series as soon as the gear module designs were complete and in manufacturing.

The dry-clutch is a commercial product targeted at the automotive racing market. It has sufficient power and speed capacity for the overall test article. Use of the selected product was also based on previous operational experience having been used in another unrelated test rig. The primary challenge in using the above type of clutch was how to actuate release since the clutch is completely within rotating hardware (i.e., low speed shaft). This is not typical in the automotive application. Details regarding this design restriction are discussed in detail further below.

The friction materials used in the dry-clutch test configuration are carbon-on-carbon. This friction material combination was recommended as the best for an application involving heavy sliding while requiring good wear characteristics. There are other friction materials available but no others were planned to be used with the dry-clutch. The configuration used is a single-disc type with power capability exceeding the rig requirements. The clutch is available in up to a four-disc configuration for significantly higher capacity that could permit significant HP increase. The commercial clutch is mounted to a clutch hub assembly which includes a rolling element bearing set providing support between the input and output shafts. Previously designed at NASA for another application, the design was adapted for this application. Changes to the existing clutch hub design included revision to the forward end to provide a flange to mate with the output flange of the gear train shaft. The above afforded an easy integration of the basic mechanical aspects of the clutch. The basic dry-clutch assembly is shown on below in Figure 14.

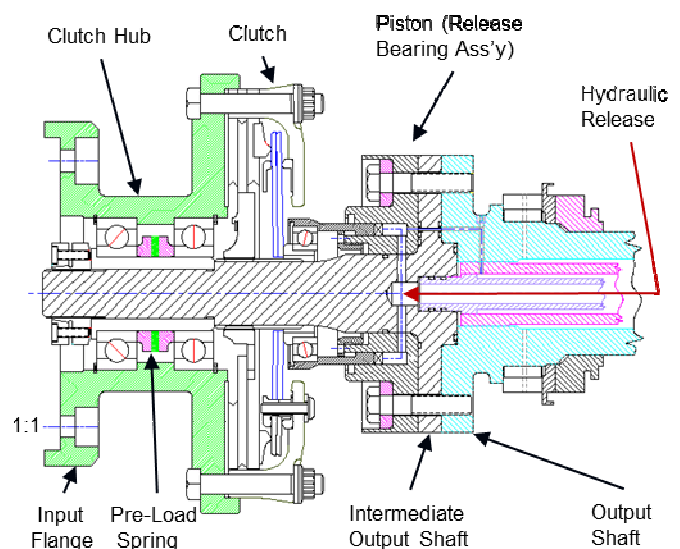


Figure 14. Dry-Clutch.

A major change to incorporate the overall clutch system into the subject test article was the design of the release bearing

system which is typically mounted to static ground. In the subject test article, the release bearing, being enclosed inside of the rotating low-speed hollow shaft, must rotate. This in turn further requires that the hydraulic connections be made through a rotating component connection. Details of the hydraulic release bearing design and related rotating feed-through are discussed later in detail as separate components.

During initial testing some issues arose with the clutch release force/pressure required for this clutch. The release force of the clutch was higher than expected. Higher release pressure caused some system control issues due to pump capacity and also fluid leakage. In addition, the high release force and hydraulic pressure negatively impacted performance of the rotating feed-through where the hydraulic release signal is passed from a static to rotational reference frame, and is directly applied to internal ring seals. To address the above, the clutch disc spring and several cover assembly parts were altered or remade to reduce the force required to release the clutch. The result was a 50% lower release force and corresponding lower hydraulic release pressure. This was beneficial for rotating feed-through performance although it did not lessen the aggressive engagement characteristics of the clutch design. The aggressive engagement characteristic is partially due to the friction coefficients as well as the mechanical ratio of the integral release mechanism geometry. While alternate friction materials with lower friction coefficients were available, the nature of use for this clutch did not warrant exploration. However, exploration of friction materials is planned with the wet-clutch and is discussed further below. Details of the resulting load reduction on the rotating feed-through are discussed below in sections discussing the rotating feed-through and seal rings.

Wet-Clutch. The wet clutch is an all new design. The basic wet-clutch configuration is shown in Figure 15.

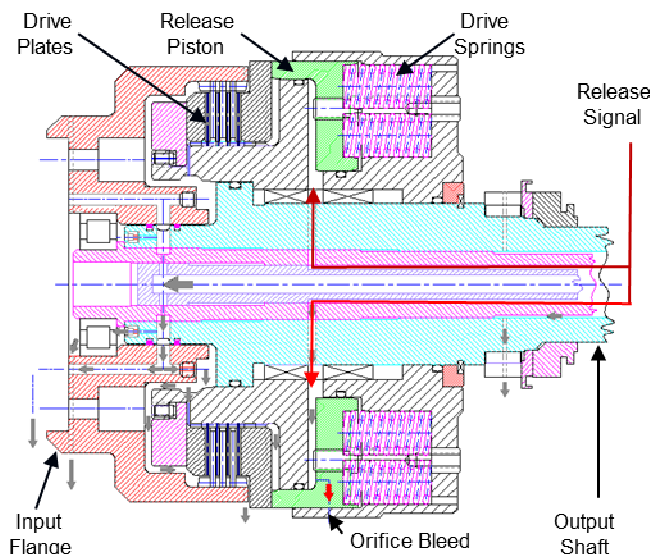


Figure 15. Wet-Clutch.

It was decided to utilize a number of commercially available clutch parts. Commercial parts include the toothed friction and steel drive plates. This saves the cost of manufacturing the drive teeth and the process of applying friction materials as well as selecting materials compatible with the required lubricant. This is certainly an area for future development and is identified as such. Due to the popularity of the base commercial transmission from which the drive plates are taken, a number of high performance friction plate materials are available. In addition a number of steel plate configurations are also available. Two multi-plate stack configurations of differing materials were selected for testing to establish a minimum baseline for shifting smoothness and wear. The stacks are of equal height and directly interchangeable, with the inclusion of one spacer plate. The two available high performance materials will be tested in the DOD-PERF-85734A synthetic ester-based fluid oil and compared. Other options are available but are described as being more of a standard performance grade. Although potentially being life limited, there is value in testing multiple materials with a goal of identifying a suitable range for friction coefficient with regards to operability and smoothness of engagement as a bench mark. Friction material durability can be addressed later as a development focus area.

