10th INTERNATIONAL SCIENTIFIC CONFERENCE
ON DEFENSIVE TECHNOLOGIES
OTEH 2022
Belgrade, Serbia, 13 – 14 October 2022

CONTROL SYSTEM DESIGN AND INTEGRATION OF A BLOWDOWN WIND TUNNEL MODEL SUPPORT PITCH MECHANISM

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Abstract: *Development of a novel system for automation of aerodynamic experiments in the VTI T-38 wind tunnel included control system design and integration of a wind tunnel model support pitch mechanism. Considering that even small errors in scaled-model pitch angle affect the final aircraft configuration, an extremely accurate position control servo system is needed, capable of positioning the model with no measurable overshoot, low settling times and very good repeatability. To study the performance of the mechanism under the extremely high aerodynamic loads acting on the wind tunnel model, a dynamic model that reproduces the behavior of the model support mechanism is developed. Then a closed-loop control system is designed, consisting of a major loop with position feedback, which is stabilized by an inner loop with velocity feedback. To compensate for the system steady-state error, a gain scheduling approach is applied by switching two linear controllers based on the system state. The pitch mechanism control system is integrated and tested covering the full wind tunnel operating envelope. It is capable of accurate trajectory control of ±0.05º throughout an entire move at speeds of up to 15º/s. The system accuracy at the endpoint is ±0.01º with no measurable overshoot.*

Keywords: *Wind tunnel, model support, pitch mechanism, system dynamics, position control, gain-scheduled control.*

1. INTRODUCTION

Wind tunnel testing applications normally require positioning of the model in the airstream to simulate flight attitudes. For that reason, a standard wind tunnel design includes a positioning system which locates a model in the test section at a series of prescribed positions.

In most tests, it is required to place the model at various angles of attack for fixed sideslip angles or at various sideslip angles for fixed angles of attack. In blowdown wind tunnels, this type of positioning is usually accomplished by combining the actions of pitch and roll model support mechanisms. The problem of model positioning in the airstream is further complicated by deflections of the model support component parts caused by aerodynamic loads, which vary depending on the structure stiffness, model attitude, Mach number and dynamic pressure. Model support position achieved by the actions of the support mechanisms is therefore mathematically corrected, based on the previous experimental determination of the expected deflections, in order to establish the model position in the airstream. In such circumstances, the extremely accurate control of model support position is of utmost importance.

Model support mechanisms are usually sturdy systems weighing up to several tons and having one or more degrees of freedom, with drives of hydraulic, electric or hybrid type. Advanced control systems are required to both control mechanisms such as these and provide synchronization with the overall wind tunnel control and data acquisition process [1]. In the past, such control systems were hardware-based, with the multitude of accessories, connection cables and software applications to support devices in use [2]. In recent years, costly and cumbersome process of integration and maintenance of hardware-based control systems is being replaced by the software approach to motion control [3], providing the user with an interactive, productive and cost-effective platform for automation, as well as superb precision performance.

In line with these trends, the traditional hardware-based control system for the VTI T-38 wind tunnel model support pitch mechanism is replaced with the software solution. A dynamic model of the pitch mechanism is developed and used in the design of a double-loop control system that performs both velocity and position control. A compromise between several conflicting requirements, namely high position accuracy, precise reference tracking, low settling times and positioning without overshoot, is achieved using a gain scheduling method based on switching two linear controllers. The pitch control system is designed and integrated into the overall wind tunnel control system in such manner to operate both independently and in synchronization with other components during a test.

2. MODEL SUPPORT DESCRIPTION

Model support in the 1.5 m ×1.5 m VTI T-38 wind tunnel is a 2 DOF system consisting of two mechanisms: the hydraulically actuated pitch mechanism and the electrically driven roll mechanism. It permits movement of the model within ±20º in pitch and ±720º in roll. The main components of the model support are shown in Figure 1.

