Results of NASA Technical Challenge to Demonstrate Two-Speed Drive for Vertical Lift Vehicle

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ABSTRACT

Currently, manned vertical lift vehicles are flown in a manner such that the rotors operate over a narrow range of rotating speed regardless if the vehicle's flight condition is one of vertical takeoff and landing, hover, or forward cruise. The propulsion systems are optimized for operation at the same, corresponding narrow range of rotor speed. However, certain missions and markets benefit greatly if the rotor speed can be adjusted over a wide range of speed to match demands of different missions and flight regimes. A vehicle that can operate with a wide range of rotor speeds would address key barriers to enable new markets and missions for vertical lift vehicles. Key barriers addressed by the wide-range of rotor speed include noise reduced via lower rpm rotor, increase of maximum forward flight speed, increased payload and range, reduced fuel burn, and lower operating costs. A new paradigm for the propulsion system is needed to enable these key benefits. One viable approach is to make use of a two-speed ratio drive system such that the engine can continue to operate over a narrow speed range, whereby engine performance is optimal, while adjusting the rotor speed as needed using the two-speed drive system. Motivated by such needs and by results of several system studies, a NASA Revolutionary Vertical Lift Technology Challenge was established to develop and demonstrate required technologies and designs for achieving a 50 percent reduction in rotor rpm via a two-speed drive system that incurs less than 2 percent power loss and maintains current power-to-weight ratios. The technical challenge work was completed and the technical objectives were achieved. This report describes the motivations, the research approach and the significant outcomes.

INTRODUCTION

An assessment study was completed to determine if a two-speed drive system for a large civil tiltrotor featuring use of composite materials to replace steel could achieve the technical objective to "maintain current power to weight ratios". The engineering assessment made use of information from References 1 to 14. The vehicle chosen for the assessment was the NASA LCTR2 vehicle, a conceptual vehicle for 90 passengers with a cruise speed greater than 300 knots and a range greater than 1,000 nautical miles. A sketch of the LCTR2 vehicle is shown in Figure 1 (Ref. 15). To achieve the desired range and forward vehicle speed, the LCTR2 converts to an airplane configuration by tilting the rotors and slowing the rotor tip speed to 350 fps while flying in cruise mode.

Therefore, the technical challenge set a number of goals. These goals included: (i) Develop and demonstrate a design capable of reducing the main rotor speed by 50 percent by means of a two-speed ratio drive; (ii) the two-speed drive power loss to be no more than 2 percent; and (iii) the two-speed ratio drive maintains current power to weight ratios.

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Many steps had to be completed to accomplish the technical challenge. The approach used was the following: (i) Specify, procure, and commission a new test facility for two-speed drive demonstration experiments. The new facility was specified to test a gearbox with 15,000 rpm input shaft speed transmitting 200 hp; (ii) Complete a conceptual design study for two-speed drives for vertical lift vehicles. Assess innovative designs and down select concepts for hardware demonstration. The selection of concepts for demonstration should consider the power loss and weight-neutral objectives of the technical challenge; (iii) Complete detailed design of the selected two-speed ratio demonstrator drives. Complete fabrication and purchasing of parts, dynamic balancing for high rotational speeds, and assembly; (iv) Demonstrate a design capable of reducing the main rotor speed by 50 percent by means of a two-speed ratio drive; (v) Research, develop, and demonstrate lightweight and low-power-loss drivetrain technologies; (vi) Assess the power loss of a demonstrated two-speed ratio drive concept and compare result to the objective of "no more than 2 percent power loss"; and (vii) Assess the weight two-speed ratio drive modules for a LCTR2 vehicle. Determine the weight that can be saved from the drivetrain using lightweight drivetrain technologies demonstrated to TRL4. Compare the weight added for two-speed functionality to weight saved using lightweight drivetrain technologies to assess the objective "maintain power to weight ratios".

The objective of this paper is to describe how the technical challenge was accomplished in support of NASA's Revolutionary Vertical Lift Technology (RVLT) project.

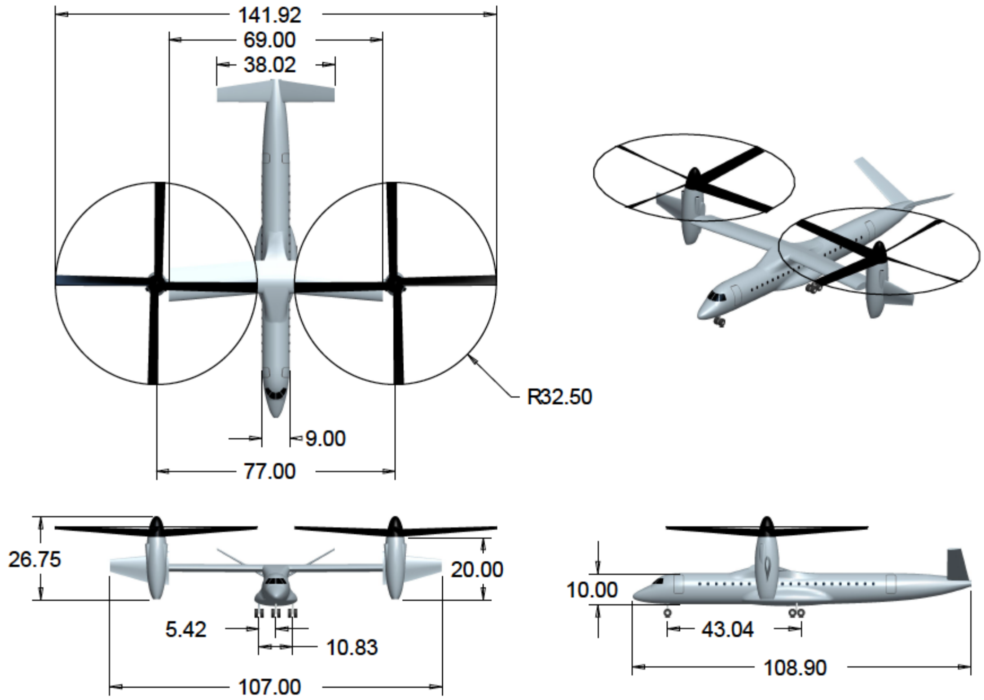


Figure 1. Large Civil Tiltrotor (Version 2) LCTR2 from Reference 15.

Test Facility for Two-Speed Drive System

To complete the demonstrations and performance experiments of two-speed ratio drive systems, a new NASA test facility was needed and established. A full-scale facility for drive systems for manned vertical lift vehicles was prohibitively expensive. To demonstrate the key technologies, it was deemed important to operate at speeds representative of flight hardware but not necessary to operate at full-scale power. To meet the research needs, a facility was specified to operate at 200 hp. The maximum shaft speeds were specified to be 15,000 rpm for the input shaft and 15,000 or 7,500 rpm for the output shaft, the output shaft speed depending on whether the test article were operating in high-speed (hover) or low-speed mode (cruise).