As previously mentioned, to meet the design requirement of failsafe to high-speed direct-drive hover mode, the driving/clamping force for the friction/steel plates is applied by mechanical springs. This mode of operation is similar to the dry-clutch (i.e., mechanical drive - hydraulic release). A number of spring types were explored, (e.g., diaphragm, spring disc, and helical coil). For both operational and load considerations, a soft spring stiffness is desired and thus eliminated the diaphragm and disc spring due to very high stiffness. Coil springs were identified as the best from the perspective of applied loads and low stiffness. Recall with the dry-clutch that high spring stiffness and quick mechanical ratio were short comings relative to controlling smoothness of engagement. Since the springs are directly loaded, the mechanical advantage is 1:1, leaving stiffness to be the controlling mechanical variable. Due to the design requirement to failsafe to high-speed direct drive output, the springs must be preloaded to the force required for the drive plates to transfer the required power. To release the clutch requires that the springs be further compressed, which results in even higher spring loads, and requires higher hydraulic release pressure. Hence a soft spring stiffness, preloaded to provide the required driving force (with suitable margin), and further compressed to release, is the lowest overall compressed force possible. That is, barring the added complexity of some mechanical leverage system. Forty three-quarter inch diameter by two inch length springs were selected. The selection of coil springs over spring disc or diaphragm springs is not without a negative aspect and that is achieving and maintaining a balanced assembly. The coil springs fit into counter bored pockets. Compressing a coil spring results in an increase in coil diameter. This in turn requires a clearance between the spring and pocket to

preclude binding and provides a small error for unbalance. Small centering bosses were integrally machined in the hub spring seats. Considering the tradeoffs, the coil spring was still selected as the best. A need to change the spring force was realized with the dry-clutch and was accomplished through modifications. For the wet-clutch, spring force is optimized but is also easily adjustable by simply adding shims to increase force or by removing springs to lower force. Despite significantly higher release forces due the lower coefficient of friction of the wet-clutch friction materials, release pressure is minimized by using large area annular piston.

Clutch Release Closed-Loop Load Path. The design for both the dry-clutch and wet-clutch incorporate a closed-loop load path such that the clutch and output shaft can be assembled in the preloaded condition and also during operation the release loading is not transferred to the main assembly bearings. During operation the above clutch release forces are contained within a closed loop load path and are not transmitted to either the input shaft bearings or output shaft bearings.

Thus in the assembled state, both clutches are preloaded to the required drive force and released with the applied hydraulic force, with both the driving force and release forces contained within the respective assemblies.

The closed-loop load paths for both the dry-clutch and the wet-clutch are shown below in Figure 16. The dry-clutch spring load is partially contained in the clutch housing assembly, and the release load is contained within the clutch-shaft assembly load path between the bearing lock nut "A" and intermediate shaft aft flange "B". The wet-clutch shaft module drive spring preload also self-contained in a closed-loop load path between the shaft forward flange "C" and aft positioned retainer ring/shaft groove "D".

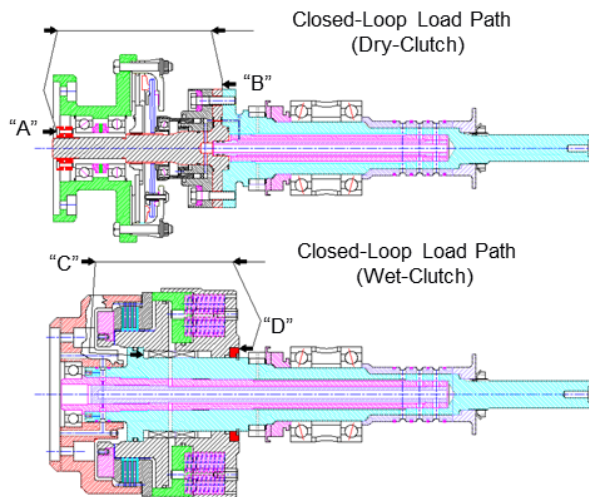


Figure 16. Clutch Release Closed-Loop Load Paths.

Hydraulic Clutch Release. Due to the basic inline drive configuration, the clutch and the release mechanism are both contained within the hollow low-speed shaft, thus the design of a clutch release system for both the dry-clutch and wet-clutch is a challenge, with each requiring somewhat differing considerations.

For the dry-clutch, the commercially available release bearing is typically mounted to a stationary housing and hydraulic lines are connected to the pressure energizing source. For the subject dry-clutch configuration, a rotating hydraulic release bearing is required which is supplied by a multi-passage output shaft, supplied by the rotating feed-through in which the fluid signal is changed from a static to rotational reference frame (discussed further down). Design of the rotating hydraulic release bearing is based on a commercial static mounted unit. The internal piston, hard stop, and rolling element bearing are used in original form. The internal geometry of the housing was redesigned for rotation, supply entry point, air bleed, and a flange to mate with that of the output shafts. The impact on centrifugal forces effecting the hydraulic passages was considered and deemed small due to the radial dimensions involved. The release system requires holding the pressurized signal during clutch release and a vent to purge any air.

For the wet-clutch, the air bleed function is achieved via replaceable twin continuous flow metered bleed orifices. The orifices ensure that the system is bled at all times during operation. Despite significantly higher mechanical spring forces as compared to the dry-clutch, a substantially larger annular release piston enables the use of a relatively low hydraulic release pressure compared to the dry-clutch where the annular piston area was approximately one square inch. The design includes positive lubricant flow within the clutch to provide cooling to the drive plates, flushing of wear debris from the area, as well as keeping drive teeth and splines flushed and lubricated. The system lubrication supply pump is drawn upon to support these minor flows continuously and can be optimized via two replaceable orifices which are different than those used to bleed the annular release piston.

