AIR MODE OPERATION OF THE U.S. NATIONAL TRANSONIC FACILITY

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Abstract

The U.S. National Transonic Facility (NTF) can operate using nitgrogen as the test gas from ambient to cryogenic temperatures or with dry air as the test gas at ambient temperatures. The air mode operation can be used to economize the NTF operations when high Reynolds number testing is not required. In the air mode, a water cooled heat exchanger is used to control the tunnel temperature. The NTF Reynolds number envelope in the air mode is restricted by fan power limit and the heat exchanger capacity. The heat exchanger system capability varies with the sump water temperature and the desired tunnel temperature set point. Whenever the cooling capacity of the heat exchanger is exceeded, the tunnel internal structure functions as a heat sink and keeps the rate of temperature rise low, allowing marginal excursions beyond the desired temperature set point. Through dynamical simulation the control laws were designed and implemented for air mode temperature control. This control system uses a proportional-integral type control law with a feed forward of the estimated fan power equivalent of the water flow. The air temperature control system can maintain ±0.4K (±1°F) about a given temperature set point.

Introduction

The U.S. National Transonic Facility (NTF) is a unique high Reynolds number wind tunnel. It is capable of providing flow Reynolds number of 120 million on a 0.25m chord aerodynamic model at transonic Mach numbers.[1,2] The NTF Reynolds number performance is nearly an order of magnitude greater than most contemporary ambient temperature transonic tunnels. The NTF achieves high Reynolds numbers by operating in a cryogenic mode at cryogenic temperatures, using nitrogen as the test gas. The cryogenic mode of operation increases the ratio of inertial to viscous forces in the tunnel flow which represents the Reynolds number. In the cryogenic mode, liquid nitrogen is injected into the tunnel thereby cooling the tunnel structure and test gas. In this mode, the tunnel can operate between ambient and cryogenic temperatures (335 to 100K) and at pressures ranging from 1 to 8.85atm.

The NTF can also operate like most other closed circuit tunnels with dry air as the test gas. In this mode the tunnel can operate at ambient temperatures (300 to 335K) and at pressures ranging from 1 to 8.85atm. The air mode Reynolds number envelope for the NTF is much smaller than the cryogenic mode envelope as shown in figure 1. Air mode operations can be used to economize the NTF operations when high Reynolds number testing is not required.

The quality of the aerodynamic data in the NTF, as in other wind tunnels, is a function of the stability of the tunnel flow parameters (Mach number, Reynolds number, and dynamic pressure) and the test section flow quality. Stability of the flow parameters to within 0.1-0.3% of the set values is considered essential for good data quality. The dynamics of the tunnel states constitute a strongly coupled multivariable process. It involves mass enthalpy interactions among tunnel resident gas, fan pressure rise, tunnel structural enthalpy, mass flow rates, and tunnel circuit loss coefficients. The flow parameters are related to tunnel states (temperature, pressure, and Mach number) in the test section. The temperature, pressure, and Mach number dynamics of the NTF in the cryogenic mode have been analyzed and closed loop control has been established.[3]

The dynamics of the air mode pressure control and Mach number control are the same as in the cryogenic mode.[2] However, the tunnel air temperature control dynamics are vastly different from the cryogenic mode. In order to maintain the air temperature during the air mode operation, a water cooled heat exchanger is located in the settling chamber. This heat exchanger is capable of removing fan induced heat from the tunnel flow and regulates the air temperature. The temperature process is relatively slow and complex because of the inherently sluggish thermal dynamics of the system with its long lines, cooling tower, and the heat exchanger. This paper provides an analysis of the air mode performance limits for the NTF. A dynamical model of the process, and the control law design for maintaining the temperature, pressure, and Mach number are also presented. These laws are fully implemented at the NTF.

Nomenclature

A_t	cross sectional area of test section, m^2
A_{vent}	pressure control valve area, m^2
b	tunnel circuit loss coefficient
$\overline{c}$	reference aerodynamic chord, m
C_h	specific heat of heat exchanger, kJ/kgK
C_m	specific heat of metal in tunnel, kJ/kgK
C_p	specific heat at constant pressure, kJ/kgK
C_v	specific heat at constant volume, kJ/kgK
C_w	specific heat of water, kJ/kgK
k	constants in text
$k_{0,1,2}$	fan drive constants
K_c	heat exchanger constant, kJ/K s
k_f	fan power constant, kJ/K s
k_{iP}	integral gain, pressure loop
k_{iT}	integral gain, temperature loop
k_{iM}	integral gain, Mach loop
k_m	Mach number constant
k_{pP}	proportional gain, pressure loop
k_{pT}	proportional gain, temperature loop
k_{pM}	proportional gain, Mach loop
M	test section Mach number
M_{set}	Mach number set point
$\dot{m}_i$	mass flow of air into tunnel, kg/s
$\dot{m}_o$	mass flow of air through exhaust, kg/s
$\dot{m}_t$	test section mass flow, kg/s
$\dot{m}_w$	water mass flow through heat exchanger, kg/s
mole	molecular weight of air
N	fan speed, rpm
N_{set}	fan speed set point, rpm
P	total pressure, atm
P_s	test section static pressure, atm
P_{set}	pressure set point, atm
R	universal gas constant (8.314 kJ/kg-mol K)
Re	Reynolds number
r	fan pressure ratio
S	Laplace operator
T	test section total temperature, K
T_i	temperature of incoming dry air, K
T_s	test section static temperature, K
T_h	average heat exchanger body temperature, K
T_m	average metal temperature, K
T_{set}	temperature set point, K
T_{win}	heat exchanger inlet water temperature, K
T_{wp}	pump exit water temperature, K
T_{wr}	heat exchanger return water temperature, K
T_{ws}	sump water temperature, K
t_a	test section plenum time constant, s
t_m	time constant associated with wall heat release, s
u	velocity, m/s
W_g	mass of air in the tunnel, kg
W_h	mass of heat exchanger and resident water, kg
W_t	mass of metal within tunnel insulation shell, kg
x	water mixing ratio
x_b	water bypass valve stroke
ρ	density, kg/m^3
μ	viscosity, $N\text{-}s/m^2$
γ	ratio of specific heats (γ=1.4)
$2\delta h$	temperature across heat exchanger walls, K
τ	transport delay in cooling water circuit, s
δ_{igv}	fan inlet guide vane angle
ΔT	steady state fan temperature rise or heat exchanger temperature drop, K
$\frac{d}{dt}$	derivative expression

