



# Variable-Speed Simulation of a Dual-Clutch Gearbox Tiltrotor Driveline

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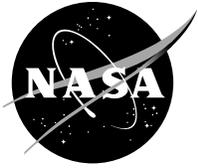
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## Abstract

This investigation explores the variable-speed operation and shift response of a prototypical two-speed dual-clutch transmission tiltrotor driveline in forward flight. Here, a Comprehensive Variable-Speed Rotorcraft Propulsion System Modeling (CVSRPM) tool developed under a NASA funded NRA program is utilized to simulate the drive system dynamics. In this study, a sequential shifting control strategy is analyzed under a steady forward cruise condition. This investigation attempts to build upon previous variable-speed rotorcraft propulsion studies by 1) including a fully nonlinear transient gas-turbine engine model, 2) including clutch stick-slip friction effects, 3) including shaft flexibility, 4) incorporating a basic flight dynamics model to account for interactions with the flight control system. Through exploring the interactions between the various subsystems, this analysis provides important insights into the continuing development of variable-speed rotorcraft propulsion systems.

## Introduction

According to a recent NASA-Army-Industry-University investigation, significant benefits to rotorcraft operational performance, effectiveness and acoustic signature could be gained through the ability to adjust main rotor speed to accommodate various flight conditions (Ref. 1). In particular, variable-speed rotor technology is critical to the slowed-rotor compound configuration concepts and would offer significant benefits to future Heavy-Lift helicopter and tiltrotor configurations (Ref. 2). Current rotorcraft propulsion systems are designed around a fixed-ratio transmission without the capability to vary rotor speed except by engine speed adjustment. Since the specific fuel consumption of modern gas-turbine engines is optimum within a relatively narrow speed range, the ability to achieve a variable-speed rotor

through engine speed adjustment is limited (Refs. 3 and 4). Therefore, to achieve effective variable rotor speed designs, a variable ratio transmission in the rotorcraft propulsion system becomes attractive. The NASA Heavy-Lift study (Ref. 1) suggests that transmission designs capable of rotor speed variations on the order of 50 percent will be sufficient to achieve significant rotorcraft performance and noise benefits.

Recently, Stevens, Handschuh, and Lewicki (Ref. 5) presented several variable and multi-speed rotorcraft drive system concepts. Based on relative simplicity and necessary requirements of positive drive (i.e., no traction or fluid-based power transmission) for rotorcraft applications, it was concluded that one promising configuration would incorporate a planetary drive with a discrete two-speed shift capability. Therefore, a two-speed dual-clutch planetary gear transmission of the type evaluated by Lewicki, et al. (Ref. 6) is considered in the current investigation. In Lewicki, et al. (Ref. 6), a model and analysis of the dual-clutch shifting dynamics including nonlinear dry clutch stick-slip friction and dual-clutch control is developed and experimentally validated on a purpose-built test rig at the NASA Glenn Research Center. Furthermore, the shift dynamics of similar dual-clutch transmissions for automotive applications were explored (Refs. 7 and 8). One advantage offered by dual-clutch transmissions (DCT) is their ability to provide continuous power transfer when transitioning from one gear ratio to another. Depending on the dual-clutch shift control strategy, a pure neutral condition can be avoided during the shifting process, a seeming obvious benefit for rotorcraft applications. Generally speaking, to accommodate a gear ratio change, it is kinematically necessary for either the engine speed or the load speed to vary. For example, during an upshift from a lower gear to a higher gear, either the load speed must increase or the engine RPM must decrease (or some combination therein). In automotive systems, since the engine rotary inertia is typically much lower than the effective vehicle inertia, the engine speed primarily changes during clutch engagement.

Once clutch lockup in the new gear ratio occurs, the engine speed is then adjusted to obtain the desired vehicle speed.

In the case of rotorcraft, since it is important to maintain powered flight throughout the shift process and not to over-speed the engine power turbines, Litt, et al. (Ref. 9) proposed an engine-at-a-time or sequential shifting control (SSC) for multi engine helicopters. This approach takes advantage of the existing drive system free-wheeling clutch connections to decouple individual engines from the transmission thus allowing sequential gear ratio changes on individual transmission paths. In particular, Litt, et al. (Ref. 9) explored this concept for a 3000 hp class, two-engine single rotor helicopter using an integrated system model consisting of a linearized engine, rigid gear transmission, rotor and idealized clutches to capture the basic dynamics. The advantages of this approach are that: 1) it allows for powered flight to always be maintained on the reduced set of engines, and 2) it avoids full power transfer across engaging clutches.

The purpose of the current investigation is to explore the variable-speed operation and shift response of a prototypical two-speed DCT tiltrotor driveline in forward flight airplane mode. Here, a Comprehensive Variable-Speed Rotorcraft Propulsion System Modeling (CVSRPM) tool developed under a NASA-funded NRA program is utilized to simulate the drive system dynamics under a two-speed shift SSC strategy for several mission scenarios.

As discussed in References 1, 3, and 4, one of the potential beneficiaries of variable-speed rotor drive-system technologies is Heavy-Lift rotorcraft. Therefore, this study considers a two-speed version of the 30,000 hp-class NASA Large Civil Tiltrotor Iteration 2 (LCTR-2) (Fig. 1, (Ref. 4)) and analyzes the transition to and from slowed rotor cruise operation. The simulations in this paper include a stick-slip dry clutch friction DCT model based on Lewicki, et al. (Ref. 6), a new fully nonlinear first principles-based transient gas turbine engine model, cross shaft flexibility effects, and an idealized uniform inflow forward flight rotor/propeller model.



Figure 1.—NASA Large Civil Tiltrotor LCTR-2 (Ref. 4).

## Two-Speed Tiltrotor Simulation

A schematic of the two-speed dual-path tiltrotor driveline considered in this investigation is shown in Figure 2. Each path consists of an engine nacelle containing a pair of gas turbine engines coupled to an initial reduction gearbox, ratio  $n_i$ , via freewheeling clutch units (FWU), see Appendix A, Nomenclature.

$$n_i = \frac{\Omega_{DCT,in}}{\Omega_{FWU,out}} = 1.87 : 1 \quad (1)$$

Speed is then further reduced by a two-speed DCT gearbox with low and high ratios

$$n_{DCT,low} = \frac{\Omega_{DCT,low}}{\Omega_{DCT,in}} = 0.383, \quad n_{DCT,high} = \frac{\Omega_{DCT,high}}{\Omega_{DCT,in}} = 0.71 \quad (2)$$

The final reduction, ratio  $n_f$ , before the main rotor is represented as a high reduction ratio planetary gearbox

$$n_f = \frac{\Omega_{MR}}{\Omega_{DCT}} = 25 : 1 \quad (3)$$

Finally, the left and right side driveline paths are linked via a cross-shaft.

The cross-shaft is located downstream from the DCT speed changing units (see Fig. 2) to prevent relative rotor windup during shift events thus ensuring that rotor indexing phase is maintained.

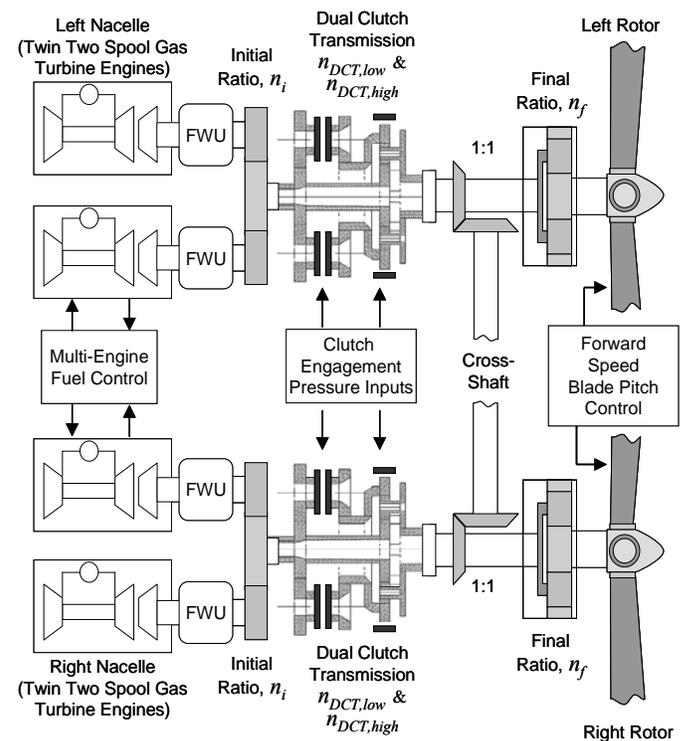


Figure 2.—Two-speed DCT dual-path tiltrotor driveline.

To simulate forward flight operation, the propulsion system model is coupled to a simplified flight dynamics block to account for vehicle altitude and forward speed regulation under variable rotor speed operation. Since all engines were considered to be identical, only one engine-per-side was simulated in the model, with appropriate output torque and inertia effects to account for both engines. Figure 3 shows the architecture of the overall propulsion system coupled with a vehicle flight dynamics block and associated closed-loop flight control systems and command inputs.

A description of the main elements and some key assumptions included in the simulation model are as follows:

- Two-spool gas turbine engine transient dynamics model with closed-loop PID-based fuel controller to regulate power turbine speed  $\Omega_{pt}$  and multi-engine load sharing.
- DCT model with nonlinear stick-slip friction and clutch lockup effects and open loop clutch engagement pressure command inputs (Ref. 6).
- Idealized rigid gearing (tooth meshing compliance neglected).

- Main rotor idealized as rigid rotational inertia with aerodynamic torque computed based on momentum theory with uniform inflow (Ref. 10).
- Torsionally flexible cross-shafting.
- Simplified vehicle flight dynamics model with horizontal and vertical degrees of freedom (two DOF) accounting for fuselage drag, vehicle weight, rotor thrust and wing lift.
- A PID-based main rotor blade pitch controller to regulate rotor thrust and track a desired forward speed set-point (controls main rotor pitch  $\theta_p$ ).
- A PID-based wing elevon pitch controller to maintain flight level or track a desired climb rate set-point (controls wing flap angle  $\theta_w$ ).

The simulation model was developed in the Matlab Simulink (The MathWorks, Inc.) environment. Appendix B shows the system and subsystem block diagrams. Further details of the propulsion system model are given in the subsequent subsections.

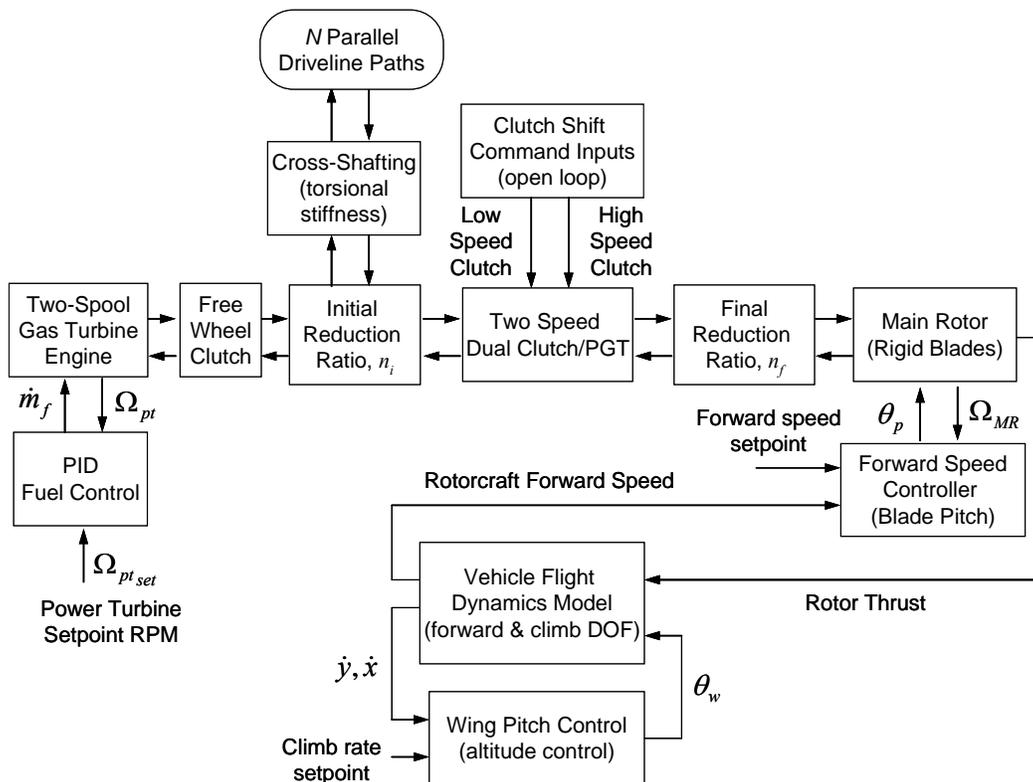


Figure 3.—Propulsion system/vehicle dynamics/flight control block diagram assembled in the CVSRPM code.