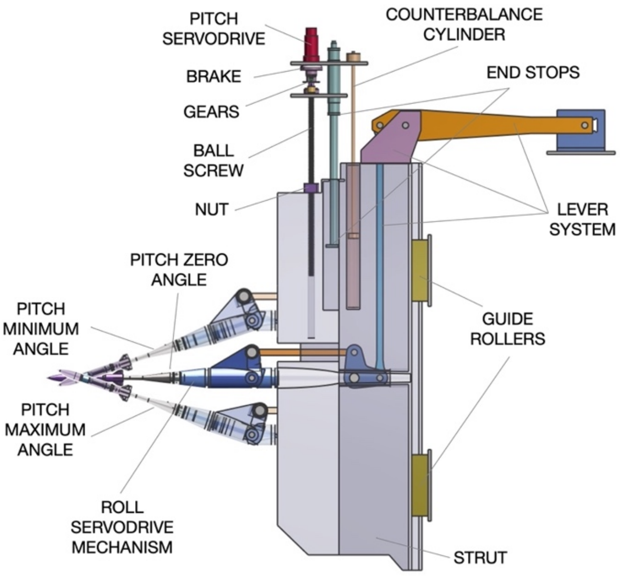

Figure 1. Model support in the VTI T-38 wind tunnel

The vertical strut is the most massive part of the pitch mechanism. Mass of the strut with linkage assembly is more than 3000 kg. Motion of the strut is restrained to vertical only by axial and side roller assemblies mounted in the wind tunnel structure. It is driven by a hydraulically powered ball screw and nut. Adjustable end stops with energy absorber and strut counterbalance cylinder are included in the system. Model pitching is accomplished by a lever system, parts of which are built into the body of the strut. The model support system is designed to withstand the maximum aerodynamic forces with suitable safety margins, that is, the normal force of 64 kN, the axial force of 18.3 kN and the pitching moment of 3.3 kNm.

Moog Series A084 hydraulic servodrive is used for the model support actuation, with a Moog 72-102 servovalve and a 9-piston, axial drive Moog-Donzelli servomotor. Around 90 percent of the mass of the model support system is pneumatically balanced via the counterbalance cylinder connected to the source of compressed air.

3. MATHEMATICAL MODEL EQUATIONS

The mathematical model of the electrohydraulic and mechanical components of the pitch mechanism is derived in order to obtain a physical model of the system.

A two-stage flow control servovalve uses an electrical torque motor, a double-nozzle pilot stage and a sliding-spool second stage. Electrical input current in the torque motor gives proportional displacement of the second stage spool, and hence, control flow proportional to the load. Control flow to the load will change with load pressure drop and electrical input, following the theoretical relationship for sharp-edged orifices, which is:

$$Q_L = C_d K_V i \sqrt{P_V} \tag{1}$$

where:

$$P_V = (P_S - P_R) - P_L \tag{2}$$

The two servovalve control ports connect to a fixed-displacement axial drive motor. An input current to the servovalve causes one control port to be connected to supply pressure, while simultaneously the other control port is connected to the return line. Pistons that are connected to supply pressure attempt to extend, so apply a force to the swash plate, resulting in a torque on the motor drive shaft. The overall result is motor shaft speed and direction proportional to the input current to the servovalve. The torque generated on the hydraulic motor shaft must overcome acceleration torque, torque due to viscous friction and load torque:

$$D_m P_L = J\alpha + B\omega + T_L \tag{3}$$

Moment of inertia on the motor shaft represents a sum of the moments of inertia of the main movable parts of the pitch mechanism:

$$J = J_m + J_s + J_L \tag{4}$$

The load torque comprises torque due to the mechanism mass unbalanced by the pneumatic system, torque due to friction and torque due to aerodynamic forces:

$$T_L = T_g + T_f + T_a \tag{5}$$

The electrohydraulic system is fully defined by adding to the relations (1) – (5) the continuity equation for the motor, assuming that the hydraulic fluid is incompressible:

$$Q_L = D_m\omega + Q_{il} \tag{6}$$

Based on the mathematical model (1) – (6), it is possible to determine the control flow for the known load conditions, ie., the required input signal from the control system.