Output Shaft (Fluid Passages). Due to the inline configuration, in order to provide the required fluid passages for both sprag and bearing lubrication, as well as providing hydraulic clutch control, a multi-passage shaft is required. The wet-clutch configuration includes a shaft designed to provide three co-axial passages. The multi-passage output shaft for the wet-clutch is shown in Figure 17. With respect to multi-passage design, the output shaft for the dry-clutch is similar and is not shown.

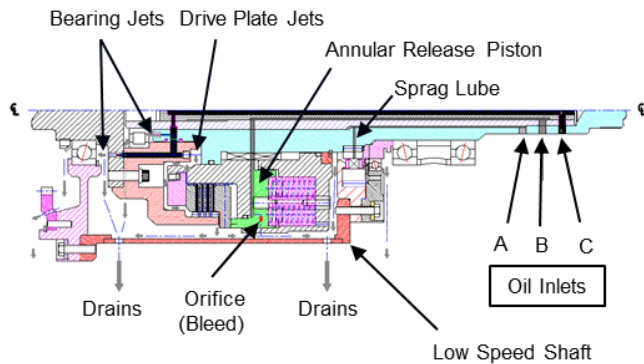


Figure 17. Output Shaft & Hydraulic Passages.

Oil flow circuit “A” provides lubrication to the sprag and aft-support bearing, “B” provides hydraulic pressure to the clutch release piston, and “C” provides lubrication to bearings and clutch friction/steel plates and drive teeth. The flows and drains are also shown. The “bleed” is a twin orifice metered flow to bleed the release piston chamber.

An output shaft with three co-axial fluid passages, requires a complex three-piece non-separable assembly consisting of a gun drilled primary outer shaft and two heat-shrink fit concentric inserts. Following the above assembly, final machining of the exterior is completed (e.g. grinding bearing journals and polygons as required). Radial passages connecting the three annular passageways to the outside diameter of the aft end at the rotating feed-through station are EDM (electro-discharge machined) to preclude trapping machining chips which may result from conventional cutting-edge machining. The shaft is balanced as an individual part. Different output shaft designs are employed for each clutch and are part of the respective clutch module.

The wet-clutch output shaft is more complicated than the dry-clutch output shaft. Added complications are primarily due to increased shaft length and the addition of a male polygon drive feature for the mating clutch. The differences in the output shafts are mainly due to the use of a number of commercial and NASA designed parts which make up the dry-clutch configuration as well as being able to decompose the dry-clutch shaft into intermediate and aft sections.

Rotating Feed-Through. A rotating feed-through is required for lubricating the sprag and local bearings, to provide the hydraulic control signal, and fluid flow other areas within the clutch (wet-clutch only). No commercial configuration could be located that met the physical size and placement requirements as well as operate at 15,000 rpm. Available products being of physical size were generally speed limited to approximately 5,000 rpm or less. Those products which had the speed capability were typically both size and channel count limited. To meet the design requirements for the two-speed drive, a three-channel design was developed using

ring seals. The higher pressure passages for the hydraulic clutch release signal are positioned at the innermost location, passage “B” in Figure 17, and the passages providing lower pressure lubrication flow are positioned outward of the above to reduce the net pressure differentials across the ring seals. Pressures for the dry-clutch are considerably higher than required for the wet-clutch. This was governed by clutch release force to overcome the mechanical springs and existing design release piston size limitations.

The rotating feed-through is comprised of the stationary support housing in which a stationary stator is mounted. The stator is sealed at the exterior via o-rings and at the interior with ring-seals. The o-rings require care during installation to avoid chafing/tearing at the longitudinal edges at the centerline (at cap to saddle interface) in the split support housing. The stator has two anti-rotation pins which engage with the main housing. A rotor is mounted on the shaft and includes exterior grooves in which the ring seals are installed. The stationary fluid connections are made to the exterior of the housing and transferred through the stator, into the rotor, and then into the shaft passages. The three-channel rotating feed through is shown in Figure 18. Oil “A”, “B”, and “C” functions below are as described earlier.

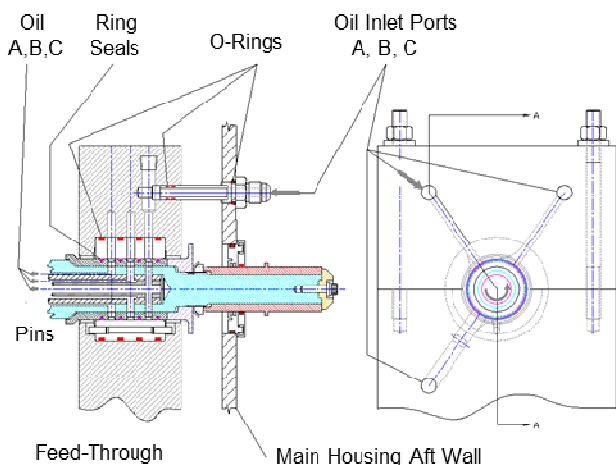


Figure 18. Rotating Hydraulic Feed-Through.

Ring Seals. Initially metal ring seals were used but were observed to experience high rates of wear. This was partly due to the initial high release pressure required for the dry-clutch and high operating speeds, both exceeding metal ring seal capabilities. Current design automotive transmissions incorporate ring seals made from Vespel®, a polyimide material, (Ref. 7). Vespel® rings seals, coupled with the lower pressures resulting from modifications to the dry-clutch, significantly reduced ring wear at the high pressure passage and nonexistent at the 80 psi lubrication passages. Pressure and tangential speed are two parameters that dictate ring seal performance. Earlier it was pointed out that the dry-clutch release force was reduced by modifying the clutch spring stiffness. Further release pressure reduction could be obtained by increasing release piston area but was

not feasible. Within the new wet-clutch design, the release piston area is increased to reduce required hydraulic release pressure to offset significantly higher clamping forces required for wet-friction drive plates due to inherently lower coefficients of friction compared to dry friction materials.

