NASA/TM—2009-215456 ARL–TR–4758

Variable/Multispeed Rotorcraft Drive System Concepts

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March 2009

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Level of Review: This material has been technically reviewed by technical management.

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Abstract

Several recent studies for advanced rotorcraft have identified the need for variable, or multispeed-capable rotors. A speed change of up to 50 percent has been proposed for future rotorcraft to improve vehicle performance. Varying rotor speed during flight not only requires a rotor capable of performing effectively over the extended operation speed and load range, but also requires an advanced propulsion system to provide the required speed changes. A study has been completed, which investigated possible drive system arrangements to accommodate up to the 50 percent speed change. These concepts are presented. The most promising configurations are identified and will be developed for future validation testing.

List of Acronyms

CAD	computer-aided design
CV	continuously variable
CVT	continuously variable transmission
SRW	Subsonic Rotary Wing

Background and Introduction

Rotorcraft propulsion is a critical part of the overall aircraft. Unlike fixed wing aircraft, the rotor and propulsion system provides lift and control as well as forward thrust. As a result, the rotorcraft engine and 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, thus limiting speed changes to approximately 15 percent maximum.

The recent NASA Heavy Lift Study (Ref. 1) has shown that variable-speed propulsion was necessary for all aircraft concepts studied. Variable-speed propulsion, without loss of efficiency and torque, is necessary to permit a high-speed operation with reduced noise. Previous NASA variable-speed transmission studies concentrated on 15 percent speed changes (refs. 2 and 3). The NASA Heavy Lift Study (ref. 1) suggests that increased speed variations of 50 percent will have a dramatic effect on reducing external noise while increasing rotorcraft performance.

To achieve this large speed variation capability, advanced variable/multispeed drive system concepts must be developed. This report summarizes an effort to identify viable concepts for both a two- and variable-speed drive transmission configuration. Efforts will culminate with laboratory testing of a reduced-scale variable/multispeed drive system to validate system level tools and concepts.

Study Objectives

This paper summarizes a concept study directed at identifying and creating multiple concepts and selecting the most viable concept(s) for future development and scale-model testing of a variable/multispeed drive for application to rotary wing aircraft. Both discrete two-speed and variable-speed configurations are considered. The primary requirement for this study is to identify and create the most viable concept for a transmission with a high-range ratio of 1:1 and a low-range reduction ratio of 2:1. The focus is specifically on identifying the most suitable mechanical drive configuration(s). Control of such transmissions in the overall rotorcraft driveline and system dynamics are beyond the scope of this paper and have been ignored. Concepts created and described in this paper are not supported by rigorous engineering analysis. This study concentrates on the creation of multiple preliminary concepts rather than maturing the design of a single concept.

Concept Creation and Development

A chronology of the concept creation, development, and directions is described below.

While all concepts created and discussed in this paper may not be purely original, they are based on basic configurations outside of the area of application with the primary goal to meet the design requirements. This approach is in contrast to developing concepts based on something existing from within the area of application and either modifying or enhancing the configuration. A goal of this approach was to search for something new. Concepts discussed in this paper were initially created and evolved individually. Once the barrier of conceiving an initial concept was overcome, additional concepts were more easily conceived. A few of the initial concepts included shortcomings resulting in a need to pursue resolution to the particular issue(s). An example of one such shortcoming is reverse output rotation, and will be overviewed and presented in further detail in the appendix. Resolution of shortcomings became an impetus for creativity. On several occasions, resolution to a concept-specific issue was introduced into other concepts, thus improving two or more concepts. Later, some concepts were developed in parallel as idiosyncrasies or improvements of a given concept came to light, which had direct application to earlier concepts sharing something in common. Through brainstorming and simultaneous concept development a welcomed synergy was realized. Rather than finish one concept, any new concept thoughts were immediately committed to sketch to preclude loss. This resulted in some parallel concept development and creation of a broader scope of concepts than planned.

Another initial goal was that any concept(s) to be created should be original. This led to review of some textbooks in the area of gearing (refs. 4 to 6) for basic ideas. Concept creation was initially directed at a variable-speed drive, being more desirable than a multispeed drive for the intended application. A variety of variable-speed devices were reviewed, though none stood out as being ideally suited. All configurations employed a traction or friction drive, or comprised components such as belts and pulleys, seemingly unsuited for the power and speeds associated with this application. The realization of a true variable-drive basis (traction or friction drive via variable geometry) would not suffice, but perhaps variable ratio could be synthesized with gearing.

An initial variable-drive concept emerged based upon a two-engine-driven planetary differential (sun-in, ring-in, and carrier-out). The configuration was thought to be adaptable to a scheme using one engine (input), with the second engine (input) replaced by an external speed controller device. With this configuration, the carrier (output) is continuously variable (CV). The concept did not appear to have much merit and was rejected at first. Additional factors discounting this concept were it was not a classic CV drive per se, having no contour surfaces, nor variable geometry, and a second external input was surely a tremendous weight liability. However, the basis was retained as a remote possibility since it was variable-speed output and positive gear drive.

Next, a schematic for an inline discrete two-speed transmission configuration was conceived and evolved into a CAD (computer-aided design) layout. The concept included a clutch, thus requiring an immediate need for a representative scale clutch. Since any conventional two-speed transmission configuration intended to change ratio during power transfer requires a means of disengaging power during the ratio change, conceptualization focused toward a clutch as it would be needed for other concepts. A concept clutch was needed. A dry multidisc automotive racing clutch was considered as a basis. The design aspect to be capitalized upon is the small-diameter multiple drive plates and high-power density compared to that of an equivalent capacity larger diameter single-plate design. Bossler (ref. 3) describes a similar multiplate clutch concept for rotorcraft application also based on the multiplate automotive racing clutch. The above clutch concept is a hydraulic over mechanical spring design, fail-safe to the mechanical spring(s) load. Hydraulic actuation seemed preferable over a mechanical linkage as being readily adaptable to solenoid control. Concept development would also conclude hydraulic actuation be the exclusive method as the clutch is enclosed by rotating gear elements. Based on the above, a hydraulic over mechanical spring clutch was configured for this study. Several clutch concepts were evolved to meet specific design requirements for various drive concepts and are all discussed in “Clutch Concepts” in the appendix.

During development of the initial inline two-speed transmission, the intended application for the concept clutch, a major shortcoming relative to the output direction of rotation between the two ratios emerged. The drive concept employs a planetary gear system with fixed star gears, sun gear input, and ring gear output. Powering in the low-speed range causes the ring gear to rotate opposite to the sun gear, and opposite to the high-speed range output, directly coupled via the clutch. To make this concept viable, an immediate challenge was to determine the best approach to reverse the output rotation for the low range. This example shortcoming, mentioned at the beginning of this section, is significant in that it became an impetus for the creation of several reversing concepts for this specific concept, as well as being applied to various other concepts requiring a rotational reversing function. In addition, the search for a suitable resolution to this shortcoming led directly to one of the final selected concepts featuring a novel gearing configuration (discussed later). Reverse rotation concepts are discussed in the appendix.

Though a specific candidate CV drive was not identified as ideally suited for an overall drive for rotorcraft, the idea of CV is highly desirable contrasted to the operability of a discrete multispeed drive. The dichotomy of CV in contrast to direct positive gear drive is a design dilemma of which avenue to pursue. At this point, focus shifted to an idea of attempting to capitalize on the desirable aspects of a CV element by synthesizing CV with a variable direct mechanical drive such as a differential or planetary differential gear system with the addition of some to-be-determined outside control device, which is analogous to the initial two-input planetary differential concept nearly dismissed. Variable control can potentially be either externally powered or internally takeoff driven. An internal takeoff-driven CV drive was given attention being perceived to be the lighter of the two options and a benefit of being driven from the existing transmission input shaft.

Another possible direction stemming from the above dichotomy of continuous variability contrasted to positive gear drive is that a two-speed transmission with a CV speed-matching element might be a viable operational configuration for this application. The above configuration overcomes some of the undesirable aspects of both the discrete two-speed drive and the variable-speed drive while capitalizing on the positive aspects from both configurations. In addition, it features positive drive for the two primary operating modes: hover and cruise. Several concepts were created for this direction.

Although power transmission via traction fluids is not perceived as desirable for quasi-full-time operation in either the high- or low-speed range, it may be possible for a CV element to take on full power requirements for a short duration, or perhaps the lesser power requirement of matching speeds, during the speed range transition. This is particularly true for the high-low speed change when the rotorcraft is transitioned from hover to cruise, exploiting fixed wing lift, and full engine power is not required. Conversely, the low to high speed change, from cruise to hover, is the more demanding transition since fixed wing lift is either in a state of quickly diminishing significance or does not exist at all. In addition, an emergency maneuver would require full power during this transition and must be readily available on demand thus requiring the drive to handle full power.

Another application in which a CV element may be applicable is in a split-power transmission configuration where the CV element is used to transmit a portion of the overall required power. During the course of this study, CV elements of both full- and half-toroidal configurations were considered as a possible speed variator. The selection and design of the specific CV element within the various concepts is to be addressed during the next stage of development for any of the concepts employing a CV element. The design of such a device would be a formidable task in itself as evidenced by the number of configurations and contributors pursuing their development. The toroidal variator configuration is thought to be the best for this application. Based on (ref. 7), a concept toroidal geometry traction drive was developed, which includes a reversing subconcept. The concept is discussed in the "Concepts Descriptions" section.

An alternative to using a CV element as a variator may be the identification of a speed controller to meet the specific requirements. The specific configuration and application will require review to determine if a suitable candidate can be identified. The speed-controlling device depicted in some of the concepts may be changed from that depicted to one of the other viable devices offered. This controller modularity concept will be capitalized upon in the "Recommendations" section.

System Considerations

Dynamics and Operation of Two-Speed Transmission Concepts

Manual shifting a transmission in a rotorcraft during flight is seemingly perilous whether from hover to cruise or the converse. For these two ratio transitions, the degree of difficulty is different, as well as a function of the urgency of the maneuver (i.e., a planned routine operation or an emergency condition). Initiating change in any situation should be equally easy and of second nature to the operator (pilot) or must be automated.

Down shifting a two-speed transmission from hover to cruise mode appears the easier transition since rotorcraft speed is nearing the point where the fixed wing takes over the function of providing lift and the engine or driveline only has to provide forward thrust. One can envision slowing the engine to a lower power level and shifting the transmission, followed by an increase in, or resuming, engine speed, though at a lower power level.

In contrast, the most difficult ratio change is the transition from cruise to hover. During hover, the rotor must provide the functions of lift, thrust, and maneuver. At cruise, with the engine at 100 percent speed and the transmission at 50 percent speed output (low range 2:1), the required operation is to affect a ratio change to 1:1, resulting in the engine at 100 percent speed and the transmission output speed also at 100 percent (high ratio 1:1). This above transition requires either reducing engine speed and torque prior to shifting or somehow increasing the speed of the transmission output shaft while maintaining engine speed. Without some powered means, to accomplish this one must realize the ratio change and slip the clutch to increase the output speed in a smooth manner while generating of heat and parts wear.

One approach offered (ref. 3) is an operational scheme for a twin rotor application that transitions power from both engines to a single engine while shifting one transmission. Note that the scheme is for significantly less ratio change (i.e., less change in ratio than 50 percent for this study). The above transition scheme still involves slipping clutches, though less severe due to the smaller ratio change. This scheme does not seem to be a desirable option for the cruise to hover transition for the ratio change in this study.

Returning to the basic issue of mismatched shaft speeds encountered during the process of shifting ratios, the following two scenarios are offered for consideration as to achieving the cruise to hover ratio change manually (though they are not applicable due to magnitude of the required engine speed changes to be realized for this study):

(1) Disengage clutch while reducing engine speed 50 percent, shift transmission to high range (hover), reengage clutch, and return engine speed back to 100 percent, thus increasing transmission output speed to that required to hover (i.e., 100 percent speed).

(2) Disengage clutch while maintaining engine speed (with reduced load), shift transmission to high range (hover), and slowly reengage (slip) clutch to increase output speed to 100 percent, resulting in generation of heat and component wear.

Clearly, the toughest challenge both in design and in the operation of a discrete two-speed transmission for rotorcraft application is that of powering though the ratio change required to transition from cruise to hover and shaft speed synchronization. With airframe speed being reduced and fixed-wing lift diminishing to negligible magnitude, during this change, the rotor must take over the tasks of providing both thrust and lift. This requires full power and speed, a significantly increased speed (i.e., from 50 to 100 percent output speed). The degree of difficulty in making the above transition may be further aggravated for both the machine and the operator or pilot in situations of emergency.

Anyone that has experienced being in the midst of manually shifting an up-ratio gear change in an automobile when an unforeseen situation of urgency required an immediate down shift to a lower ratio for immediate acceleration will be aware of either the extreme time lag and/or extreme driveline shock that can result. Lag being attributed to operator and/or mechanism reaction time period and shock due to unsynchronized clutch engagement resulting from distressed operator reaction.

Clearly, such operational schemes take too long to execute and not tolerable in an aircraft application. Controlled speed ratio changes are highly desirable, if not absolutely required. If a discrete two-speed transmission is truly to be employed for this application, system control will need to be intensely studied to verify the suitability.

The probable best approach for using a CV element as a ratio-changing device in a two-speed drive might be to only use the CV element for up shifting (accelerate rotor and shaft speed matching) and to use the clutch for down shifting. Operation in this manner would serve to reduce the required service duty of both the CV element and the clutch.

Dynamics and Operation of Variable-Speed Transmission Concepts

As stated earlier, the controllability aspect of discrete versus continuous variability is a significant design consideration. Each configuration has both merits and liabilities. The discrete ratio drive is the most straight forward, reliable, cost effective (initial manufacture), and can be based on current design methodologies and state-of-the-art technologies. However, from a perspective of application to flight, the two-speed drive is less desirable than a drive with the ability to provide continuous variable output speed.

With a CV drive, engine speed can be maintained nearly constant while the output speed is decreased based on power demand. A big advantage is that the ratio could continue to vary in the given direction or, if emergency situation warranted, the ratio could be reversed and varied toward the opposite direction in a smooth and continuous manner without any abrupt changes in torque or speed. With a CV drive, the period, or time, to execute a speed change is dramatically more flexible than that of a discrete two-speed drive where smoothness of the speed change is a strong function of the transition period. Although increasing the period would tend to improve smoothness, it must be done quickly to maintain forward velocity with minimal internal heat generation.

Should an unforeseen emergency condition arise during an in-progress speed range change, the speed change may be required to return quickly to the original setting. Such a scenario might be transitioning from hover to cruise during which time the aircraft encounters an emergency condition requiring conversion quickly back to hover. This situation would result in an abrupt change in torque transfer or mismatch in speed, which also results in an abrupt loading or unloading condition. The condition is created in the discrete two-speed drive because something mechanical must be engaged or disengaged to initiate or permit the change in ratio and something must be synchronized to continue power transfer. This situation is in direct contrast to normal operation striving to obtain a smooth transition. Any event that disrupts the disengagement-reengagement periods would undoubtedly result in an abrupt change in power flow.

Based upon the above, the system dynamics of speed changes within a fixed multispeed transmission is a significant concern based on the high speeds and horsepower being transmitted and the potential resultant shock loads that may occur due to unsynchronized speeds during the transitions to and from the discrete speed ranges. At high power and speed, even a small speed mismatch can introduce significant shock loads within the driveline as well as the engine(s) and/or rotor(s).