The test facility is based on an electrically-regenerative principle. The facility includes two high-speed electric machines that can operate as a motor or generator. The facility implements a four quadrant, flux vector, high-performance drive system containing a common direct-current (DC) bus. The common DC bus approach allows the power being absorbed from electric machine operating as a generator to be utilized by the driving motor. This permits testing at high power levels but drawing only enough line power to overcome friction and electrical resistance losses. Figure 2 is an artist's sketch of the test facility's rotating equipment, test transmission, and base-plates. The large baseplate depicted in Figure 2 has a length of 16 ft.

The test facility also includes auxiliary systems, namely power electronics, controllers, lubrication and hydraulic fluid system, cooling water, sensors, and data acquisition. The electric machines can be controlled to operate with conditions of constant speed, torque, or power depending on the research test requirements. The lubrication system provides the test transmission with filtered lubricating oil at controlled oil temperature and pressures. A separate lubrication system provides oil-mist lubrication of the high-speed bearings of the two facility motors.

Concept Design Study and Selection of Demonstrator Configurations

Currently, manned vertical lift vehicles are flown such that the rotors operate over a narrow range of rotational speeds regardless if the vehicle's flight condition is one of vertical takeoff and landing, hover, or forward cruise. Therefore, existing drivetrain configurations have a fixed ratio of engine speed to rotor speed. The technical challenge required to operate the main rotor at two discrete speed, the slower speed a 50 percent speed reduction. Therefore, innovation was required to devise the drivetrain demonstrator configurations.

A concepts study was completed for demonstrator configurations for the technical challenge (Refs. 16 and 17). Looking at concepts from a vehicle-level perspective, the drivetrain concepts naturally grouped into three candidate gearing configurations: (1) inline planetary gears, (2) offset compound gears, and (3) planetary differential gear configurations. Also, three methods were identified for accomplishing the speed shift transitions: (a) direct speed-shift using clutches, (b) controller assisted speed shifts requiring a second prime mover, and (c) variator-assisted speed shift requiring a continuously-variable speed traction/friction drive.

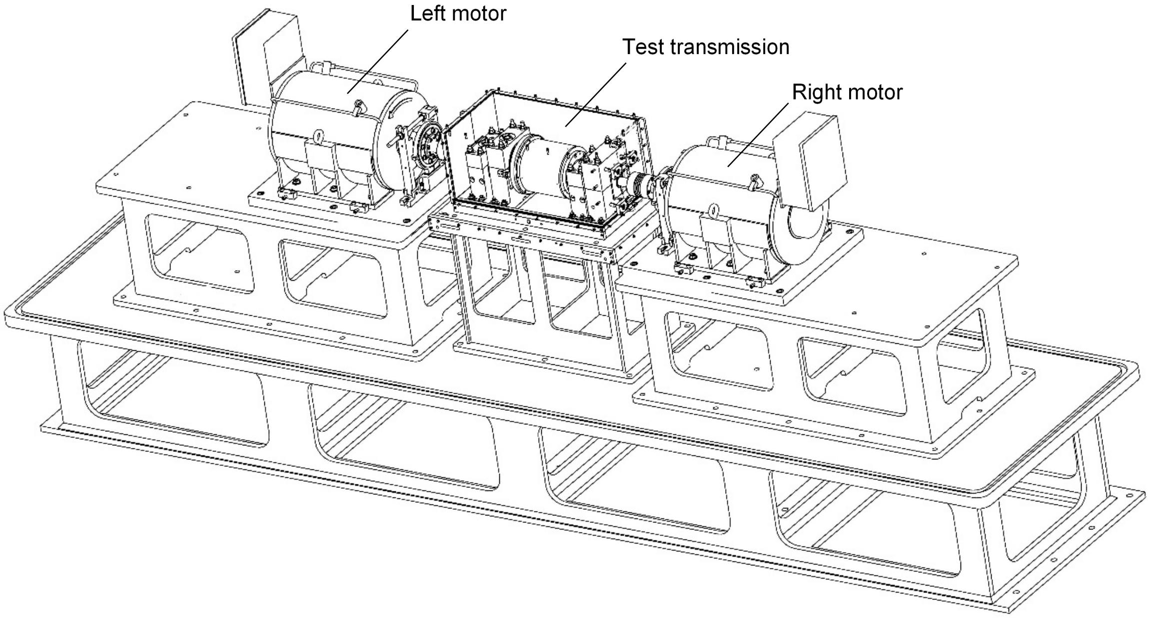


Figure 2. Test facility for two-speed drivetrain demonstrations and performance experiments.

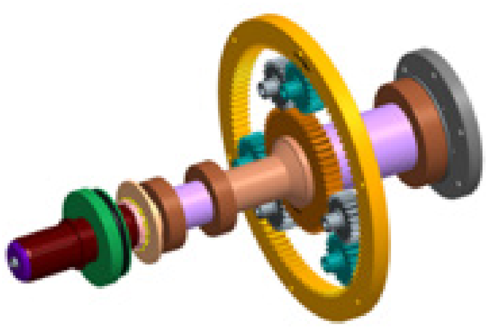


Figure 3. Dual Star Idler Planetary

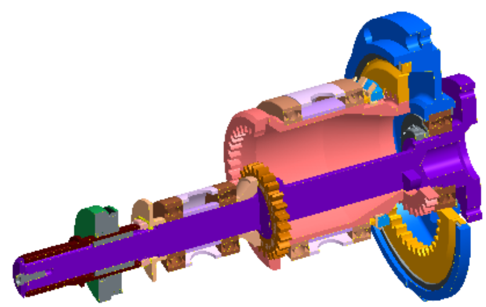


Figure 4. Offset Compound Gear Drive

Qualitative evaluations were made of the nine possible demonstrator candidate configurations considering the technical challenge objectives. The objective to "maintain current power to weight ratios" was influenced by the selection of demonstrator concepts from among the candidate configurations. Two gearing concepts, the dual star idler (Figure 3) and offset compound gear configurations (Figure 4) were selected for detailed design and demonstration. The dual-star idler configuration is similar to the gearing used for modern geared-fan engines that make use of so-called "star" gearing configurations. For the geared fan application it is permissible for the output member to rotate in the opposite direction from the input member. However, for the two-speed drive, the output member must rotate in the same direction whether in low-speed or high-speed mode. The star configuration could be used by introduction of a second "idler" star gear at each star-gear location to achieve proper rotation directions. The offset-compound idler is another unique gearing configuration devised to achieve the proper rotation direction for both high- and low- speed modes. While gear pairs with external teeth on both members will rotate in opposing directions, gear pairs using internal teeth on one member rotate in the same direction. Using two sets of external-internal gear pairs with proper center distances produced an innovative two-speed drive solution with desired coaxial input and output shafts. The two-speed offset compound gear concept is the subject of two patents (Refs. 18 and 19).