Air Mode Tunnel Process

The NTF is a closed-circuit single-stage fan driven pressure tunnel designed to operate with nitrogen as the test gas, at temperatures ranging from 100 to 335K. The tunnel can also operate with dry air at temperatures ranging from 300 to 335K, depending upon the heat exchanger water supply temperature. The Reynolds number operational envelope of the tunnel in the air mode is restricted by heat exchanger limitations and fan drive limitations. The Reynolds number envelopes of the NTF operation in cryogenic and in air mode are shown in figure 1. Figure 2 shows the tunnel circuit and the locations of the fan, the rapid diffuser leading to the heat exchanger, settling chamber, contraction, test section and diffuser segments of the closed circuit tunnel.

During air mode opeation dry air is injected into the tunnel to buildup the tunnel air mass and control the tunnel pressure. This air is derived from a 1800psig high pressure dry air supply with a dew point of about 230K. The inlet mass flow is in the range of 1 to 10kg/s at a typical ambient temperature of 320K. The inlet mass flow is set to a constant bias flow under steady state operating conditions. Excess air is discharged through the pressure control valves to the atmosphere through the vent stack. A closed loop pressure control law manages mass flow through the pressure control valves. To save dry air, the bias flow is kept to a low value such that the pressure control valves function at their minimum linear range.

The fan rotation creates a pressure ratio across the fan which results in continuous mass flow around the tunnel circuit. The contraction section accelerates the flow isentropically into the test section providing the desired test velocity. The pressure ratio can be controlled either by fan speed control or by varying the fan inlet guide vane angle. The adiabatic work done by the fan results in heat generation. The fan induced heat, unless removed from the tunnel flow, progressively increases the tunnel gas and structural temperatures. During air mode operation the fan induced heat is removed from the flow by the water cooled heat exchanger. To reduce the tunnel circuit losses, the heat exchanger is located in the settling chamber at the exit of the rapid diffuser where the flow velocity is the lowest.

The heat exchanger consists of multi-path finned copper tube bundles carrying cooling water. The tubes span the cross section of the settling chamber in two uniform layers of eighteen bundles each. Figure 3 shows how the water flows from bottom to top in the upstream layer of tubes and from top to bottom in the down stream layer of tubes. Water mass flow in tube bundles have been adjusted using throttling valves to provide uniform air temperature at the heat exchanger exit. The external surface of the tubes provides a large exposed area to the tunnel air flow. This exposure allows efficient heat transfer from the warm air to the cooling water in the heat exchanger.

The heat transfer between air and water is a function of air temperature, velocity, pressure, exposed area of tubing, conductivity of the copper tubes, mass flow of water, and temperature of the cooling water. The returning warm water is cooled in a cooling tower with an up-draught fan.

With a heat exchanger system the quantum of heat removed at any given tunnel condition can be controlled by varying the mass flow of water through the heat exchanger at a fixed inlet water temperature or by maintaining a constant mass flow and varying the inlet water temperature.

The varying mass flow control would require a 1:4 rangebility in water mass flow because the NTF fan power varies form 10 to 41MW during air mode opeation. Varying the mass flow using a constant displacement type water pump is inefficient and difficult to realize the 1:4 flow ranges. Because of varying water velocities, the spatial profile of air temperature in the settling chamber will also vary with tunnel power.

Alternately, the heat removed from the tunnel can be controlled by varying the inlet water temperature with a fixed water mass flow. Constant water mass flow makes the water pump work at an optimal condition. The inlet water temperature control is realized by mixing a fraction of the heat exchanger return water with cooler sump water. In such a case, the inlet water temperature depends on the mixing ratio and is limited to the sump water temperature at low end and the heat exchanger exit water temperatures at high end. The details of the thermal system are discussed in the process modeling section. This concept of constant mass flow cooling yields better spatial profile of air temperature in the settling chamber than the varying mass flow method.

The heat exchanger circuit at the NTF for controlling the inlet water temperature with a fixed mass flow is shown in figure 3.

Since the heat exchanger system reuses the same water in a closed cycle (with limited new water makeup capability), the sump water temperature is dictated by seasonally varying ambient wet bulb temperature, the total mass of water in the system, and the cooling tower thermal loading history.