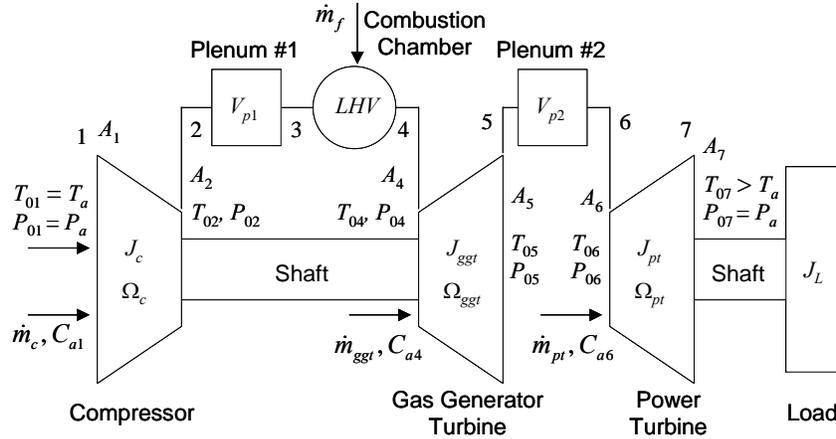


Figure 4.—Two-spool gas turbine engine model components and analysis stations.

## Gas Turbine Engine Model

To model the gas turbine engines and fuel control dynamics during transient operation, a generic two-spool gas turbine engine model was developed. The components of this model, shown in Figure 4, include the axial compressor, combustion chamber, gas-generator turbine, power turbine and two mass accumulators (plenums). A dynamic model is formulated by considering the torque balances between the compressor and gas generator turbine as well as between the power turbine and load. These torques are computed based on first principles mechanical and thermodynamic laws following a similar approach as in reference (Ref. 6). The thermodynamic model is based on the standard Joule/Brayton cycle (Ref. 11). The main assumptions made in this analytical model can be summarized as follows:

Engine model assumptions:

- The air mass flows through the system are all one-dimensional at the mean line.
- Air with ideal gas law behavior is assumed.
- All expansion and compression is adiabatic.
- The inlet stagnation temperature and pressures are constant.
- All power transfer occurs in the rotor stages, the stators do no work.
- The instantaneous compressor mass flow, pressure ratio and efficiency are determined via pre-calculated beta maps.
- Gas generator and power turbines are in a state of choking flow (Ref. 11)
- No air bleed occurs from the compressor.
- The combustion is modeled as isobaric heat addition.
- The specific heat of air was assumed constant throughout the engine model.

The power required to drive the compressor is

$$W_c = \dot{m}_c c_p (T_{02} - T_{01}) \quad (4)$$

Where  $\dot{m}_c$  is the compressor air mass flow rate,  $c_p$  is the specific heat of air and  $T_{01}$  and  $T_{02}$  are the stagnation temperatures at the compressor entrance and exit respectively. Furthermore the stagnation temperature ratio is related to the compressor pressure ratio by

$$\frac{T_{02}}{T_{01}} = 1 + \frac{1}{\eta_c} \left[ \left( \frac{P_{02}}{P_{01}} \right)^{\frac{\gamma-1}{\gamma}} - 1 \right] \quad (5)$$

Where  $\eta_c$  is the compressor isentropic efficiency,  $\gamma$  is the specific heat ratio, and  $P_{01}$  and  $P_{02}$  are the stagnation pressures at the compressor entrance and exit respectively. Based on (4) and (5), the torque required to drive the compressor is

$$TQ_c = J_c \dot{\Omega}_c + \dot{m}_c c_p \frac{T_{01}}{\Omega_c \eta_c} \left[ \left( \frac{P_{02}}{P_{01}} \right)^{\frac{\gamma-1}{\gamma}} - 1 \right] \quad (6)$$

Where  $\Omega_c$  is the compressor rotation speed and  $J_c$  is compressor rotational inertia. Also, in the above equations the stagnation temperature and pressures at the compressor entrance are assumed to be the ambient atmospheric pressure,  $P_{01}=P_a$ , and temperature  $T_{01}=T_a$  values.

The basic engine parameters, summarized in Table 1, are selected based on the NASA LCTR-2 heavy lift concept engine prototype (Refs. 12 and 13).

TABLE 1.—ENGINE NOMINAL PARAMETERS

Parameter	Value
Ambient temperature, $T_a$	288° K
Ambient pressure, $P_a$	1.023 bar
Nominal output (load) power, $W_L$	7500 hp
Nominal compressor mass flow, $\dot{m}_{cn}$	13.5 kg/s
Nominal compressor pressure ratio, $pr_{cn}$	28
Nominal compressor axial velocity, $C_{an}$	150 m/s
Number of compressor stages, $N_s$	8
Compressor mean radius, $R_c$	0.119 m
Compressor inlet hub-to-tip ratio, $ht_c$	0.35
Nominal compressor speed, $\Omega_{cn}$	26,500 RPM
Power turbine mean radius, $R_{pt}$	0.1811 m
Power turbine inlet hub-to-tip ratio, $ht_{pt}$	0.65
Nominal power turbine speed, $\Omega_{pt}$	12,500 RPM
Compressor rotational inertia, $J_c$	0.641 kg m <sup>2</sup>
Gas generator turbine rotational inertia, $J_{ggt}$	0.144 kg m <sup>2</sup>
Power turbine rotational inertia, $J_{pt}$	1.04 kg m <sup>2</sup>
Plenum chamber volume, $V_p$	0.1 m <sup>3</sup>
Combustion chamber loss coefficient, $\eta_{cc}$	0.85
Combustion chamber time constant, $\tau_{cc}$	$5.0 \times 10^{-4}$ sec
Fuel lower heating value, $LHV$	43.1 MJ/kg
Nominal fuel rate, $\dot{m}_{fn}$	0.225 kg/sec
Maximum fuel rate, $\dot{m}_{fmax}$	0.338 kg/sec

Furthermore, the compressor mass flow rate,  $\dot{m}_c$ , pressure ratio,  $pr_c$ , and efficiency characteristics,  $\eta_c$ , are computed by way of so-called beta maps (Ref. 14) with functional dependencies,

$$\begin{aligned} pr_c &= f_{pr}(T_a, P_a, \Omega_c, \beta_c) \\ \dot{m}_c &= f_m(T_a, P_a, \Omega_c, \beta_c) \\ \eta_c &= f_\eta(T_a, P_a, \Omega_c, \beta_c) \end{aligned} \quad (7-a)$$

Where  $\beta_c$  is the compressor beta parameter which maps the mass flow to pressure ratio as

$$pr_c = \beta_c \dot{m}_c^2 \quad (7-b)$$

In this study, the compressor pressure and efficiency maps are generated numerically based on the nominal compressor design values. Figure 5 shows the pressure-ratio versus mass flow characteristic curves and beta lines for the 7500 hp gas turbine engine considered in this study and Figure 6 shows the compressor adiabatic efficiencies  $\eta_c$ .

To drive the compressor and load, separate gas generator and power turbines are present in the two-spool gas turbine engine (Fig. 4). The power generated from these turbines is

$$W_{ggt} = \dot{m}_{ggt} c_p (T_{04} - T_{05}) \quad \text{and} \quad W_{pt} = \dot{m}_{pt} c_p (T_{06} - T_{07}) \quad (8)$$

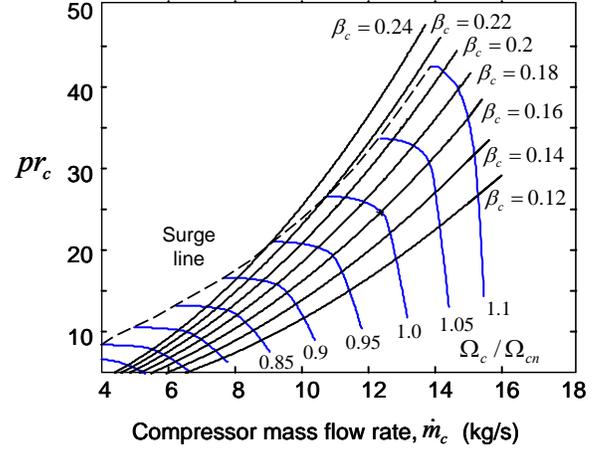


Figure 5.—Compressor pressure ratio versus air mass flow.

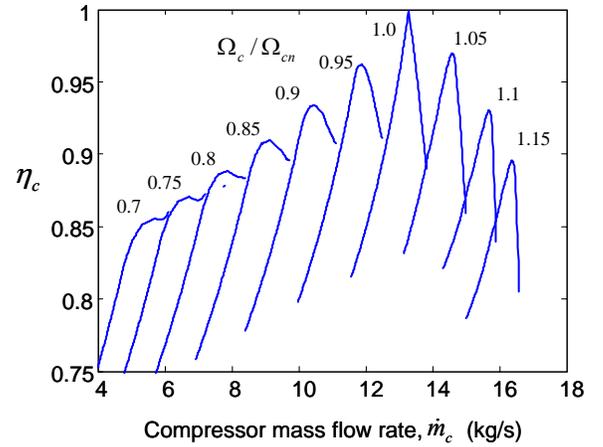


Figure 6.—Compressor adiabatic efficiency vs. mass flow.

Where  $\dot{m}_{ggt}$  and  $\dot{m}_{pt}$  are gas generator and power turbine mass flows, respectively, and  $T_{04}$ ,  $T_{05}$ ,  $T_{06}$  and  $T_{07}$  are turbine entrance and exit stagnation temperatures as defined in Figure 4. Furthermore, the turbine pressure ratios based on isentropic expansion through the turbines are

$$\frac{P_{04}}{P_{05}} = \left( \frac{T_{04}}{T_{05}} \right)^{\frac{\gamma}{\gamma-1}} \quad \text{and} \quad \frac{P_{06}}{P_{07}} = \left( \frac{T_{06}}{T_{07}} \right)^{\frac{\gamma}{\gamma-1}} \quad (9)$$

and the turbine mass flows are computed based on the assumption that both turbines are operating in the condition of choking flow (Ref. 11). Therefore

$$\dot{m}_{ggt} = \varphi \frac{A_4 P_{04}}{\sqrt{T_{04}}} \quad \text{and} \quad \dot{m}_{pt} = \varphi \frac{A_6 P_{06}}{\sqrt{T_{06}}} \quad (10-a)$$

with

$$\varphi \equiv \sqrt{\frac{\gamma}{R} \left(1 + \frac{\gamma-1}{2}\right)^{\frac{\gamma+1}{1-\gamma}}} \quad (10-b)$$

Where  $R$  is the ideal gas constant. Next, since the gas generator turbine and compressor are on a common spool, we have  $\Omega_c = \Omega_{ggt}$ . Also, the torque difference between the compressor and gas generator turbine is

$$(J_c + J_{ggt})\dot{\Omega}_c = \frac{1}{\Omega_c} (W_{ggt} - W_c) \quad (11)$$

and the torque difference between power turbine and load is

$$(J_{pt} + J_L)\dot{\Omega}_{pt} = \frac{1}{\Omega_{pt}} (W_{pt} - W_L) \quad (12)$$

Where  $W_L$  is the power consumed by the load and  $J_c$ ,  $J_{ggt}$ ,  $J_{pt}$  and  $J_L$  are rotational inertias of the compressor, gas generator turbine, power turbine, and the load, respectively.

During steady-state operation the air mass flow rates through the compressor and turbines are equal,  $\dot{m}_c = \dot{m}_{ggt} = \dot{m}_{pt}$ . However, during transient operation, differential mass flows can develop between the components due to volume packing (Refs. 14 and 15). To account for such transient effects, two plenum chambers (Plenum no. 1, situated between compressor and combustion chamber, and Plenum no. 2 between the two turbine units) are included in the model as depicted in Figure 4.

The plenum chambers are modeled as ideal mass accumulators with no losses or heat transfer. Figure 7 shows a schematic of the plenum chamber element with input and output mass flow rates  $\dot{m}_{in}$  and  $\dot{m}_{out}$  generated by the upstream and downstream elements. Due to the differential mass flows, the rate of change of density inside a plenum,  $\dot{\rho}_p$ , is given by

$$\dot{\rho}_p = \frac{\dot{m}_{in} - \dot{m}_{out}}{V_p} \quad (13)$$

Where  $V_p$  is plenum chamber volume. Furthermore, through differentiation of the ideal gas law and assuming adiabatic compression, the following differential equation for plenum chamber pressure  $P_p$  is obtained

$$\dot{P}_p(t) = \frac{\gamma(\dot{m}_{in} - \dot{m}_{out})}{m_{pa}(t)} P_p(t) \quad (14-a)$$

with

$$m_{pa}(t) = \int (\dot{m}_{in} - \dot{m}_{out}) dt + m_{pa0} \quad (14-b)$$

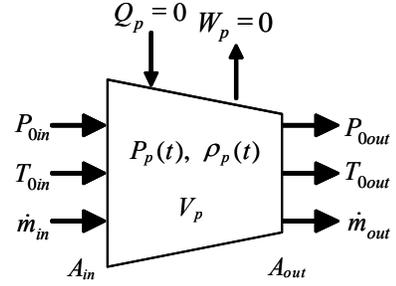


Figure 7.—Plenum chamber element.