Detailed Design and Fabrication of Two-Speed Demonstrator Drives

The detailed design of the demonstrator drives made use of a modular test article design approach. Module interfaces were established that included a number of fixed stations for bearings and mating flanges. It was further decided to forego a flight-like housing configuration and instead design a single common unit to permit more of the budget to be used for rotating components rather than to multiple housings. The common housing approach was also deemed more practical for integration of research sensors and allowed for maximum accessibility. With the modular approach each gear module could be assembled with any clutch module. Detail design and fabrication was accomplished for two gear concepts and three clutch concepts, and the combinations of modules could produce six demonstrator configurations. The modular design concept maximized flexibility for demonstration and performance testing. Flight systems would be packaged differently to minimize volume and weight while optimizing performance and meeting all aircraft integration constraints.

Detailed design was completed for two gear concepts, the dual star idler and offset-compound gear. Three clutch concepts were designed and fabricated, a dry-clutch, alternative dry-clutch, and wet-clutch. The designs are combinations of unique parts and commercial off-the-shelf parts. For example, the dry-clutch used for demonstration made use of an available automotive racing clutch with some minor modification made to mate with the gear module output flange. Figure 5 depicts the process of combining the offset-compound gear module with the dry-clutch module to produce one of the six possible two-speed drive test demonstrator.

Demonstration of Two-Speed Drives With 50 Percent Speed Reduction

As has been mentioned, the modular approach for the test articles allowed for six possible demonstrator designs. Three of the six possible designs have been tested. All designs had concentric input and output shafts and were designed to provide 1:1 and 2:1 output speed reduction ratios. All were designed for 200 hp and 15,000 rpm maximum input shaft speed. The three configurations tested were: (1) the dual star idler (DSI) gear with dry clutch, (2) the offset-compound gear (OCG) with dry clutch, and (3) the OCG with wet clutch. All of the tested designs performed successful transitions from high-speed (1:1 ratio) mode to low-speed (2:1 ratio) mode and then transitioned back to high-speed mode.

References 20 and 21 have provided detailed reporting of the results of testing of the two configurations using the dry clutch. Following is a summary of the testing scope and the research findings. Shift tests were performed on the demonstrator transmissions at input speeds of 5,000, 8,000, 10,000, 12,500, and 15,000 rpm. Both the DSI and OCG configurations successfully performed speed shifts at full rated 15,000 rpm input speed. The transient shifting behavior of the OCG and DSI configurations were very similar. As the transient behaviors of the OCG and DSI tests were very similar, it is deduced that the shift clutch (same dry carbon clutch used with both configurations) had more of an effect on shifting dynamics than did the configuration of the gears. For both configurations, the low-to-high speed shifts were accomplished with a small transmitted mean applied torque to prevent overloads on the transmission due to transient torque spikes. It is believed that the relative lack of appreciable slippage of the dry shifting clutch was the primary cause of the transient torque spikes. For the low-to-high speed shifts, the output speed ramp-up time slightly increased and the peak output torque transient peak slightly decreased by varying clutch-control pressure in a more gradual manner. This desired reduction of the transient torque spike peak was a result of more clutch slippage produced when the commanded clutch pressure ramp-down rate was decreased. Some of these details are explained more fully in the text to follow.

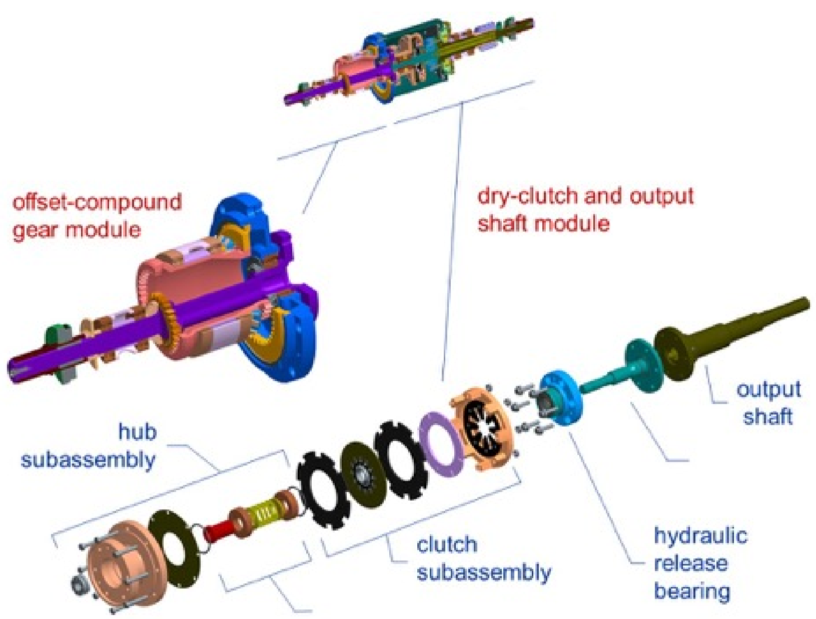


Figure 5. Illustration of process of combining gear module with a clutch and shaft module to produce a two-speed drive demonstrator test article (Offset Compound Gear Drive shown).