To fully describe the pitch mechanism, geometric description is needed for motion of mechanical components of the system. Rotational motion of the motor is converted to translational motion of the strut via leadscrew, based on the following kinematic relation:

$$\varphi = \frac{2\pi}{p}x \tag{7}$$

Kinematic relation for the strut translational motion and the mechanism pitch angle is the following:

$$x = 1.75 sin\theta \tag{8}$$

System of equations (1) – (8) represents the mathematical model of the pitch mechanism.

4. PHYSICAL MODELLING AND SIMULATION

A physical model of the pitch mechanism is developed using Simscape in the Matlab-Simulink environment, as presented in Figure 2. The model is developed to be used in the simulations, with the aim to obtain the response of the pitch mechanism, and to use simulation data in control design and integration. The parameters of the components defined by model equations are coded into the Simscape model and presented in Table 1.

Table 1. System parameters

Parameter	Symbol	Unit	Value
System pressure	P_S	bar	130
Valve maximum opening	-	m	0.005
Flow discharge coefficient	C_d	-	0.7
Critical Reynolds number	-	-	12
Valve sizing constant	K_V	m^3/sA	0.265
Valve maximum current	i_{max}	A	0.01
Motor displacement	D_m	m^3/rad	12.88 10^{-6}
Motor maximum velocity	ω_{max}	rad/s	146.6
Motor volumetric efficiency	-	-	0.92
Motor total efficiency	-	-	0.8
Motor moment of inertia	J_m	$kg \cdot m^2$	0.06
Damping coefficient	-	Nm/(rad/s)	0.4
Ball screw lead	-	m	0.0254
Ball screw efficiency	-		0.9
Ball screw velocity threshold	-	m/s	0.6
Coulomb friction force	-	N	500

A fixed-displacement axial piston pump that is used as a source of hydraulic energy in combination with four accumulators is considered powerful enough to maintain required supply pressure at its outlet regardless of the load and it is represented by an ideal hydraulic pressure source in the Figure 1. The flow from the pump supplies the flow control valve, which regulates the amount of fluid entering the hydraulic motor. The ball screw that is used to convert rotation of the motor shaft to translation of the vertical strut is modelled taking into account friction losses. The sensors for position and velocity are used in the model under assumption that their inertia, friction and delays can be neglected. The outputs of the sensors are coded as relative, with the possible ranges of [-1, 1].

The nonlinearity in this type of system can arise from different sources, including flow deadband and saturation, relation between pressure and flow, friction between moving parts or changes in supply pressure and load [5]. The response of the pitch drive mechanism is analyzed in order to determine if the system can be linearized in the entire position range. The analysis is done by presenting an impulse function to the input of the simulation model in Figure 2, assuming the maximum aerodynamic forces.

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Based on the simulation data, transfer function of the motor shaft angular velocity (AV) to the servovalve input current (I) is approximated by the second-order system (V) with 99.02% fit to estimation data, as is shown in (9).

$$V(s) = \frac{AV(s)}{I(s)} = \frac{43.06s+800.9}{s^2+57.64s+748.1} \tag{9}$$

In addition, transfer function of the wind tunnel model angular position (AP) to the motor shaft velocity (AV) is represented by the first-order system (P) with 95.68% fit to estimation data, as is shown in (10).