Gear mesh and rolling element bearing efficiency is high, thereby ring seal drag becomes a significant contributor to overall system drag and was particularly evident with the initial high release pressure of the dry-clutch. Reduction in clutch release pressure directly reduces overall system drag.

In the wet-clutch design, release pressure is further reduced, and is on the order of twice the bearing lubrication pressure, or 145 psi (1MPa), yielding a delta-pressure drop across the ring seal nearly equal to that of the pressure drop of the lubrication passage to atmosphere. This reduction was achieved by an increased release piston area. The above yields an ideal pressure distribution across all ring seals and is expected to operate at nearly the same delta-pressure. The ring seals for the wet-clutch are anticipated to perform well with reduced pressure differentials. The above results in a significant improvement to the boundary conditions and within capability of Vespe[®] ring seals.

Sprag (Overrunning Clutch). An automotive sprag with a torque capacity exceeding the design requirement for the two-speed drive was selected as the initial basis. The sprag is a sixteen element configuration. Inner and outer races were designed by NASA. The installation includes a generous geometric envelope for the sprag and races for future development as the sprag was identified as requiring future focus development. Sprag lubrication is provided at the inner race by a dedicated direct pressure-feed from a passage in the output shaft. Design of the rotating feed-through assembly and output shafts are clutch dependent, and include separate passages for sprag and bearing lubrication permitting independent flow adjustments. The sprag installation is shown in below in Figure 19.

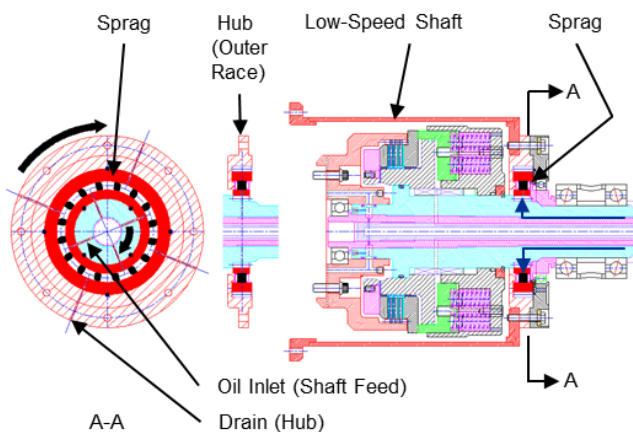


Figure 19. Sprag (Low Speed Drive).

Initial checkout testing indicated that the lubricant flow to the sprag was insufficient and was increased. In addition, the lubricant flow was observed to pulse slightly. It is believed that this condition is due the number sprag elements being an integer multiple of the lubrication passages (i.e., 16 elements vs. four lubricant supply passages). The above geometry causes the lubricant inlet and outlet passages to be intermittently blocked. Future design will increase the number of supply feeds at the inner race as well as a slight increase in the number of drain passages at the outer races. In the test rig hardware, additional supply passages were added at a one-half sprag element angular spacing for the wet-clutch design. This was readily accomplished by the addition of a circumferential groove at the inside diameter of the inner race and the addition of four radial holes at one-half sprag element spacing for the wet-clutch. For the dry-clutch design, a dam/deflector is added inside the sprag mounting hub to minimize lubricant entrance to the clutch local area. Increased drain capacity would be beneficial to aid in keeping lubricant from the vicinity of the dry-clutch. This consideration does not apply to the wet-clutch.

A shortcoming was identified in the design of the original sprag installation which allowed non-uniform loading both in the driving (loaded) and free-wheeling modes of operation, when used with the OCG Offset Compound Gear Drive. Due to the single mesh engagement of the OCG, an asymmetric load is transferred into the ring gear, thus generating an overturning moment which transferred through the low-speed shaft to the sprag outer race. Transfer of this load to the sprag was not recognized during initial design. To investigate low speed shaft movement due to the overturning moment, radial proximity instrumentation was added to monitor the low speed shaft. From tests, a change in low-speed shaft position was measured when the OCG transitioned from free-wheeling to driving mode.

Since the initial sprag installation design did not include straddle bearing support, the configuration was revised to include the needed aft bearing at the sprag. Using the available design envelope space to advantage, an aft bearing support was incorporated. The low-speed drum with ring gear at the forward end and sprag on the aft end, is now supported on a duplex bearing set. All other duplex arrangements in the overall drive design employ the back-to-back arrangement to maximize moment resistance to maintain input and output shaft alignment. Due to the overall length of the low speed shaft assembly and other geometric considerations, the face-to-face arrangement is employed. This duplex bearing set is axially preloaded as are all other duplex bearing locations in the overall design. In this instance, a spring is used to apply the preload whereas in other locations, universally ground inner and outer spacer tubes. Looking back, the single forward bearing and lack of an aft bearing at the sprag was a significant oversight and should have been included for both gear trains.

Offset Compound Gear Drive. The OCG innovation is based on a novel inline gear train which provides a 50 percent speed reduction in two stages, or meshes, utilizing only three gears (Ref. 8 and 9).

The heart of the concept is the offset cluster gear, which uses internal teeth on the input and external teeth on the output end, thus allowing it to simultaneously mesh in series with both a smaller external tooth input sun gear and a larger internal tooth output ring gear. Within this geometry, the cluster gear rotates on a separate axis offset from the input gear and output gear, which both rotate on the machine axis. The cluster gear replaces six planet gears required in a comparable simple planetary stage. The offset cluster eliminates the need for a reversing function as required with a conventional planetary gear train. See Figure 20 below.

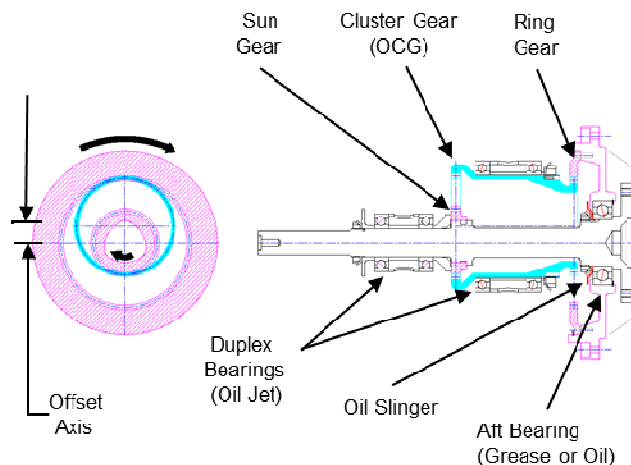


Figure 20. OCG Gear Module.