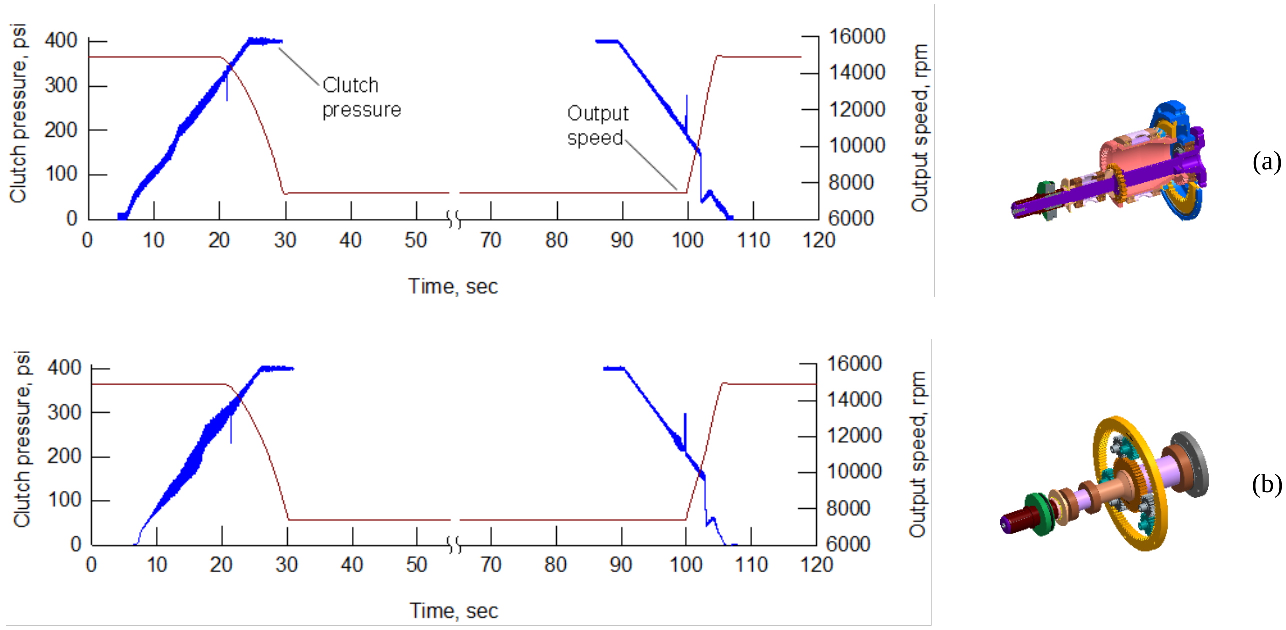


Figure 6. Transmission clutch pressures and output speed during transient shift events. (a) Offset compound gear with dry clutch configuration. (b) Dual star idler gear with dry clutch configuration.

The trends of the output shaft speed during shifting events when using the dry clutch are provided in Figure 6. The plots depict both the down-shift and up-shift events for both the OCG and DSI gear configurations. The data shown are for the transmissions operating at full rated 15,000 rpm input shaft speed. The data begins with the transmission operating in high-speed mode (output shaft speed of 15,000 rpm). A down-shift event is initiated at time corresponding to 5 sec. The clutch pressure, depicted by the blue-colored line, is commanded to increase linearly with time. Eventually the clutch pressure is large enough that the pressurized piston overcomes the spring forces engaging the friction clutch plates. At approximately time equal to 20 sec, the friction clutch disengages and power to the output shaft is released. The output shaft speed coasts down until the output shaft speed equals the speed of the slowest-speed gear (at time of approximately 30 sec) and a sprag clutch engages to power the output shaft at 50 percent reduced speed of 7,500 rpm. The low output shaft speed is obtained in a stable manner. Next the up-shift event is initiated at time of approximately 90 sec as the clutch pressure is commanded to decrease linearly with time. At time approximately 100 sec, the friction clutch reengages and the power flow abruptly changes load-path from the slowest speed gear to the highest speed input gear. The output shaft speed must quickly increase to match the 100 percent input shaft speed. The shaft-speed behavior was stable and nearly identical for both the OCG and DSI transmissions.

The speed-shifting events depicted in Figure 6 produced transient shaft torques during the speed transitions of the output shaft. The output shaft torques were measured and the data plots are provided in Figure 7. First the output shaft torques during the downshift event as the output shaft condition transitions from high-speed to low-speed mode will be discussed. Once the friction clutch disengaged at time of 20 sec, the output shaft torque dropped to zero since both the friction clutch and overrunning sprag clutch were disengaged. The zero output-shaft torque condition lasted for approximately a ten second duration while the disengaged electric machine (acting as a generator) coasted down in speed. The output shaft speed reduced from 15,000 rpm reaching 7,500 rpm at time of approximately 30 sec. At this time the sprag clutch reengaged, the output shaft torque increased briefly to about 25 ft-lb, and then the output shaft torque decreased to a steady-state condition of 15 ft-lb that matched the required constant-power condition of the driven electric machine. Next in the testing sequence an upshift event was commanded, with minimal mean torque applied, at time of 90 sec. The friction clutch engaged at a time of approximately 100 sec. The speed of the electric machine driven by the output shaft is increased over a brief period of time from 7,500 rpm to 15,000 rpm. There was a transient torque on the output shaft reaching a value of about 38 ft-lb. The shaft-speed behavior was nearly identical for both the OCG and DSI transmissions.

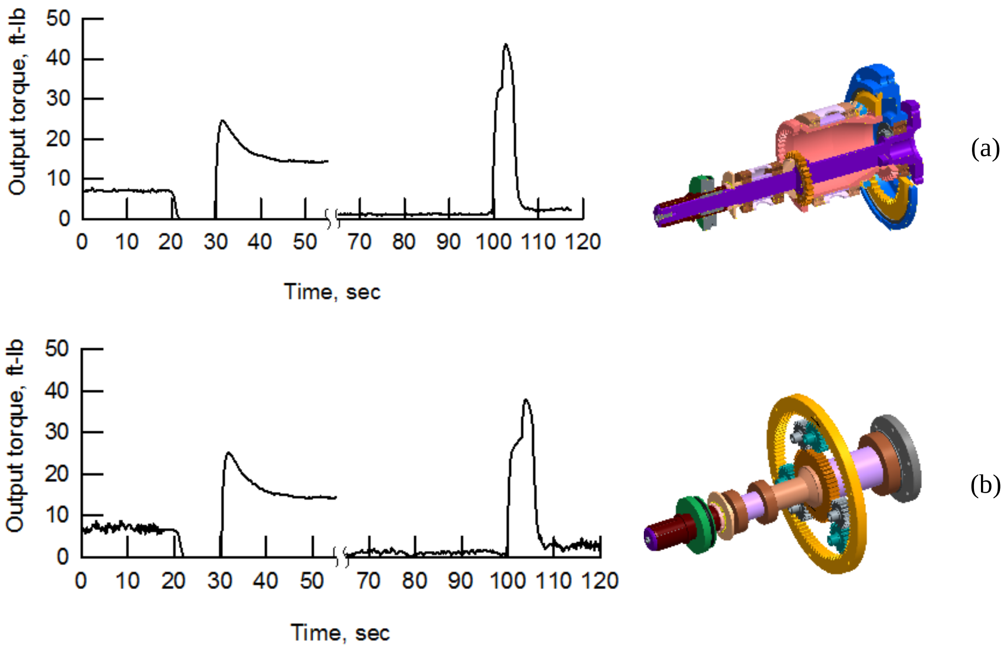


Figure 7. Transmission output torque during transient shift events. (a) Offset compound gear with dry clutch configuration. (b) Dual star idler gear with dry clutch configuration.