$$P(s) = \frac{AP(s)}{AV(s)} = \frac{0.9316}{s} \tag{10}$$

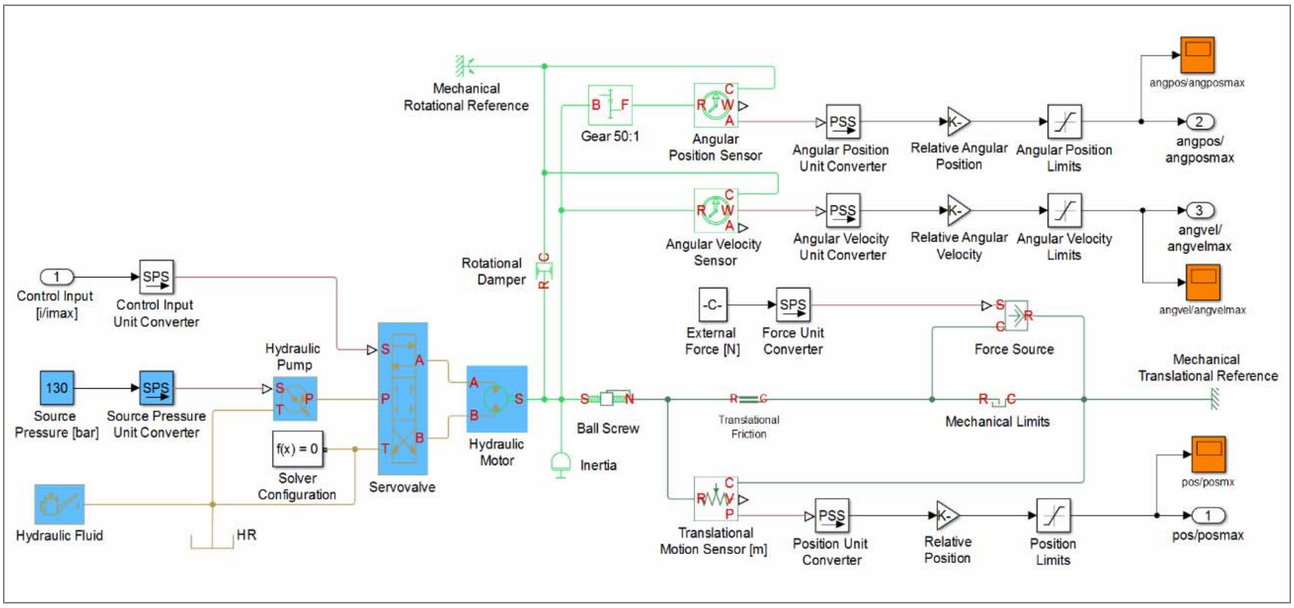

Figure 2. Simulink/Simscape model of the pitch drive mechanism

Equations (9) and (10) represent the linearized simulation model in Figure 2. Note that the motor and the load here are modelled separately. Since they are rigidly connected via the common shaft, it would be possible to develop a combined lumped parameter model and use the single transfer function of the angular position to the servovalve input current. However, due to highly variable nature of aerodynamic loads acting on the wind tunnel models it was decided to consider the motor and the load separately in control system design. This approach enables better capturing of how the dynamics of the motor and the load is distributed [6] and consequently, the more effective compensation by the control system.

4. CONTROL SYSTEM DESIGN

Control system for the pitch mechanism is designed based on the requirement to have precise model positioning in all wind tunnel test conditions, including those with extremely high and very variable aerodynamic loads. Consequently, the torque required to move the load is also variable, and it needs to be generated while maintaining the desired shaft speed and the model support mechanism position. Such requirements imply the need for the multi-loop control system that would be able to provide both velocity control of the motor shaft and position control of the mechanism.

Based on the above reasoning, the cascade control system is designed, consisting of the inner loop for the motor shaft angular velocity control with a tachogenerator as a velocity feedback, and the outer loop for the mechanism position control with an encoder as a position feedback. Block diagram of the suggested cascade control system applied to the linearized dynamics of the motor (9) and the load (10) is presented in Figure 3.