A CV drive features highly desirable fully synchronized output speeds throughout the speed range resulting in the smoothest range changes with little or no driveline shock due to speed mismatch between the driving and driven elements. However, in a CV traction drive configuration, any realized speed mismatch could result in internal slippage and heat generation within the traction fluid and drive surfaces due to the huge inertia of the rotors. Traction fluid properties are generally highly dependent upon fluid temperature with any significant heat rise resulting in unstable fluid properties as well as low reliability in the traction coefficient (ref. 9). This has the potential to deteriorate to the point of total instability and loss of traction. While not desirable for any power drive, this could be catastrophic on an airframe. For the above reasons, full power transmission through fluid traction and friction is not foreseen as viable for the rotary wing applications.

Seemingly intuitive, a discrete multispeed transmission is not the best for the given application, nor would a pure fully continuously variable transmission (CVT) utilizing power transmission via fluid traction, for which the performance is so tightly coupled to a fluid and its temperature. In addition, the majority of any mission probably comprises operation primarily in either the upper or lower speed range, not transitioning between them.

The probable best approach for this application is a perceived split-power transmission with a CV element, or variator, operating between two fixed ratio speed ranges.

Concept Studies

Concepts created for this study include both two-speed and CV drives. The concepts were reviewed and ranked as discussed in the next few sections. Several concepts, or variations thereof, were generated for each of the above directions. Only overviews of the top candidates for each category are presented below.

Concept Descriptions

Concept descriptions for all clutches, rotation reversal schemes, two-speed drives, and variable-speed drives created during this study are presented in the appendix, as well as some concepts and designs created by others. Figures are included for each concept. Some concepts include a number of variations on the basic concept. Original concepts are in the same scale (detail gearing analysis may indicate otherwise) and employ many standard configurations and representative bearing and shaft sizes. Original concepts depict gear pitch diameters based on minimum diametral pitch of 12 and a pressure angle of 25° (ref. 3) and are limited to consideration of spur tooth, helical, or double-helical gears.

Concepts are created in two-dimensional CAD using a commercially available drafting design tool (ref. 11). CAD is used to provide a more realistic image of the concept basis and maintain consistency of scale in contrast to schematic representations, which do not bring out details such as the introduction of lubrication, sealing, and assembly and mounting features as readily as CAD representations.

Assessment of Concepts and Selection Process

Concepts developed during this study, as well as some by others, were reviewed to identify those meriting further development. A ranking process was initially planned that included specific metrics against which the concepts would be rated. During selection discussions, several points were recognized within the concepts, evaluation metrics, and selection process, which resulted in an evolution of the evaluation metrics that resulted in a favorable strategy for development and testing and is discussed below.

Reviewing the concepts from a top-level perspective, concepts fall into three groups:

(1) Inline discrete two-speed configurations
(2) Dual-input planetary differential configurations
(3) Variable-speed multishaft split-power configurations

From the above groups, a very distinct demarcation of simplicity versus complexity emerges. The simplicity of the discrete two-speed drives and planetary differentials is contrasted to the complexity of the multishaft split-power and variable-speed drives. Complex concepts possess an obvious increase in the number of gears, shafts, bearings, along with associated weight gains. Also, there are increased manufacturing and maintainability costs and lower efficiency due to increased gear meshes and surface rotational area. In addition, design complexities increase difficulty in assessing overall concept merit.

In contrast, simple two-speed configurations are the lightest, cheapest, and easiest to manufacture, assemble, and maintain; although, not necessarily the overall best for the application. A simple design with well developed configurations having field service heritage can be easily made to be more robust with inherent high reliability. In a good design, simplicity and robustness led to other desirable traits such as high reliability and high efficiency. Though a few inferences in the above are thought to be debatable by some, they should be considered as trends, not statements, of design laws or rules.

Two of the several two-speed configurations rose to the top as being the best based on simplicity. As eluded to in the "Concept Development" section, the concepts created are readily evolvable due to the manner in which conceptual work was synergistically created through simultaneous concept creation. It was observed that any of the two-speed concepts can be extended into quasi-variable-speed configurations with the addition of a parallel controller and variator. The function of controller device would take on somewhat different rolls within the various concepts with differing power and speed requirements. However, the basic theme of adding variability to a discrete two-speed device is the key idea to be capitalized upon. This concept will be later exploited.

Concept Selection

Group discussion led to a plan that ranked the concepts based primarily on the "simplicity" metric. This plan will advance consideration and design of multiple- and variable-speed transmissions for rotary wing application through the development and testing of multiple test articles, applicable to a broader range of aircraft and providing the availability of the results, through a wider array of study than initially thought possible.

The resulting plan is to proceed with parallel development of both discrete two- and variable-speed adaptable configurations, which are based on basic simple design concepts along with a second parallel study and development of controller and variator devices able to be incorporated into the above driveline concepts. This assures that the outcome will meet the NASA Subsonic Rotary Wing (SRW) requirements. The development and test plan is as follows.

Strategy Resulting From the Selection Process

(1) Develop the two simplest two-speed configurations and test. Test results will lead to identification of the best overall configuration with respect to a wider range of metrics. Evaluations will be measured against the scale test models and actual hands-on experience rather than paper studies and intuition.

(2) Develop off-shoot variations to the above discrete two-speed configurations, which are variable-output capable (meaning able to accept a modular controller and variator device).

(3) Identify, develop, and test several concepts of variable control devices both takeoff driven (variator) and externally powered (controller). Apply both in the same applications for direct comparison.

Test articles will be of modular configurations, which can be integrated into the above discrete two-speed transmission configurations.

(4) Test the basic two-input differential planetary configuration to determine power levels required for a 100 to 50 percent speed range and compare with the two-speed variable input-capable configurations to identify the best device to develop as the end configuration.

(5) Combine the results from items 1 to 4 above into the overall best configuration and test. The outcome of these development steps is intended to yield both a discrete two-speed and a variable-speed configuration either of which can be the basis for incorporation into specific airframe applications based upon prevailing overall system requirements.

Selected concepts and overall implementation plans are presented in the "Results" section.

Conclusions

Conclusions—Two-Speed Transmission Concepts

Initiating and executing a required ratio change within a discrete two-speed transmission in an aircraft application is not the same as for most other transmission applications. For rotorcraft application, power transmission must be smooth and continuous, at high output power levels, and relatively high rotational speeds.

The toughest challenge both in design and in the operation of a discrete two-speed transmission for the intended application is that of powering through the ratio change required to transition from cruise to hover and maintaining shaft speed synchronization between input and output. With airframe speed being reduced and fixed-wing lift diminishing to negligible magnitude during this change, the rotor must take over the tasks of providing both thrust and lift. This requires full power and speed, a significantly increased speed (i.e., from 50 to 100 percent output speed). The degree of difficulty in making the above transition may be further aggravated for both the machine and the operator or pilot in emergency situations.

For the two-speed with CV shift assist concepts developed for this study, it seems that the best approach for using a CV element in this capacity as a synchronization device would be to only use the CV element for up shifting (cruise to hover, speeding up the rotor) and to use the clutch for down shifting (hover to cruise, slow down rotor). Operation of the CV element and clutch in this manner would serve to reduce the required service duty of both the CV element and the clutch resulting in maximization of service life.

Conclusions—Variable-Speed Transmission Concepts

Shifting between discrete gear ratios is not a desirable flight operation. Power must be smooth, continuous, and performed at both high power levels and relatively high speeds. From a controls perspective, a CV drive is better suited to rotorcraft application than a discrete two-speed drive. It is evident that performance of a traction drive device of any configuration is going to be highly, if not solely, dependent on the specified traction fluid (refs. 9 and 10), which is likely to be susceptible to degradation in performance with increased temperature. Slippage, causing local friction heating, may impact performance or even basic operation of the device. Thermal and flow management of the traction fluid is potentially a tricky area that would most likely require a significant amount of additional supporting hardware (added weight) to maintain fluid temperature. The tradeoff may not be as attractive as a discrete speed transmission without these idiosyncrasies. For the above reasons, full power transmission through fluid traction or friction is not foreseen as viable for the rotary wing applications.

Another shortcoming of attempting to employ the CV toroidal drive concept as a variator in a split-power drive configuration is that output rotation is reverse of input, requiring a reversing stage of some sort to match output to input rotation. Output of the CV element must mesh with the remainder of the split-power drive. The required reversing stage is a substantial weight penalty for no other benefit than to correct the reversal shortcoming. Although four conventional reversing approaches utilizing gearing were considered for the toroidal variator concept, none were suited in creating an overall lightweight variator.

A concept that synthesizes continuous variability through employment of differential or epicyclical gearing achieves the desirable aspects of a CV element with the benefits of positive mechanical drive while shedding the unfavorable aspects of the traction drive. Within this task two versions surfaced to the top, the differential drive and the planetary (epicyclic) drive. Based upon this study, the selected approach to achieve a positive-driven variable output with the highest degree of reliability and robustness is determined to be a configuration that exploits basic two-input planetary differential gearing.

Recommendations

Based on the conceptual configurations included in this study and tradeoff comparisons made, the following configurations are identified to be the best for further consideration.

Two-Speed Transmission Configurations

Concept 2A, "Inline Two-Speed With Double Star/Idler Reversing Stage," and concept 3, "Offset Compound Gear," are selected as being the simplest and possessing the potential for the highest degree of reliability of the configurations considered. The influence of two basic metrics, simplicity and reliability, are indicative of a configuration possessing highly desirable traits such as lightest weight, easiest to manufacture and maintain, lowest cost, and most robust and can also be indicative of a high-efficiency device based on minimization of losses due to bearing and gear friction, windage, and rotating and static mass. Design and testing the above two concepts will lead to the best two-speed drive.

Variable-Speed Transmission Configurations

Concept 5/5A, "Planetary Differential With Variable Controlled Ring Gear," is selected as the simplest and most viable of the variable-speed drive configurations considered. The basic approach to obtain variable output through the use of a positive drive two-input differential will be exploited. In addition, each of the above selected two-speed drive configurations has the potential to be a variable-speed drive configuration when used in conjunction with a parallel path variator and controller device, thus creating a high-reliability variable-speed drive without relying on traction drive (fluid friction).

Summary

Two inline two-speed drive configurations will be pursued further, the offset compound gear drive, and the double star/idler (reversing stage) drive. These configurations will be developed both as discrete two-speed configurations and as variable output speed capable, meaning able to accept a modular controller or variator. In addition, a planetary differential (carrier output) will be investigated relative to power required to vary and control ring gear speed, in effect providing CV carrier output speed with minimized input power. The above will be considered both as an individual research area (i.e., dual-input planetary differential concept) and also with consideration as being applied to the two-speed configurations as either a variator or speed controller. Testing will enable determination of the required power for either a controller or variator device and lead to a selection of the configuration with the highest viability (i.e., power takeoff driven or externally powered). Together this group of research test articles will lead to future development of both a two-speed drive configuration and a positive drive variable-speed drive configuration. For detailed descriptions of the concepts and figures, see the appendix.

Test Rig Demonstration

A scale model transmission rig test is currently being designed to test the mechanical portion of the above recommended candidate transmission configurations. Overall control system design is beyond the scope of the above. Rig test demonstration will be primarily manually controlled by the test cell operator and research team. This is well suited to test off-design and abrupt transient loading and speed variations. The adaptable test rig configuration will be developed on which a planetary differential gear system can be tested with focus on evaluating requirements for the second input, be it a controller, synchronizer, or CV traction drive. The configuration will be tested measuring the speed and power required of a second motor. Test objective is to determine how power requirements to control the output ratio change and maintain the speed ranges over a mission scenario for a given input power. By varying the second motor power and speed, controller rating requirements can be established. This will permit the variable-speed configurations to evolve in the direction of including either an internally driven variable-speed device powered by a gear takeoff on the input end of the transmission, or a separate externally powered speed controlling device. NASA Glenn is currently designing the shared test facility to be used for testing the selected conceptual test articles as well as being adaptable to test others. It is anticipated that the selected concepts will be developed and tested in the 2010 to 2011 timeframe.

References

1. Johnson, W.; Yamauchi, G.K.; and Watts, M.E.: NASA Heavy Lift Study. NASA/TP—2005-213467, 2005.
2. Kish. J.: Vertical Lift Drive System Concept Studies Variable Speed/Two-Speed Transmissions. NASA/CR—2002-211564 (ARL–CR–495), 2002.
3. Bossler, R.B., Jr.: Vertical Lift Drive System Concepts Studies. NASA/CR—2002-211563, 2002.
4. Dudley, Darle W.: Gear Handbook—The Design, Manufacture, and Application of Gears. First Edition, McGraw-Hill Book Company, 1962.
5. Dudley, Darle W.: Handbook of Practical Gear Design (Revised Edition of Practical Gear Design First Edition, 1954). McGraw-Hill Book Company, 1984.
6. Sclater, N.; and Chironis, N.P.: Mechanisms and Mechanical Devices Source Book. McGraw-Hill, Third Edition, 2001.
7. Dodge, A.Y.: Variable Speed Transmission. U.S. Patent 2,164,504, Serial No. 29,797, July 05, 1935.
8. Goi, Tatsuhiko: Variable Rotor Speed Transmission With High Speed Traction Drive. Proceedings of the 55th AHS Annual Forum, May 1999.
9. Raynard, A.E.; Kraus, J.H.; and Bell, D.D.: Design of Toroidal Traction CVT for Electric Vehicles Final Report. NASA CR−159803 (DOE/NASA/0117−80/1, AiResearch 80−16762), 1980.
10. Loewenthal, S.H.; Anderson, N.E.; and Rohn, D.A.: Advances in Traction Drive Technology. NASA TM−83397 (AVRADCOM Technical Report 83−C−5), 1983.
11. AutoCAD 2002.

Appendix—Concept Descriptions and Figures

Designation	Concept Title and Description
Component Concepts Created and Used in Driveline Concepts	
Clutches	
0	Clutch, Four-Plate (Hydraulic over Mechanical Spring)
	Clutch, Eight-Plate Dual-Acting (Hydraulic over Mechanical Spring)
Rotation Reversal	
00	Reversing, Planetary—Double Star-Idler
	Reversing, Idler Train
	Reversing, Opposing Ring Gear and Idler (Face or Bevel Gear)
Driveline Concepts	
Differential Transmission	
1	Dual-Input Planetary Differential Drive (Redundant Drive Concept)
Planetary Transmission	
2	Inline Two-Speed Planetary
2A	Inline Two-Speed Planetary (Modified—Double Star/Idler)
2B	Two-Stage Counter-Rotating Planetary
2C	Inline Two-Speed Planetary (Modified—Opposing Ring Gear and Idlers)
Novel Transmission	
3	Two-Stage Offset Compound Gear
Double Clutch Transmission	
4A	Double Clutch Drive (Variable Control—Alternate Version)
4B	Double Clutch Drive (Variable Control—Original Basis)
Differential—Variable Transmission	
5	Differential Drive
5A	Differential Drive (Co-Rotating and Counter-Rotating)
5B	Differential Drive (Variable Control and Clutch)
6	Rotorcraft Two-Speed Drive (Compound Gear)
Drive Concepts by Others	
7	Bossler Two-Speed Planetary (ref. 3)
CV Variator Concept	
999	Torus (Toroidal) Traction Drive
Other—Planetary Differential Variator	
999G	Variator (half-toroidal) Planetary Differential (Goi)

Clutch Concepts

Clutch Concepts 0

This section describes several clutch configurations developed for various driveline concepts requiring a clutch. A single-output element and two dual-output element clutch configurations are described.

A disengaging and engaging device, or clutch, is needed to select between two ratios or load paths. While a clutch permits the selection between load paths, it does not synchronize shaft speeds for a smooth transition. The synchronization function must be provided by other methods, either mechanical or operational. Methods and devices for synchronization are discussed elsewhere.