Next, the influence of the clutch pressure ramp rates on the output shaft torque will be discussed. The test data are provided in Figure 8. Showing first the downshift event whereby the shaft speed is decreased 50 percent, and because the driven electric motor was controlled for constant power, the torque must increase by a factor of two after completion of the shift event. The event was initiated at approximately time 20 sec, and the output shaft torque first decreases as the sprag clutch begins to overrun while some torque is still transferred through the slipping friction clutch. Note that the output shaft torque never decreases fully to zero torque, a desired and different behavior than that shown in the data of Figure 7 where output shaft torque was zero for some duration. Next we discuss the behavior during upshift speed transition commanded to occur at time approximately 100 sec. The slower pressure ramp rates, progressively slower from Figure 8(a) to 8(b) and again slower to 8(c), resulting in smaller output-shaft transient peak torque values for slower ramp rates. The slower rates allow for some friction clutch slippage increasing the time for the driven electric machine to increase in speed from 7,500 to 15,000 rpm. It is evident that even for the simple clutch-pressure control employed for these tests, open-loop control using linear shaped pressure profiles, the output shaft transient torque peak can be managed. For a vertical lift vehicle the control of the speed-shift transition will have more possibilities for control. This control will be more complicated and likely involving the of engine fuel, engine speed, rotor blade pitch, and clutch pressure in a coordinated manner to be able to have desired outcomes for vehicle speed, altitude, and component loads during the speed-shift transitions. The demonstrator transmissions all successfully completed shaft-speed transitions for 50 percent reduction of the rotor speed, and the data shows that the mechanical behavior can be optimized by careful control of the shifting events.

Research, Develop, and Demonstrate Lightweight and Low-Power-Loss Drivetrain Technologies

The technical challenge set objectives to demonstrate goals for low drivetrain weight and low power loss. Technologies were investigated to support these goals. In the text that follows, the approach and significant results will be presented.

To achieve lowest possible weight for two-speed functionality, components such as gearing and clutches were located in the drive system where rotating speeds are high and torque is relatively low. Since the parts will rotate at high speed, there is the possibility that such designs could have a large windage power loss, depending on design details. Windage loss is produced by high-speed rotating surfaces interacting with an oil-air environment. In recent years, much progress has been made toward using computational fluid dynamics (Ref. 22) and advanced gear tooth meshing models (Ref. 23) to understand gear windage losses. Progress toward fruitful application of such advanced computation methods is perhaps slow because there is a lack of experimental data for verification and validation of these new computation methods. Therefore, in-house research was concentrated on power loss experiments to provided suchvalidation data. Also, work was established with a university with the purpose to further understand the gear windage phenomena analytically. The NASA Glenn Research Center (GRC) experimental work has resulted in recent publications of gear windage data (Refs. 24 and 25). An especially significant finding is shown in the experimental data of Figure 9. The data shows a non-linear increase of windage power loss as a function of surface speed for a pair of meshed gears for three different gear shroud configurations (data plotted using diamond symbols). Also plotted, using circle symbols, are two data points, one each for the gear pair members rotating individually and so without the gear tooth meshing action. An expectation may be that the power loss for meshing gears could be found by

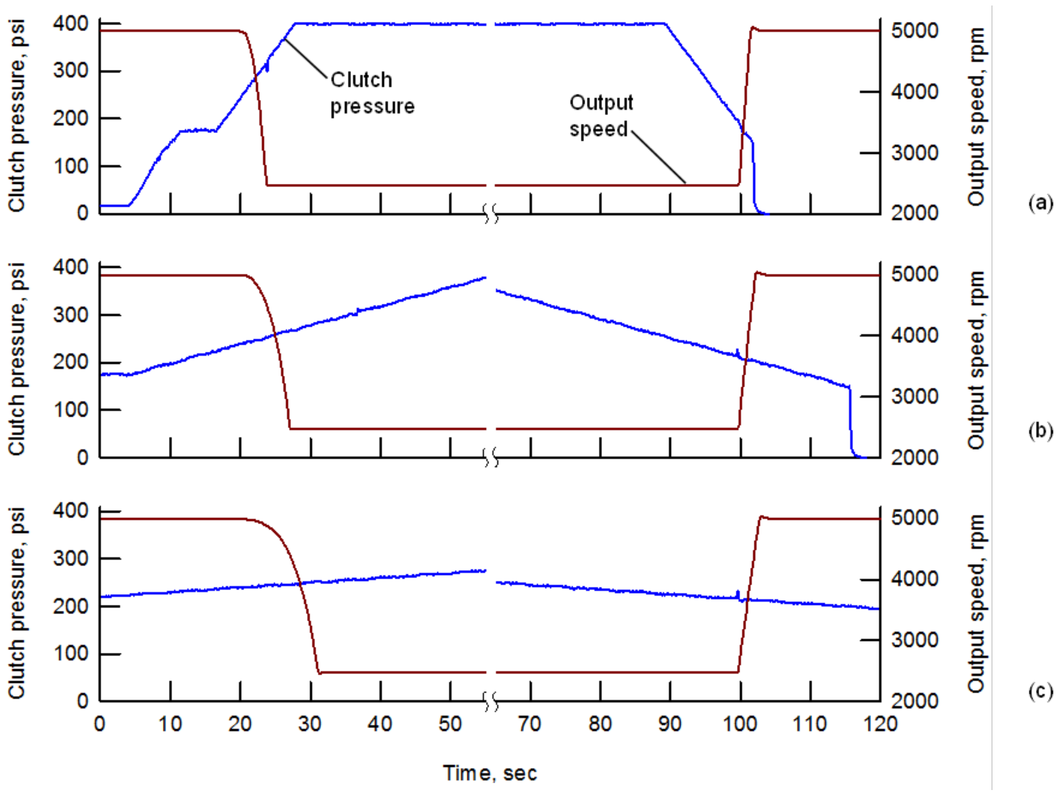


Figure 8. Transmission output speed and clutch pressure for DSI power control shift tests. (a) 20 psi/sec clutch pressure ramp rate. (b) 4 psi/sec clutch pressure ramp rate. (c) 1 psi/sec clutch pressure ramp rate.