Process Modeling

Mass Flow. Given the density of air as 339P/T, and assuming isentropic expansion of air as a perfect gas, the steady state test section mass flow ($\dot{m}_t$) can be estimated as:

$$\dot{m}_t = \rho u A_t = 339 \frac{P_s}{\sqrt{T_s}} \sqrt{\frac{R\gamma}{\text{mole}}}\, M A_t$$

Fan Power. The tunnel circuit loss coefficient dictates the fan power and fan pressure ratio required to move the air around the closed circuit. The fan pressure ratio is related to spatial dynamic pressure profile. It is usually considered to be invariant with Reynolds number in the relation $r \propto M^2$, expressed as $r = 1 + bM^2$. However, in the NTF, the fan pressure ratio does vary weakly with tunnel Reynolds number which modifies the fan pressure ratio expression to $r \propto M^{1.5}$. The fan pressure ratio and the fan temperature ratio are related under isentropic flow conditions as:

$$r = 1 + bM^{1.5} = \left(\frac{T}{T - \Delta T}\right)^{\frac{\gamma}{\gamma-1}}$$

Therefore the temperature rise across the fan (ΔT) can be represented as:

$$\Delta T \simeq T \left\{ \frac{r^{\left(\frac{\gamma-1}{\gamma}\right)} - 1}{r^{\left(\frac{\gamma-1}{\gamma}\right)}} \right\}$$

The adiabatic work done at the fan, which results in heating of tunnel resident gas, can be evaluated assuming fan mass flow to be the same as test section mass flow.

$$\text{fan power} = C_p\, \dot{m}_t\, \Delta T$$

$$= k_f \frac{P\sqrt{T} M^{2.5}}{\left(1 + \frac{\gamma-1}{2} M^2\right)^{\frac{\gamma+1}{2(\gamma-1)}}}$$

where $k_f = 339\sqrt{\frac{R}{\text{mole}}}\, bC_pA_t \frac{\gamma-1}{\gamma}$

Heat Exchanger Water Mixing. The schematic of the heat exchanger system is shown in figure 3. The water pump is operated at a constant mass flow of $\dot{m}_W$. The heat exchanger return water is mixed with the cooler sump water at the suction of the pump in a mass flow ratio (1-x):x. This mixing is controlled by operating the bypass valve. Assuming one dimensional flow and ideal mixing, the pump exit water temperature (T_{wp}) can be expressed as:

$$\dot{m}_W C_W T_{wp} = [x\dot{m}_W] C_W T_{wr} + [(1-x)\dot{m}_W] C_W T_{ws}$$

$$T_{wp} = xT_{wr} + (1-x)T_{ws}$$

where $0 \leq x \leq 1$

Heat Exchanger. In the proposed heat exchanger model, the air temperature drops from the entry value of $T + \Delta T$ to T at the exit of heat exchanger. The water temperature increases from the entry value of T_{win} to an exit value of T_{wr}. Hence the mean value of air temperature in the heat exchanger is $T + \frac{\Delta T}{2}$.

Though the profile of heat exchanger body temperature varies from the entry point of water to exit, the average temperature of heat exchanger metal mass can be taken as T_h. The exit water temperature (T_{wr}) is taken to be ($T_h - \delta h$) in the present analysis.

Heat conduction occurs from air to water through the thickness of the copper tubes, with a wall temperature difference of $2\delta h$. If the mass of heat exchanger is W_h (with equivalent tube resident water mass included) with a specific heat of C_h (including equivalent water specific heat), and the tunnel mass flow is $\dot{m}_t$, the dynamics of the heat exchanger can be expressed as:

$$W_h C_h \frac{dT_h}{dt} = \dot{m}_t C_p \left\{ \left(T + \frac{\Delta T}{2}\right) - (T_h + \delta h) \right\} - C_W \dot{m}_W \left\{ (T_h - \delta h) - T_{win} \right\}$$

This expression accounts for quantum of heat absorbed in the body of the heat exchanger. The conduction across the walls of the heat exchanger is the mean of entry and exit heat rates:

$$2\delta h\ K_c = \frac{1}{2}\left\{\dot{m}_t C_p\left\{(T+\frac{\Delta T}{2}) - (T_h+\delta h)\right\} + C_w \dot{m}_w\left\{(T_h-\delta h) - T_{win}\right\}\right\}$$

This expression accounts for the heat transfer of tunnel fan heat through the walls of the heat exchanger to water in the conduction mode.

Transport Delay. The length of the water pipe from the pump outlet to the heat exchanger is large, as is the return pipe length from the heat exchanger exit to mixing valve and pump inlet. The heat exchanger inlet temperature has a transport time delay relative to water pump outlet. This transport delay is lumped into one as:

$$T_{win} = T_{wp} e^{-\tau S}$$

Transport delay (τ) can be estimated from the volume of pipe divided by the mass flow rate, which is invariant because of constant mass flow operation.

The previous equations correspond to a quasi-steady dynamic representation of the heat transfer from air to water. In this model, the steady state values have been used to develop the basic thermal dynamics of the system, without going in for true dynamical representation of the complex heat flow. This model does not account for the cooling tower dynamics, which involves the mass of water in the sump, the dew point of atmospheric air, cooling tower fan, and water film area characteristics.

Temperature. The quasi-steady dynamics of the air temperature can now be evaluated, utilizing the other mass enthalpy terms in the tunnel circuit. They are, the thermal mass internal to the tunnel insulation, air enthalpy in and out of the tunnel, heat absorbed by the heat exchanger and the fan work done. The air temperature dynamics are represented as:

$$\frac{dT}{dt} = \frac{1}{W_g C_v}\left\{k_f \frac{P\sqrt{T}M^{2.5}}{(1+0.2M^2)^3} + (\dot{m}_i-\dot{m}_o)C_p T - W_t C_m \frac{T-T_m}{t_m} - \dot{m}_t C_p\left\{(T+\frac{\Delta T}{2}) - (T_h+\delta h)\right\}\right\}$$

$$\frac{dT_m}{dt} = \frac{T-T_m}{t_m} \text{ where } t_m \propto \frac{1}{\dot{m}_t}$$

Under equilibrium conditions, the fan heat cancels the water cooling capacity, the net dry air based enthalpy tends to zero and the tunnel structural absorption tends to zero.