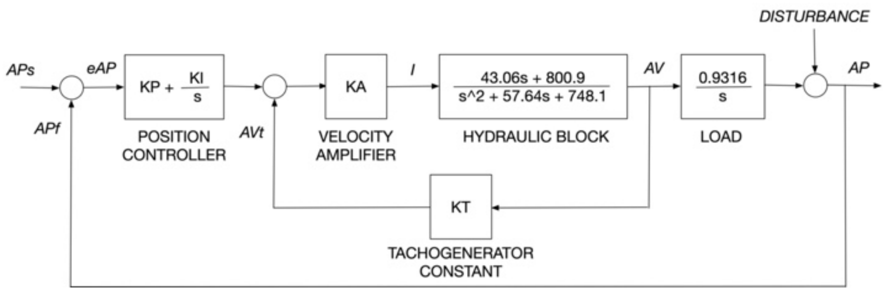

Figure 3. Block diagram of the pitch mechanism with cascade control

Both loops are tuned using Simulink linear analysis tool.

The tachogenerator constant $KT = 0.8$ is determined based on its relative output voltage relation to the relative motor shaft speed. The actual feedback and setpoint velocities in the inner loop are fed into the velocity amplifier. Any difference between the two velocities is amplified and the output serves as the relative current input to the hydraulic block. The amplifier gain is tuned to $KA = 4.671$. Figure 4 compares the motor shaft velocity frequency response of the system with inner loop (red line) to the response of the system without inner loop (blue line).

Closed-loop control of motor shaft velocity apparently brings several benefits, the most important being the larger bandwidth. The cutoff frequency without speed control loop of 7.45 Hz is increased to 32.2 Hz in a closed-loop system, which increases bandwidth more than four times. The higher is the bandwidth of the velocity loop, the quicker system responds to changes in the velocity command [7]. For example, a load disturbance causing an acceleration or deceleration of the motor shaft and the rise of velocity error can be compensated in the inner loop, which results in maintaining the motor speed without error propagation to the outer position loop.

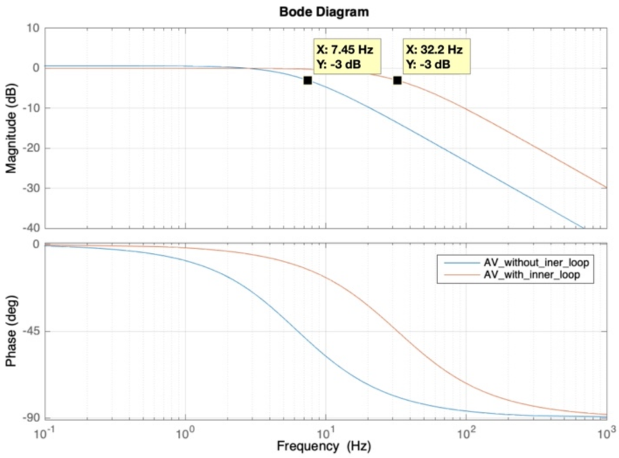

Figure 4. Bode plot of the motor shaft velocity responses without and with velocity loop

The position controller of the PI type selected for the outer loop was tuned using the previously determined parameters of the inner loop. In addition to high positioning accuracy and low settling times, the main requirement was to achieve positioning without measurable overshoot and with the maximum rise time, which amounts to ~ 0.85 s for the unit step response of the system (expressed as relative position, with step function from 0 to 1).

Figure 5 compares step responses of the system for several proportional (P-only) controllers, with *KP* =1,2,3,4,5.