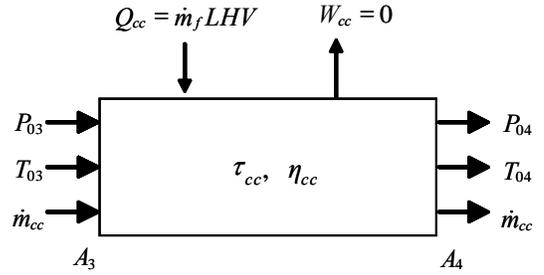


Figure 8.—Combustion chamber element.

Here  $m_{pa}$  is the mass of air accumulating inside the plenum and  $m_{pa0}$  is the initial air mass.

It is assumed that plenum pressure is uniform throughout its volume and that velocity effects and energy accumulation inside the plenum are negligible resulting in

$$P_{0in} = P_p \text{ and } P_{0out} = P_p \quad (15)$$

and

$$T_{0in} = T_{0out} \quad (16)$$

Where  $P_{0in}$ , and  $P_{0out}$  are plenum entrance and exit stagnation pressures and  $T_{0in}$ , and  $T_{0out}$  are the plenum entrance and exit stagnation temperatures.

Finally, the combustion chamber (entrance station 3, exit station 4 in Figs. 4 and 8) is modeled as a pure isobaric heat addition process. With heat addition,  $Q_{cc}$ , based on the idealized combustion model

$$Q_{cc} = \dot{m}_f LHV \quad (17)$$

Where  $\dot{m}_f$  is fuel mass flow rate and  $LHV$  is the fuel lower heating value.

As a result of an energy balance, the combustion chamber exit stagnation temperature  $T_{04}$  is determined as a solution to

$$\tau_{cc} \frac{dT_{04}}{dt} = \dot{m}_{cc} c_p (T_{03} - T_{04}) + \dot{m}_f LHV \quad (18)$$

Where  $\tau_{cc}$  is the combustion chamber time-constant and  $\dot{m}_{cc}$  is the air mass flow through the combustion chamber which equals the mass flow drawn by the gas generator turbine  $\dot{m}_{cc} = \dot{m}_{ggt}$ . Furthermore, the combustion chamber exit stagnation pressure,  $P_{04}$ , is computed from the input stagnation pressure  $P_{03}$  using,

$$P_{04} = \eta_{cc} P_{03} \quad (19)$$

Where  $\eta_{cc} < 1$  is the combustion chamber pressure drop ratio. Also, at steady-state the combustion temperature rise  $\Delta T_{0cc} = T_{04} - T_{03}$  is

$$\Delta T_{0cc} = f \frac{LHV}{c_p} \quad \text{with} \quad f = \frac{\dot{m}_f}{\dot{m}_{cc}} \quad (20)$$

Where  $f$  is the fuel-air ratio. Based on the equations presented in this subsection, the transient response of the two-spool gas turbine engine can be simulated. Please refer to the Appendix B for the full engine model block diagrams.

## Dual-Clutch Gearbox Model

As mentioned in the previous section, a two-speed DCT gearbox is utilized as the ratio changing element in each transmission path of the tiltrotor drive system (Fig. 2). A schematic of the DCT considered in this study is shown in Figure 9 along with the corresponding kinematic diagram in Figure 10. Here the carrier is the output. In the DCT, high ratio,  $n_{DCT,high}$  is achieved by engaging the multiple clutch disks that couple the planet shafts (clutch 1).

$$n_{DCT,high} = \frac{Z_i Z_{cg2} Z_r}{2(Z_{cg1} Z_{cg3} + Z_s)(Z_p + Z_s)} \quad (21-a)$$

Furthermore, low ratio,  $n_{DCT,low}$ , is achieved by fully engaging, a cylindrical clutch (clutch 2) which clamps the ring gear.

$$n_{DCT,low} = \frac{Z_s}{2(Z_p + Z_s)} \quad (21-b)$$

Clutch engagement is controlled by applied pressures,  $p_{cl1}$  and  $p_{cl2}$ , normal to the clutch friction surfaces. Due to the DCT kinematics, the only condition under which both sets of clutches can be engaged and locked simultaneously corresponds to the zero speed or fully stalled condition. When both clutches are applied simultaneously there is a negative recirculating power ratio and clutches act as a braking system. Thus, during steady-state constant speed operation, only one clutch is fully engaged and locked ( $p_{cli} = p_{clmax}$ ) while the other clutch remains open ( $p_{clj} = 0$ ).

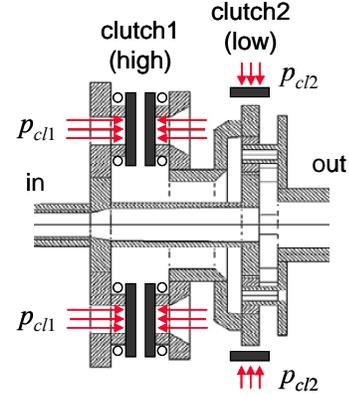


Figure 9.—Dual-clutch transmission.

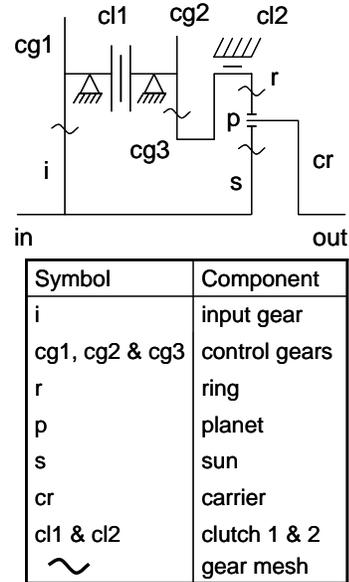


Figure 10.—DCT kinematic diagram.

To account for the DCTs in the driveline simulation, a rigid body dynamics model is developed which accounts for the DCT kinematics, the gear rotational inertias, and the nonlinear clutch friction torques.

$$\mathbf{M}\ddot{\mathbf{q}} = \mathbf{F}(\ddot{\mathbf{q}}, \dot{\mathbf{q}}, p_{cl1}, p_{cl2}) \quad (22-a)$$

with corresponding DOF vector

$$\mathbf{q}(t) = [\phi_i, \phi_r]^T \quad (22-b)$$

Where  $\phi_i$  and  $\phi_r$  are the DCT input and ring gear rotations. Also, the system inertia matrix is given by

$$\mathbf{M} = \begin{bmatrix} J_{in} + J_i + J_s + \frac{J_{cg1} N_{cg} Z_i^2}{Z_{cg1}^2} + \dots & \frac{Z_r Z_s}{4} \left( \frac{J_{cr} + J_{out} + N_p m_p R_{cr}^2}{(Z_p + Z_s)^2} - \frac{J_p N_p}{Z_p^2} \right) \\ \frac{Z_s^2}{4} \left( \frac{J_{cr} + J_{out} + N_p m_p R_{cr}^2}{(Z_p + Z_s)^2} + \frac{J_p N_p}{Z_p^2} \right) & \\ \frac{Z_r Z_s}{4} \left( \frac{J_{cr} + J_{out} + N_p m_p R_{cr}^2}{(Z_p + Z_s)^2} - \frac{J_p N_p}{Z_p^2} \right) & J_r + J_{cg3} + \frac{J_{cg2} N_{cg} Z_{cg3}^2}{Z_{cg2}^2} + \dots \\ \frac{Z_r Z_s}{4} \left( \frac{J_{cr} + J_{out} + N_p m_p R_{cr}^2}{(Z_p + Z_s)^2} + \frac{J_p N_p}{Z_p^2} \right) & \frac{Z_r^2}{4} \left( \frac{J_{cr} + J_{out} + N_p m_p R_{cr}^2}{(Z_p + Z_s)^2} + \frac{J_p N_p}{Z_p^2} \right) \end{bmatrix} \quad (22-c)$$

and the system generalized force vector is

$$\mathbf{F} = \begin{bmatrix} 1 & \frac{-Z_s}{2(Z_p + Z_s)} & \frac{-N_{cg} Z_i}{Z_{cg1}} & 0 \\ 0 & \frac{-Z_r}{2(Z_p + Z_s)} & \frac{N_{cg} Z_{cg3}}{Z_{cg2}} & -1 \end{bmatrix} \begin{bmatrix} TQ_{in} \\ TQ_{out} \\ TQ_{cl1} \\ TQ_{cl2} \end{bmatrix} \quad (22-d)$$

Where  $TQ_{in}$  and  $TQ_{out}$  are the torques on the DCT input and output shafts and where  $TQ_{cl1}$  and  $TQ_{cl2}$  are the transmitted clutch torques. To account for the possibility of clutch sticking (lock-up) or slipping, the clutch torques are computed based on a so-called combined Coulomb / viscous model (Refs. 6, 16, and 17)

$$TQ_{cl1} = \begin{cases} \tau_{cl} (\Omega_{cg2} - \Omega_{cg1}) & |\Omega_{cg2} - \Omega_{cg1}| < \varepsilon \\ p_{cl1} \mu_{cl} \frac{4\pi R_{cl1}^3}{3} \text{sgn}(\Omega_{cg2} - \Omega_{cg1}) & |\Omega_{cg2} - \Omega_{cg1}| \geq \varepsilon \end{cases} \quad (23)$$

$$TQ_{cl2} = \begin{cases} \tau_{cl} \Omega_r & |\Omega_r| < \varepsilon \\ p_{cl2} \mu_{cl} 2\pi R_{cl2}^2 L_{cl2} \text{sgn}(\Omega_r) & |\Omega_r| \geq \varepsilon \end{cases} \quad (24)$$

Here,  $\Omega_{cg1}$  and  $\Omega_{cg2}$  are the upstream and downstream clutch disk rotation speeds of clutch no. 1 and  $\Omega_r$  is the ring gear rotation speed at clutch no. 2. Here  $\mu_{cl}$  and  $\tau_{cl}$  are clutch friction and viscous torque coefficients (Ref. 16). Also, in clutch no. 1 (disk-type clutch) the torque is a function of the disk radius  $R_{cl1}$ , while for clutch no. 2 (cylindrical clutch) the torque is depends on clutch radius,  $R_{cl2}$ , and axial length  $L_{cl2}$ .

In each case, the clutch torque transitions from a viscous mode to a Coulomb friction mode based on a clutch slip velocity threshold parameter  $\varepsilon$ . Here, when clutch slip speed exceeds  $\varepsilon$ , the torque transitions from a viscous mode into the slipping mode (Coulomb friction). By proper selection of  $\varepsilon$  and  $\tau_{cl}$ , the viscous mode numerically approximates clutch lockup without no need for explicit model DOF reduction (Ref. 6). The DCT system parameters used in this investigation are summarized in Table 2.

TABLE 2.—DCT PARAMETERS.

Parameter	Value
Number of gear teeth	$Z_i=40, Z_{cg1}=42, Z_{cg2}=29, Z_{cg3}=52, Z_r=74, Z_p=14, Z_s=46$
Gear radii, m	$R_i=0.1, R_{cg1}=0.106, R_{cg2}=0.074, R_{cg3}=0.132, R_r=0.19, R_p=0.036, R_s=0.12, R_{cr}=0.152$
Rotational inertias, kg m <sup>2</sup>	$J_i=0.064, J_{cg1}=0.078, J_{cg2}=0.039, J_{cg3}=0.402, J_r=1.766, J_p=0.002, J_s=0.264, J_{cr}=0.848$
Number of planets	$N_p = 8$
Planet mass, kg	$m_p = 3.576$
Planet clutch radius (clutch 1), m	$R_{cl1} = 0.238$
Number of planet clutches (clutch 1)	$N_{cg} = 2$
Ring clutch dimensions (clutch 2), m	$R_{cl2} = 0.197, L_{cl2} = 0.1$
Clutch friction coefficient	$\mu_{cl} = 0.45$

### Forward Flight Rotor (Propeller) Model

To include the rotor torque and resulting thrust for the tiltrotor in forward flight airplane mode with vehicle speed  $v_c$ , a rigid propeller model based on a standard blade element theory approach with quasi-steady aerodynamics (Ref. 10) is utilized (see Fig. 11). Here the differential rotor thrust  $dFT_{MR}$  and torque  $dTQ_{MR}$  at blade radial location  $r$  are expressed as

$$\begin{aligned} dFT_{MR} &= N_b (dL \cos \phi - dD \sin \phi) \\ dTQ_{MR} &= N_b r (dL \sin \phi + dD \cos \phi) \end{aligned} \quad (25)$$

Where  $dL$  and  $dD$  are differential blade lift and drag force increments along with the number of blades  $N_b$ .