Note in the above that all bearings are oil-jet lubricated except the aft bearing, which is grease lubricated when used with the dry-clutch and also includes a forward oil slinger to protect against the mesh oil jets, and is replaced with an oil-jet lubricated bearing when used with the wet-clutch.

The OCG configuration is a simple gear arrangement with a number of benefits. Compared to a planetary gear train of equivalent ratio, the OCG configuration features significantly reduced gear and bearing count. The OCG intermediate gear and bearings spin at significantly lower speed compared to intermediate planet gears. The OCG cluster gear teeth do not experience reversed bending, common to planet gears in a planetary gear train, thereby potentially extending intermediate gear tooth life. The OCG configuration allows for both simplified optimal mesh lubrication and cooling since it requires only four total oil jets to provide for mesh-in flow and mesh-out flow at each of the two meshes. In addition, bearing oil jet lubrication requirements are minimal since the configuration essentially requires only four main bearings. In the test rig design,

bearing lubrication points were single point. Whereas in a follow on, higher power design, the bearing jet count would likely be increased for reliability from a single jet to two or more lubrication jets as a function of bearing rolling element pitch circumferential length, and is easily accomplished. The temperature of each bearing in the test rig is monitored using a single spring loaded thermocouple located circumferentially at the load reaction line. The lubrication points and thermocouple locations are shown for both the input and output gear meshes in Figure 21 below.

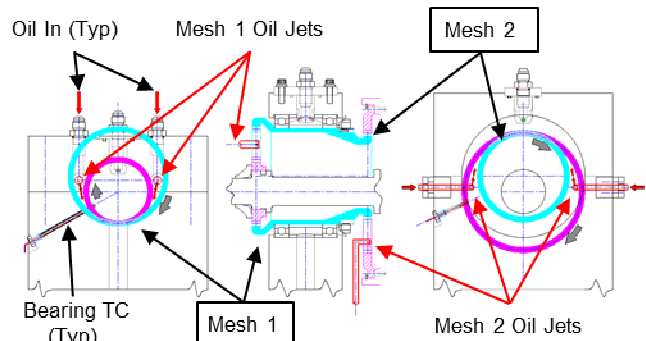
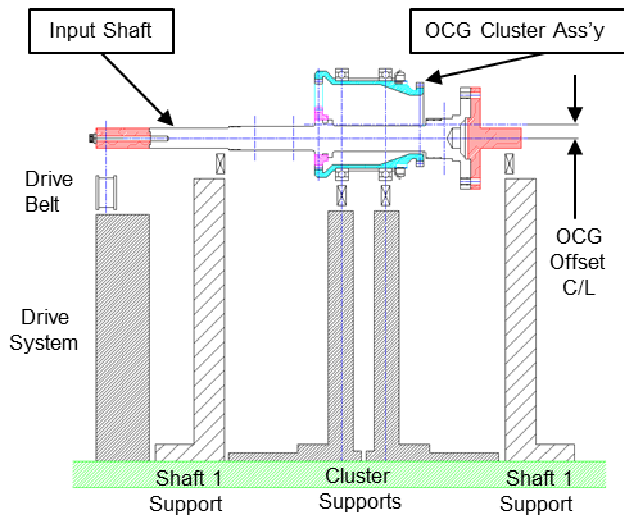


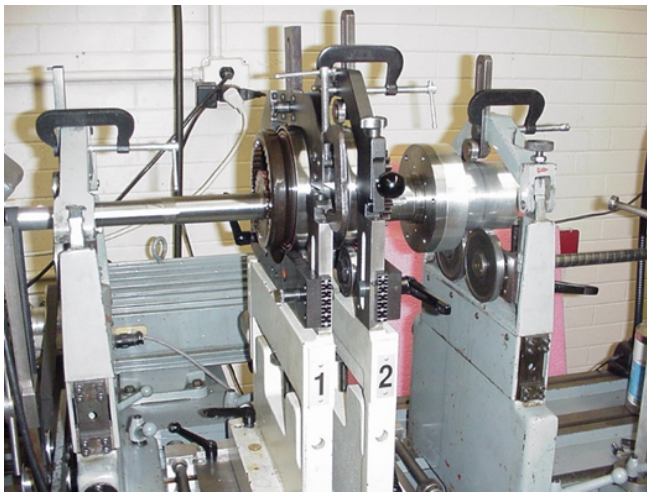
Figure 21. OCG Gear and Bearing Lube & Temps.

The OCG is not without some negative aspects. The large diameter of the cluster bearings result in increased pitch circumferential speed of the balls and cage. The loading on the cluster support bearings is increased due to the position of the gear teeth loads being outboard of the bearings, and the offset axis of the cluster gear relative to the central machine axis. However, comparing a single offset axis to the multiple axis of planet gears, a single offset axis is not be all that negative as will be shown below in the description of the complementary planetary gear module.

During the course of establishing a balance plan for the OCG module it became apparent that the cluster gear assembly did not lend itself to a simple balance set up on any of the balance machines available in-house at NASA GRC. Since this configuration must be balanced for high-speed operation, a major concern arose. This was the first potential shortcoming for the OCG and a potential dead end for follow on consideration if not resolved. A solution to this dilemma was devised using the input shaft and input gear to drive the cluster gear, both individually mounted on separate rollers, on two axis separated by the OCG offset, for a balance check in lieu of traditional direct belt drive. The above set up required separate bearing supports for the input shaft and an additional bearing supports, adjusted for the offset axis, as well as the sensor supports for the cluster, to obtain the mass unbalance readings required to make any needed mass correction. The initial concept layout is shown in Figure 22a and a photograph of the balance machine set up is shown in Figure 22b. A setup like this had never been done in the GRC balance shop. The balance was successful and resulted in relatively smooth operation in the test rig.



a) CAD Layout.



b) Balance Shop Set-Up.