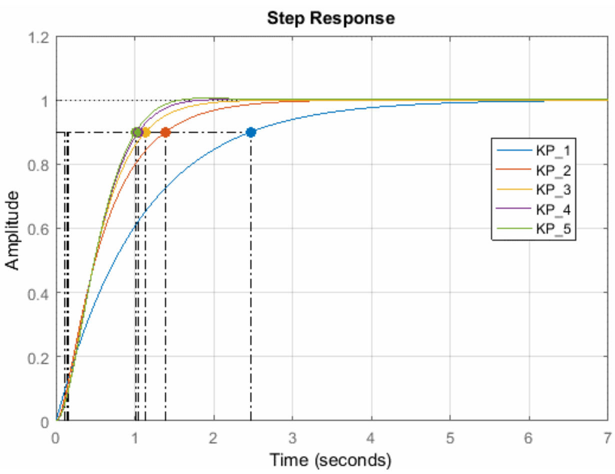

Figure 5. Step responses and rise times of the system position for several P-only controllers

The main characteristics of the system responses shown in Figure 5 are given in Table 2. Rise time is defined from 10% to 90% of the response, and settling time within 2% of the setpoint.

The rise time for *KP* = 4 is close to maximum for the system dynamics. Further increasing of *KP* does not bring a noticeable improvement of the rise time and settling time, and contrary to the control system design goals, it increases overshoot. Thus, the proportional *KP* = 4 controller appears to fulfil requirements concerning the rise time and settling time, but it still brings a mild overshoot and, as it is expected for this control method, results in a steady-state error. Since wind tunnel tests bring variable and frequent disturbances and the model support positioning application requires tighter set point tracking, proportional-only control does not appear to be sufficient.

Table 2. Step response characteristics

Controller	Rise time [s]	Settling time [s]	Overshoot [%]	Steady-state error [%]
KP = 1	2.35	4.19	0.00	0.02
KP = 2	1.25	2.28	0.00	0.00
KP = 3	0.98	1.70	0.00	0.01
KP = 4	0.88	1.44	0.16	0.08
KP = 5	0.85	1.34	0.56	0.30

The proportional-integral (PI) form of the controller provides correction for steady-state error and decreases rise time. However, adding integral term comes to the cost of introducing overshoot. In the case of the pitch mechanism, maximum physically possible rise time is achieved with P-only controller. Integral term thus cannot improve the system rise time. Tuning was attempted by adding relatively weak integral actions (*KI* = 0.1 and *KI* = 0.01) to two P controllers (*KP* = 4 and *KP* = 2.5), the idea being to avoid overshoot that would have been brought by higher values of the integral term. The step responses of the corresponding four PI controllers are given in Figure 6.

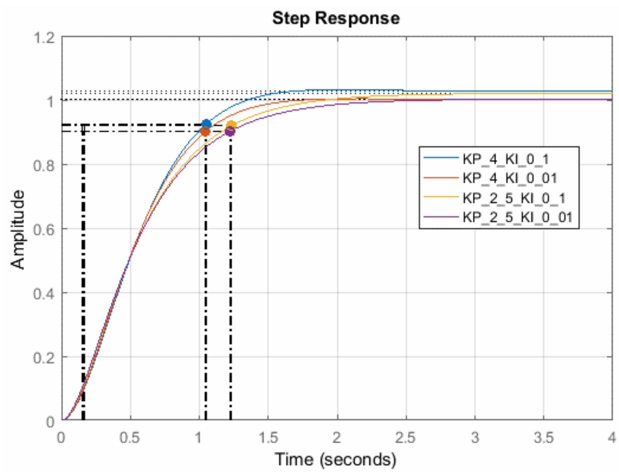

Figure 6. Step responses and rise times of the system position for several PI controllers

The main characteristics of the system responses shown in Figure 6 are shown in Table 3.

Table 3. Step response characteristics

Controller	Rise time [s]	Settling time [s]	Over-shoot [%]	Steady-state error [%]
KP = 4.0; *KI* = 0.1	0.88	1.40	0.471	2.80
KP = 4.0; *KI* = 0.01	0.88	1.44	0.182	0.25
KP = 2.5; *KI* = 0.1	1.08	1.94	1.19	1.19
KP = 2.5; *KI* = 0.01	1.08	1.94	0.24	0.24

It is evident that adding integral action results in larger overshoots, without the effect on rise times. In fact, proportional term appears to have a dominant effect on the pitch system response, given that KP = 2.5, KI = 0.1 and KP = 2.5, KI = 0.01 controllers resulted in the increased rise times despite the integral terms. Larger overshoots brought by PI controllers are explained by the fact that the integral term action is proportional not only to the error, but also the time for which it has persisted. Thus, the error accumulation during the move causes overshoots even for very low values of integral coefficients. In addition, since the performance of the pitch system are close to maximum with P-only controller, adding integral action does not have an expected effect to steady-state error, which is even higher with PI controllers.