Based on tangential blade speed,  $u_T = \Omega_{MR} r$ , together with  $v_c$  and the induced inflow velocity  $v_{ind}$ , the total inflow angle  $\phi$  is

$$\phi = \phi_c + \phi_{ind} \quad (26-a)$$

With

$$\phi_c = \tan^{-1} \left[ \frac{v_c}{\Omega_{MR} r} \right] \text{ and } \phi_{ind} = \frac{v_{ind}}{\sqrt{v_c^2 + \Omega_{MR}^2 r^2}} \quad (26-b)$$

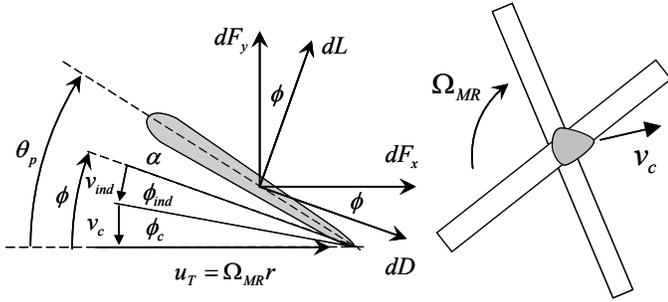


Figure 11.—Propeller in forward flight.

Where the induced inflow angle,  $\phi_{ind}$ , is assumed small. Now, assuming uniform induced inflow velocity  $v_{ind}$  together with uniform blade pitch,  $\theta_p$ , the following expression for total rotor thrust is obtained as

$$FT_{MR} = \frac{\rho_{air} a_{cL} c N_b \Omega_{MR}}{2} \int_0^{R_b} \Psi_c (\phi_c + \phi_{ind} - \theta_p) (v_c \phi_{ind} - r \Omega_{MR}) r dr \quad (27)$$

Similarly the corresponding rotor torque is

$$TQ_{MR} = \frac{\rho_{air} c_D c N_b \Omega_{MR}}{2} \int_0^{R_b} \Psi_c (r \Omega_{MR} - v_c \phi_{ind}) r^2 dr \dots - \frac{\rho_{air} a_{cL} c N_b \Omega_{MR}}{2} \int_0^{R_b} \Psi_c (\phi_c + \phi_{ind} - \theta_p) (v_c + r \Omega_{MR} \phi_{ind}) r^2 dr \quad (28)$$

Both with

$$\Psi_c = \sqrt{1 + \frac{v_c^2}{r^2 \Omega_{MR}^2}} \quad (29)$$

With air density  $\rho_{air}$ , rotor radius,  $R_b$  and blade chord  $c$ . Also  $c_D$  is the blade section drag coefficient and  $a_{cL}$  is blade section lift coefficient slope.

Next, by equating the thrust expression in (27) with an expression based on momentum theory (Ref. 10), we obtain the following equation for the induced inflow ratio  $\lambda_{ind}$

$$a_\lambda \lambda_{ind}^2 + b_\lambda \lambda_{ind} + c_\lambda = 0 \quad (30-a)$$

with coefficients

$$a_\lambda = 48 \ln \left[ \frac{\gamma_c}{\lambda_c} \right] \lambda_c^2 + 6 a_{cL} \sigma \ln \left[ \frac{1 + \gamma_c}{\lambda_c} \right] \lambda_c - 24 \quad (30-b)$$

$$b_\lambda = 24 \sin^{-1} \left[ \frac{1}{\lambda_c} \right] \lambda_c^3 + 6 a_{cL} \sigma \ln \left[ \frac{\gamma_c}{\lambda_c} \right] \lambda_c^2 + \dots (-24 \gamma_c + 6 a_{cL} \sigma \tan^{-1} \lambda_c - 6 a_{cL} \sigma \theta_p) \lambda_c - 3 a_{cL} \sigma \quad (30-c)$$

$$c_\lambda = a_{cL} \sigma \left( \pi - 2 \theta_p - \ln \left[ \frac{1 + \gamma_c}{\lambda_c} \right] \right) \lambda_c^3 \dots - 2 a_{cL} \gamma_c \sigma \tan^{-1} [\lambda_c] \lambda_c^2 \dots - a_{cL} \gamma_c \sigma \lambda_c + 2 a_{cL} \gamma_c \sigma (\gamma_c^2 \theta_p - \tan^{-1} \lambda_c) \quad (30-d)$$

With induced and forward speed inflow ratios

$$\lambda_{ind} = \frac{v_{ind}}{R_b \Omega_{MR}}, \quad \lambda_c = \frac{v_c}{R_b \Omega_{MR}} \quad \text{and} \quad \gamma_c = \sqrt{1 + \lambda_c^2} \quad (31)$$

and rotor solidity ratio

$$\sigma = \frac{c N_b}{\pi R_b} \quad (32)$$

Based on (30)-(32), the induced velocity,  $v_{ind}$ , is determined from the roots of (30-a). Using this result in (27)-(28), the rotor thrust and torque values can be computed for a given rotor speed, forward speed and blade pitch angle as shown in the rotor thrust and torque simulation block in Figure 12.

Using (1)-(3) and (21), the main rotor speed in high and low ratio are

$$\Omega_{MR,high} = n_f n_{DCT,high} n_i \Omega_{pt} \quad (33)$$

$$\Omega_{MR,low} = n_f n_{DCT,low} n_i \Omega_{pt}$$

The rotor parameters used in this investigation are summarized in Table 3.

TABLE 3.—ROTOR PARAMETERS

Parameter	Value
Rotor speed (high), $\Omega_{MR,high}$ .....	190 RPM
Rotor speed (low), $\Omega_{MR,low}$ .....	102.5 RPM
Number of blades, $N_b$ .....	4
Rotor radius, $R_b$ .....	9.91 m
Rotor solidity, $\sigma$ .....	0.13
Blade mass, $m_b$ .....	372 kg
Rotor rotational inertia, $J_{MR}$ .....	$4.874 \times 10^4$ kg m <sup>2</sup>
Section lift coefficient, $c_D$ .....	0.012
Section drag coefficient, $a_{cL}$ .....	5.056

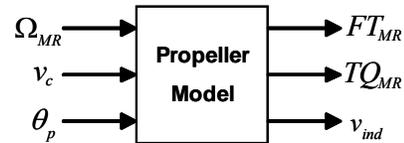


Figure 12.—Rotor thrust and torque simulation block.

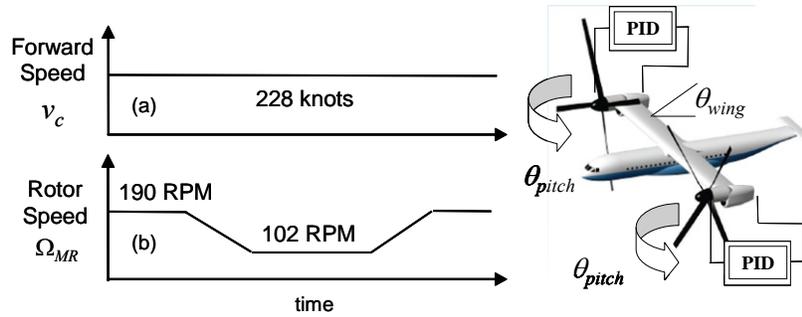


Figure 13.—Mission scenario; downshift and upshift in steady forward cruise using SSC shifting.

## Two-Speed Shift Simulation Results

Utilizing the model summarized in the previous section, this portion of the paper presents a two-speed shift simulation case study for the tiltrotor DCT drive-system (Fig. 2) operating in airplane mode at an intermediate steady forward cruising speed. The operating conditions and parameters used in the simulation (see Tables 1 to 3) are based on the LCTR-2 (Refs. 12 and 18). Specifically, the simulated mission scenario, depicted in Figure 13, is a rotor speed downshift followed later by a rotor speed upshift both while operating at the constant forward speed setpoint,  $v_{c,set} = 228$  knots.

To avoid shifting under full power, the SSC strategy, presented in (Ref. 9), is utilized in both downshift and upshift phases. Fundamentally, the SSC method utilizes the freewheel clutches at the power turbine output to disengage each engine sequentially wherein shifting along the particular driveline path can be performed in an unloaded condition. Since the FWU prevents the power turbine from being driven by the load, it will disengage when the load speed becomes larger than that of the power turbine. In a multi-engine driveline equipped with a variable ratio gearbox there are essentially two situations which can cause a specific FWU to disengage; 1) by lowering the power turbine setpoint speed of that engine or, 2) by downshifting the gearbox in that engine's driveline path. In the first situation, the remaining engines maintain the system speeds as the particular engine speed is lowered. In second the situation, the commanded downshift acts as an upshift on the gearbox input side thus increasing the load speed at the FWU. Since the engine setpoint speed remains fixed, the FWU in this driveline path will disengage. Both of these situations arise in the SSC shifting method. The downshift and upshift portions of this simulation are presented separately in the following two subsections. Here the effects of shift rate on the engine response will be explored in detail.

### Forward Cruise SSC Downshift

The timeline of the SSC downshift procedure along with a brief description of each event is given in Table 4.

TABLE 4.—SSC DOWNSHIFT TIMELINE\*

Time	Commands	Comments
$t=12$ sec	1) Disable engine torque sharing control loop 2) Start left side DCT downshift (disengage clutch no. 1 and engage clutch no. 2). $\dot{p}_{cl1} = -30$ psi/sec $\dot{p}_{cl2} = +4$ psi/sec	Start of SSC downshift procedure Increases left DCT input shaft speed which disengages left engine FWU
$t=32$ sec	3) Start right engine power turbine setpoint ramp-down (-600 RPM/sec)	Main rotor speed reduces from 190 to 102.5 RPM Left engine re-engages FWU at end of ramp-down
$t=52$ sec	4) Start right side DCT downshift (disengage clutch no. 1 and engage clutch no. 2). $\dot{p}_{cl1} = -30$ psi/sec $\dot{p}_{cl2} = +4$ psi/sec	Increases right DCT input shaft speed which disengages right engine FWU
$t=72$ sec	5) Start right engine power turbine setpoint ramp-up (+600 RPM/sec)	Right engine re-engages FWU at end of ramp-up
$t=82$ sec	6) Re-enable engine torque sharing control loop	End of SSC downshift procedure

\*Under constant forward speed setpoint  $v_{c,set} = 228$  knots.

Figures 14 to 19 give the forward cruise SSC downshift response of the two-speed DCT tiltrotor system. By examining Figures 14(a) and (b), it is apparent that the rotor speed is successfully lowered with very little deviation in forward velocity. Also note in Figure 14(b) that the left and right rotor speeds are nearly identical throughout the shift. Figures 14(c) and (d) show the open-loop DCT clutch pressure command inputs used to sequentially downshift the left and right side DCTs (step nos. 2 and 4 in Table 4). Furthermore, the engine power turbine speeds are shown in Figure 14(e) and (f). In Figure 14(e) it is observed that, even though the left side engine setpoint speed remains constant, the power turbine transiently increases due to the left side DCT downshift. This is a result of the rapid unloading due to FWU disengagement (see step no. 2 in Table 4). According to step nos. 3 and 5 in Table 4, Figure 14(f) shows the ramp-down and ramp-up of the right engine power turbine. Similarly, when the right side DCT is downshifted, the right engine power turbine experiences a transient speed increase above its setpoint as its FWU disengages (see step no. 4 in Table 4).

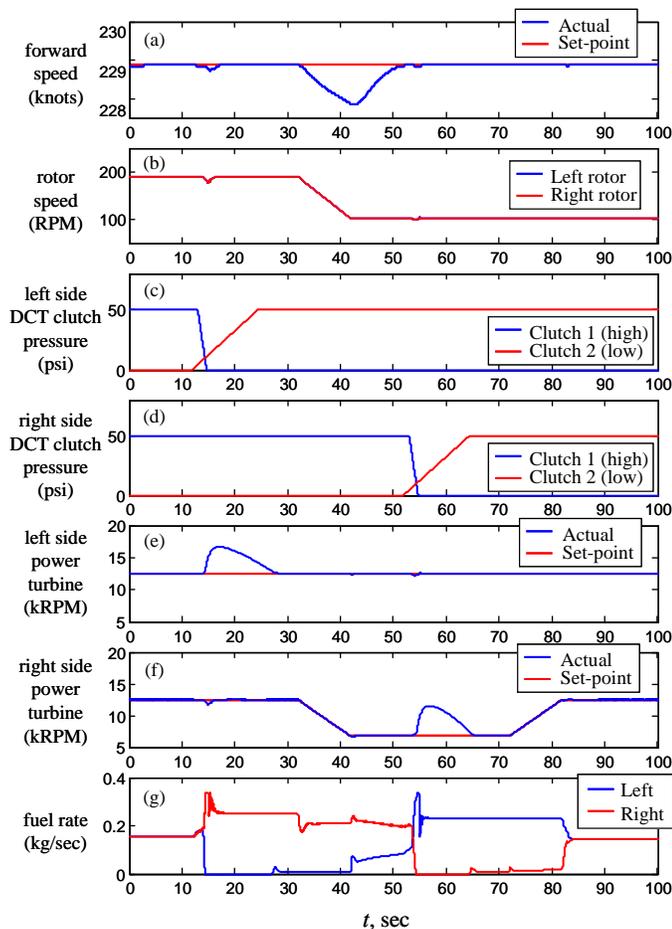


Figure 14.—Forward cruise SSC downshift simulation; (a) forward speed, (b) rotor speed, (c) left DCT shift commands, (d) right DCT shift commands, (e) left power turbine speed, (f) right power turbine speed (g) fuel rates.