Figure 22. OCG Cluster Gear Balancing.

Dual Star-Idler Planetary Drive. The DSI planetary is shown below in Figure 23.

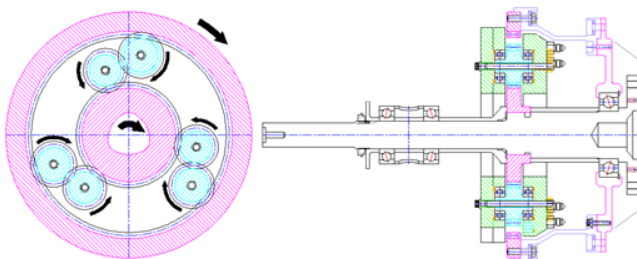


Figure 23. DSI Gear Module.

The design of the DSI planetary gear train and support carrier is more complicated than that of the OCG due to the significant increase in gear and bearing parts count and the required lubrication points and thermocouple temperature measurement points. There are significantly more axis of revolution for the DSI planet star gears as compared the single offset axis for the OCG cluster gear. To achieve the desired accuracy in planet axis positions, the carrier assembly requires simultaneous match machining to assure that the locations for the planet bearing bores and other features crossing the three-plate construction are in the proper positional relationship to the machine central axis as well as being individually coaxial thorough the three plates.

The number of gear meshes and bearings in the DSI increases the complexity of the lubrication delivery system. The star-idler planet gears require oil jets at each of the twelve support bearings. The six star gears require nine mesh lubricating and cooling oil jets. Some of the mesh oil jets were not incorporated due the more important need to provide thermocouple temperature monitoring for each of the twelve planet bearings (i.e., nine mesh jets is not optimal). Providing bearing temperature monitoring took precedence over providing optimal gear mesh lubrication. A spring-loaded thermocouple is located circumferentially at the bearing load reaction centerline. An exploded view of the carrier plate assembly and lubrication hardware assembly side-elevation is shown in Figure 24.

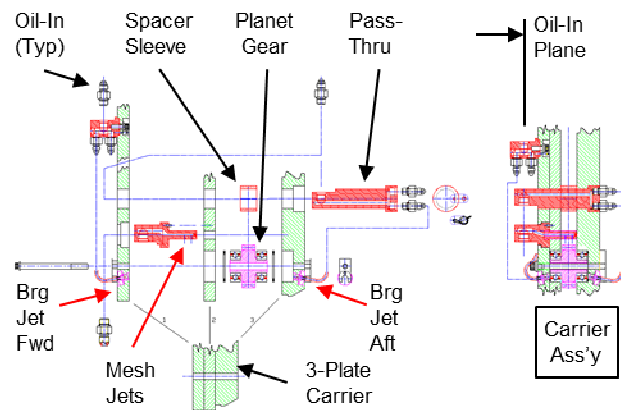


Figure 24. DSI Carrier Ass'y and Lubrication.

Significant reduction in planet bearing loads are realized by operating the gear train in the design direction of rotation, compared to operating in the opposite direction. The DSI design minimizes the planet bearing loads. A nominal load reduction to 50% at the star and reduction to 67% at the idler is shown in Figure 25. Reduced loads are due to a portion of the planet bearing loads reacted by the sun and ring gear teeth whereas operating in the opposite direction the gear tooth loads are reacted solely by the planet bearings.

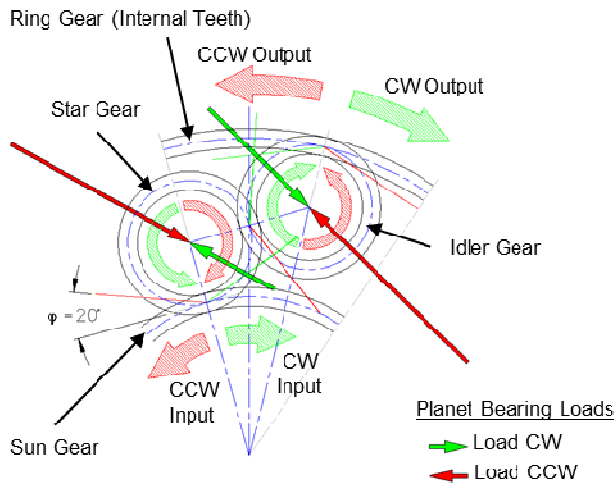


Figure 25. DSI Planet Bearing Loads vs. Rotation

Comparison of Design Parameters for OGC vs. DSI. OGC gear parameters are shown below in Table 3 and DSI gear parameters are shown in Table 4. The difference in output ratio of the two gear sets is due to adjusting tooth counts to obtain a hunting-tooth gear configuration for improved life.

Table 3. OCG Gear Parameters.

Material: 9310, Backlash: 0.006-0.011 inch, Width: 0.375 inch, Contact Angle 20°

Gear	Pitch	Pitch Dia (inch)	N _{teeth}	Rpm
Input	8.727	2.865	25	15,000
2	8.727	4.240	37	10,135
3	8.0	3.875	31	10,135
Ring	8.0	5.250	42	7,481

Table 4. DSI Gear Parameters.

Material: 9310, Backlash: 0.010-0.015 inch, Width: 0.600 inch, Contact Angle 20°

Gear	Pitch	Pitch Dia (inch)	N _{teeth}	Rpm
Sun	12	4.1667	50	15,000
Star	12	1.5833	19	39,474
Idler	12	1.6667	20	37,500
Ring	12	8.4167	101	7,426

Bearings - Input / Output Shaft and Intermediate. Bearings at the input and output shafts for the various modules are the same respectively. A dedicated set of bearings is used for each module since they become part of the overall rotating assembly, precluding the need to fully disassemble a prior configuration to reuse bearings. Radial loads at the input shafts differ due to the respective gear modules. This difference is minor. From the tables above it is seen that the speeds of the intermediate gear elements differ significantly

due to mesh count (load sharing), axisymmetric vs. non-axisymmetric geometry, and physical size.