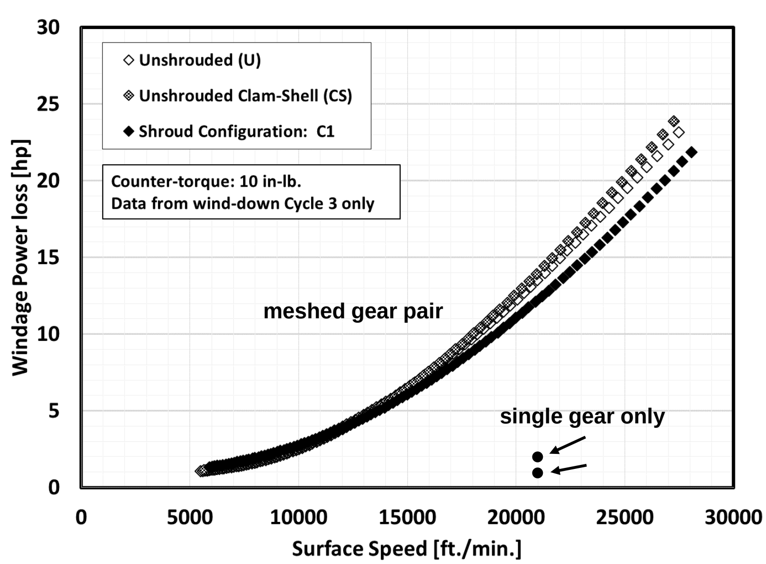


Figure 9. Windage power loss as a function of surface speed for meshed gear pairs and for single gears.

simple addition of the values for each gear operating individually. However this data shows that the gear tooth meshing action is actually the most significant power-loss phenomena, far exceeding the air-stirring action of gear teeth away from the gear-meshing zones. The meshing losses found here (sometimes termed "pocketing" losses in the literature (Ref. 23) have magnitudes approximately six times greater than single gear windage power losses. Proper shrouding of the gears is typically employed to mitigate such windage losses. Novel experiments and computational fluid dynamic analyses help to guide shrouding optimizations. Such findings will be important for optimizing designs to achieve low power loss gear systems.

Next, the research approach and significant results toward reducing the weight of a vertical lift vehicle's drivetrain will be discussed. Three main topics were investigated with aim to reduce drivetrain weight, namely: (1) loss-of-lubrication research aimed toward elimination of auxiliary and emergency lubrication systems, (2) finite-life and flaw-tolerant design methods to be alternatives to the current safe-life design approach, and (c) replacement of heavy steel with composite materials. Some progress was made concerning the first two topics (Refs. 26 to 29). However, based on initial research findings, the most promising approach proved to be the third topic, use of composite materials to replace steel for gear bodies and shafts, and resources became focused toward that topic to support the objective to reduce drive system weight.

The first feasibility demonstration of composite materials for gear bodies made use of an existing all-steel, aerospace quality 3.5 in. pitch diameter gear. The gear was first separated into three pieces, the inner hub that had a keyslot feature for mounting on a test rig, the gear body, and a rim with the involute gear teeth. The steel gear body was discarded and replaced with composite material. The steel parts were machined with hex-shaped features to create geometries able to transfer torque across the steel to composite interfaces. The tested hybrid gear concept is illustrated using an exploded view in Figure 10. The composite material was made using a (0°/+60°/ 60°) braided prepreg and compression molding. The gear body used layers of composite material that were assembled and cured in a specialized fixture. The gear was tested at 10,000 rpm and 87 hp (Figure 11) and survived 1×10^9 cycles with no indications of fatigue or other impending failure modes. This feasibility test demonstrated that the material had sufficient durability at reasonable operating stresses in an aviation gearing environment (i.e. temperatures, vibrations, exposure to lubricating oil).

Having demonstrated the hybrid gear feasibility on a small scale, next the hybrid gear concept was scaled up in size and power. The next demonstration gear made use of an existing ~16.5 in. pitch diameter, aerospace quality test gear having a bolted connection for installation into the test rig. The full-scale, high-power hybrid test gear consisted of three main components: an outer steel adapter, an inner steel adapter, and a composite portion that interfaces with both the inner

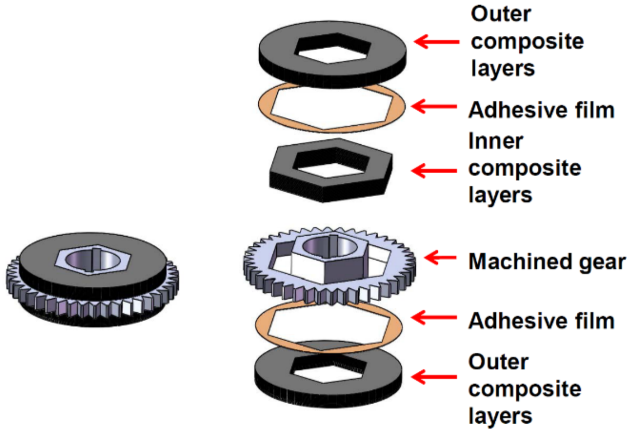


Figure 10. Exploded view of hybrid gear concept.

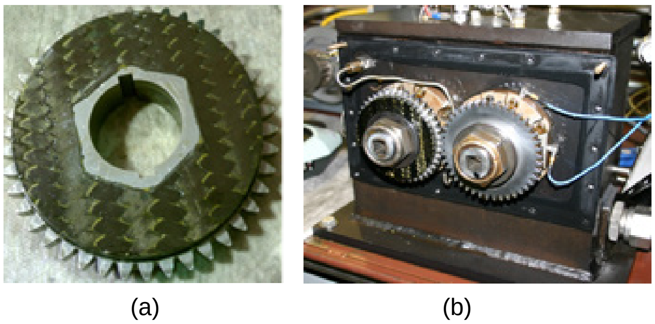


Figure 11. Feasibility demonstration of hybrid gear technology. (a) Hybrid composite gear. (b) Hybrid gear installed onto gear fatigue test rig mated with an all-steel gear.

and outer adapters. The composite portion of the web consists of three sections. The inner section includes an inner and outer sinusoidal lobed pattern used as a torque interlock. This center section is captured on either side with additional composite sections referred to as capture plies which contain the inner section axially and provide an additional bond surface at the steel/composite interfaces perpendicular to the axis of rotation. The test article design and the assembled test article are depicted in Figure 12. A second test article using only steel was also built and tested for performance comparisons.

Experiments were completed in the GRC High-Speed Helical Gear Rig as follows. First the lubrication system was heated to an oil inlet temperature of approximately 120 °F. The temperature was chosen as a conservative starting temperature that would not cause issues with the integrity of the composite and adhesive bond (lubricant exit temperature less than 250 °F). The gearshaft orbit and housing vibration magnitudes were comparable to the steel baseline configuration. There was concern that the addition of the composite webbed bull gear would adversely affect the thermal behavior of the gearbox. This concern was investigated by monitoring the change in oil temperature at the oil exit compared to the oil inlet temperature, and the steel baseline and hybrid gear exhibited comparable performance (Ref. 30). The hybrid bull gear of Figure 11 was successfully tested up to 3,300 hp (2,460 kW). A follow-on effort using a modified composite web was successfully tested to 5,000 hp (3,725 kW). These tests demonstrated the key technologies for a hybrid composite gear to TRL-4.