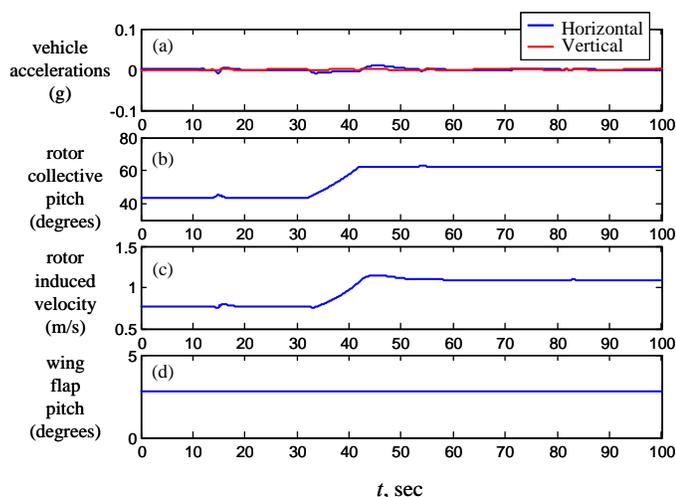


Figure 15.—Forward cruise SSC downshift—flight dynamics; (a) vehicle acceleration, (b) collective pitch, (c) induced velocity, (d) wing flap angle.

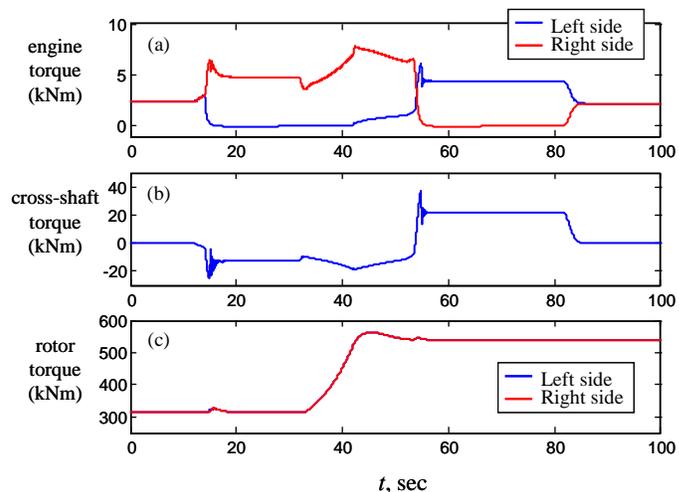


Figure 16.—Forward cruise SSC downshift—torque response; (a) engine torques, (b) cross-shaft torque, (c) rotor torque.

Figure 14(g) shows the corresponding fuel rates commanded by the closed-loop engine fuel control for both the left and right side engines. Note, the left and right side engine fuel rates are equivalent prior to disabling ( $t=12$  sec) and after re-enabling ( $t=82$  sec) the multi-engine torque sharing control loop (see step nos. 1 and 6 in Table 4).

The flight dynamics are summarized in Figure 15. Figure 15(a) shows that both the horizontal and vertical vehicle accelerations are minimal. Thus, no forward speed or altitude is lost during the downshift. In order to maintain the constant forward speed setpoint under the decreasing rotor speed, the closed-loop blade pitch controller increases the blade collective pitch to keep constant thrust as shown Figure 15(b). This results in a corresponding increase in rotor induced velocity is seen in Figure 15(c). Finally, as seen in Figure 15(d), under this constant thrust condition, there is no appreciable change in wing flap angle.

The driveline torques are summarized in Figure 16. As observed in the engine fuel rate response in Figure 14 (g), the left and right side power turbine torques are identical prior to  $t=12$  sec and after  $t=82$  sec. During the downshift, the engine torques become dissimilar and are due to both rotor aerodynamic torque as well as acceleration torque demands.

As a result of the dissimilar left and right engine torques, the cross-shaft also becomes loaded as shown in Figure 16(b). Finally, Figure 16(c) shows the increasing rotor torque which is a result from the increasing collective pitch and constant thrust requirement (see Fig. 15(b)). Note that the left and right rotor torques are identical throughout the shift (Fig. 16(c)).

Other important aspects of the two-spool gas turbine engine response are shown in Figures 17 and 18. In particular, by plotting compressor pressure ratio versus air mass flow, the left and right engine transient operating lines during the SSC downshift are shown in Figure 18.

Figure 18 shows that both engines experience a wide range of operation during the SSC process and that both engines transiently pass over the surge line. Since Figure 18 does not show temporal information, the engine surge margin (SM) versus time is examined. Here, the working definition of SM used is

$$SM = \frac{pr_{c,surge} - pr_c}{pr_{c,surge}} \times 100\% \quad (34)$$

Therefore  $SM > 0$  indicates no compressor surging. In this analysis, three different DCT shift rate cases (slow, medium and fast) as given in Table 5 are considered. Note the medium case corresponds to the values used thus far in Figures 14 to 18.

TABLE 5.—DCT DOWNSHIFT RATE CASES

Case	DCT clutch pressure engagement and disengagement rates (psi/sec)	
Slow	Clutch no. 1 disengagement rate, $\dot{p}_{cl1} = -15$ psi/sec	Clutch no. 2 engagement rate, $\dot{p}_{cl2} = 4$ psi/sec
Medium	Clutch no. 1 disengagement rate, $\dot{p}_{cl1} = -30$ psi/sec	Clutch no. 2 engagement rate, $\dot{p}_{cl2} = 4$ psi/sec
Fast	Clutch no. 1 disengagement rate, $\dot{p}_{cl1} = -300$ psi/sec	Clutch no. 2 engagement rate, $\dot{p}_{cl2} = 4$ psi/sec

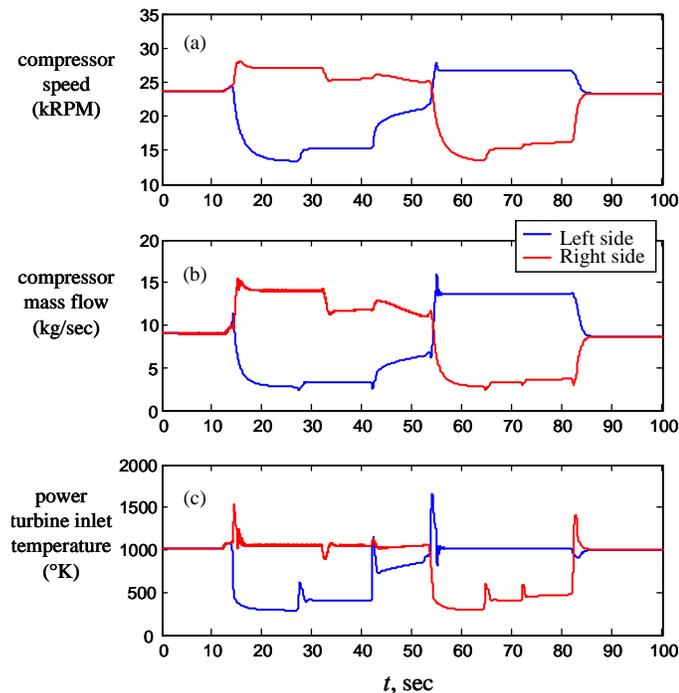


Figure 17.—Forward cruise SSC downshift—engine response; (a) compressor speed, (b) mass flow, (c) power turbine inlet temperature.

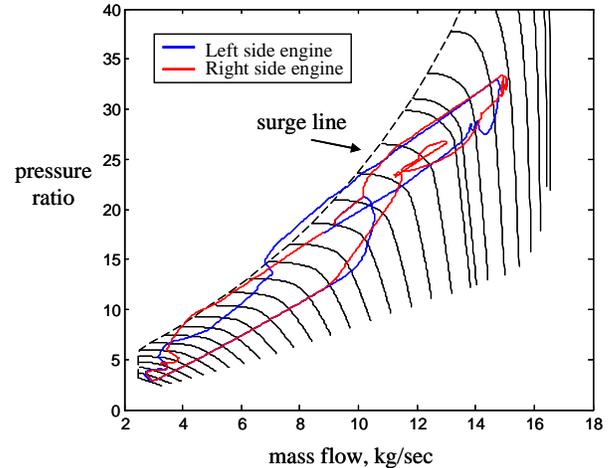


Figure 18.—Forward cruise SSC downshift—two-spool gas turbine engine transient running lines.

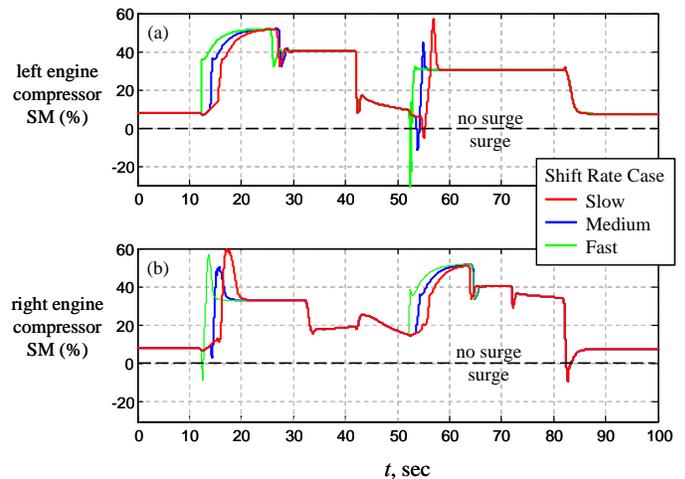


Figure 19.—Forward cruise SSC downshift—engine compressor surge margins for three DCT shift rates.

The resulting SM values for the left and right engines are shown in Figures 19(a) and (b). In each case, the surge line is crossed very briefly. Here, Figure 19(a) shows that the left engine crosses the SM immediately after  $t = 52$  sec as the right DCT begins to downshift (step no. 4, Table 4). Thus, the left engine surge is a result of the transient shock due to the unloading the right engine FWU.

Furthermore, Figure 19(b) shows that the right engine experiences a brief surge near the start and end of the downshift process for similar reasons as the left engine. Specifically, the first surge of the right engine is due to the initial disengagement of the left engine FWU (step no. 1, Table 4) and the second surge occurs when the left engine re-engages the FWU at the end of its ramp-up (step no. 5, Table 4). Figure 19 shows that slower DCT shifting rates are more beneficial for the engine surge margins in the SSC downshift procedure.

## Forward Cruise SSC Upshift

Next, the simulation results of the SSC upshift procedure, which is essentially the process reverse of the SSC downshift, are described. The SSC upshift timeline is given in Table 6.

TABLE 6.—SSC UPSHIFT TIMELINE\*

Time	Commands	Comments
$t=7$ sec	1) Disable engine torque sharing control loop 2) Start right engine power turbine setpoint ramp-down (-600 RPM/sec)	Start of SSC upshift procedure Disengages right engine FWU
$t=27$ sec	3) Start right side DCT upshift (disengage clutch no. 2 and engage clutch no. 1). $\dot{p}_{cl1} = +4$ psi/sec $\dot{p}_{cl2} = -30$ psi/sec	Decreases right DCT input shaft speed. Right engine FWU becomes re-engaged
$t=47$ sec	4) Start right engine power turbine setpoint ramp-up (+230 RPM/sec)	Main rotor speed increases from 102.5 to 190 RPM Speed increase disengages left engine FWU
$t=82$ sec	5) Start left side DCT upshift (disengage clutch no. 2 and engage clutch no. 1). $\dot{p}_{cl1} = +4$ psi/sec $\dot{p}_{cl2} = -30$ psi/sec	Decreases left DCT input shaft speed which re-engages left engine FWU
$t=92$ sec	6) Re-enable engine torque sharing control loop	End of SSC upshift procedure

\*Under constant forward speed setpoint  $v_{c,set} = 228$  knots.

Figures 20 to 25 show the forward cruise SSC upshift response of the two-speed DCT tiltrotor. By examining the forward speed and the rotor speed in Figures 20(a) and (b), it is apparent that rotor speed is successfully increased with little deviation in forward velocity. Figures 20(c) and (d) show the open-loop DCT clutch pressure command inputs used to sequentially upshift the right and left side DCTs (step nos. 3 and 5 in Table 6). Furthermore, the engine power turbine speeds are shown in Figure 20(e) and (f). In Figure 20(e) it is observed that the left power turbine speed transiently increases due to the right engine ramp-up. This is due to the unloading of the left engine FWU. (See step no. 4 in Table 6). According to step nos. 2 and 4 in Table 6, Figure 20(f) shows the ramp-down and ramp-up of the right engine power turbine. Figure 20(g) shows the corresponding fuel rates commanded by the closed-loop engine fuel control for both the left and right engines. Note, the left and right side engine fuel rates are equivalent prior to disabling ( $t=7$  sec) and after re-enabling ( $t=92$  sec) the multi-engine torque sharing control loop (see step nos. 1 and 6 in Table 6).