The bearing location, speeds, and sizes are summarized below for the various configurations. OCG bearing parameters are shown in Table 5. DSI bearing parameters are shown below in Table 6. In both tables, D_{brg} is the bearing outside diameter, and d_{brg} is the bearing bore, units in millimeters.

Table 5. OCG Bearing Parameters.

Site	Rpm	Size	D_{brg}	d_{brg}	dN factor
Input	15,000	206	62	30	450,000
2	10,135	1822	140	110	1,114,850
3	10,135	1822	140	110	1,114,850
Ring	7,481	210	90	50	374,050

Table 6. DSI Bearing Parameters.

Site	Rpm	Size	D_{brg}	d_{brg}	dN factor
Sun	15,000	206	62	30	450,000
Star	39,474	202	35	15	592,110
Idler	37,500	202	35	15	562,500
Ring	7,426	210	90	50	371,300

From the above tables it is observed that the OCG cluster gear bearings rotate at a slower speed compared to the DSI planet bearings. However, due to their physical size, the resulting dN values for the OCG bearings are substantially higher than those of the DSI planet bearings. The extreme speeds of the DSI star-idler gears limits the length of service due to the shorter time to accumulate cycles.

Duplex bearing sets at the input and output shafts utilize machined equal-length spacer sleeves at inner and outer races to achieve their axial preload. All central shaft bearings are oil jet lubricated with a single oil jet, but are not located at the angular orientation of maximum radial load. Each bearing is instrumented at the outer race at the load reaction angular location with spring loaded thermocouples. For a follow on application design, the above bearing oil jet count would be increased. For the test rig it would be an easy revision to increase the oil jets at all shaft locations other than DSI planet gear bearings but testing indicated it was not necessary. Main bearing drains are oversized.

Duplex bearing sets at the OCG cluster use machined equal-length spacer sleeves at inner and outer races to achieve their axial preload. Duplex bearing sets at the DSI planets utilize conical bearing springs to provide axial preload. Drains for the OCG cluster gear support structure are equally ample replicating those in the main bearing configuration used for the input and output shafts. The drains at the DSI planetary are limited and additional drain holes would be required.

Retaining Rings. Retaining rings are used in the overall design to reduce costs of fabricated parts, reduce weight, and reduce the use of commercial fasteners, as well as to improve/simplify assembly and disassembly. A negative aspect of using retaining rings is the need to account for the inherent unbalance. This is particularly a problem for high-speed hardware when using large diameter rings where the unbalance can be substantial (e.g., OCG cluster gear). Balanced rings are commercially available but are expensive and often larger quantities of rings must be ordered. Since a significant number of retaining rings are used in the modular design, with some quite large, an in-house balance methodology was developed to eliminate the need for clocking & match marking retaining rings during assembly. The ring unbalance mass correction was calculated and machined. No further corrections were required. A typical retainer ring unbalance mass correction of predefined drilled holes is shown below in Figure 26.

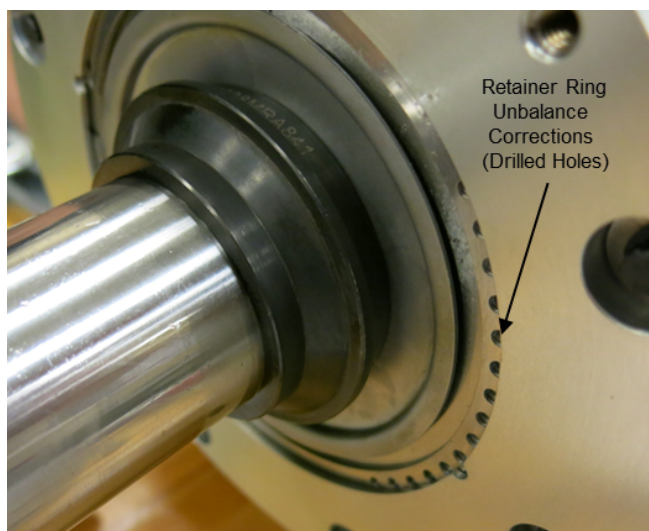


Figure 26. Retaining Ring Unbalance Mass Correction.

LESSONS LEARNED (FROM ABOVE DESIGNS)

Housing Considerations. Specific to the test article housing, the housing side walls leaked at the base. O-ring provisions at these locations were intentionally excluded for cost reduction due to this joint not being one that would be disassembled following the initial build. It was thought that minor leaks would not be a concern in this area. Leakage was minor, but o-ring grooves and o-ring stock, as on other joints should be added.

Oil control. For the OCG, when used with the dry-clutch, the oil slinger located at input shaft aft end is insufficient and allows oil to enter aft bearing as discussed and shown in Figure 20. This applies only when used with the dry-clutch since the aft bearing for this configuration is a grease-

pack/sealed bearing. In both the OCG and DSI gear trains the ability to also use oil-jet lubricated bearing exists when used with the wet-clutch due to incorporation of oil jet supply passages in the wet-clutch design. Although no service issues were encountered, the slinger outside diameter should be increased in such a configuration. No changes are planned due to the planned short term use of the dry-clutch.

Sprag. The current sprag and race hardware will be used with the wet-clutch configuration. However, a custom made-for-application sprag and sprag races should be considered in future development, such as the design of an intermediate power level drive. An optimized design sprag configuration for intended operation as well as materials and surface enhancements for added durability required for a mid-power level model demonstration should be used.

The feed and drain holes in the inner and outer races should be changed from present design and not coincide as an integer multiple of the number of sprag elements to prevent simultaneous fluid flow blockage of elements, at the inner and outer races. The above should be increased by a factor of two such that the second additional set of feed holes is fed from the same supply and are located at one-half the sprag element angular spacing to completely preclude passage blockage and increase oil distribution during free-wheeling.