Power Loss Assessment

One of the demonstrated two-speed drive concepts that successfully achieved the 50 percent rotor speed reduction, the offset compound gear with a dry clutch, was assessed to determine the power loss. The assessment was by a thorough engineering evaluation for operation at 1,000 hp.

The assessment was done primarily by analysis since well-validated analysis procedures have been established for most of the power-loss phenomena. However, two phenomena were identified as unique to the two-speed drive configuration, and as such reliable analysis methods were not available. To assess and quantify the power loss from ring seals of the rotating lubricant feed through and sprag clutches operating in the overrunning condition, dedicated experiments were completed. Typical results of the power loss experiments are provided in Figure 13.

The experimental data of Figure 13 was combined with analysis results. Losses from gear tooth rolling and sliding were calculated using the "Load Distribution Program" (Ref. 31). Gear tooth pocketing loss was calculated by The Ohio State University using the methods of Reference 23. Gear windage losses were calculated using empirically derived methods (Refs. 32 to 34). Rolling element bearing losses were calculated using well established handbook methods and manufacturing specifications.

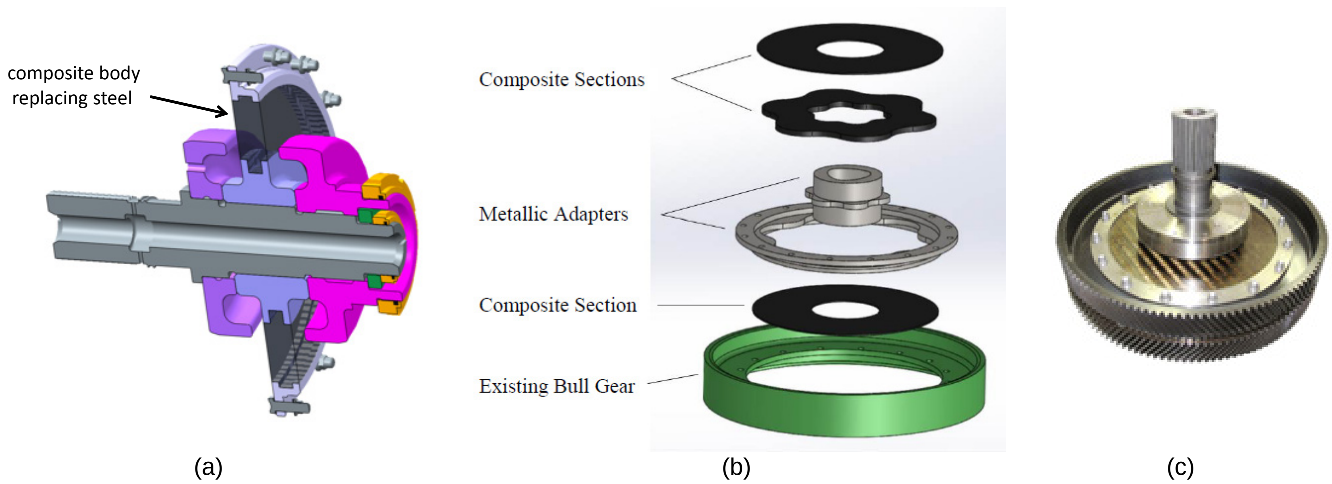


Figure 12. Full scale high-power hybrid composite test gear. (a) Solid model of overall design integration. (b) Details of adapters using lobed torque interlocks. (c) Assembled hybrid gear.

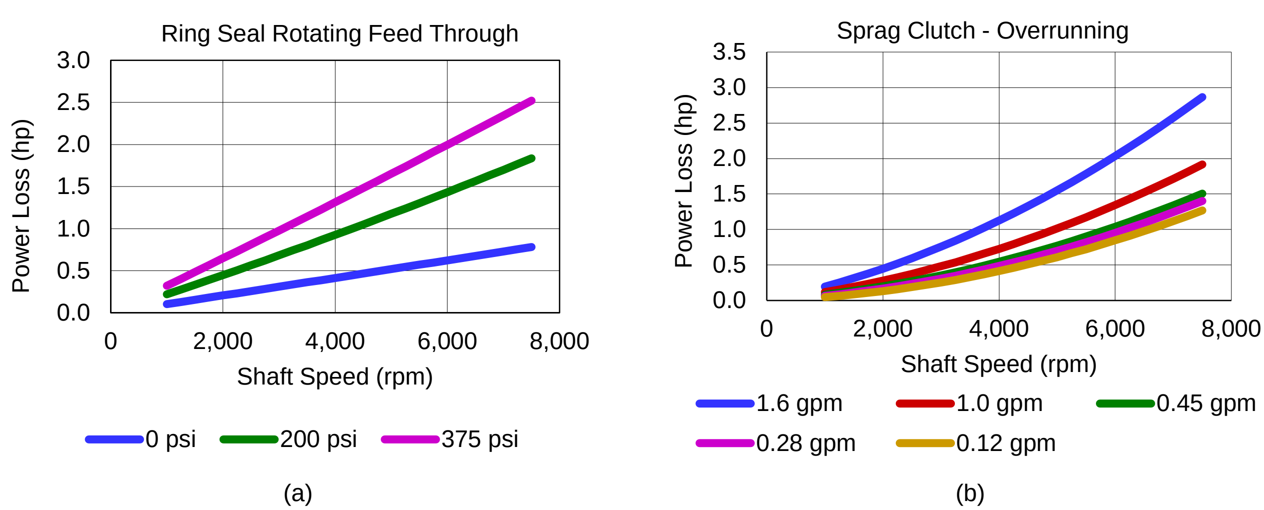


Figure 13. Experimentally measured power loss of components of the two-speed ratio drive. (a) Power loss of ring seals as function of shaft speed for three hydraulic pressures. (b) Power loss of sprag clutches as function of shaft speed for five flow rates of lubricant oil.