Also note that the fuel rate is essentially flat during the rotor speed acceleration phase (step no. 4, Table 6). This is due to the fact that as rotor speed increases, the collective pitch is automatically decreased by the forward speed pitch controller to maintain constant thrust as further shown in Figure 21.

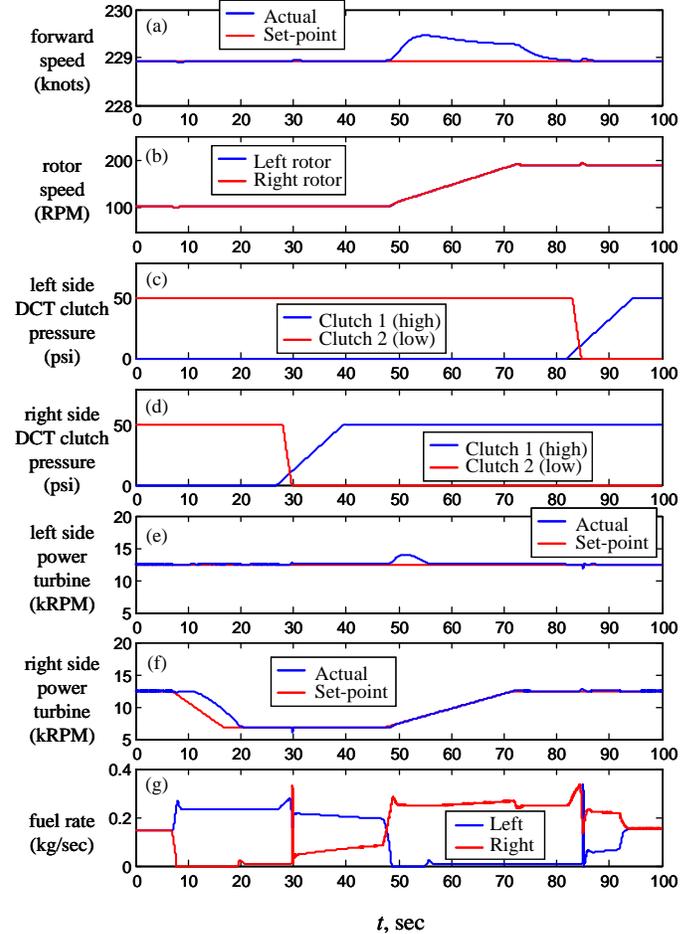


Figure 20.—Forward cruise SSC upshift simulation; (a) forward speed, (b) rotor speed, (c) left DCT shift commands, (d) right DCT shift commands, (e) left power turbine speed, (f) right power turbine speed (g) fuel rates.

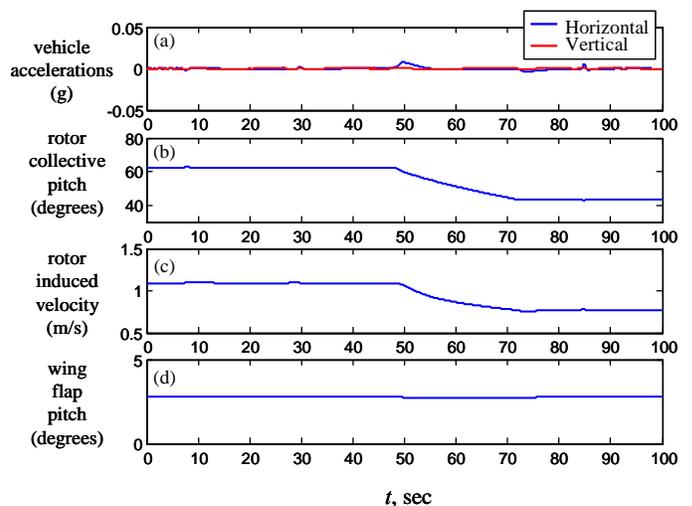


Figure 21.—Forward cruise SSC upshift—light dynamics; (a) vehicle acceleration, (b) collective pitch, (c) induced velocity, (d) wing flap angle.

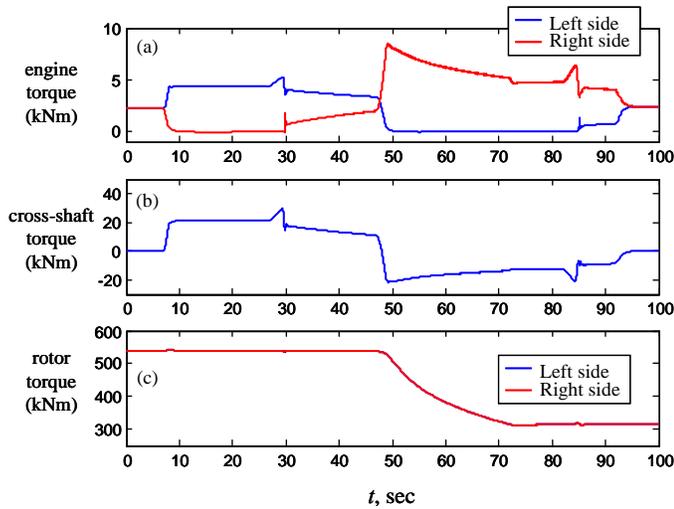


Figure 22.—Forward cruise SSC upshift—torque response; (a) engine torques, (b) cross-shaft torque, (c) rotor torque.

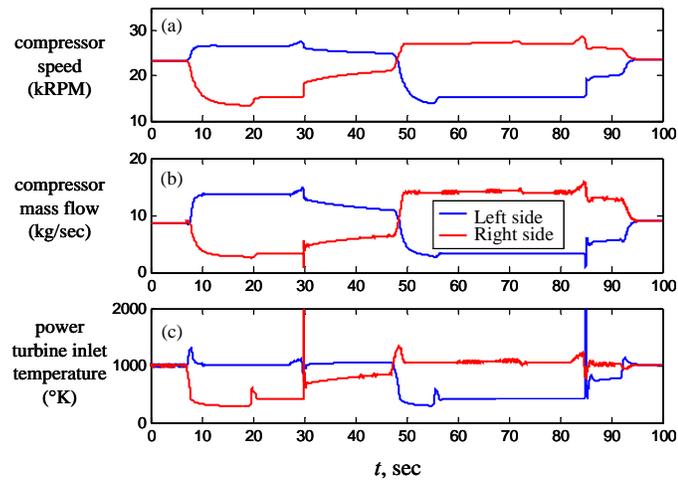


Figure 23.—Forward cruise SSC upshift—engine response; (a) compressor speed, (b) mass flow, (c) power turbine inlet temperature.

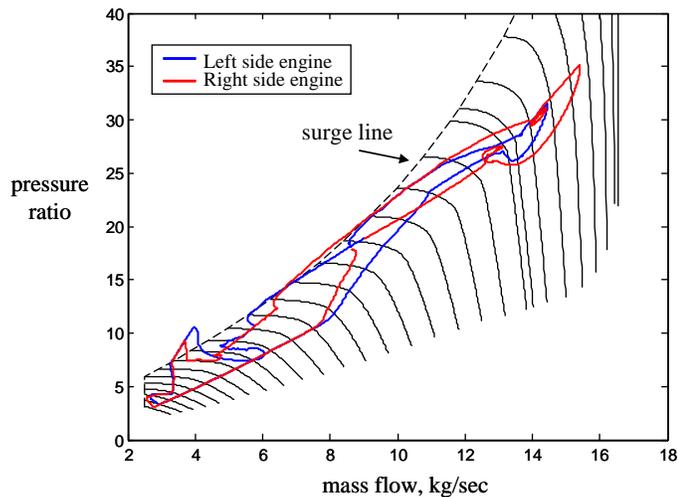


Figure 24.—Forward cruise SSC upshift—two-spool gas turbine engine transient running lines.

Similar to the downshift case, Figure 21(a) shows that both the horizontal and vertical vehicle accelerations are minimal thus with no forward speed or altitude lost during the upshift. To maintain constant forward speed under the increasing rotor speed, the closed-loop blade pitch controller decreases the blade collective pitch to keep constant thrust as shown Figure 21(b). This causes the drop in induced velocity as seen in Figure 21(c). Finally, as seen in Figure 21(d), under the constant thrust condition, there is no appreciable change in wing flap angle.

The driveline torques are summarized in Figure 22. As observed in the engine fuel rate response in Figure 20(g), the left and right side power turbine torques are identical prior to  $t=7$  sec and after  $t=92$  sec. During the downshift, the engine torques become dissimilar and are due to both rotor aerodynamic torque as well as acceleration torque demand.

As a result of the dissimilar left and right engine torques, the cross-shaft also becomes loaded as shown in Figure 22(b). Here, the cross-shaft is subjected to a fully reversing torque load. This is also true in the downshift case. Finally, Figure 22(c) shows the decreasing rotor torque which is a result from the decreasing collective pitch.

Other important aspects of the two-spool gas turbine engine response are shown in Figures 23 and 24. As seen in Figure 24, similar to the downshift case, the transient running lines of each engine undergo small excursions past the compressor surge line during the SSC upshift process.

Next, as in the downshift case, three different DCT shift rates (slow, medium and fast), given by Table 7, are examined. Note the medium case corresponds to the values used in Figures 20 to 24.

TABLE 7.—UPSHIFT RATE CASES

Case	DCT clutch pressure engagement and disengagement rates (psi/sec)	
Slow	Clutch no. 1 engagement rate, $\dot{p}_{cl1} = 4$ psi/sec	Clutch no. 2 disengagement rate, $\dot{p}_{cl2} = -15$ psi/sec
Medium	Clutch no. 1 engagement rate, $\dot{p}_{cl1} = 4$ psi/sec	Clutch no. 2 disengagement rate, $\dot{p}_{cl2} = -30$ psi/sec
Fast	Clutch no. 1 engagement rate, $\dot{p}_{cl1} = 4$ psi/sec	Clutch no. 2 disengagement rate, $\dot{p}_{cl2} = -300$ psi/sec

The resulting SM values for each shift rate are shown in Figure 25. In each case, the surge line is crossed briefly. Figure 25(a) shows that the left engine crosses the surge line when the FWU re-engages due to the left side DCT upshift (step no. 5, Table 6). Furthermore the right engine experiences a surge near  $t = 30$  sec and  $t = 48$  sec. The first surge of the right engine is due to re-engagement of the right engine FWU as a result of the right side DCT upshift (step no. 3, Table 6). The second surge, near  $t = 48$  sec, is due to the right engine power turbine set-point ramp-up (step no. 4, Table 6) which causes the left engine to disengage from its FWU.

Finally, Figure 25 shows that faster DCT shifting rates are more beneficial for the engine surge margins in the SSC upshift procedure.

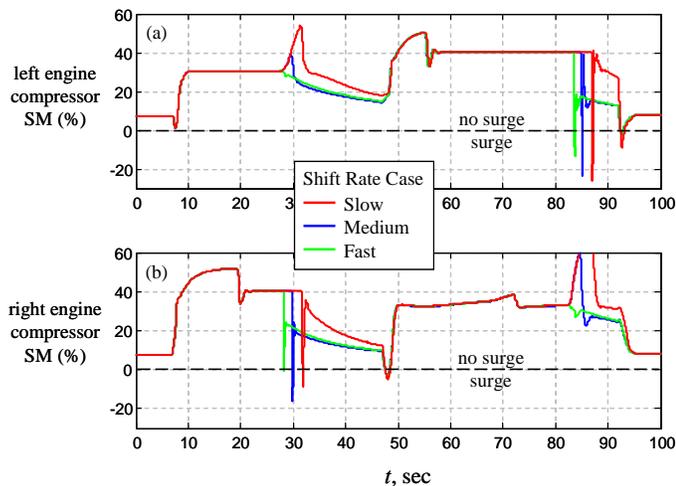


Figure 25.—Forward cruise SSC upshift—engine compressor surge margins for three DCT shift rates.

## Conclusions

This investigation explores the variable-speed operation and transient shift response of a prototypical two-speed dual-clutch transmission tiltrotor driveline based on the NASA LCTR-2 in forward flight airplane mode. A comprehensive variable-speed rotorcraft propulsion system modeling tool developed under a NASA-funded NRA program is utilized to simulate the drive system dynamics. In this study, a sequential shifting control strategy is analyzed under a steady cruise condition. This investigation attempts to build upon previous variable-speed rotorcraft propulsion studies by 1) including a fully nonlinear transient gas-turbine engine model, 2) including clutch stick-slip friction effects, 3) including shaft flexibility, and 4) incorporating a basic flight dynamics model to account for interactions with the flight control system. Sequential shifting provided smooth rotor speed changes while maintaining a constant forward speed setpoint and a constant altitude for the tiltrotor studied. However, it was also found that transient shocks produced by engine engagements and disengagements from their freewheeling clutches tended to produce brief excursions of the engine compressor surge limit. It was found that proper choice of dual-clutch transmission shift rate could minimize these excursions. In the sequential downshift case, it was found that slower shift rates tended to reduce the engine surge effects while the opposite trend was found for the upshift case. Through exploring these interactions between the various subsystems, this analysis provides important insights into the continuing development of variable-speed rotorcraft propulsion technology.