Description of Conceptual Four-Plate Clutch

A concept for a clutch was created based on multiplate clutches as employed in high-performance automotive racing applications. The multiplate configuration permits increased torque capacity without increasing the overall diameter of the unit. The configuration was further enhanced based on a proposed design described by Bossler in reference 3. The aforementioned device is a hydraulic over mechanical spring configuration with the failure mode for this configuration being fail-safe to the mechanical spring load. A failure, or loss of hydraulic signal, is determined best to fail-safe to the high range or cruise mode (i.e., loss of hydraulic pressure would engage the clutch resulting in the high-range power path). This is also the logic of the unit described in reference 3.

The Bossler concept was reviewed and an enhancement was incorporated relocating the introduction of the hydraulic release signal from passing through a face seal arrangement to the lowest practical radial location by relocating the hydraulic signal passage transition between the stationary and rotating components. Sealing between the stationary and rotating parts was changed from a face seal element to a radial seal element at the shaft. The basis being to reduce peripheral speed at the seal(s) for increased sealing reliability.

The hydraulic release circuit enters the shaft via a sealed rotating pass-through somewhat analogous to the oil supply configuration of an automotive engine cross-drilled crankshaft main journal, only with end seals. This pass-through, where the hydraulic release circuit transitions from static housing structure to the rotating shaft element(s), is a design detail to be addressed in the next level of design when specific design parameters are known. The primary design detail is the sealing configuration. Recall that the location of this feed-through has been intentionally positioned at the lowest radial point to minimize tangential speed with the intent to optimize sealing potential. If specific driveline design warranted, hydraulic circuit entry could be configured to enter the output end instead of the input end by reversing orientation of the hydraulic passage as the shaft is one piece.

Mechanical description.—The four-plate concept design is shown in figure 1. Friction plates are stacked with alternating configurations splined at either the inside or the outside diameter. An applied axial load transfers power across the friction faces from the splines located at either the inside or outside diameter. Axial clamping is provided by mechanical spring diaphragm and released via a hydraulic pressure circuit.

Spring versus diaphragm and hydraulic piston configuration.—Axial force application may be accomplished via individual helical springs, multiple conical disc springs, or alternatively, a large central conical disc spring, or diaphragm. Conical disc springs offer higher reliability compared to helical springs with the latter also offering the highest load potential. Design of the axial force application element(s) is a function of the required torque capacity and friction materials to be employed. At present, it is not known if the design will be a dry or wet clutch. The above are dependent upon driveline concept selection and determined at the next level of design.

The hydraulic release piston is an annular ring configuration (disc with central hole) housed within an annular volume, which is fed by multiple radial passages branched off from the central shaft feed. Employing multiple passages assures that the annular volume is pressure balanced for uniform hydraulic release force the release ring (piston) to preclude tilting and potential jamming.

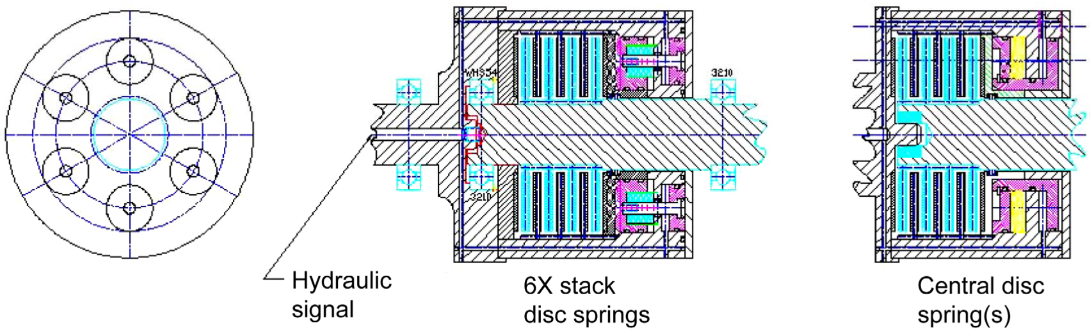


Figure 1.—Four-plate clutch concept.

Pilot bearing.—To ensure radial alignment between the two rotating shafts and allow for differential relative rotation, a rolling-element bearing is employed in the design. During clutch engagement, both of the rolling-element bearing races spin with the clutch assembly without relative motion (i.e., in a locked mode where the inner and outer races rotate at the same speed and the balls do not spin relative to the races). During periods of clutch disengagement, the bearing realizes the relative differential speeds of the decoupled shafts up to 50 percent input speed, or 7500 rpm. This leads to a unique situation where the bearing centrifugal loads are higher for the given relative speed. In some of the drive concepts the bearing will be required to operate for the major portion of the mission in this manner. This bearing will most likely require special internal clearances to assure longevity and mission requirements may necessitate positive through-flow lubrication.

Presently, the bearing is conceived as a grease-packed sealed configuration to permit employing the optimal lubricant. If a wet clutch design is employed, the clutch fluid may not be the best choice for lubrication of the bearing, whereas if a dry clutch is employed, no lubricant source will be present for the bearing. The possibility exists that a wet clutch design could be employed and that the lubricant would be suitable for bearing lubrication in which case the bearing would be jet lubricated. A detailed bearing study during driveline detailed design would dictate the lubrication requirements.

Description of the Conceptual Eight-Plate Dual-Acting Clutch

During the creation of one of the driveline concepts, a concept evolved in which two clutches are employed on a single shaft to direct power flow. In the configuration, two mutually exclusive power inputs were to be directed to a single power output. Based upon the input selected, the output ratio would be different. A unique switching clutch was conceived for the application, an eight-plate dual-acting clutch, to replace two independent four-plate clutches and simplify control by permitting a switched mutually exclusive input to power a single output shaft.

Each end of the above dual-acting clutch has a four-plate disc arrangement based upon the basic four-plate design. The operation of the basic multidisc elements is the same as the clutch described above. By combining two clutches into one, the number of hydraulic release signals is simplified from two to one. The configuration of this clutch permits directing power to one of two paths from within a single unit in lieu of controlling two separate clutches.

The clutch is operated as a hydraulic over mechanical mechanism meaning the hydraulic signal releases, or opens, one output element while engaging the other. Releasing the hydraulic signal causes the mechanical spring force to engage the opposite output element. In the initial concept, the left side (low speed) is hydraulically activated and the right side (high speed) is mechanically activated with the spring(s). The clutch is fail-safe to the mechanical spring(s) side. The function of this clutch is either the right or left drive is engaged while the opposite end is disengaged and fail-safe to high-speed hover range.

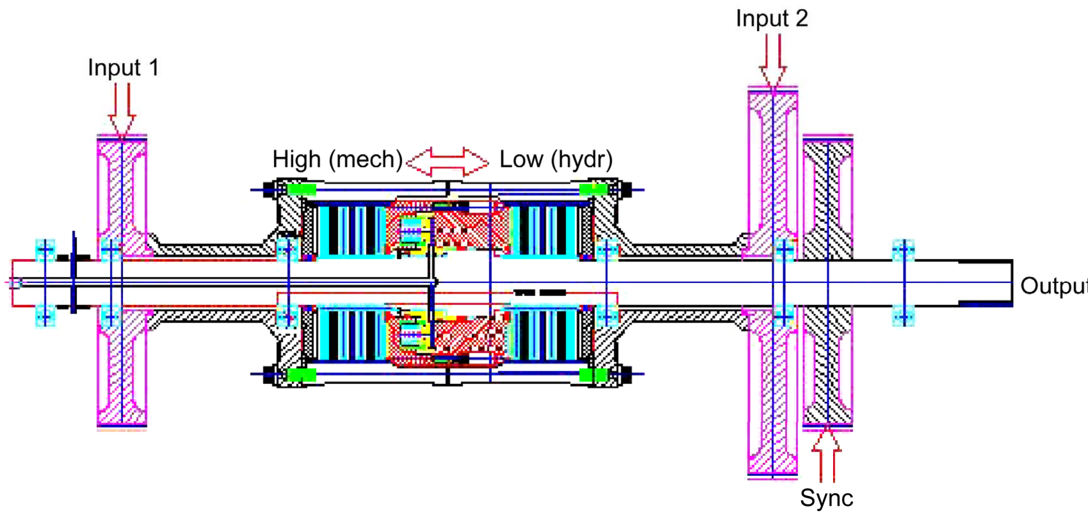


Figure 2.—Clutch configuration used in concept 4B, double clutch (original basis).

This clutch has no internal means of synchronizing output to input shaft speeds. The period, or duration, for range transition and clutch shifting is to be addressed and refined during the actual design of the clutch concept for the intended end use application. Initially, this would be appropriately scaled and designed to be variable as an area of study for a scale test rig demonstration. The period and smoothness of the range transition could be adjusted and tuned by varying damping in the hydraulic circuit of the clutch engagement and disengagement system and the associated controller. During this transition period there would be clutch slippage and heat generation. Thus, in a real application this is most likely a controlled variable based on revolutions per minute and power loads.

Mechanical configuration.—The concept configuration for an eight-plate dual-acting clutch design, shown in figure 2, is made up of two four-plate designs.

As in the single four-plate clutch described earlier, plates are designed such that plates are alternately stacked with every other disc splined at the inside diameter or at the outside diameter. An axial load transfers power across the friction faces from splines located at either the inside or outside diameter. Thrust reactions of the clutches are transferred directly to the shaft from both clutches to a combination of shoulders and retainer rings without additional loading to the main shaft bearings.

As in the basic four-plate design above, axial clamping is provided by a mechanical spring or diaphragm and is released via hydraulic pressure. At this time it is not known if the best choice would be a dry or a wet clutch. This will depend upon the driveline concept in which it employed and will be determined at the next level of design.

A notable difference in the eight-plate dual-acting clutch is where and how the axial clamping mechanism is attached. In the basic four-plate design, the mechanical spring and hydraulic release mechanism were contained within the outer structure of the clutch. The hydraulic release circuit enters the clutch via the input shaft and around the outer structure of the clutch. Hydraulic flow to the release piston is essentially entering the clutch radially inward through the clutch outer structure. In the eight-plate dual-acting clutch, this system must be contained on the shaft and hydraulic flow to the release piston is radially outward. The above functionality required a significant reconfiguration.

Like the four-plate clutch, the eight-plate clutch also requires a pilot bearing. However, in the eight-plate design, two pilot bearings are required and their function is somewhat different. Instead of being between the shafts, they are located at the extremities of the clutch assembly. In the four-plate clutch design, the clutch couples two inline, or in series shafts, and the pilot bearing carries only radial load between the two shafts. In the eight-plate clutch design, the clutch alternately couples two pairs of concentric or in-parallel shafts though the clutch unit itself resides entirely on a single shaft. In this configuration these bearings maintain concentricity and allow for differential rotational speeds. However, in addition, when the particular clutch output element is engaged, they also react the clutch axial clamping load and do so without internal relative motion (i.e., loaded with axial thrust in a quasi-static condition and spinning in locked mode without internal rotations). This is a unique loading condition for a rolling-element bearing and will warrant close consideration.

Alternate Variations of the Eight-Plate Dual-Acting Clutch

Two different configurations for an eight-plate dual-acting clutch concept were created to meet specific requirements of the respective driveline concept. In addition, the first configuration may be used in-reverse by applying input and outputs at alternate locations. A figure of the later, though not used for any of the driveline concepts described in this paper, is included at the end for completeness. The first two configurations are employed in two different driveline concepts. The second eight-plate dual-acting clutch configuration is discussed below highlighting specific differences to the first.

The second concept variant, shown in figure 3, is an inline configuration having a different power flow path than the previous concept.

In this configuration, the “central shaft” comprises two shafts in a collinear arrangement whereas the previous configuration contained a single central shaft. Power is input to the shaft forward end and the output is directed to either the forward (high-speed) or aft (low-speed) clutch output elements, both of which are directed to the same output shaft. Another difference in this configuration is the addition of a sprag at the low-speed path power reentry gear (i.e., at the output shaft). The sprag is the avenue of directing the two possible power paths to the single output shaft, and also completely decouples the low-speed gear train when not being powered. Thus, in the high-speed range, the low-speed gear train is disengaged at both ends, due to the sprag, and does not spin.

Input is applied to one shaft end and transferred either through the clutch (1:1) or to a low-speed gear train path when the output speed is reduced, as in our application, or increased for some other application.

To ensure radial alignment between the two rotating shafts and allow for differential relative rotation, rolling-element bearings are employed. During engagement of the high-speed clutch element, the rolling-element bearing spins with the clutch assembly but has no internal relative motion (i.e., in a locked mode where the inner and outer races rotate at the same speed, the balls do not spin relative to the races). During engagement of the low-speed clutch element, the bearing spins at the relative differential speeds of the decoupled shafts. The differential speed being up to 50 percent input speed. Bearing reactions are converse when operating in the alternate modes. Thrust reactions of the clutch are transferred directly to the shaft from both clutches to a combination of shoulders, retainer rings, and bearings.

Presently, the bearings are conceived as grease-packed sealed configuration for the same reasons as those discussed for the previous configurations. Detailed study of the bearing during a detailed design will dictate the lubrication requirements.

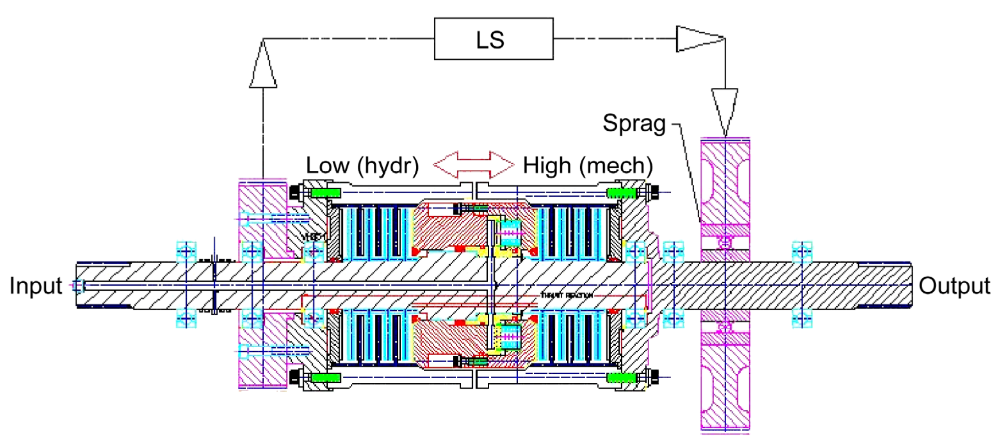


Figure 3.—Configuration used in concept 6, "two-speed drive (CG)."

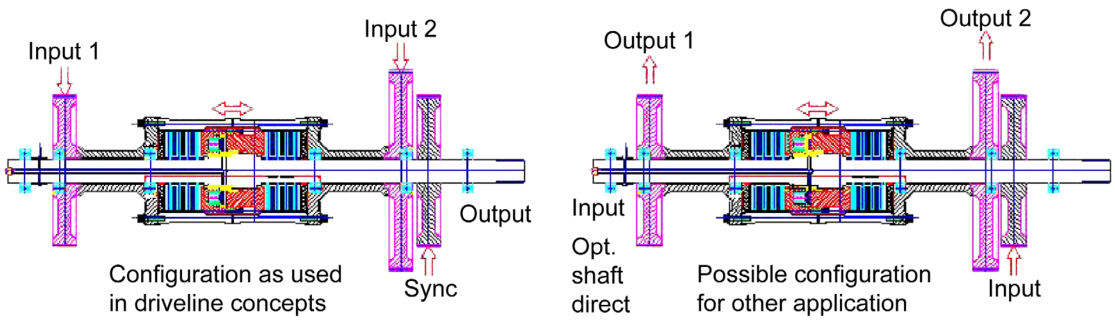


Figure 4.—Eight-plate clutch alternate power flow scheme.