In order to entirely eliminate overshoot and achieve steady-state error below 0.01%, the attempt was made with gain-scheduling approach. The idea was to use P-only controller during the pitch move, and switch to PI controller only when position is within narrow range around the setpoint. The final gain-scheduled controller obtained by the tuning procedure as the best fit is the following:

$$|eAP| > 2\%\text{:}\ \ KP = 4.5;\ KI = 0$$

$$|eAP| \leq 2\%\text{:}\ \ KP = 4.5;\ KI = 0.01 \tag{11}$$

Thus, the integral action is triggered only when angular position error (eAP) is within ±2% of the setpoint. Figure 7 presents the step response of the system position with the gain-scheduled controller defined by (11).

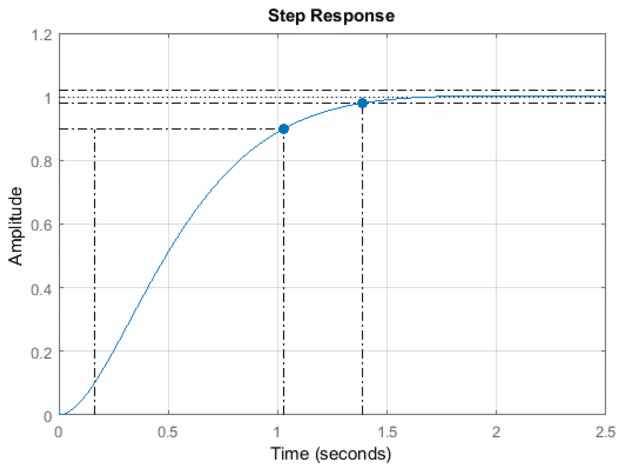

Figure 7. Step response of the system position with the final gain-scheduled controller

The response of the selected controller has the similar characteristics as for the KP = 4 controller. Its rise time is 0.88 s and the settling time is 1.44 s. However, it has not a measurable overshoot, and its steady state error is 0.008%, which is within required accuracy of 0.01%.

The frequency response of the outer position loop of the pitch system is presented in Figure 8. The outer position loop cutoff frequency is 0.386 Hz, which is two orders of magnitude smaller than the bandwidth of the inner velocity loop (see Figure 4). This bandwidth difference is an indicator of the quality of the cascade control design.

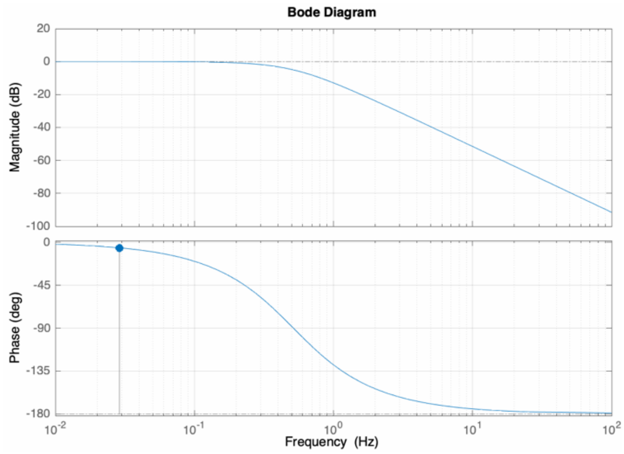

Figure 8. Bode plot of the position loop response

In addition, the closed-loop system appears stable, with phase margin of 174 degrees at 0.029 Hz.