Based on the addition of an aft sprag support bearing, a better fully-integrated straddle duplex bearing arrangement to support the OCG overturning moment imposed on the low-speed shaft as well as prevent directional loading of sprag during free-wheel should be used. Although the DSI gearing does not create the same overturning moment as the OCG gearing, the aft-sprag bearing, thus forming a duplex bearing set for the low speed shaft assembly (ring gear, shaft, sprag hub) is required for any follow on designs.

Rotating feed-through. Optimize the diameters of the rotor and stator to increase net area in contact with face side of seal to reduce net pressure on sealing face and reduce the effective pressure on the contact area, thereby reducing wear and improving service life. Low wear, high-speed ring seals such as the Vespel material are required. Ring seal drag is a significant contributor to the overall system drag. Reduced clutch release pressure directly reduces overall system drag.

Shafts and Couplings (Input/Output). Positive drive key and grooves at input/output should be used in lieu of taper friction lock as initially used in the test rig. The taper friction lock configuration was employed because of apparent advantages of interchangeability and reduced balance mass corrections, and ease of assembly/disassembly. No further issues with slippage of the taper locks were realized once the motor control and torque limits were revised from initial settings. The use of taper lock hubs was considered only for this test article as a perceived simplification and not for follow on higher power designs, which would be designed for specific interfaces.

FUTURE DESIGN

Planetary Differential (Leveraging the DSI Design).

A dual-input planetary differential has merit as a possible variable-speed positive drive and was the third selected concept (Ref. 4) for possible future development. Recently a paper-design for a configuration based off of the NASA DSI planetary configuration was initiated. A brief overview of the configuration is provided below.

During the design of the DSI dual star-idler fixed carrier gear train, a significant challenge was providing oil-jet mesh lubrication and mesh cooling, as well as planet bearing oil-jet lubrication. Due to the number of gears and bearings, any planetary gear train, even a simple planetary is considerably more complicated to lubricate than the OCG configuration or a parallel shaft configuration.

For the subject dual-input planetary differential drive, providing oil-jet lubrication to a rotating carrier with planet gears and bearings mounted within is a significant increase in complexity over the fixed-carrier DSI.

A starting point in the development of a possible dual-input planetary differential configuration is to leverage the matured design basis of the DSI planetary gear train for the basic planetary differential. Immediately the question arises as to what is the best approach, carrier-control with ring-output, or, ring-control with carrier-output? Following the creation of a few concept study layouts of both configurations, it appears employing carrier-control with ring-output is less complex to develop. That is assuming intent to utilize some of the existing DSI test hardware.

For the present, consideration of the requirements for a second input and identification of a suitable second input is deferred. A secondary low level task, by others, is the consideration and possible identification of a suitable second input source but is not within current scope. Attention is solely focused on the gear train.

The initial challenge to provide carrier rotation and required bearing lubrication. Having the initial design basis in hand could then permit further development of the above into a more complete design. From this, a carrier controlled (brake/release), or a dual input (fully variable speed control) device, can be realized.

The second challenge is to provide lubrication to the planet gears and bearings on the moving carrier. The best approach is to provide direct jet lubrication supplied from within the rotating shaft, directed radially outward to the required lubrication points. This is in contrast to attempting to spray lubricant inward on a rotating object (splash lubrication). Also part of the second challenge is supplying oil to gears which are spinning on their own axis and also simultaneously revolving around the primary machine axis (i.e. star and idler gears on a revolving carrier). As was done

in the DSI design, it is advantageous to provide separate channels/circuits for bearing lubrication and gear lubrication.

It is believed that a dual-input planetary differential configuration can be leveraged from the NASA DSI planetary gear drive, coupled with leveraging and advancing of few other aspects of the overall drive design. The basis of the rotating feed-through, ring seals, can be leveraged to provide an integral lubrication system for the planet gears and bearings supplied through a multi-passge shaft. Since there are planet bearings fore and aft of the sun and ring gears, lubrication must be provided individually to both to the forward and aft planet bearings. Gear mesh lubrication and cooling can be provided from either side, but the obvious choice is from the forward end due to the low-speed output shaft blocking access to the aft planet bearings.

A possible design of the gear train for a 1:1 to 2:1 ratio range dual-input planetary differential with carrier-control ring-output is shown in Figure 27.

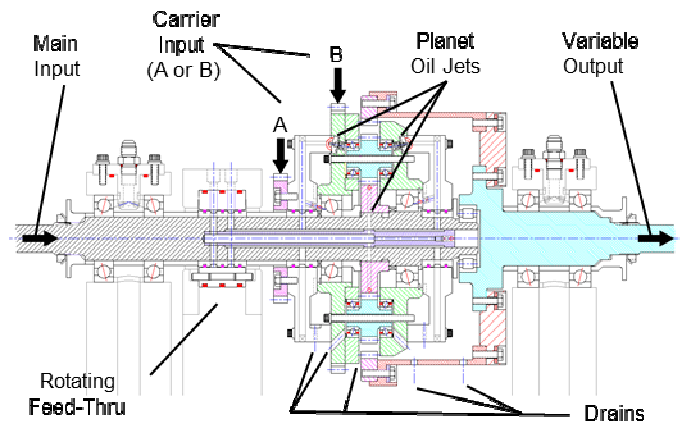


Figure 27. Planetary Differential CAD Layout.

CONCLUDING REMARKS - RECOMENDATIONS

This paper has summarized the conceptualization, design, manufacture, and initial testing of inline two-speed drives for advanced design tilt-rotor helicopters to provide reduced rotor speed. Two gear trains are offered for consideration.

A number of areas were identified for detailed development or possible alternative designs. Advancing these component level areas is critical to the success of an overall system.

In addition, a number of lessons learned from the design and initial testing of the first two configurations of our two-speed drive rig. Some lessons can be used in the design of an intermediate power version while some may be applicable to other designs using similar components.

Finally, the basis for a gear train concept was presented that could be used as part of a variable-speed gear drive system based on a dual-input planetary differential drive.

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