Pressure. The enthalpy entering the tunnel as dry air is $\dot{m}_i C_p T_i$. This enthalpy, interacts with tunnel resident gas mass. The tunnel discharge enthalpy is $\dot{m}_o C_p T$. Assuming $T_i \simeq T$, the pressure dynamics of the tunnel can be written as a function of mass flow rates and the rate of change of tunnel gas temperature:

$$\frac{dP}{dt} = \frac{P}{T}\frac{dT}{dt} + \frac{P}{W_g}(\dot{m}_i - \dot{m}_o)$$

Mach Number. The tunnel Mach number varies with the mass flow in the subsonic and transonic regime where as the test section geometry controls the supersonic behavior. The tunnel Mach number dynamics are represented as:

$$\frac{dM}{dt} = \frac{k_0}{k_1-k_2\delta_{igv}}\ \frac{N}{t_a\sqrt{T}} - \frac{M}{t_a}$$

$$M = \frac{N}{\sqrt{T}}\frac{k_0}{(k_1-k_2\delta_{igv})}$$

The fan inlet guide vane angle and speed are used to control the tunnel Mach number. This dynamic model is a small perturbation linearization of the true fan map of the NTF.

The experimentally determined fan map for the NTF is shown in figure 4. This map shows the temperature normalized fan speed (rpm/$\sqrt{K}$) as a function of Mach number for two extreme limits of the inlet guide vane settings. The plot is closed with one end at very low fan speed and the other end when the tunnel is about to choke at M=1. The fan map is also illustrated in figure 5 as a carpet plot with inlet guide vane steps of 1° from 30° to -20° and the normalized fan speed in steps of 0.555 from 2.22 to 21.7rpm/$\sqrt{K}$. This fan map has been used in the evaluation of various air mode performance envelopes including the power limits imposed by the drive motors in the following section.

The NTF fan is driven by two wound rotor induction motors in tandem which are capable of delivering a maximum of 41MW at 840 motor revolutions per second. These motors are driven by a variable frequency power supply between 70 and 840rpm. At a gear ratio of 1.4, the maximum fan speed corresponds to 600rpm. With a gear ratio of 2.33, the maximum fan speed corresponds to 360rpm. The maximum power that can be delivered is linearly related to the motor speed with a constant current limit.

Reynolds Number. The following expression is used to estimate the flow Reynolds number.

$$R_e = \frac{\rho \, \overline{c} \, u}{\mu} = \frac{339 \dfrac{P_s}{\sqrt{T_s}} \sqrt{\dfrac{R\gamma}{\text{mole}}} \, M \, \overline{c}}{0.1082 T_s^{0.9}}$$

$$= k \, \overline{c} \, \frac{P}{T^{1.4}} \frac{M}{(1+0.2M^2)^{2.1}} 10^6$$

This expression assumes that air and nitrogen behave similarly as ideal gases.

Heat Exchanger Performance

The design specifications of the NTF heat exchanger system are shown in table 1. A schematic diagram of the heat exchanger water flow paths is shown in figure 3. The gross performance of the system can be made from the steady state power identity, obtained by assuming perfect conduction in the copper tubes and a quasi-steady identity expressed as:

$$\dot{m}_t C_p \left\{ T + \frac{\Delta T}{2} - T_h \right\} = \dot{m}_w C_w \left\{ T_h - T_{ws} \right\}$$

$$T_h = \frac{\dot{m}_t C_p (T + \frac{\Delta T}{2}) + \dot{m}_w C_w T_{ws}}{\dot{m}_w C_w + \dot{m}_t C_p}$$

Table 1. Design Specification of the NTF Heat Exchanger

Capacity: 35.17 MJ/s (1.2×10^8 BTU/hr)	
Heat Transfer Coefficient: 0.09 to 0.125 kJ/s/m^2/K	
Air:	Inlet Temp.: 351.3 K
	Outlet Temp.: 338.9 K
	Flow Rate: 2820 kg/s
	Pressure Drop: 60 to 100 psf
	Surface Area: 9800 m^2
	Tube Length: 13000 m (36 paths)
Water:	Inlet Temp.: 302.78 K
	Outlet Temp.: 321.67 K
	Flow Rate: 510 kg/s (fixed)
	Pressure Drop: 25 psid
	Velocity: 1 to 1.5 m/s
Mass:	Tube Metal: 22610 kg
	Water in Tubes: 3700 kg

Performance Envelopes

The NTF air mode performance envelope is bounded by the tunnel maximum pressure, the fan drive capacity, and the heat exchanger capacity. Estimates of the performance boundary have been made and are presented in figures 4 to 9.

Figure 4 and 5 show the fan map of the NTF both as a temperature normalized speed vs Mach number plot and a carpet plot of temperature normalized fan speed, Mach number, and inlet guide vane angle.

Figure 6 shows the air mode Reynolds number envelope. This envelope is bounded by the tunnel maximum pressure limit and the heat exchanger limits for sump water temperatures of 294.4K (70°F) and 300K (80°F). This envelope shrinks when the sump water temperature rises to 300K. The envelope also shows the limits imposed by fan drive motors for the choice of gear ratio and the loci of inlet guide vane limits. The need for the 2.33 ratio gearing at high Reynolds number is also illustrated.

Figure 7 shows the maximum usable pressure associated with the Reynolds number envelope. The boundaries are the tunnel maximum pressure, heat exchanger limits for two sump temperatures and the loci of inlet guide vane limits for the choice of gear ratios.