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## Appendix A.—Nomenclature

$A_i$ [ $i=1-7$ ]	engine stage cross-section areas, $m^2$	$\dot{m}_{f,\max}$	maximum fuel flow rate, kg/s
$a_{cL}$	blade lift coefficient slope	$\dot{m}_{fn}$	nominal fuel flow rate, kg/s
$a_\lambda, b_\lambda, c_\lambda$	inflow ratio solution coefficients	$\dot{m}_{ggt}$	gas generator mass flow rate, kg/s
$C_a$	compressor axial flow velocity, m/s	$\dot{m}_{in}$	plenum chamber input mass flow rate, kg/s
$C_{an}$	nominal compressor axial flow velocity, m/s	$\dot{m}_{out}$	plenum chamber output mass flow rate, kg/s
CVSRPM	comprehensive variable-speed rotorcraft propulsion system modeling	$\dot{m}_{pt}$	power turbine mass flow rate, kg/s
$c$	blade chord, m	$N_b$	number of main rotor blades
$c_D$	blade drag coefficient	$N_{cg}$	number of DCT planet clutches
$c_p$	specific heat of air, kJ/(kg K)	$N_p$	number of DCT planets
DCT	dual-clutch transmission	$N_s$	number of compressor stages
DOF	degrees of freedom	$n_{DCT,\text{high}}$	DCT high speed ratio
$dD$	differential blade element drag, N	$n_{DCT,\text{low}}$	DCT low speed ratio
$dF_x, dF_y$	differential blade element force components, N	$n_f$	final gearbox ratio
$dL$	differential blade element lift, N	$n_i$	initial reduction gearbox ratio
$f$	fuel-air ratio	$P_a$	ambient pressure, bar
<b>F</b>	DCT generalized force vector	PGT	planetary gear transmission
$FT_{MR}$	main rotor thrust, N·m	PID	proportional-integral-derivative
FWU	freewheeling clutch units	$p_{cli}$ [ $i=1-2$ ]	clutch engagement pressures, $N/m^2$
$ht_c$	compressor inlet hub-to-tip ratio	$p_{cl\max}$	maximum clutch engagement pressure, $N/m^2$
$ht_{pt}$	power turbine inlet hub-to-tip ratio	$P_p$	plenum chamber pressure, $N/m^2$
$J_c$	compressor rotational inertia, $kg\ m^2$	$P_{0i}$ [ $i=1-7$ ]	engine stagnation pressures, $N/m^2$
$J_{cgi}$ [ $i=1-3$ ]	DCT control gear rotational inertias, $kg\ m^2$	$pr_c$	compressor pressure ratio
$J_{cr}$	DCT carrier rotational inertia, $kg\ m^2$	$pr_{cn}$	nominal compressor pressure ratio
$J_{ggt}$	gas generator turbine rotational inertia, $kg\ m^2$	<b>q</b>	DCT degree-of-freedom vector
$J_i$	DCT input gear rotational inertia, $kg\ m^2$	$Q_{cc}$	combustion chamber heat addition, W
$J_{in}$	DCT input-side rotational inertia, $kg\ m^2$	$r$	blade element radial location, m
$J_{out}$	DCT output-side rotational inertia, $kg\ m^2$	$R$	ideal gas constant, J/(mol K)
$J_p$	DCT planet gear rotational inertia, $kg\ m^2$	$R_b$	blade radius, m
$J_{pt}$	power turbine rotational inertia, $kg\ m^2$	$R_c$	compressor mean radius, m
$J_r$	DCT ring gear rotational inertia, $kg\ m^2$	$R_{cgi}$ [ $i=1-3$ ]	DCT control gear radii, m
$J_s$	DCT sun gear rotational inertia, $kg\ m^2$	$R_{cli}$ [ $i=1-2$ ]	clutch radii, m
$L_{cl2}$	ring gear clutch length, m	$R_{cr}$	DCT carrier radius, m
LHV	fuel lower heating value, J/kg	$R_i$	DCT input gear radius, m
<b>M</b>	DCT inertia matrix	$R_p$	DCT planet gear radius, m
$m_b$	blade mass, kg	$R_{pt}$	power turbine mean radius, m
$m_p$	DCT planet mass, kg	$R_r$	DCT ring gear radius, m
$m_{pa}$	plenum chamber air mass, kg	$R_s$	DCT sun gear radius, m
$\dot{m}_c$	compressor mass flow rate, kg/s	SM	compressor surge margin, percent
$\dot{m}_{cc}$	combustion chamber mass flow rate, kg/s	SSC	sequential shifting control
$\dot{m}_{cn}$	nominal compressor mass flow rate, kg/s	$t$	time, s
$\dot{m}_f$	fuel flow rate, kg/s	$T_{0i}$ [ $i=1-7$ ]	engine stage stagnation temperatures, K

$T_a$	ambient temperature, K	$\gamma_c$	forward speed inflow parameter
$TQ_c$	compressor torque, N·m	$\eta_c$	compressor isentropic efficiency
$TQ_{cl,i}$ [i=1-2]	clutch torques, N·m	$\eta_{cc}$	combustion chamber loss coefficient
$TQ_{in}$	DCT input shaft torque, N·m	$\varphi$	choking mass flow parameter, $s/(m\sqrt{K})$
$TQ_{out}$	DCT output shaft torque, N·m	$\lambda_c$	forward speed inflow ratio
$u_T$	blade element tangential speed, m/s	$\lambda_{ind}$	induced inflow ratio
$v_c$	forward speed, knots	$\mu_{cl}$	clutch dry friction coefficient
$v_{c,set}$	forward speed setpoint, knots	$\theta_p$	main rotor pitch angle, rad
$v_{ind}$	induced velocity, knots	$\theta_w$	wing elevon flap angle, rad
$V_p$	plenum chamber volume, m <sup>3</sup>	$\rho_{air}$	ambient air density, kg/m <sup>3</sup>
$W_c$	compressor power, W	$\rho_p$	plenum chamber gas density, kg/m <sup>3</sup>
$W_L$	load power, W	$\sigma$	rotor solidity ratio
$W_{ggt}$	gas generator turbine power, W	$\tau_{cc}$	combustion chamber time constant, s
$W_{pt}$	power turbine power, W	$\tau_{cl}$	clutch viscous torque coefficient, kg m <sup>2</sup> /s
$x$	vehicle horizontal position coordinate, m	$\Psi_c$	total speed ratio
$y$	vehicle vertical position coordinate, m	$\Omega_c$	compressor speed, RPM
$Z_{cgi}$ [i=1-3]	number of DCT control gear teeth	$\Omega_{cn}$	compressor nominal speed, RPM
$Z_i$	number of DCT input gear teeth	$\Omega_{DCT}$	two-speed transmission output speed, RPM
$Z_p$	number of DCT planet gear teeth	$\Omega_{DCT,high}$	two-speed transmission high output speed, RPM
$Z_r$	number of DCT ring gear teeth	$\Omega_{DCT,in}$	two-speed transmission input speed, RPM
$Z_s$	number of DCT sun gear teeth	$\Omega_{DCT,low}$	two-speed transmission low output speed, RPM
$\alpha$	angle of attack, rad	$\Omega_{FWU,out}$	freewheeling clutch output speed, RPM
$\beta_c$	compressor beta parameter, s <sup>2</sup> /kg <sup>2</sup>	$\Omega_{cg1,2}$	DCT planet clutch speeds, RPM
$\Delta T_{0cc}$	combustion chamber temperature rise, K	$\Omega_{MR}$	main rotor speed, RPM
$\varepsilon$	clutch slip velocity threshold, RPM	$\Omega_{pt}$	power turbine speed, RPM
$\phi$	total inflow angle, rad	$\Omega_{ptn}$	nominal power turbine speed, RPM
$\phi_c$	forward speed inflow angle, rad	$\Omega_{pt,set}$	power turbine setpoint speed, RPM
$\phi_i$	DCT input gear rotation, rad	$\Omega_r$	DCT ring gear speed, RPM
$\phi_{ind}$	induced inflow angle, rad	.	Note, if not explicitly stated, a dot above a variable represents change with respect to time.
$\phi_r$	DCT ring gear rotation, rad		
$\gamma$	specific heat ratio		

## Appendix B.—Comprehensive Variable-Speed Rotorcraft Propulsion System Model

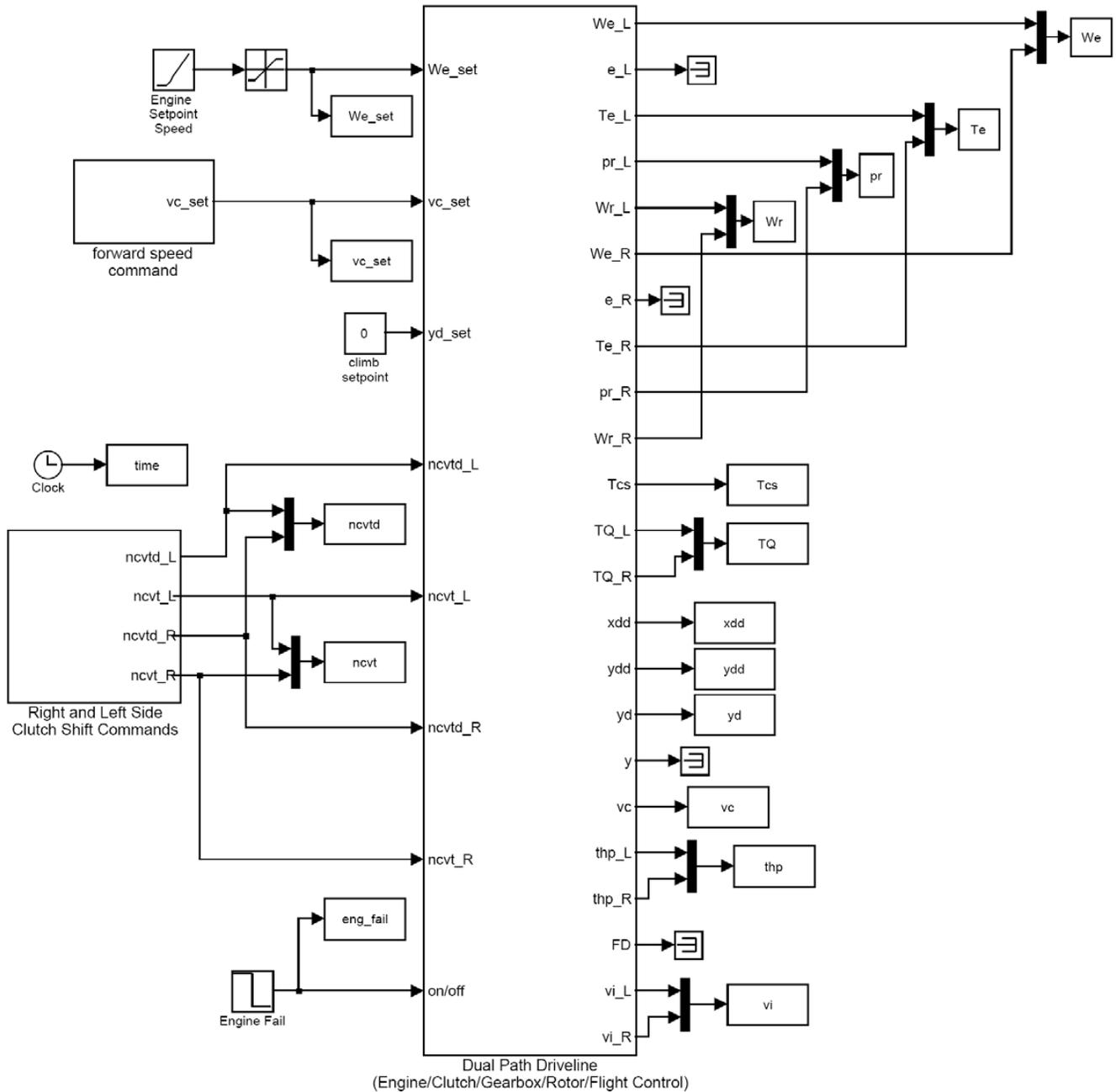


Figure B-1.—Two-speed rotorcraft driveline comprehensive simulation—main block.