In the driveline concept in which this clutch is employed, the high-speed range is achieved by engaging the right-hand portion of the clutch. During this mode, the left side of the clutch is automatically disengaged. Low-speed range is achieved by engaging the left-hand portion of the clutch. During this mode, the right side of the clutch is automatically disengaged. For the driveline concept application high-speed operation, power is transferred through the high-speed clutch and the output ratio is 1:1. For low-speed operation, the low-speed clutch engaged with power transferred through a two-stage gear reduction (two-stage 8.5- and 12.0-in. pitch diam.). As stated earlier, this clutch has no means of achieving input and output shaft speed synchronization.

The third clutch configuration is presented side by side with the first clutch configuration in figure 4, highlighting an alternate power flow possibility.

This variant configuration is identical to initial eight-plate clutch configuration, differing in only how the power flow is input and output. This configuration is not employed in any driveline configuration presented but is included for completeness of the effort expended in configuring a conceptual clutch for this effort and as an offering for other applications. In this variant, the power flow is opposite to that of the first presented with the input being applied to the shaft and the outputs taken from either of the two gears.

Speed range changes for discrete two-speed transmission.—A procedure for operating a pair of clutched two-speed transmissions in a tilt rotor twin engine aircraft is described in reference 3 (sec. 6.1, p. 14).

Table 1 summarizes speed and torque requirements for transmission and clutch components.

TABLE 1.—REFERENCE DESIGN PARAMETERS

Transmission and clutch design requirements	
[a]Speed and torque required	
High range	15 000 rpm, 21 000 in.-lb, 5000 hp
Low range	7500 rpm, 31 500 in.-lb, 3750 hp
Maximum torque	31 500 in.-lb
Max. hydraulic pressure	1000 psi (clutch release)

[a]Conditions per July 24, 2007, meeting RFH/DGL/MAS.

Reversing Concepts

Reversing Concepts 00

During the creation of possible driveline concepts, a need for reversing direction of rotation between two rotating elements arose on a few occasions. In both instances the concept looked viable except for a major flaw in that the output direction was opposite of that desired. Except for the reversal issue, the concept was viable. A correction to reverse rotation for the one operating speed range was needed. The second reverse rotation issue arose with the consideration of a continuously variable (CV) speed configuration, which for this endeavor was a CV toroidal drive. The geometry of the toroidal variator configuration is such that the output rotation is reverse of the input direction of rotation. For this instance, the need for rotation reversal would be necessary at all times. Initially reverse rotation was not seen as a problem, but later, the idea of employing the above in a split-power configuration led to a need to reverse the output so that it could be integrated into the overall concept.

Several schemes for reversing the direction of rotation were considered during the course of this task, but the following three methods for reversing output rotation emerged: First, a double star idler gear arrangement. The second, an offset parallel shaft spur and helical gearing arrangement. The third, opposing face ring gears with multiple idlers or a similar configuration employing bevel gear geometry.

The first, the double star idler is shown in figure 5.

The double star idler configuration, though appearing simple, is highly geometrically restrictive since the star gears and idler gears must fit between the sun and ring gears and properly mesh. In addition to basic pitch diameter restrictions, it is further restricted by diametral pitch and the limitations, which are placed on the number of teeth in mesh. As with any planetary system, even basic assembly may be an issue. The highly restricted aspect was immediately obvious during fitting to the given concept. Provisions for adjusting backlash could be addressed with a pivoting cam arrangement but would be costly and less than ideal with respect to locking the adjustment. Concepts of how to provide backlash adjustability were considered, but not developed due to the perceived low probability of overall viability. The configuration is not readily applicable to any ratio and diametral pitch. For the above reasons, viability in application is highly limited.

Secondly, the offset parallel-shaft spur or helical gear configuration is shown in figure 6.

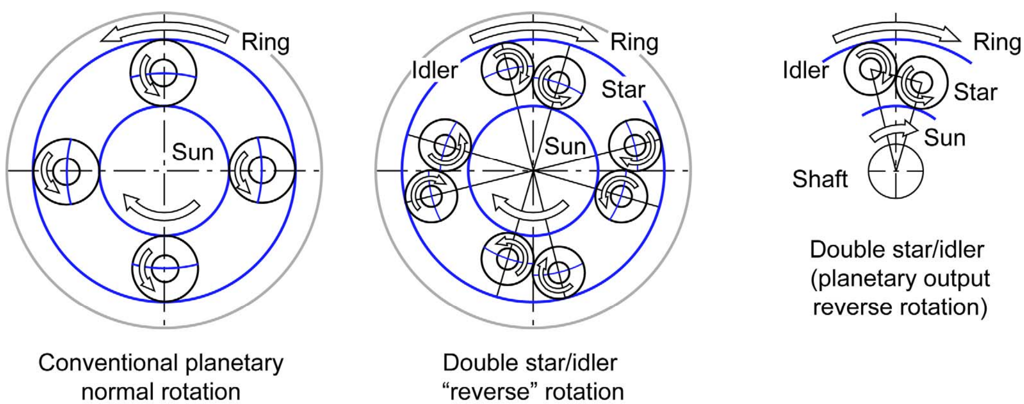


Figure 5.—Double star idler reversing concept.

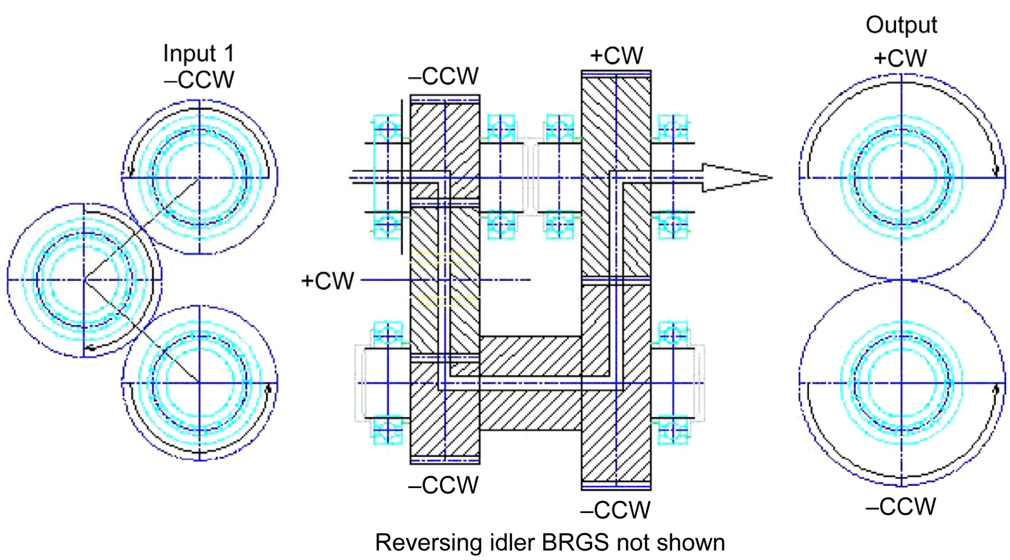


Figure 6.—Parallel-shaft idler reversing concept.

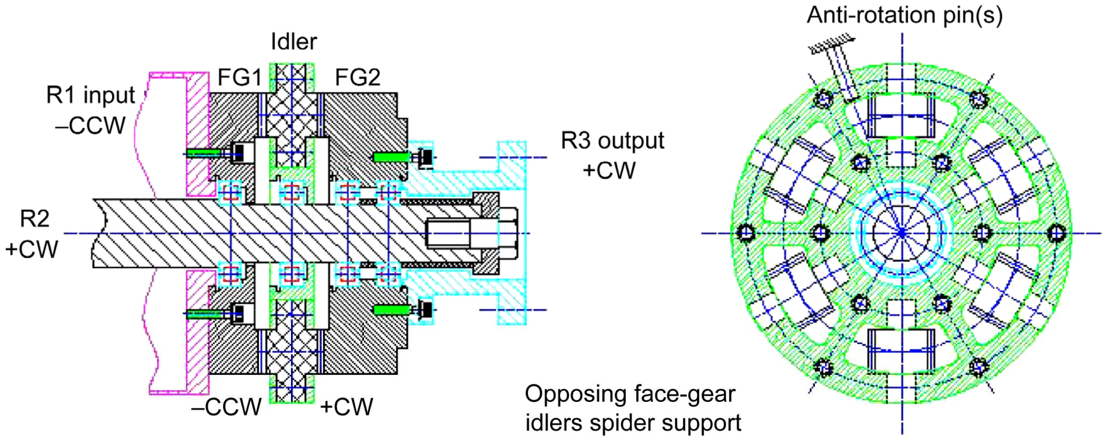


Figure 7.—Opposing ring gear idler reversing concept.

The offset parallel-shaft spur and helical gearing configuration employs a high number of gears, shafts, and bearings. Adjustability of backlash is easily addressed. However, the heaviness of this configuration makes it undesirable for a flight application.

Lastly, opposing ring gears driving through multiple idlers is shown in figure 7.

This configuration comprises opposing ring gears with multiple idlers utilizing either face gear geometry or possibly bevel gear geometry, is the most advantageous of the above. The face gear geometry is a proven configuration, relatively light, load distributing, and highly flexible with respect to the number of idlers that reduce gear tooth loads through load sharing. The configuration could potentially accommodate multiple power takeoffs should the application warrant. The idler configuration may either be spider supported as depicted above or the idlers could be of cantilevered pinion configuration. Although a basic configuration for cantilevered idlers was created, only the spider configuration is presented.

Backlash adjustability for individual idlers is easily addressed with bevel gear geometry whereas it may be more difficult with face gear geometry. In addition, face gear geometry is more restrictive with respect to gear tooth strength than bevel gear geometry where tooth width can generally be increased reducing bending and contact stresses.

The above is ideally suited to application as a post or secondary reversal stage to a variator path such as a toroidal variator. In contrast to the toroidal variator, which requires significant axial loads, the opposing ring face gear idler arrangement must be axial (separation) load limited. The design must address control and limiting the thrust loading within the gearing while conversely addressing the opposing requirement of high axial loads for the toroidal variator.

Summary of Reversing Mechanisms

From the above configurations, the following is concluded: The opposing ring gears with either the face gear or bevel gear geometry should be incorporated into the final selection driveline concept(s) if required.

Driveline Concepts

Driveline Concept 1—Dual-Input Planetary Differential (Redundant Drive Concept)

This concept, as shown in figure 8, is based on a dual-input, single-output planetary differential. In basic form, the concept uses two inputs powering a single output. The primary input, engine number one, drives the sun gear while the secondary input, engine number two, drives the idler gear, spinning the ring gear. The configuration has rotational output when both or either individual input is spinning. The configuration is used in fail-safe lifting and towing devices.

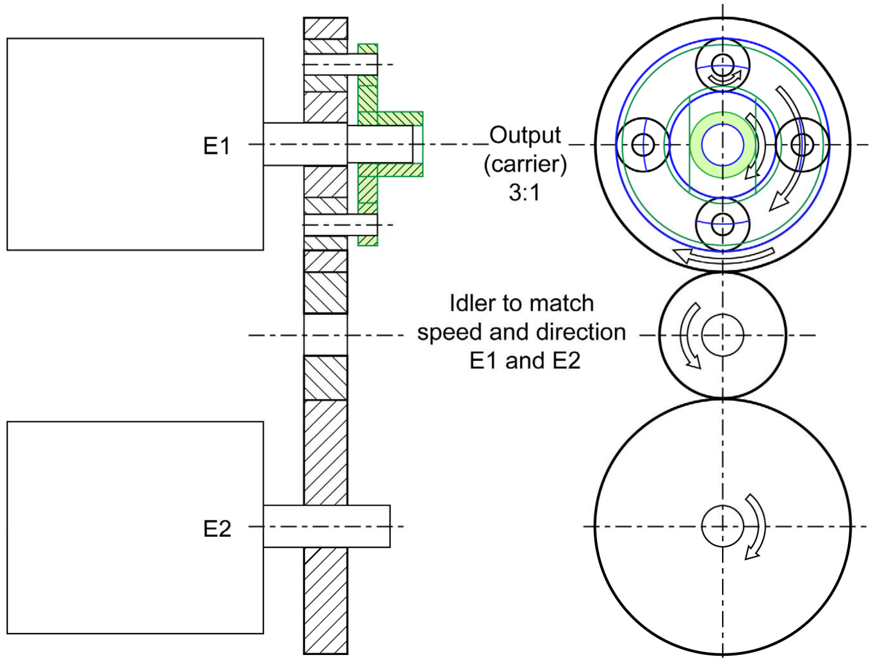


Figure 8.—Two dual-input planetary gear drive.

The basic concept capitalizes on a planetary differential in which primary power is input to the sun gear, power is output from the carrier, and output speed variation is achieved by varying the speed of a special ring gear from zero to full speed with a variable-speed controller device. The ring gear is special in that it has both an internal pitch diameter and an external pitch diameter in an integral ring. As depicted, the ring gear speed is varied from full to zero speed by a second input. The second input drives the external pitch diameter of the ring gear. As depicted, drive ratio is 1:1 but may be varied. As depicted, the second input rotates in the same direction as the engine via an idler.

It was initially conceived that the above configuration could be adapted to be a variable-speed output drive if the second input were replaced with a controller.

Thus, the concept is an engine being the primary input, driving the sun gear, and a controller, the secondary input, driving the ring gear. This configuration is a basic variable-speed device where the engine is operated at full speed and output variability is achieved by varying the ring gear speed with a controller. The controller is operated between full speed (engine speed), or some reduced speed, if a gear speed increasing gear train is included, and zero or near zero speed. At a controller speed of zero, the carrier output speed ratio is 3.0 resulting in an output speed of one-third of that of the engine.

If the ring gear is rotated, the carrier output speed is increased. If the controller is operated at a low speed equal to one-sixth of the engine speed, then the carrier output speed would be one-half the engine speed. The configuration is a variable-speed device with a positive method of power transfer, gear teeth.

This concept was an impetus toward the development of some other concepts so it is presented primarily in the light of being a basis for other concepts. The speed controller may be a variety of possible devices, either externally powered and controlled, or internally driven from a power takeoff gear with only the output speed varied. The power takeoff may be a CV speed device as discussed elsewhere. Possible configurations for a speed controller are also discussed in another section. Also, a possible configuration for a planetary differential drive with integral CV control is presented in a later section.

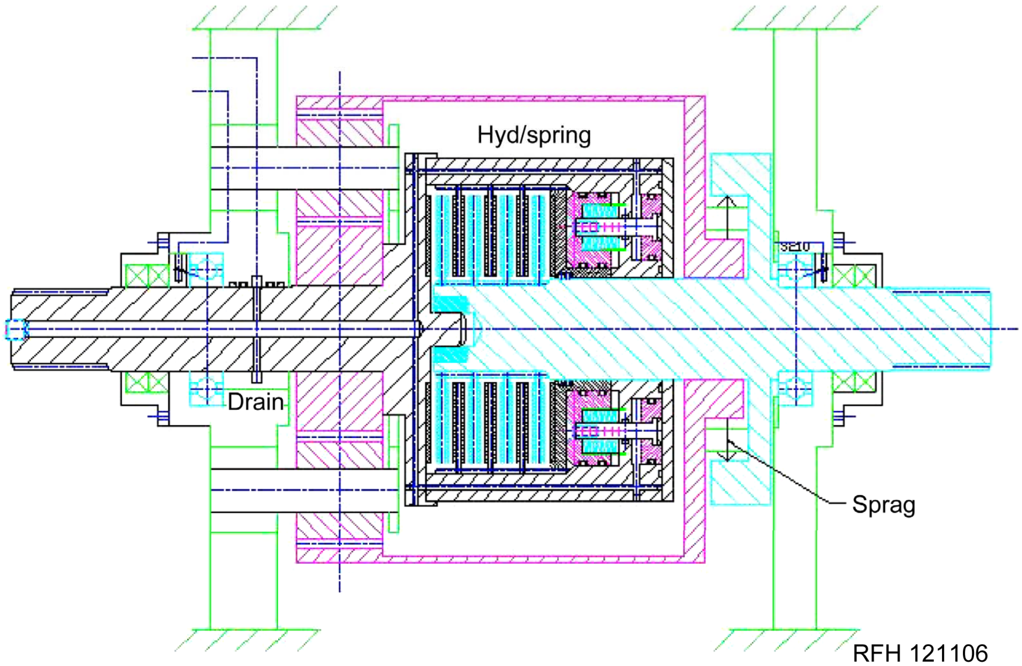


Figure 9.—Inline two-speed planetary drive.