The air mode power envelope is shown in figure 8. It is bounded by same constraints of maximum tunnel pressure, heat exchanger capacity limits for the two sump temperatures and the loci of inlet guide vane with associated gear ratios. Though the motors are rated at 17MW each, they have an overload capability of 20.5MW each, and 41MW is used in determining the power envelope.

Figure 9 shows the air mode fan speed envelope. Shown is the relationship between fan speed and Mach number for extreme and zero inlet guide vane angles. This is a closed plot indicating the range of speed to inlet guide vane relationship. The use of the gear ratio of 1.4 and 2.33 are shown.

Figure 10 shows a pressure-Mach number plot for a typical NTF constant Reynolds number testing. In both the cases, the operating point crosses the limit imposed by 300K sump water temperature. In such a case, the tunnel internal metal slowly soaks up the extra heat and the temperature control is lost. The testing does not cross the 294.4K envelope limit, indicating that the temperature control authority would be available if the sump temperature were 294.4K.

The NTF internal metal mass is nearly 378,000kg with a specific heat of 1.015kJ/kgK at ambient temperatures. This mass can absorb 384MJ for a temperature rise of 1K. When the bypass valve runs out of authority, the excess heat is absorbed by the metal mass. The resultant temperature rise for the gas and internal metal mass is very slow. Each degree rise in temperature allows the heat exchanger system to remove an extra 800kW. An excess power of 1MW beyond what the heat exchanger system can remove results in about 0.15K/minute temperature rise for the internal metal mass. The NTF has been run many times in this mode up to nearly 10-20 minutes with about 2-3K gas temperature rise in the tunnel. Figure 10 shows two typical experiments from NTF, where the cooling power boundary have been crossed.

The transport delay (τ) in the water circuit is experimentally estimated by performing a power spectral density analysis on tunnel temperature and water temperature time histories. Shown in figure 11 is a sample of temperature data covering a transient in the bypass valve that is recorded for analysis. The power spectral density analysis shown in figure 11 yields a spectrum with a strong peak at 0.0116Hz, corresponding to about 86 seconds of circuit time delay.

Dynamic Simulation

The five nonlinear process differential equations for P, M, T, T_m, and T_h representing the air mode dynamics can be solved using Runge-Kutta-Ferlberg integration routines to generate time trajectorie responses of the system on simulation software.[4] This simulation was used to evaluate candidate closed loop control laws for air mode temperature control. The pressure and Mach number closed loop control laws are the same as the ones used for cryogenic mode operation.

Initially, open loop system responses were studied. This led to a closed loop control scheme consisting of proportional-integral type law with a feedforward of the estimated fan power equivalent of the water flow.

Figure 12 illustrates a simualtion of the air temperature control for a test program at 6.96 million/chord Reynolds number with Mach number ranging from 0.5 to 0.78. During this set of runs, the tunnel pressure has been set to yield a desired constant Reynolds number. The fan power exceeds the cooling capacity of the heat exchanger when the tunnel Mach number exceeds 0.78. At this point the tunnel gas and internal metal temperatures slowly increase. This rate of increase in temperature is slow because of the large thermal mass of the internal metal structure. The bypass valve remains fully closed at this point and looses its authority to control the tunnel temperature.

Air Mode Temperature Control

The closed loop control law derived from the above simulation study was implemented on the NTF. The control law called for a power feedforward which is derived on the basis of the tunnel temperature (T) and the sump water temperature (T_{ws}) and is described below.

$$x_b = k_{pT}(T - T_{set}) + k_{iT}\int (T - T_{set})dt + \frac{\frac{k_f P\sqrt{T}M^{2.5}}{(1+0.2M^2)^3}}{\dot{m}_w C_w (T_{set} - T_{ws})}$$

where $0 \le x_b \le 100\%$

$$0 \le k_{iT}\int (T - T_{set})dt \le 100\%$$

$$T > T_{ws} \text{ and } T_{set} > T_{ws}$$

Pressure Control

The closed loop control law for tunnel pressure control in the air mode is the same as the one used in cryogenic mode of operation. The input mass flow ($\dot{m}_i$), from the dry air supply is usually kept constant. This quantity can be varied by the operator if desired. The exhaust mass flow ($\dot{m}_o$) is controlled using the feedback control law to regulate the pressure control valves to hold the pressure accurately. The pressure

$$\dot{m}_o = \frac{1}{T}\left\{k_{pP}(P - P_{set}) + k_{iP}\int (P - P_{set})dt\right\} = k\frac{P}{\sqrt{T}}A_{vent}$$

where $0 \le k_{iP}\int (P - P_{set})dt \le 100\%$

In the cryogenic mode, the liquid nitrogen mass flow ($\dot{m}_i$) varies from 0 to 500kg/s where as in the air mode it is a fixed mass flow of less than 10kg/s. The pressure control law works for both modes of operation.

Mach Number Control

The closed loop control law for tunnel Mach number control in the air mode is the same as the one used in cryogenic mode of operation. Mach number control is accomplished by fan speed control and fan inlet guide vane control. The closed loop speed control law uses fan speed command at the existing guide vane angle setting. This is a simple on-off control law based on a 5rpm error band command to speed control:

$$\frac{dN}{dt} = 2 \text{ when } |(N_{set} - N)| < 5$$

$$\frac{dN}{dt} = -2 \text{ when } |(N_{set} - N)| > 5$$

$$\frac{dN}{dt} = 0 \text{ when } |(N_{set} - N)| \le 2$$

The fan inlet guide vane angle based on Mach number closed loop control can be invoked at any fan speed. The control law is:

$$\delta_{igv} = \frac{N}{\sqrt{T}}(k - k_M M)\left\{k_{pM}(M - M_{set}) + k_{iM}\int (M - M_{set})dt\right\}$$