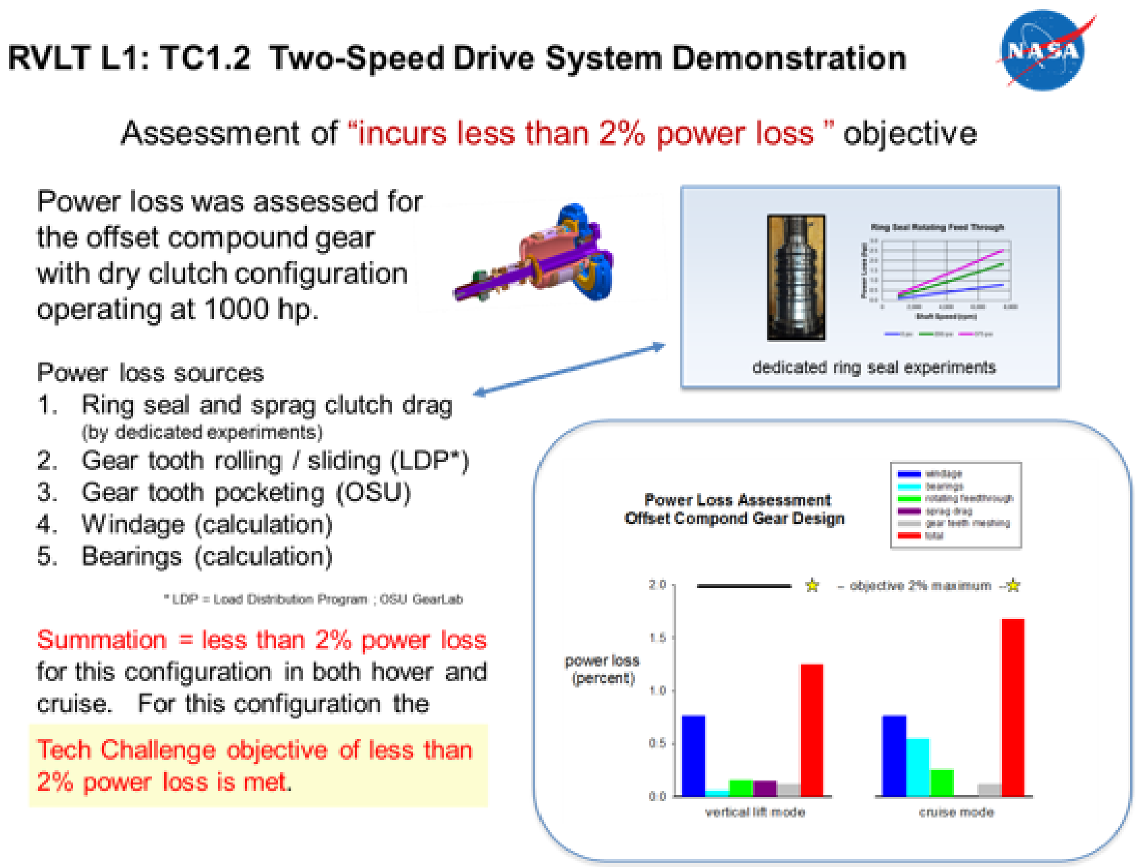


Figure 14. Summary of assessment of power loss objective, approach and results.

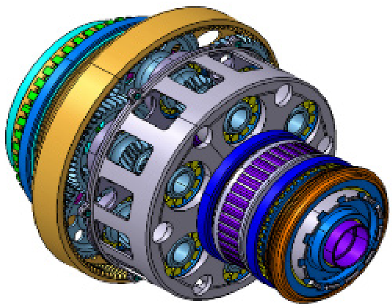
The assessment arrived at the conclusion that the power loss was less than 2 percent for both high-speed main rotor speed (for vertical lift and hover) and for the 50 percent reduced rotor speed (for forward flight), exceeding the technical challenge objective. A key performance requirement of the two-speed drive, less than 2 percent power loss, was demonstrated. The assessment procedure and results are summarized in the briefing chart of Figure 14. The results of the power loss evaluation revealed, as anticipated, that the largest contributor to total power loss for the two-speed drive system was the windage loss created when rotating surfaces interact with the oil-air environment. The windage power loss was determined to be 0.55 percent for both the high-speed and low-speed modes of operation. For certain other phenomena, the power loss values differ for operation in low- or high- speed mode. The total power loss for the offset compound gear and dry clutch configuration operating at 1,000 hp were determined to be 1.3 percent in high-speed mode (hover) and 1.7 percent in low-speed mode (forward flight), exceeding the technical challenge objective of less than 2.0 percent power loss. One can anticipate that an offset compound gear module with a dry clutch, optimized for a flight vehicle, would have an even lower power loss than the demonstrator design. For example, while the demonstrator test article used commercial off-the-shelf bearings, flight hardware would make use of highly optimized, low-power-loss bearings.

Power to Weight Ratio Assessment

The first step of the assessment of this aspect was to determine the added weight to provide the drivetrain with the two-speed function. The two-speed drive module weight was derived from a detailed design including a solid model. The study found that a two-speed drive system using only traditional materials would weigh 572 lb more than a single-speed ratio drive system (Refs. 1, 9 to 12).

Next, the influence of replacing steel with composite materials at selected locations was determined. The study result determined that the weight of the two-speed drive system could be reduced by 633 lb via the use of composite materials. Therefore, the weight is less than that of a single-speed drive using only traditional materials. The results are shown in Figure 15.

The objective to maintain power to weight ratio was met and exceeded, and the key enabling technologies were demonstrated to the required TRL-4. The assessment procedure and results are summarized in the briefing chart of Figure 15.



(a) LCTR2 Two Speed Configuration Module.

Added weight for two-speed (lb)	572
Cross shafting (lb)	–97
Input modules (lb)	–110
Rotor shafts (lb)	–190
Prop rotor gearboxes (lb)	–211
Tilt axis gearboxes (lb)	–17
Midswing gearbox (lb)	–8
Total savings (lb)	**–633**
Net (lb)	**–61**

Overall there is a weight reduction.
Goal has been achieved.

(b) Two Speed Gearbox weight reductions using composites.

Figure 15. Summary of assessment of power to weight results.

CONCLUSIONS

The goals of the two-speed drive technical challenge were achieved. Significant accomplishments and technology developments included the following:

- Demonstrated three mechanical assembles that achieved the required 50 percent reduction of rotor speed. All three mechanical assemblies were experimentally demonstrated to TRL-4 with successful transitions from high-speed (vertical take-off) mode to low-speed (forward flight) mode and then transitioning back to high-speed (vertical landing) mode.
- Demonstrated two-speed drive concepts that successfully achieved the required 50 percent rotor speed reduction, the offset compound gear with a dry clutch, was assessed to determine the power loss. The assessment was done primarily by analysis. The assessment arrived at the conclusion that the power loss was less than 2 percent for both high-speed main rotor speed (for vertical lift and hover) and for the 50 percent reduced rotor speed (for forward flight), exceeding the technical challenge objective. The results of the power loss evaluation revealed that the largest contributor to total power loss for the two-speed drive system was the windage loss created when rotating surfaces interact with the oil-air environment.
- Replacing steel with composite materials results in significant weight savings. By analysis, the two-speed drive system using composite materials at select locations weighs less than constant-speed drive system using only traditional materials.
- The technical objective was met and exceeded, and the key enabling technologies were demonstrated to the required TRL-4.

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