Driveline Concept 2—Inline Two-Speed Planetary Drive

This concept, as shown in figure 9, was one of the earliest conceived during this effort. The configuration is a discrete two-speed unit. Operation is in either of two discrete speeds. The configuration uses one clutch, one sprag, and a star planetary gear train to achieve the 50 percent speed reduction for the low-speed cruise mode.

The power flow during high-speed output mode is straight-through with the main clutch engaged and the output ratio is 1:1. The planetary gear train free-wheels or overruns the overrunning clutch (sprag). The power flow during a low-speed operation is directed through the star planetary gear train into an overrunning clutch (sprag) by disengaging the main clutch. With the clutch disengaged, transfer of power through the planetary is directed to the sprag, which is now the driving element. The star planetary achieves a 2:1 output ratio. Power requirements for the high-speed range is full power with reduced power being required for the low-speed range, thus power routing is optimally directed.

The primary shortcoming of this concept is that the direction of rotation for the low-speed range is opposite to that of the high-speed range. Though not explicitly stated at the beginning of this task, it was assumed that the input and output rotation would be the same direction. If however, they are opposite, this does not necessarily pose a show-stopping problem, especially for new application design. However, when the output of one of the two speeds is opposite to the other, this is a show stopper.

Three concepts, 2A, 2B, and 2C, are based on the above concept except that they incorporate potential resolutions to the reverse output issue utilizing three different possible approaches to make the concept viable as a discrete two-speed transmission.

Driveline Concept 2A—Inline Two-Speed Planetary (Modified—Double Star Idler)

This configuration, shown in figure 10, evolved out of the need to reverse the output direction of rotation for a planetary stage with fixed planets, the basic gear train of the previous concept. The overall concept of this configuration is the same as the inline two-speed configuration discussed in the last section. The basic problem with using a planetary gear train with input via the sun and output via the ring, with the star gears being fixed, is that the output direction of rotation is reverse of the input rotation.

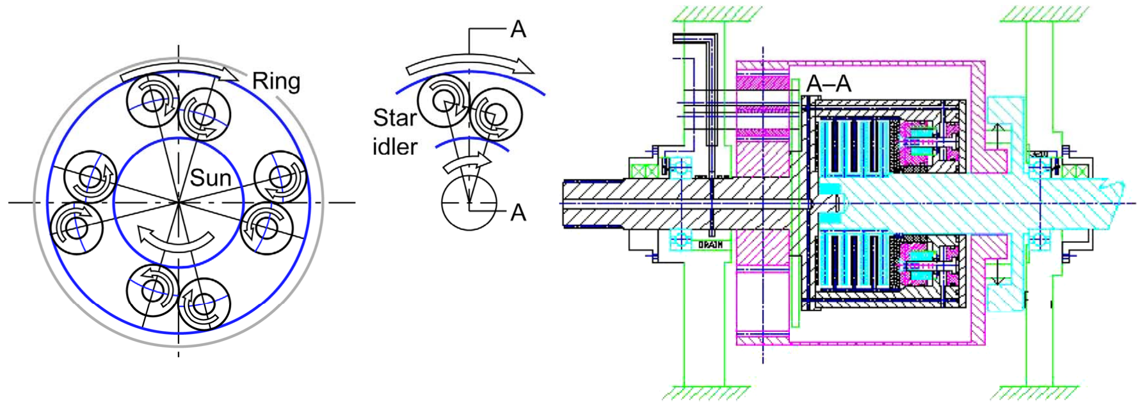


Figure 10.—Inline two-speed planetary drive (modified with addition of double star idler reversing concept).

The most basic method to reverse the output direction of two gears in mesh is through the addition of an idler gear. An idler reverses rotation through the first mesh and re-reverses it through the second mesh without changing the speed ratio of the original two gears.

For the planetary train, this means adding a second star gear to act as an idler. By adding a second star gear, serving as an idler, the direction of rotation of the output rings is reversed. The second star gear is mounted in an unconventional manner allowing it to act as an idler gear between the first star gear and the output ring gear. As an idler, the output direction of rotation of the driven gear is reversed.

The double star idler configuration is a highly restrictive design. It is difficult to achieve the desired overall ratio, simultaneously obey diametral pitch constraints, and obtain a set of gears that assemble and mesh properly. When one is fortunate enough to get the desired ratio, obtain proper pitch meshing, and maintain reasonable idler speeds, a very lightweight configuration for reversing the output direction of rotation in this form of planetary gear train is achieved. Further discussion of the double star idler is contained in a separate section of reversing concepts.

Driveline Concept 2B—Two-Stage Counter-Rotating Planetary

The two-stage counter-rotating planetary drive configuration, as shown in figure 11, also evolved out of the need to reverse the output direction of rotation for a planetary stage with fixed star gears. The basic concept of this configuration is the same as the inline two-speed configuration except that it employs two stages of planetary gearing to achieve the necessary output ratio and change in direction of rotation. In this concept, the basis is to split the required overall ratio into two identical stages of input sun, fixed star, and output ring. The output from each stage does a rotation reversal. By combining two identical stages in series, the result of the end output is rotation in the required direction, the same as the input.

The configuration employs standard gear meshing configuration and should be easily designed to be robust. The drawback to this configuration is the increased number of gears, shafts, and bearings. The gears and shafts may ultimately be made integral to reduce bolted connections and weight resulting in high costs.

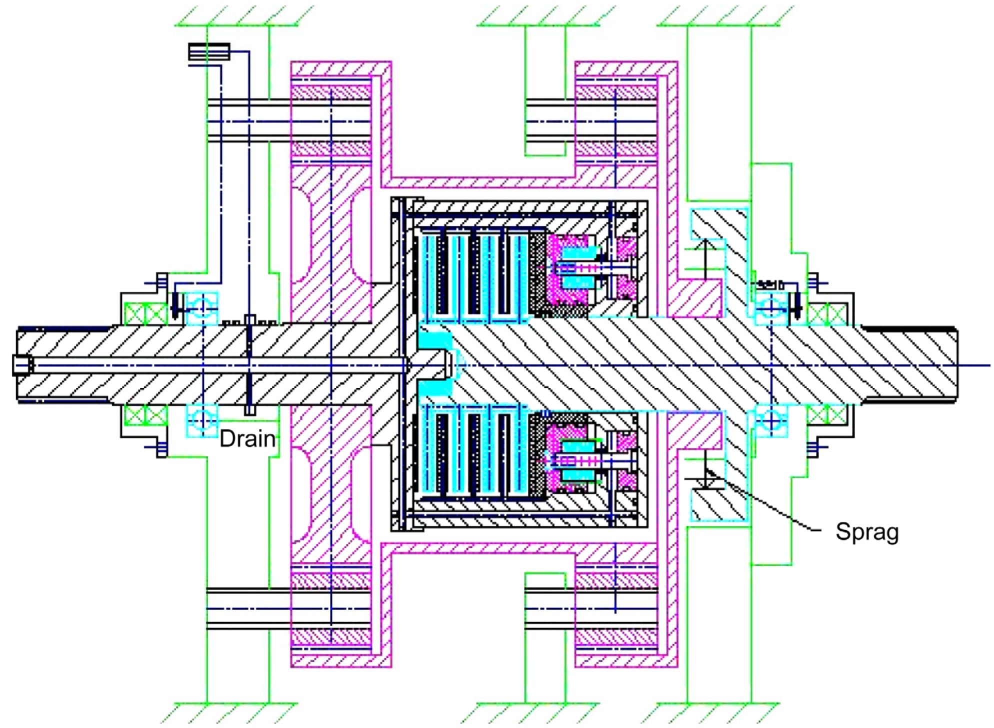


Figure 11.—Inline two-stage counter-rotating planetary drive.

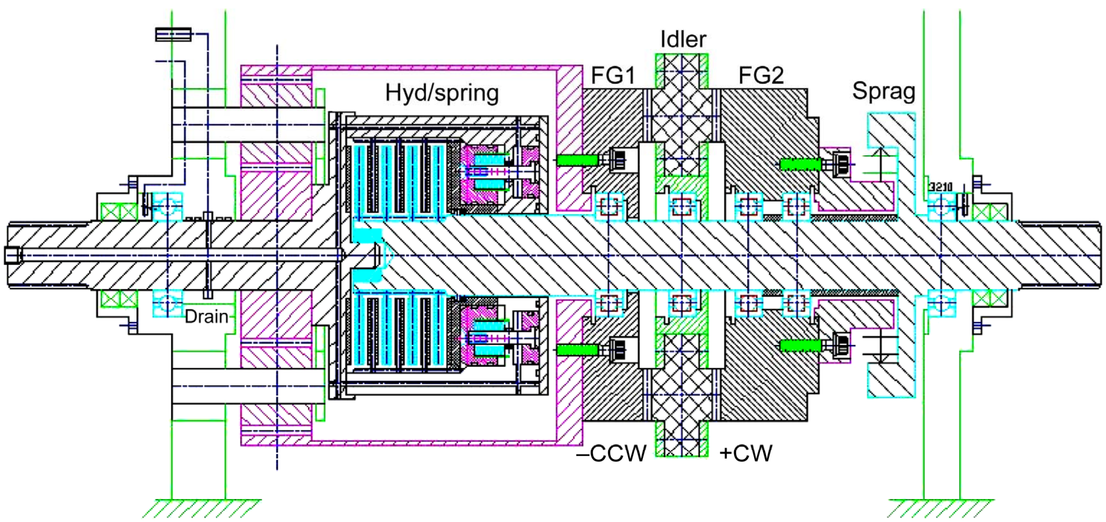


Figure 12.—Inline two-speed advanced rotorcraft drive (modified—opposing ring gear and idler reversing concept).

Driveline Concept 2C—Inline Two-Speed Drive (Modified—Opposing Ring Gear and Idler Reversing Stage)

This configuration, as shown in figure 12, also evolved out of the need to reverse the output direction of rotation for a planetary stage with fixed star gears. The overall concept of this configuration is the same as the basic inline two-speed configuration except that rotation reversal is achieved through the use of the basic idler coupled with face gear technology. The basis of this concept is to employ two identical face gears oriented in opposition transferring power through multiple idlers. By powering through the idler arrangement, the face gears rotate in opposite directions just as the case of a pair of spur gears separated by an idler. The primary difference is the respective orientations and locations of the gear axes. The face gear arrangement permits input-output to be coaxial whereas the spur gear arrangement permitted a parallel offset axis arrangement. The face gear idler arrangement is a compact reversing stage. The configuration employs proven face gear geometry meshing configuration and it should be possible to design this as a robust reversing stage.

Driveline Concept 3—Offset Compound Gear

The offset compound gear drive, as shown in figure 13, is an inline discrete two-speed device. This concept was evolved from the double reversing idler concept which employed a double idler(s) between the pitch diameter of a smaller and larger gear pair.

The configuration is based on a novel approach of off-setting and embedding a gear mesh. The heart of the concept is the offset compound gear, which uses identical pitch diameter in both an internal configuration on the input end and an external configuration on the output end, thus allowing it to mesh with both a smaller external gear and a larger internal gear in series resulting in an inline reduction gear set. The above geometry permits the compound gear to be offset and mesh with the input gear and the output gear both of which are on the same centerline. The configuration provides a 50 percent reduction in two stages, or meshes, utilizing only three gears replacing multiple gears in a conventional planetary stage. The concept will require very robust and ultimately wide gears. Although the concept is simplistic from the gearing perspective, the challenge is how to support the offset compound gear on bearings.

Low-speed operation is accomplished in two meshes: a 5.0-in.-pitch-diameter input gear to 7.50-in.-pitch-diameter intermediate gear (0.667 reduction mesh) and a 7.50-in.-pitch- diameter intermediate gear to a 10.00-in.-pitch-diameter output gear (0.750 reduction mesh). The resultant low speed ratio is 2:1, (output speed = 0.500 = 0.667 stage one reduction by 0.750 stage two reduction). The input and output shafts spin on rolling-element bearings while the intermediate gear shaft (the offset compound gear) spins on fluid film journal bearings. Power transferred through the gear train drives a sprag. During this mode of operation, the main clutch is disengaged.

The high speed, 1:1 ratio, is direct drive through the primary clutch. During this mode of operation, the gear train free-wheels an overrunning sprag. A slight reduction in input speed is required to overrun the sprag. The above gear train always spins. An alternative to spinning these gears in both speed ranges might employ the sprag at the forward end of the gear mesh allowing the gear train to quasi-idle when not transferring power.

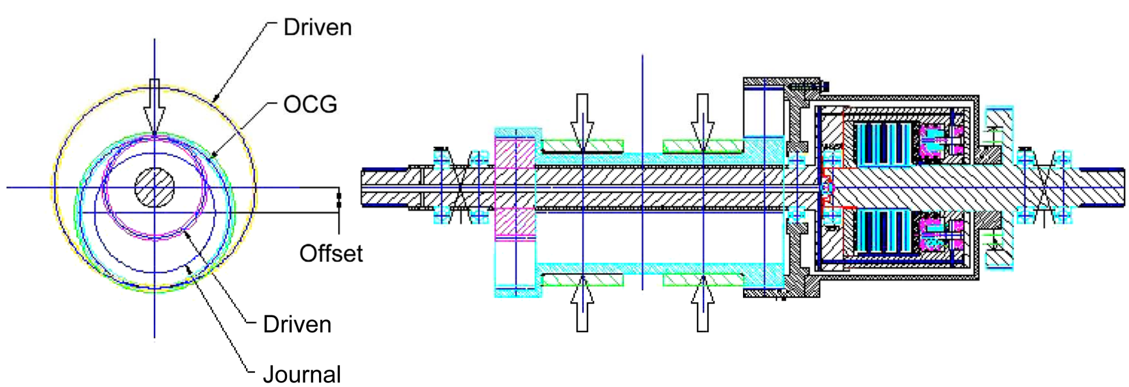


Figure 13.—Offset compound gear drive.

The concept was initially conceived with v-groove rollers supporting the offset compound gear both radially and axially. External rollers led to an overall large cross section. Internal roller support was also explored. The size of the rollers relative to the gears being supported also led to rotational speed and load issues. Whether roller support was external or internal—roller support was not well received due to potential compressive stress (contact) fatigue issues. Alternate methods of support considered included direct support via idler gears. Once again, the concept was not well received. Gas foil bearings were considered but were not determined to have the required load capacity. A latter attempt, to stretch the forward end of the concept to permit the introduction of fluid film journal and thrust bearings, seems workable.

While the concept was initially conceived as a discrete two-speed device, it is adaptable to a quasi-variable-speed drive using a speed synchronizer to power and speed up the output shaft to match the speed of the input shaft. With some minor modifications the above configuration can be explored as a variable-speed transmission. The proposed configuration is depicted in figure 14.

Driveline Concept 4A—Double Clutch Drive (Variable Control—Alternate Version)

Two configurations were based on the basic dual clutch configuration shown in figure 15. Figure 16 shows the first of two concepts based on the schematic in figure 15.