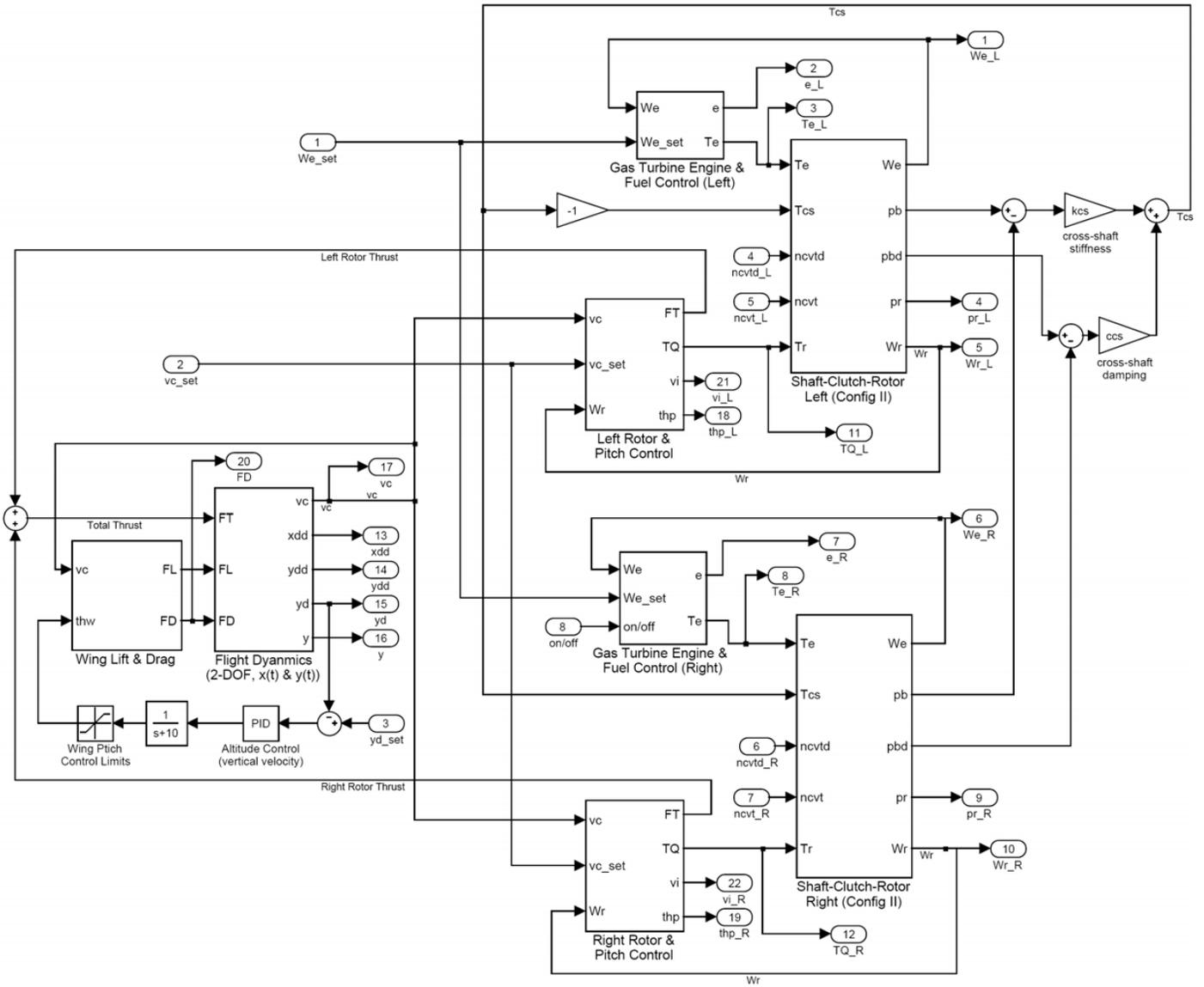


Figure B-2.—Two-speed tiltrotor driveline/engine/clutch/rotor/fuel control/flight dynamics block.

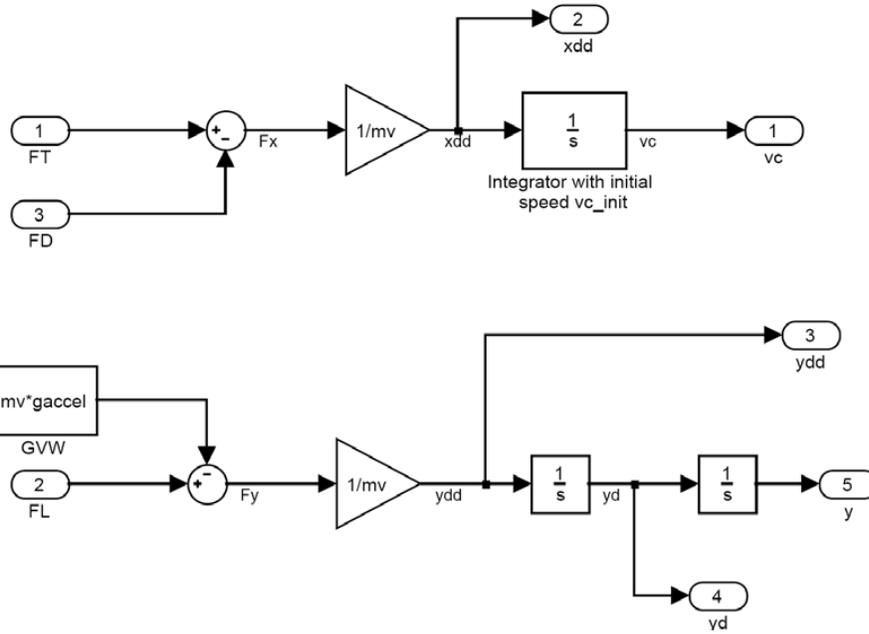


Figure B-3.—Two-DOF (altitude and forward speed) vehicle flight dynamics block.

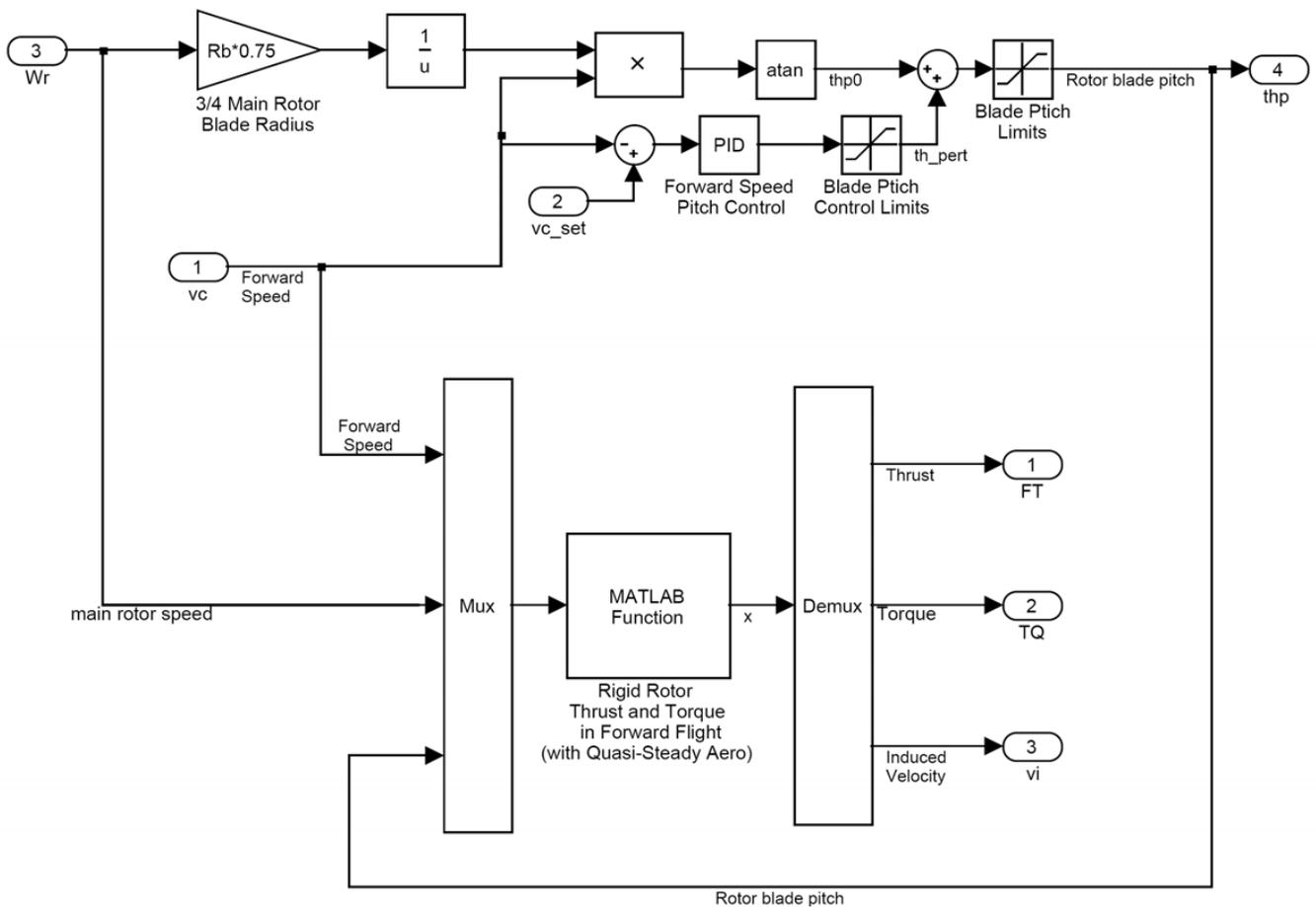


Figure B-4.—Rigid rotor and blade pitch forward speed controller block.

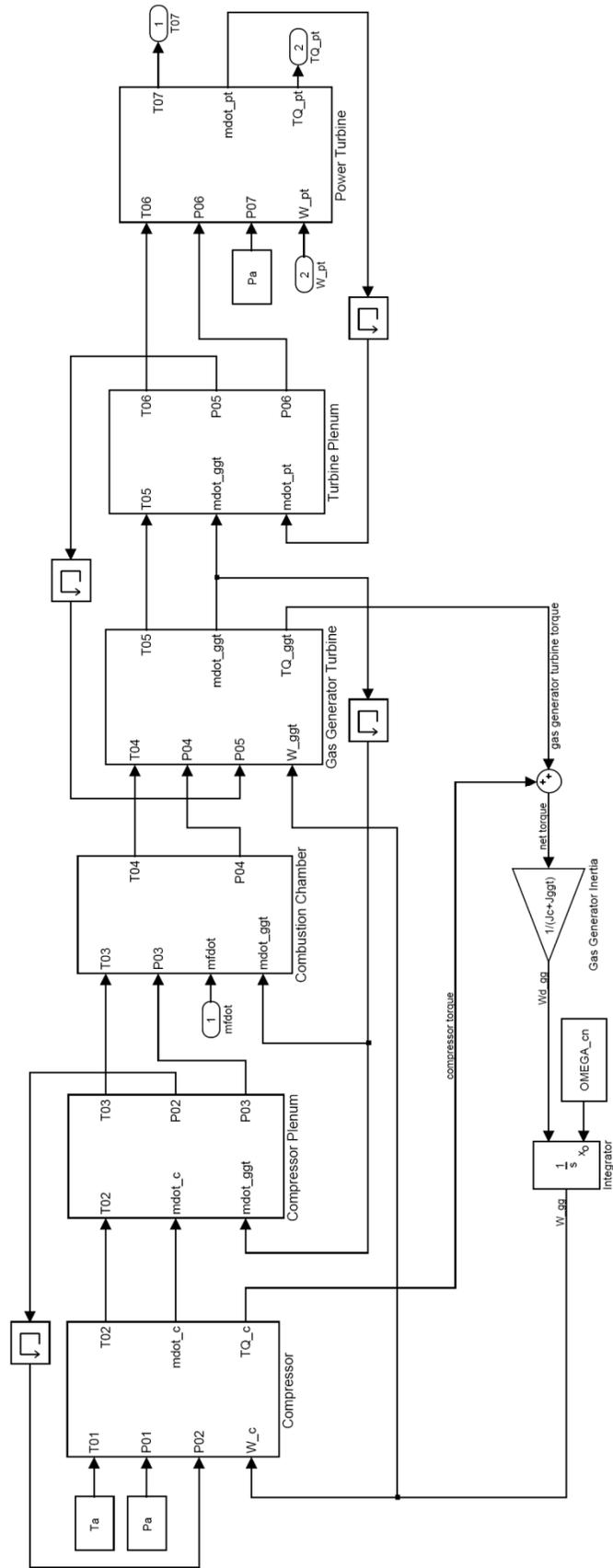


Figure B-5.—Two-spool gas turbine engine simulation block.



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<b>14. ABSTRACT</b> This investigation explores the variable-speed operation and shift response of a prototypical two-speed dual-clutch transmission tiltrotor driveline in forward flight. Here, a Comprehensive Variable-Speed Rotorcraft Propulsion System Modeling (CVSRPM) tool developed under a NASA funded NRA program is utilized to simulate the drive system dynamics. In this study, a sequential shifting control strategy is analyzed under a steady forward cruise condition. This investigation attempts to build upon previous variable-speed rotorcraft propulsion studies by 1) including a fully nonlinear transient gas-turbine engine model, 2) including clutch stick-slip friction effects, 3) including shaft flexibility, 4) incorporating a basic flight dynamics model to account for interactions with the flight control system. Through exploring the interactions between the various subsystems, this analysis provides important insights into the continuing development of variable-speed rotorcraft propulsion systems.					
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