where $0 \le k_{iM}\int (M - M_{set})dt \le 100\%$

$$-20^\circ \le \delta_{igv} \le 30^\circ$$

Results

The air temperature control law has provided control of temperature in the tunnel as illustrated in figure 13 and 14. The temperature response is sluggish for all disturbances, and shows the effect of long transport delay as an initial oscillatory response for large power changes. The temperature difference between the tunnel temperature and return water temperature grows with increasing power. In one case, the tunnel power has exceeded control authority and the tunnel temperature gradually increases. The pressure and Mach number control are also shown. The pressure control rate for pressure buildup is slow because of $\dot{m}_i$ limit, where as the pressure control rate for pressure decrease is faster. The inlet guide vane angle Mach number control system is the fastest responding system of all three control systems. This system is capable of provideing step like responses to Mach number set point changes. The system also is capable of maintaining Mach number during model attutude changes in the test section.(3)

The control laws are capable of regulating the pressure to ± 0.005 atm, temperature to ± 0.4K in air mode (± 0.2K in cryogenic mode) and Mach number to 0.0015 of the desired value.

Concluding Remarks

The NTF, though designed for cryogenic nitrogen operation, can be operated at ambient temperatures with dry air as the test gas. In the air mode, a water cooled heat exchanger is used to control the tunnel temperature. The NTF Reynolds number envelope in the air mode is restricted by fan power limit and the heat exchanger capacity. The heat exchanger system capability varies with the sump water temperature and the desired tunnel temperature set point. Whenever the cooling capacity of the heat exchanger is exceeded, the tunnel internal structure functions as a heat sink and keeps the rate of temperature rise low allowing marginal excursions beyond the desired temperature set point. The air temperature control system can maintain $\pm 0.4K$ ($\pm 1°F$) about a given set point. This air temperature control law uses a proportional-integral type law with a feed forward of the estimated fan power equivalent of the water flow.

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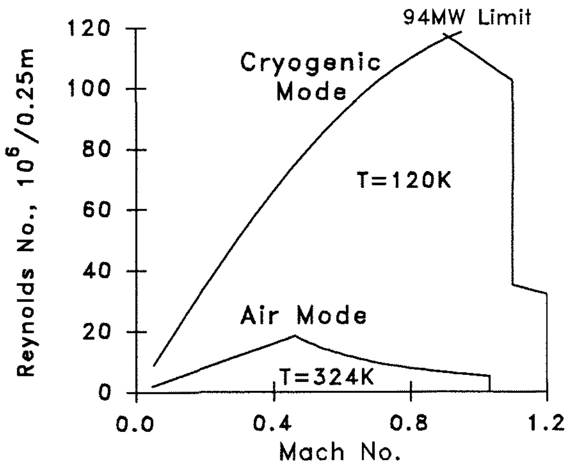

Figure 1. NTF Operational Envelopes.

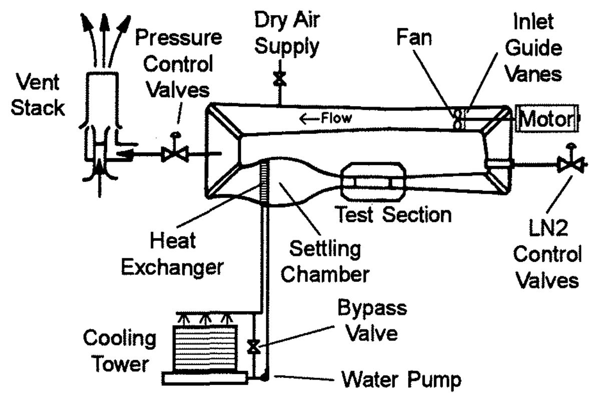

Figure 2. NTF Tunnel Schematic.

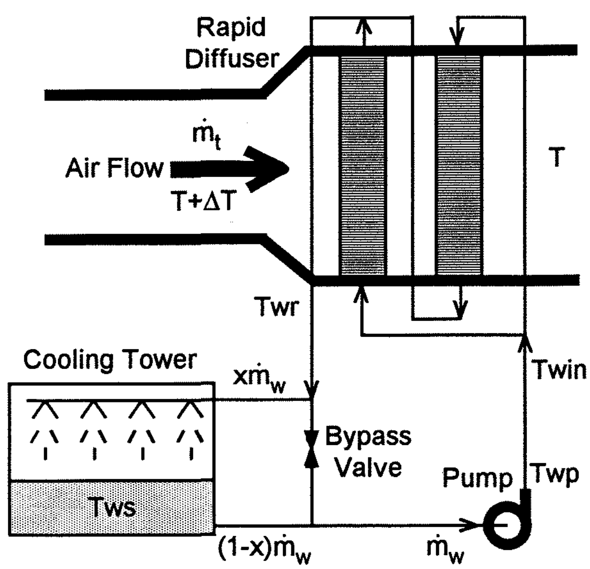

Figure 3. Heat Exchanger System.

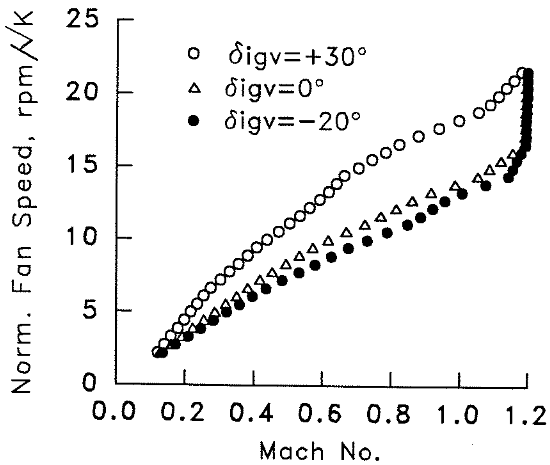

Figure 4. NTF Fan Map.