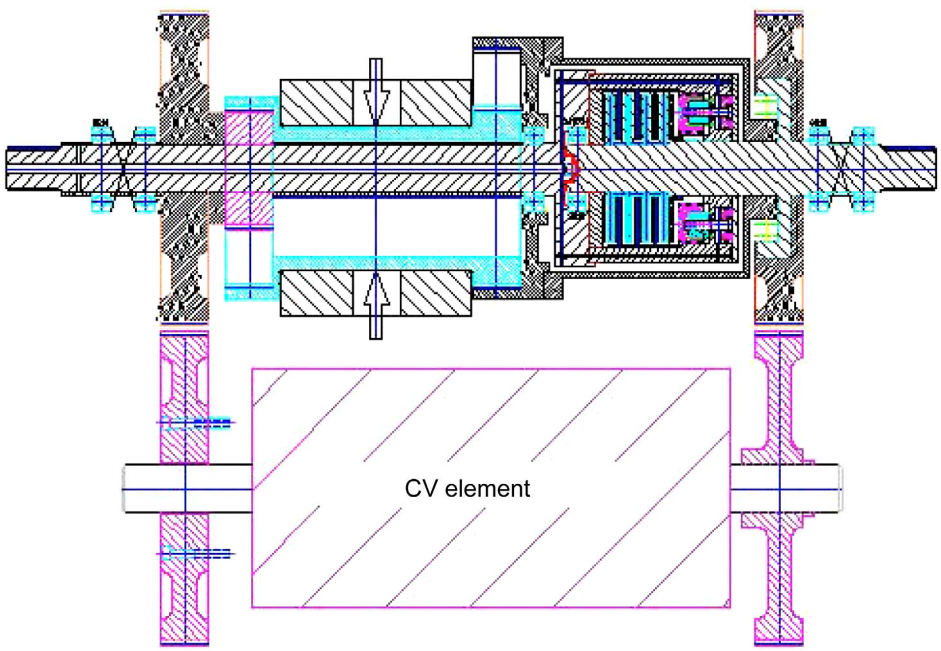


Figure 14.—Offset compound gear drive with variator.

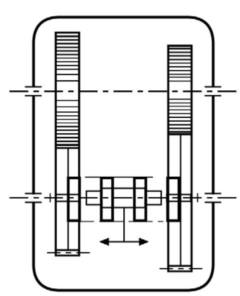


Figure 15.—Dual-clutch drive (ref. 6).

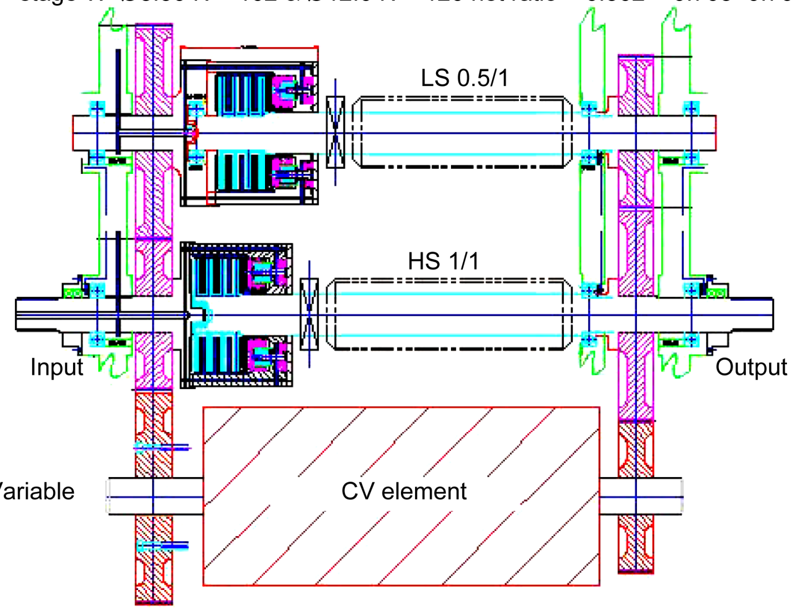


Figure 16.—Dual-clutch variator split power drive (alternate configuration).

The dual-clutch variator split power drive (alternate configuration), shown in figure 16, is an inline configuration. This is a significant departure from the basic concept (fig. 15). Gear reduction is accomplished using two-stage by 1.412 ratio (two-stage by 8.5-in.-pitch diam. and 12.0-in.-pitch diam.) reduction. Gears on both the input shaft (high-speed shaft) and loop shaft (low-speed shaft) are rigidly coupled and spin in rolling-element bearings. A third shaft carries a CV element, which is used to transition between the high- and low-speed ranges. This configuration uses two clutches, each on a separate shaft. High-speed range is achieved with the central high-speed clutch engaged and the low-speed clutch disengaged. Conversely, low-speed range is achieved with the low-speed clutch engaged and the high-speed clutch disengaged. During high-speed operation, power is transferred through the high-speed clutch and the output ratio is 1:1. During low-speed operation, the low-speed clutch is engaged and power is transferred through two-stage gear reductions of 1.412:1 (8.5-in.-pitch diam. and 12.0-in.-pitch diam.). While in either the high- or low-speed range, one of the clutches is always engaged while the other is disengaged. Other than during transition, power is either transferred through directly (main clutch engaged) or through the gear mesh (opposite clutch engaged). Clutches operate mutually exclusively. During transition between high and low speed, power is transferred through the CV element until input and output shaft speed synchronization is achieved.

A positive aspect of this configuration is that during high-speed operation, or hover mode, power is transferred via direct drive without going through a gear mesh. During low-speed operation, or cruise mode, where full power is not required, power is passed through the gear train. Power requirements for the high-speed range are full power with reduced power required for the low-speed range, thus power routing is optimally directed.

Driveline Concept 4B—Dual Clutch (Variable Control—Original Basis)

The dual-clutch variator split power drive (original basis configuration), shown in figure 17, is the second of two variations based on a basic dual clutch configuration.

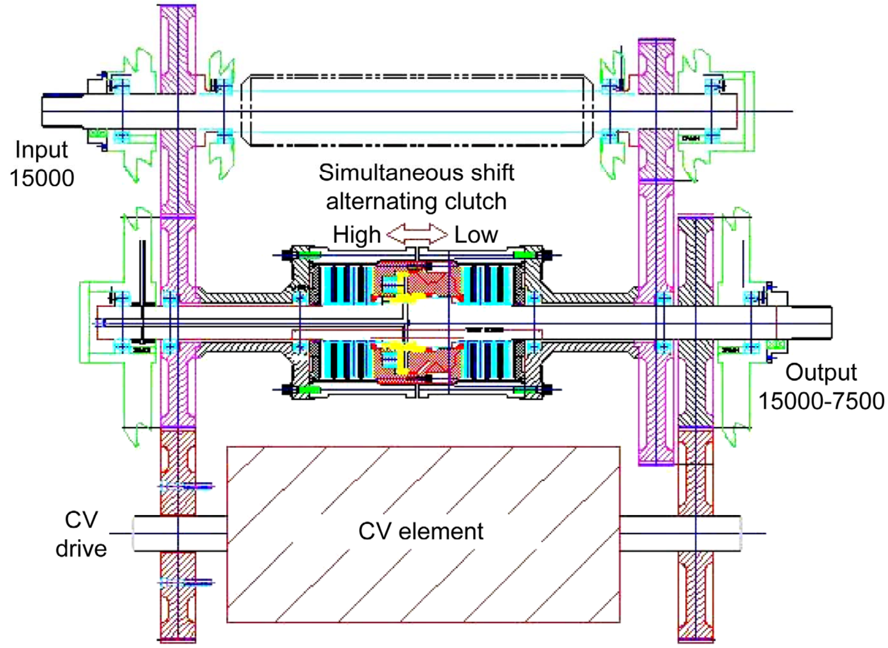


Figure 17.—Dual-clutch variator split power drive (original basis configuration).

In this variation, input and output shafts are offset and gear reduction is accomplished by a single-stage 2.0:1 ratio (8.0-in.-pitch diam. and 16.0-in.-pitch diam.). The gears on the input shaft are rigidly coupled to the shaft whereas the gears on the central shaft ride on rolling-element bearings and are coupled to a central clutch. A third shaft carries a CV element, which is used to transition between the high- and low-speed ranges. Other than during transition, power is transferred through the gear trains.

Whereas the previous configuration employed two clutches, each on a separate shaft, this configuration employs a unique switching clutch located on the central shaft. The configuration of this clutch permits directing power to one of two paths from within a single unit in lieu of controlling two separate clutches. The clutch channels power to either the right or left side. Clutch design is such that either the right or left side of the clutch is exclusively engaged while the opposite side is automatically disengaged. The right side, or low-speed side, is hydraulically activated whereas the left side, or high-speed side, is mechanically activated with the spring(s) being fail-safe to the high-speed side.

While in either the high- or low-speed range, one side of the clutch is always engaged while the other is disengaged. During transition between high and low speed, power is transferred through the CV element until input and output shaft speed synchronization is achieved. The period for the range transition and the clutch shifting will need to be refined during the actual design of the clutch concept. This would be accomplished by adjusting and tuning the damping in the hydraulic circuit of the clutch.

Speed changes are made via a CV traction drive to achieve speed synchronization of the central shaft with the desired gear train (power path). High-speed operation—the power is transferred through the high-speed clutch and the output ratio is 1:1. During low speed operation, the low-speed clutch is engaged and power is transferred through a single-stage 2.0:1 (8.0-in.-pitch diam. and 16.0-in.-pitch diam.).

For high-speed operation (hover mode) and low-speed operation (cruise mode), power is always transferred through gear mesh. This is in direct contrast with the other dual clutch configuration in which power transfer for high-speed mode is directly through the main clutch and not through a gear mesh.

Driveline Concept 5—Differential Drive (Basic)

The differential planetary drive, shown in figure 18, is a basis for a family of concepts that capitalizes on the output variability of a dual- to single-output differential with one input acting as a controller.

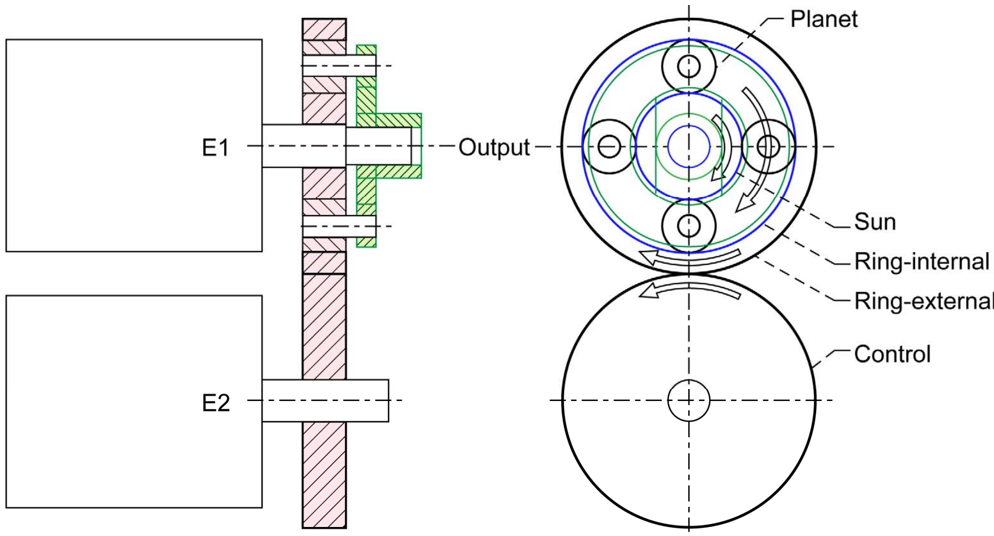


Figure 18.—Differential planetary drive.

The basic concept is a planetary differential in which primary power is input to the sun gear, power is output through the carrier, and speed variation is achieved by varying the speed of a special ring gear from zero to full speed with a variable-speed controller device. The ring gear is special in that it has both an internal tooth pitch diameter and an external tooth pitch diameter contained within an integral ring (i.e., teeth on both the inside and outside diam.). As depicted, the ring gear speed is varied from zero to full engine speed by a speed controller. The speed controller drives the external pitch diameter of the ring gear. As depicted, the controller ratio is 1:1 but may be varied in design permitting selection of the optimal power and speed range. As depicted, the controller rotates in the opposite direction of the primary input but may be the same if an idler is employed. The speed controller may be a variety of possible devices either externally powered and controlled or driven from a power takeoff from the primary power with only output speed being controlled. The power takeoff may be a CV speed device as discussed elsewhere. A possible configuration for a planetary differential drive with integral CV control is presented in a later section. Possible configurations for a speed controller are also discussed in another section.

The speed controller is operated between zero and full engine speed or it may operate a maximum speed lower than that of the engine if a suitable gear train is incorporated.

A primary advantage of the two-input planetary differential configuration is that it is a variable-speed device with power transfer through gear teeth. Unknowns at this point are the optimal power and speed requirements for the controller. If this avenue is pursued, prudent testing to aid in the identification and/or design of a controller element would be in order in the form of a basic test rig, which would mount a planetary train consisting of a sun-star-ring gear train and determine the power requirements to control the ring speed.

Operation of the unit is as described below:

High-speed mode

- Engine 1 and controller 2 rotate at the same speed—output speed is same as E1 and C2.
- Planetary stage rotates as a locked unit without relative motion between the gears.
- Output speed equals whatever speed that both E1 and C2 are operated.

Low-speed mode

- E1 rotates at full speed and C2 speed is reduced.
- As C2 speed is reduced, the output speed is also reduced.
- When C2 speed is zero, the carrier output speed is one-third that of E1.

By varying the C2 speed, output speed is continuously varied between the high- and low-speed ranges.

This configuration should be tested for the desired ratio while measuring the speed and power required. By varying the second motor power and speed, the speed controller rating requirements can be established. This would permit the concept to evolve in the direction of either an internally driven variable-speed device powered by a gear takeoff on the input end of the transmission or a separate externally powered speed controlling device.

As with concept 1, the concept as shown has 3:1 ratio output when ring gear is at rest; one can obtain 50 percent speed output with controller rotating at 16.67 percent of E1.

Driveline Concept 5A—Differential Drive (Co- and Counter-Rotating)

The differential drive concepts, shown in figure 19, permit either a co-rotating (same direction) or counter-rotating configuration dependent upon application requirements.

This concept retains much of the advantage of the basic two-input differential configuration in that it is a variable-speed device with power transfer through gear teeth. In the previous concept, ring gear speed was varied with an external variable-speed controller device. In this concept, speed variation is achieved internally through the use of a sprag and brake.

The sprag provides a basic automatic shifting function based on input speed. Above the sprag engagement speed, a condition of split power flow is realized with a portion transferred through the sun gear and a portion through the sprag and on to the ring gear. By slowing input speed near and below the sprag design speed, power is entirely transferred to the sun gear. With the aid of a brake, the ring gear is slowed down resulting in reduced planetary carrier output speed. Ultimately stopping the ring gear permits the carrier output speed to achieve the low-speed output range. At this point, all power transfer is completely through the planetary system. Although full power is being passed in through the planetary gear train, it is being done so at a reduced level. Reduced power requirement is a result of the vehicle speed being sufficient to realize wing lift. Thus, the engines need only provide power for forward flight as the power required for lift is transferred from the rotors to the wings.

Transitioning from cruise to hover requires a slow release of the brake and simultaneous increase in input speed to engage the sprag, thus reinitiated split power flow. Although the mechanical aspects of the range changes are present with this concept, the dynamics of the range changes are unknown. With close control of input speed change, coupled with rate of braking in both application and release, smooth and continuous transitions should be realizable. If the required degree of transitional smoothness be unacceptable, the concept is adaptable to accommodate the addition of a speed controller to aid in improving synchronization of shaft speeds during changes in speed range.