4. CONTROL SYSTEM INTEGRATION

The pitch mechanism cascade controller is integrated in the VTI T-38 wind tunnel overall system for automation of aerodynamic experiments. It is implemented using the LabVIEW development environment on the NI CompactRIO hardware platform. The loops are closed on the FPGA level of the CompactRIO, at a fixed frequency of 1 kHz. A resolver that was previously used as position sensor is replaced by a 14-bit digital encoder, giving a position resolution of 0.002º.

Typical pitch movement profiles of a model, sampled with frequency 1 kHz, in several wind tunnel tests at Mach 2 are shown in Figure 8. After the flow is established, the model is moved from zero to the initial pitch position (-5º in these tests) at maximum speed (15º/s), then it is swept at the desired speed (from 2º/s to 12º/s) to the final pitch position (19º), and in the end, it is moved to the zero-pitch position at maximum speed before the flow is stopped.

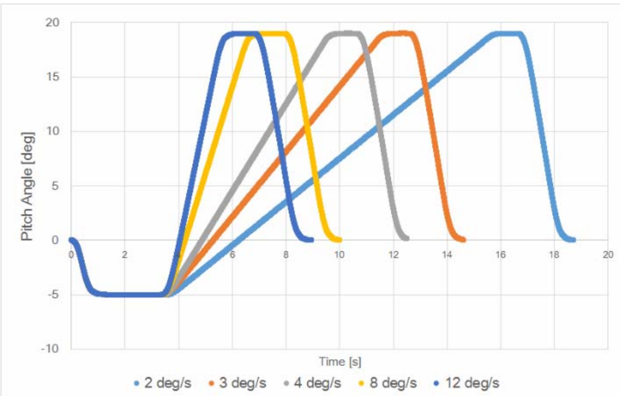

Figure 8. Typical pitch movement of the model in several wind tunnel tests

Reference tracking for all pitch speeds was within ±0.05º throughout an entire move, with the accuracy at the endpoint within ±0.01º in all wind tunnel tests.

The pitch mechanism can be operated both independently and in synchronization with other wind tunnel components during a wind tunnel run.

5. CONCLUSION

The pitch mechanism in the T-38 wind tunnel is an electro-mechanical system which is intended for high-accuracy positioning of the model in the airstream. Due to massive aerodynamic forces, the power amplification is realized through a spool valve and a hydraulic motor. In this paper, design and integration of a highly accurate gain-scheduled cascade control system for the pitch mechanism is described. It is capable of trajectory control within ±0.05º throughout an entire move at speeds of up to 15º/s. The steady-state accuracy is better than ±0.01º with no measurable overshoot. Since its integration, the system is successfully used in several wind tunnel test campaigns.

Symbols

B	[Nms]	motor viscous friction
C_d	[-]	flow discharge coefficient
D_m	[m^3/rad]	motor displacement
i	[A]	input current
J	[kg·m^2]	moment of inertia on the motor shaft
J_L	[kg·m^2]	moment of inertia of the load
J_m	[kg·m^2]	moment of inertia of the motor
J_s	[kg·m^2]	moment of inertia of the ball screw
K_V	[m^3/sA]	valve sizing constant
p	[m]	lead of the ball screw
P_L	[Pa]	load pressure drop
P_R	[Pa]	return pressure
P_S	[Pa]	supply pressure
P_V	[Pa]	valve pressure drop
Q_{il}	[m^3/s]	motor internal leakage
Q_L	[m^3/s]	control flow
T_a	[Nm]	torque due to aerodynamic forces
T_g	[Nm]	torque due to the unbalanced mass
T_f	[Nm]	torque due to friction
T_L	[Nm]	load torque
x	[m]	strut position
α	[rad/s^2]	motor angular acceleration
φ	[rad]	motor angular position
θ	[rad]	pitch angle
ω	[rad/s]	motor angular speed

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