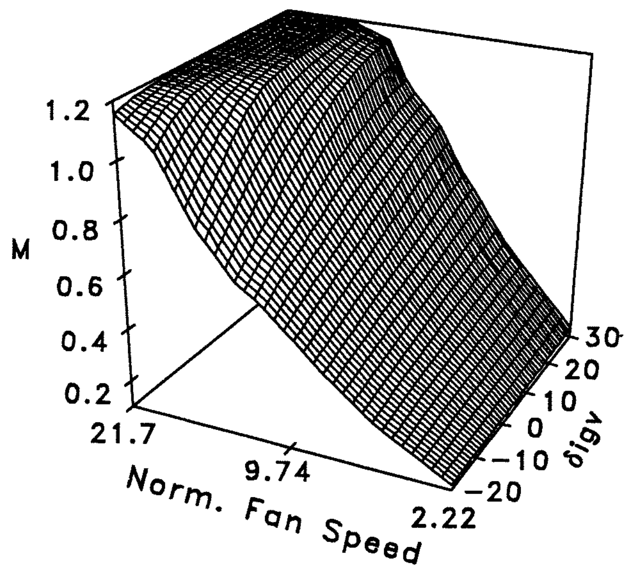

Figure 5. NTF Fan Map.

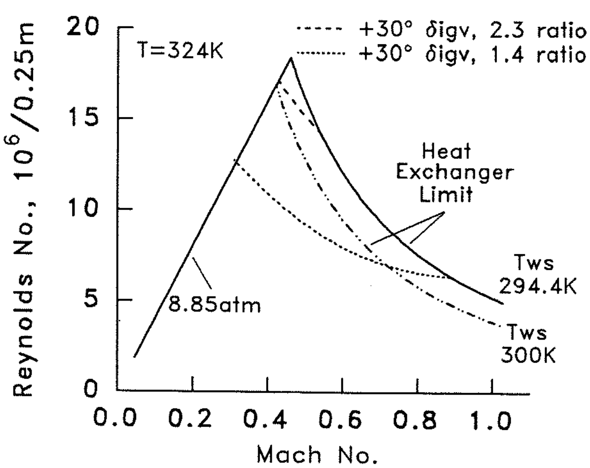

Figure 6. Air Mode Operational Envelope

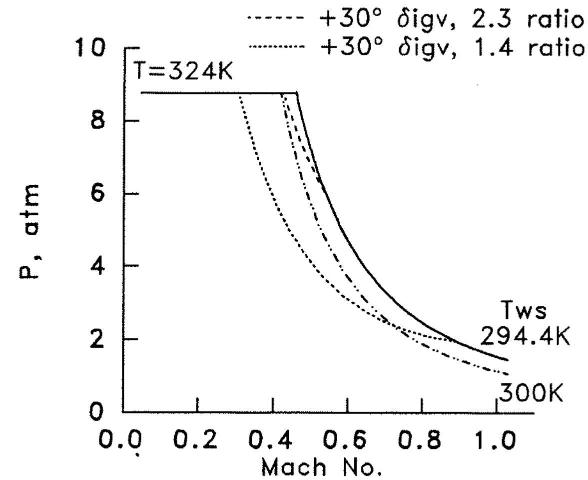

Figure 7. Air Mode Pressure Envelope.

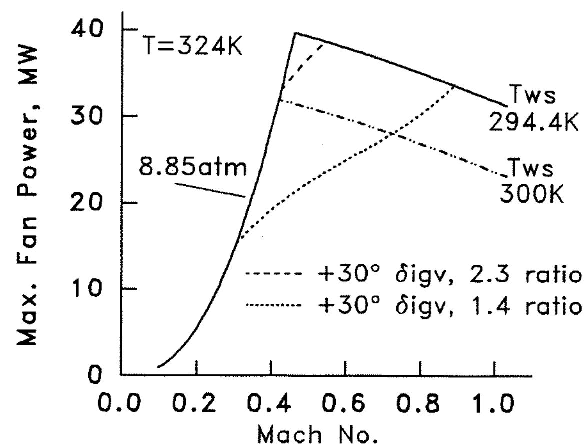

Figure 8. Air Mode Power Envelope.

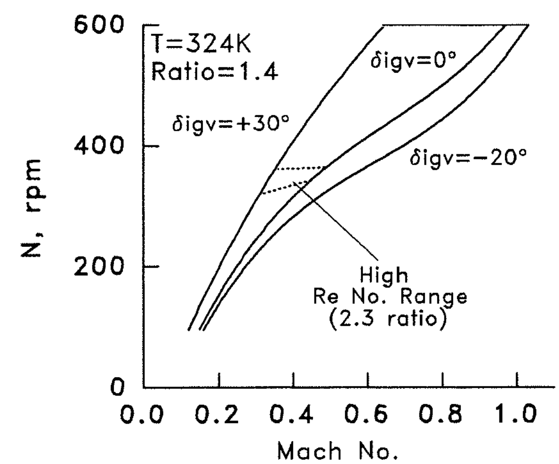

Figure 9. Air Mode Fan Speed Envelope.