Unknowns at this point: What is the power split and just how smooth are the transitions between the speed ranges? Some prudent experiments to aid in the identification and/or design of such a controller

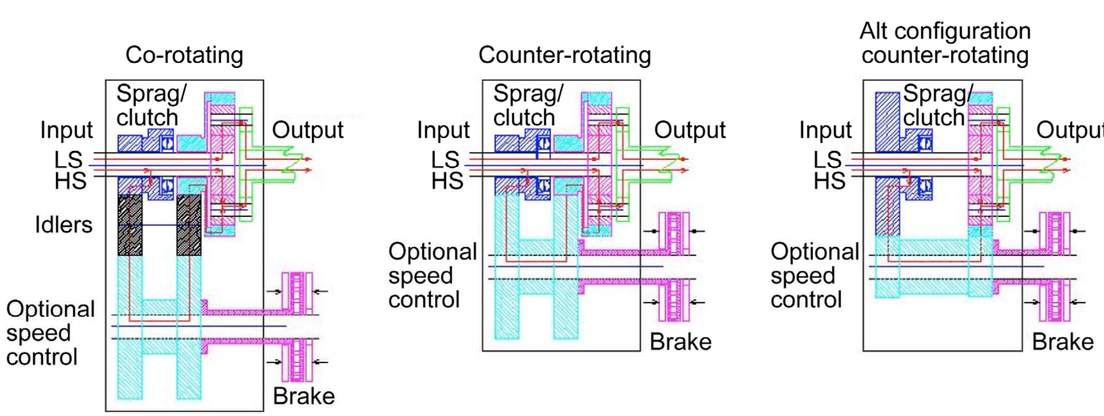


Figure 19.—Differential planetary drive (co- and counter-rotating).

element would be to set up a basic test rig, which would mount a planetary train consisting of a sun-star-ring gear train and determine the power requirements. This configuration should be tested for the desired ratio with the speed, power required measurement, as well as dynamics measurements range change dynamics. This would permit the concept to evolve in the direction of either a speed controller—less relying on the controlled engagements of the sprag and brake or if a speed control device is required.

As with concept 1, the concept as shown has 3:1 ratio output when ring gear is at rest; one can obtain 50 percent speed output with ring gear rotating at 16.67 percent of input speed.

Driveline Concept 5B—Differential Drive (Variable Control)

The differential drive with CV element, shown in figure 20, is a split-power parallel three-shaft system. The figure represents the concept as being coplanar for presentation purposes only; it is not. It comprises a central input power shaft that drives a sun gear on which a centrifugal under-running sprag is mounted on the input end. A secondary clutch shaft is driven by the above sprag in parallel with the main shaft. The output of secondary clutch shaft drives the ring gear in the planetary gear system. The output of the planetary system is the carrier. With the above basic parallel shaft system, two speeds can be achieved by either driving in parallel through the clutch and sprag for high-speed 1:1 drive, or directly through the clutch while the sprag overruns. The later providing a low speed 3:1 drive when the ring gear is stationary. The basic speed change is a function of the ring gear speed. A brake is included to positively lock in the low-speed range. The above configuration allows for two different output speeds. By introducing a tertiary shaft with a takeoff-driven CV element, the above parallel shaft system becomes a CVT (i.e., a two-input differential planetary) with a speed range of 100 to 50 percent of the input speed. Basic operation is outlined below:

High speed—In the high-speed mode, power is split between the central shaft and the main clutch shaft and planetary gear train. In this mode; however, the gear train rotated as a locked train without any internal relative motion. Power is combined at the planetary gear train and output through the carrier.

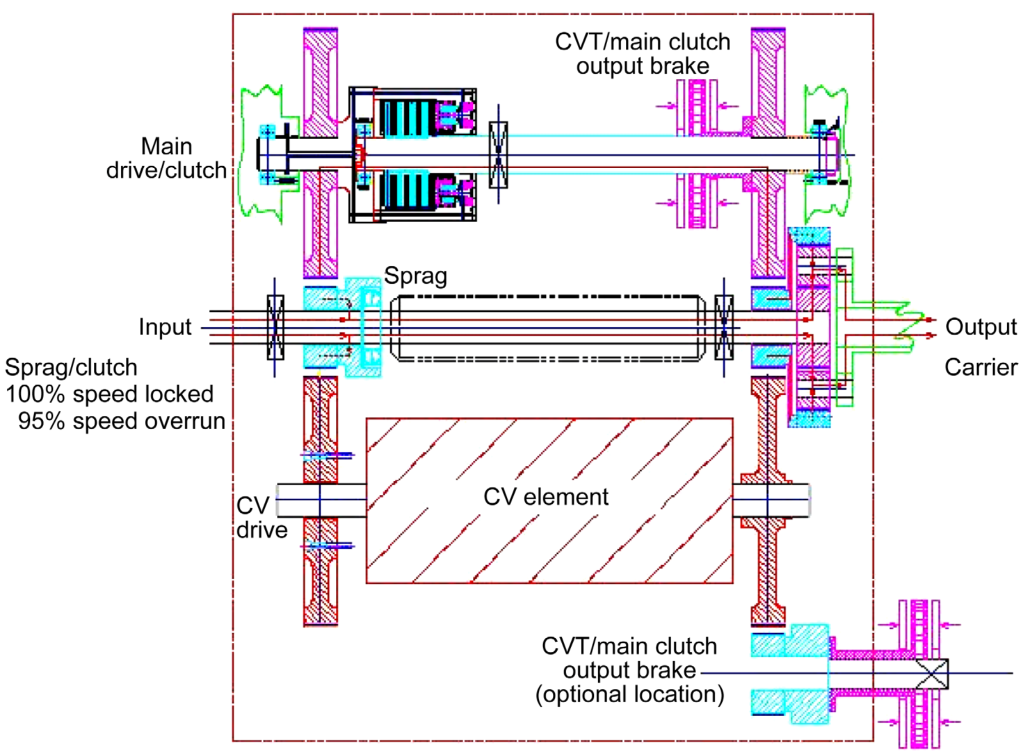


Figure 20.—Differential drive with continuously variable element.

Low speed—In the low-speed mode, power is transferred through the central shaft and planetary gear train. In this mode, only the ring gear is locked and power is transferred through the planetary system with the output through the carrier.

Speed transition—In the transitional mode, power is split between the main shaft and the CV element combining in the planetary gear train. Partial power is transferred from the main shaft driving the sun gear and partial power is transferred from sprag to CV element to the ring gear. The above power split is combined at the planetary gear train and output through the carrier.

As with concept 1, the concept as shown has 3:1 ratio output when ring gear is at rest and can obtain 50 percent speed output with ring gear rotating at 16.67 percent of input speed. To achieve 50 percent would require the CV element to provide the required power.

Driveline Concept 6—Two-Speed Drive (Compound Gear)

The two-speed drive (compound gear) concept, shown in figure 21, is a discrete two-speed configuration, which includes a speed synchronizer device (to be determined) to match shaft speeds when engaging and disengaging clutches to achieve the smoothest speed range transitions possible for two-speed discrete device. This configuration utilizes a conceptual dual-output element clutch configuration, which has the capability to direct output power to either of the two output elements controlled from within a single clutch unit. It operates in a mutually exclusive manner. The clutch configuration depicted in this concept was initially conceived for another transmission concept was reconfigured for this one. Clutch operation is described below as it pertains to this concept. Several concept clutches are discussed in more detail in the previous dedicated section.

This is an inline configuration. Low-speed gear reduction is accomplished using a two-stage by 1.412 ratio (8.5-in.-pitch diam. and 12.0-in.-pitch diam.) reduction. Power is transferred through a clutch, a double-reduction gear train, and then through the sprag. High-speed (1:1) output is transmitted directly through the clutch. The gear at the input shaft (high-speed shaft) rides on rolling-element bearings and is coupled directly to the high-speed output element of the concept dual-output element clutch. The gears on the loop shaft (low-speed shaft) are rigidly coupled to their shaft, which spins in rolling-element bearings. A third shaft, identical to the above loop shaft, is coupled to a speed synchronizing device, which is used to smooth the transition between the high- and low-speed ranges.

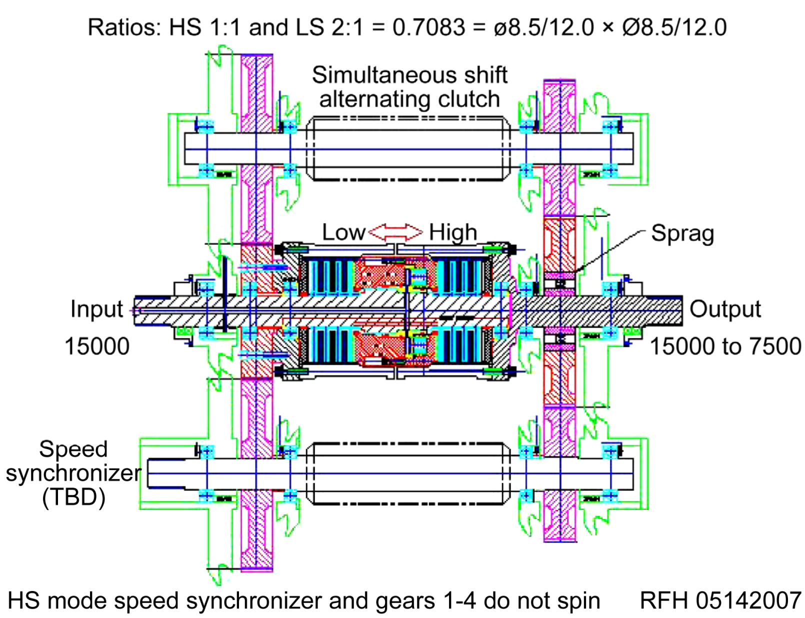


Figure 21.—Two-speed drive (compound gear).

A high-speed range is achieved by engaging the clutch high-speed output element while the low-speed output element is disengaged. Conversely, a low-speed range is achieved with the clutch low-speed output element engaged and the high-speed output element disengaged. During high-speed operation, the power is transferred through the clutch high-speed output element and the output ratio is 1:1. During low-speed operation, the clutch low-speed output element is engaged and power is transferred through two gear reductions of 1.412:1 in series (8.5-in.-pitch diam. and 12.0-in.-pitch diam.). Speed matching is achieved through shaft speed matching using a synchronizing device (e.g., CV traction drive) to achieve speed synchronization between the clutch power input shaft and either of the two power output elements. While in either high- or low-speed range, one of the clutch output elements is always engaged while the other is disengaged. Clutch design dictates engagement of only one range while the other is automatically disengaged. Other than during transition, power is either transferred through directly (high-speed output element engaged) or through the gear mesh (low-speed output element engaged).

The only exception to mutual exclusive operation of the output elements is the short duration of time during the shift. In the concept this period, or duration, and the resulting smoothness is a key factor that will need to be tuned in the analysis of the overall system dynamics. Smoothness can be improved at the cost of an increase in clutch slippage. Ideally, during transition between high and low speed, power is transferred through the speed synchronizer (CV element) until input and output shaft speed synchronization is achieved. If the speed synchronizing device is incapable of passing the required power, then clutch slippage can be tuned to approach a smoother transition. The transition from low to high speed being the most severe of the two speed changes as this would commonly be a full-power transition.

Positive aspects of this configuration are that during high-speed operation, or hover mode, power is transferred via direct drive without going through a gear mesh. In addition, the sprag is being overrun, thus the gears within the low-speed loop do not spin. During low-speed operation, or cruise mode, where full power is not required, power is passed through the gear train. Thus gearing may possibly be of lighter duty resulting in lower weight as well as lower cost in hardware and maintenance. Power requirements for the high-speed range are full, and reduced power is required for the low-speed range, thus, power routing is optimally directed. This assumes that the primary mission is short range.

Proposed operation for a tilt rotor application (per Robert F. Handschuh)

Start engine with hydraulic clutch #1 engaged

- Input speed = output speed
- Sprag clutch overruns
- Speed synchronize and gears 1, 2, 3, and 4 do not spin

Transition from helicopter mode to cruise

- Feather back the rotor speed (reduced torque required)
- Start to spin up the speed synchronizer and gears
- Throttle back the engine
- When sprag starts to drive, release clutch #2
- When engine input speed matches clutch #1 speed, lock up clutch #1
- Increase engine speed back to full throttle

Transition from cruise to hover

- Reduce torque at propeller (feather propeller)
- Reduce engine speed
- When engine speed equals output shaft speed, release clutch #1, engage clutch #2, and overrun sprag

Driveline Concept 7—Two-Speed Planetary Drive (ref. 3)

The Bossler concept is shown in figure 22. The high- and low-speed power flows are shown in figure 23.

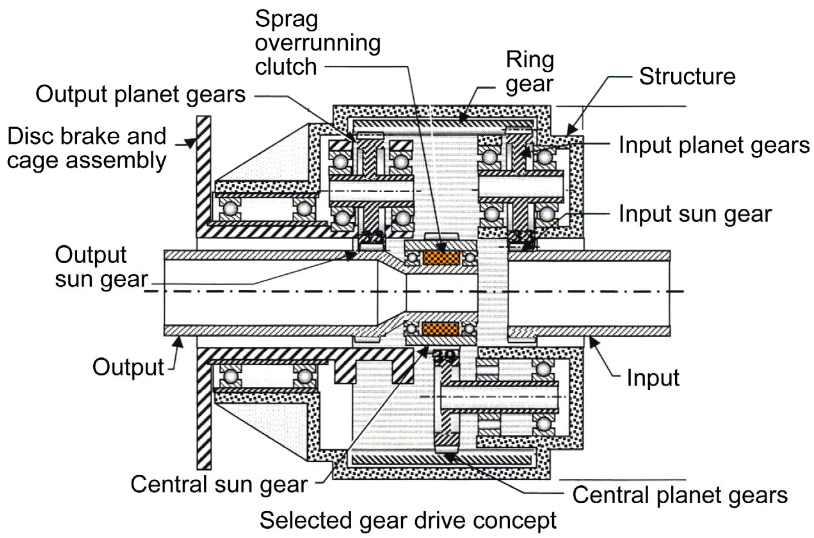


Figure 22.—Bossler planetary drive concept.

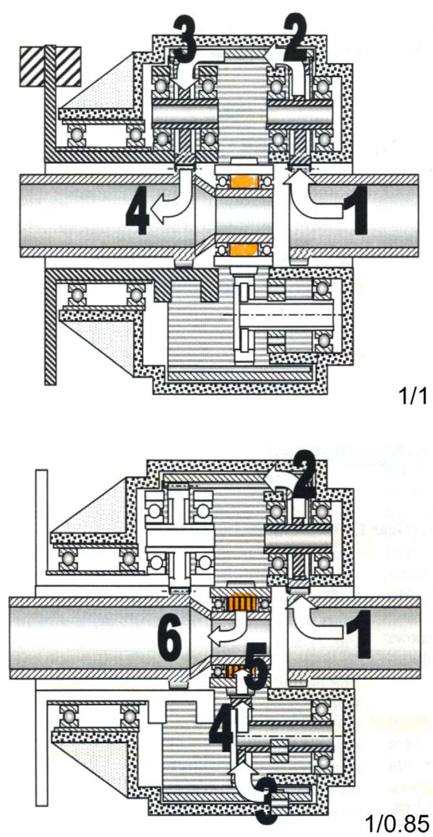


Figure 23.—High- and low-speed power flow.