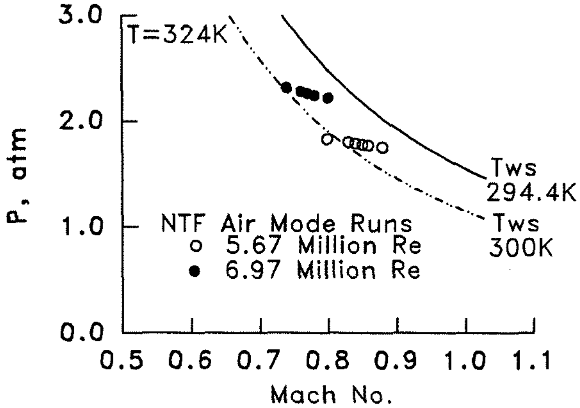

Figure 10. Air Mode Operation.

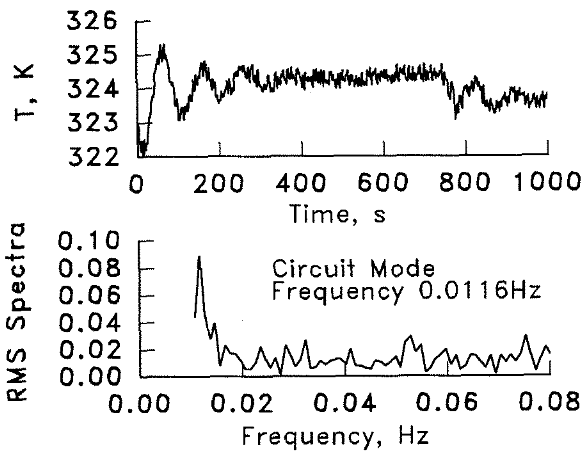

Figure 11. Air Temperature Response.

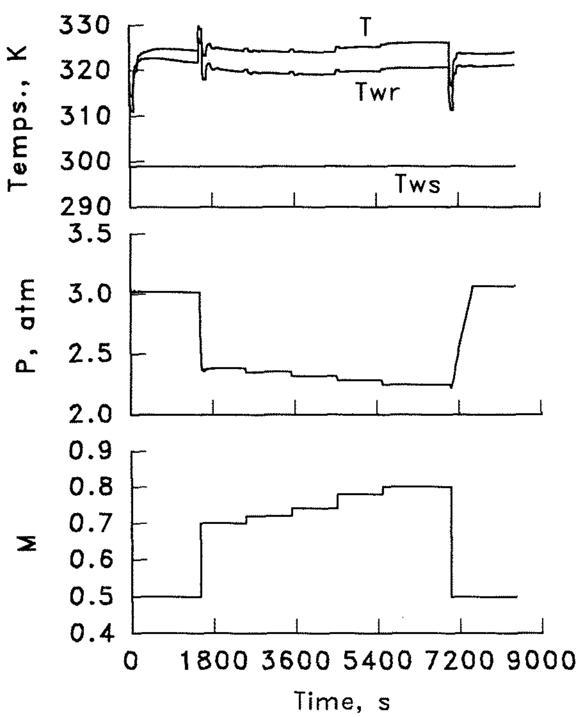

Figure 12. Air Mode Simulation.

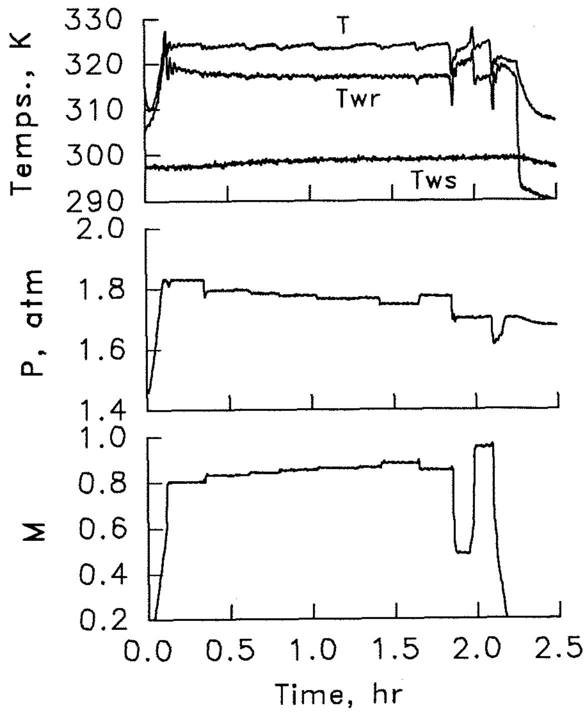

Figure 13. Air Temperature Control.

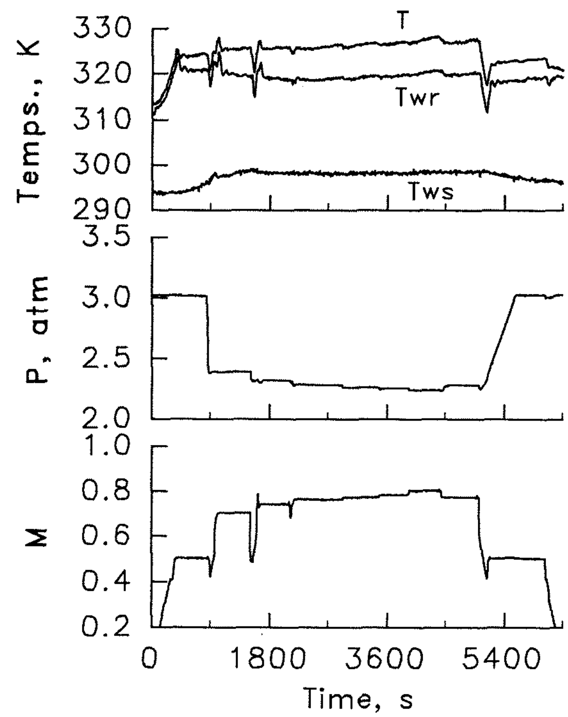

Figure 14. Air Temperature Control.