Bossler Planetary

One Glenn concept is similar in some respects yet quite different with advantage

- Bossler
  - 100 percent/85 percent
- Glenn
  - 100 percent/50 percent (Concept 2B, “Two-stage counter-rotating planetary”)
- Similarities
  - Both utilize multiple planetary stages
  - Both would use similar mode of changing speed ranges (clutch and brake)
- Differences
  - Bossler uses a brake and Glenn uses a clutch; both require slippage
- Glenn advantages
  - Hover power 1:1 going through the locked multidisc clutch—not gear train
    - Highest loading does not go through the gear train in Glenn concept
    - Highest loaded gears are sun gears in the Bossler concept
- Glenn disadvantages
  - Complex hydraulic system required to release clutch (seals prone to failure)
    - Clutch is disengaged during cruise (hydraulic lines pressurized)
    - Desirable to have the ability to lock out the clutch

Traction Drive Variators

Variator Concept 999—Variator (Toroidal)

Utilizing a CV drive as a speed-changing device, or variator element, in a split-power two-speed transmission is considered a viable configuration to meet the requirements of this task. The CV drive (CV element) is used to transition between two fixed speeds of a two-speed device. The benefit of this configuration is to capture the highly desirable aspects of a CV traction device with its disassociation from the constraints of gear tooth geometry, fixed ratios, yet exploit the power capacity and reliability of gearing. In addition, capitalizing on other benefits of variable-speed ratio such as smooth and continuous transition yet having positive mechanical drive for the major portion of the mission (i.e., high- and low-speed output range). Operation in the high and low speeds comprises the majority of any mission. The speed range transition period, a minor portion of the operational mission, is primarily important as a highly desirable operational characteristic, that being smooth and continuous speed range changes.

Based upon review of many variable-speed mechanisms, a configuration of the toroidal-drive-based variator is selected as the most plausible to meet the objectives of the above configuration. The development of a toroidal variator for this task is limited to geometric concepts and integration into the concept drivelines. A basic conceptual layout of the variator was made to the scale of the driveline concepts being pursued. The initial concept for a toroidal unit was based on a textbook schematic in figure 24.

The configuration comprises two opposed concave toroidal contoured discs between multiple rollers. The volume contained between the two concave surfaced discs is in the form of a torus. The rollers, located within the torus volume between the two discs, transfer torque from one disc to the other via traction or friction force. In application, power is transmitted through fluid traction or fluid shear, not metal-to-metal friction. The rollers are mounted on individual trunnions, which are synchronized, thus permitting them to be simultaneously pivoted. When the rollers are oriented parallel to the axis of rotation of the discs, contacting points on each disc are at the same radius and the resulting drive ratio is 1:1. By pivoting the rollers toward the aft disc (clockwise), the contact radius on the forward disc is greater than when the rollers are parallel to the axis of rotation and conversely the contact radius at the aft disc if smaller. This results in the aft disc spinning faster than the forward disc. When the rollers are pivoted the opposite direction (counter clockwise), the aft disc spins slower than the forward disc. The ratio is proportional to the angle of the roller pivot.

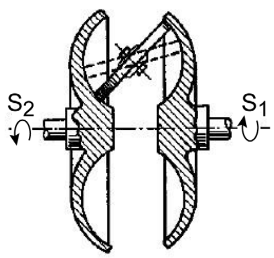


Figure 24.—Toroidal drive (ref. 6).

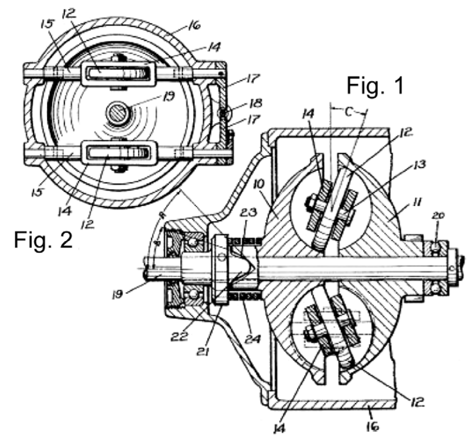


Figure 25.—Toroidal drive, Dodge (ref. 7).

Initially, a basic scaled concept was created based on a two-roller configuration with a crank and bevel gear arrangement to angle the rollers. Although this unit in itself was not deemed acceptable for full-power transfer, it was felt that it could be used as a transitioning element with a differential gear train of some sort in a split-power configuration. With that application in mind, the concept was evolved into a three-roller configuration. Roller pivoting was again accomplished with bevel gearing.

Upon continued search of the open literature, it was discovered that a variable-speed transmission based upon toroidal drive was patented by Adiel Dodge from 1932 to 1939. Figure 25 shows a small portion of the artwork depicting the basic two-roller configuration.

It should be noted that the above is much more complete, insightful, and elegant than the image above might suggest. Dodge's considerations included configurations for a three-roller system, and reversing the output rotation that were independently paralleled.

The above concept is considered a full toroidal configuration. In the open literature, similar variable-drive configurations can be found employing a half-toroidal geometry where the rollers are contoured such that the contact (line of action) is at an angle with the roller axis of rotation and contacts the torus below its inflection point (lowest point on the contour) on the forward and aft contoured discs. The geometry of the half-toroidal is smaller, therefore, is lighter and has less inertia. In practice, it may not necessarily be able to transmit the most power. It would seem that the geometry is such that side loads on the rollers are higher than that of the full toroidal configuration. The contact force acts normal to the surface and the vector does not pass through the axis of the roller thus side loads are much higher than that of the full-toroidal variator. The rollers employed in a half-toroidal variator configuration rotate on a smaller contact radius than that of a full toroidal. The angle through which the rollers are rotated to achieve the output ratio is less of an angle and as a result has less overall ratio range potential. In both the full- and half-toroidal configuration, the neutral position of the rollers, or 1:1 ratio position, is when the rollers are oriented parallel to the axis of rotation of the discs, (i.e., their pivot axis is perpendicular to the overall drive axis).

As in the multiplate clutch, where the torque capacity is increased by adding plates, the capacity of the toroidal drive can similarly be increased by stacking (ref. 8). Though not necessarily for this application, a potential advantage of the twin toroidal drive is that power may be taken off between the twins in lieu of the end. The down side is that the output rotation is again reversed. Though, if output is taken in this manner and passed through a planetary gear train, the output rotation direction can be reversed. A configuration such as this was considered but is not presented.

Applying a CV drive as a variator in the split-power two-speed configurations created to meet the objectives of this effort requires that the CV element output be the same as the input. A major drawback in applying the basic torus drive configuration as a variator in a split-power transmission configuration is that the output disc rotates opposite to that of the input disc. The drawback being a reversing feature is needed if the concept is to be employed in parallel as part of a power split shaft system. The reversing feature is ultimately additional complexity, weight, and frictional losses. A few concepts for reversing the direction of output rotation were considered for this specific application and another. These are discussed in another section.

Figure 26 shows a conceptual torus-based variator employed in a few of the driveline concepts developed for this task. This configuration, one of several generated, is an end-output configuration with a reversing gear system also depicted. Design and development of a toroidal variator will be a formidable task in itself.

Variator Concept 999—CVT Goi, Tatsuhiko

A search of the open literature revealed another variator drive concept (ref. 8). It is described as "…equipped with a power split type CVT, which consists of a high-speed traction drive variator (half-toroidal type) and a planetary differential gear unit."

The above paper presents a CV drive element, which incorporates an inline twin CV configuration. The benefit of the above is that it doubles the torque capacity of the CV element by increasing the number of traction rollers. Doubling the torque capacity is accomplished and at the cost of also nearly doubling the weight of the CV element. Although a severe penalty in weight, this configuration is very desirable due to the low torque capacity of the CV element drive in general. The above uses the CVT as a range transitioning element in lieu of attempting a configuration which is a full-time traction drive CVT.

An intriguing aspect of the paper is that the basic configurations share some similarities to NASA concepts generated independently. Although Goi's work precedes the NASA concepts by over 10 years and the maturities of the work are vastly different, it is reinforcing to see that some of the directions are somewhat similar in design philosophy. This is taken to suggest the best configuration for the application will most probably incorporate these features.

- Incorporating a planetary differential is the right path to pursue.
- Striving for the most robust CV element possible and using it only during transition or in a split-power transmission mode.

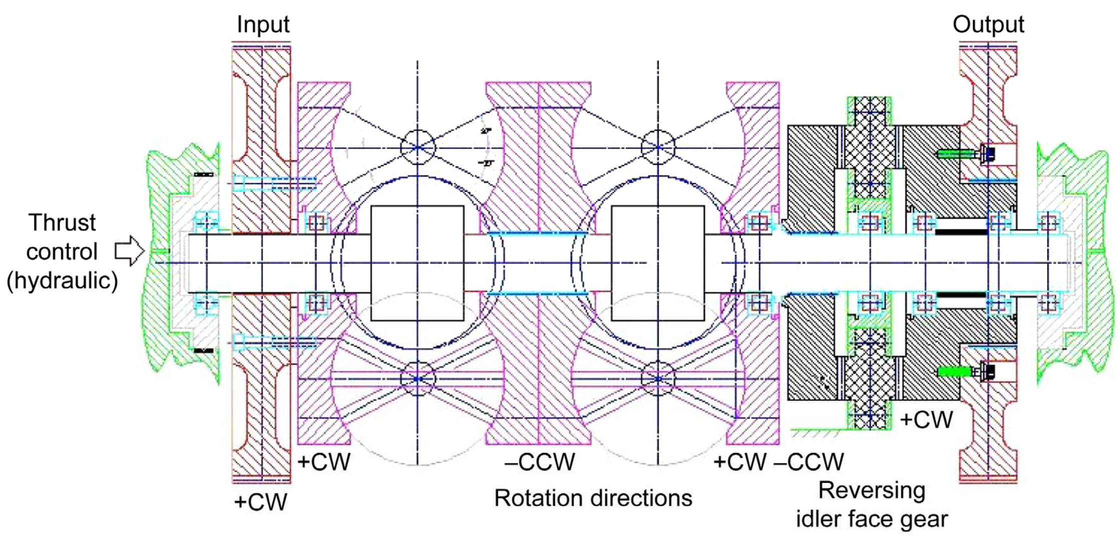


Figure 26.—Variator (twin-toroidal drive) concept.

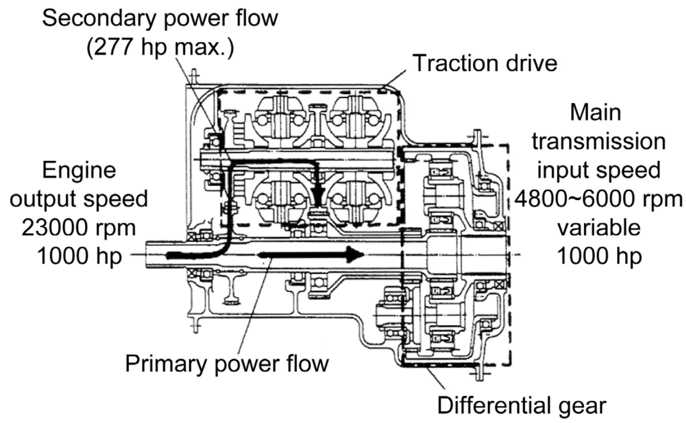
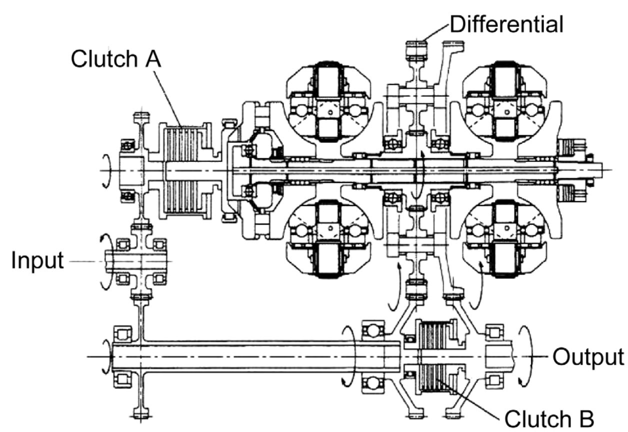


Figure 27.—Speed variator module (SVM), Goi (ref. 8).



Straight through mode: clutch A: OFF
(100% rotor speed) clutch B: ON

Straight variable mode: clutch A: ON
(variable rotor speed) clutch B: OFF

High-efficiency SVM

Figure 28.—High-efficiency speed variator drive (ref. 8).

Similarities between the Goi and NASA CV concepts.—Goi (fig. 27) always runs power through the planetary train and CV drive. The NASA planetary differential concept uses a single clutch and spins gears for high speed. Goi (fig. 28) uses two clutches and has a direct drive for high speed. NASA's dual clutch concept uses two clutches and also has direct drive for high speed. Goi uses the half-toroidal drive and NASA uses the full-toroidal drive. Why? Full-torus appears better than half torus considering axial forces—but why doesn't Goi use it?

In lieu of a true CVT traction drive, a key to a viable CVT may be to synthesize variability with a planetary gear system and a variable-speed element to control system output speed (e.g., a sun drive with carrier output and variable ring speed).

REPORT DOCUMENTATION PAGE	*Form Approved* *OMB No. 0704-0188*

1. REPORT DATE *(DD-MM-YYYY)*	2. REPORT TYPE	3. DATES COVERED *(From - To)*
01-03-2009	Technical Memorandum	

4. TITLE AND SUBTITLE
Variable/Multispeed Rotorcraft Drive System Concepts

5a. CONTRACT NUMBER

5b. GRANT NUMBER

5c. PROGRAM ELEMENT NUMBER

6. AUTHOR(S)
Stevens, Mark, A.; Handschuh, Robert, F.; Lewicki, David, G.

5d. PROJECT NUMBER

5e. TASK NUMBER

5f. WORK UNIT NUMBER
WBS 877868.02.07.03.01.01

7. PERFORMING ORGANIZATION NAME(S) AND ADDRESS(ES)
National Aeronautics and Space Administration
John H. Glenn Research Center at Lewis Field
Cleveland, Ohio 44135-3191

8. PERFORMING ORGANIZATION REPORT NUMBER
E-16649

9. SPONSORING/MONITORING AGENCY NAME(S) AND ADDRESS(ES)
National Aeronautics and Space Administration
Washington, DC 20546-0001
and
U.S. Army Research Laboratory
Adelphi, Maryland 20783-1145

10. SPONSORING/MONITORS ACRONYM(S)
NASA, ARL

11. SPONSORING/MONITORING REPORT NUMBER
NASA/TM-2009-215456; ARL-TR-4758

12. DISTRIBUTION/AVAILABILITY STATEMENT
Unclassified-Unlimited
Subject Categories: 07 and 37
Available electronically at http://gltrs.grc.nasa.gov
This publication is available from the NASA Center for AeroSpace Information, 301-621-0390

13. SUPPLEMENTARY NOTES

14. ABSTRACT
Several recent studies for advanced rotorcraft have identified the need for variable, or multispeed-capable rotors. A speed change of up to 50 percent has been proposed for future rotorcraft to improve vehicle performance. Varying rotor speed during flight not only requires a rotor capable of performing effectively over the extended operation speed and load range, but also requires an advanced propulsion system to provide the required speed changes. A study has been completed, which investigated possible drive system arrangements to accommodate up to the 50 percent speed change. These concepts are presented. The most promising configurations are identified and will be developed for future validation testing.

15. SUBJECT TERMS
Rotorcraft; Variable speed; Propulsion

16. SECURITY CLASSIFICATION OF: a. REPORT	b. ABSTRACT	c. THIS PAGE	17. LIMITATION OF ABSTRACT	18. NUMBER OF PAGES	19a. NAME OF RESPONSIBLE PERSON
U	U	U	UU	44	STI Help Desk (email:help@sti.nasa.gov)
					19b. TELEPHONE NUMBER *(include area code)* 301-621-0390