MODELS FOR THE PREDICTION OF TRANSIENTS IN CLOSED REGENERATIVE GAS-TURBINE CYCLES WITH CENTRIFUGAL IMPELLERS

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ABSTRACT

This paper presents transient-flow component models for the prediction of the transient response of gas turbine cycles. The application is to predict the transient response of a small solar-powered regenerative gas-turbine engine with centrifugal impellers. The component sizes are similar to those under consideration for the solar-powered Space Station, but the models can easily be generalized for other applications with axial or mixed-flow turbomachinery. New component models for the prediction of the propagation of arbitrary transients in centrifugal impellers are developed. These are coupled with component models for the heat exchangers, receiver and radiator. The models are based on transient applications of the principles of conservation of mass, energy, and momentum. System transients driven by sinusoidal and double-step inputs in receiver salt temperature are presented and discussed. The new turbomachinery models and their coupling to the heat-exchanger models simulate disturbance-propagation in the components both upstream and downstream from the point of generation. This permitted the study of the physical mechanisms of generation and propagation of higher-frequency contents in the response of the cycle.

INTRODUCTION

The steady-flow performance of various gas-turbine engines can be predicted to any desirable degree of model complexity, depending on available information. Models for the prediction of transient incompressible fluid flow through simple stationary passages are relatively easy to develop, because of the constant density assumption, and are readily available (Moody, 1990). Similarly, models for the prediction of transient compressible fluid flow through simple stationary passages are relatively easy to develop by predicting with characteristics the time of propagation of transients. Transient-flow models for axial and
centrifugal compressor and turbine passages, where the passages are rotating and work is added to or extracted from the flow, are harder to develop. The time currently required for computational fluid dynamics (CFD) transient computations makes them unsuitable for predicting the transient response of gas-turbine components and power plants through speed, load, and mass transients.

Non-CFD models suitable for predicting the transient performance of compressors and turbines fall into three main categories. The first category predicts the transient performance by obtaining equations resembling transfer functions in control theory, using phenomenological analyses of component behavior (for example, Kuhlberg et al. 1969; Kalinitsky and Kwatny, 1981). In the second category the unsteady conservation equations (mass, momentum, and energy) are written in a lumped-parameter approach (or alternatively finite volumes are considered along a mid streamline) through one or more components. Many of the previous models in the second category use some simplifying assumptions to minimize the effects of compressibility, or the effects of mass storage or mass depletion inside the component through the transient (for example, Adams et al., 1965; Corbet and Elder, 1974; Macdougall and Elder, 1982). In the third category assumptions are made for the delay of transport of perturbations from component inlet to outlet, based on mechanical analogues (for example, Ray and Bowman, 1976; Pink, Campsty and Greitzer, 1991). All three categories may incorporate routines for predicting relations between inlet and outlet properties from the component performance maps.

A previous paper (Korakianitis, Hochstein, and Zou, 1993) presented a first attempt on developing models of the second category (without simplifying assumptions for compressibility, mass storage, etc.) for the prediction of the propagation of arbitrary transients through regenerative-gas-turbine cycle components. The three-point lumped-parameter models for the turbomachinery components resulted in a stiff system of algebraic and differential equations. Numerical stability considerations and multiple iteration passes through the turbomachinery components imposed very restrictive upper and lower time-step limits in the integration. They also imposed limits on the amplitude of the disturbances for which the system model could be run. The allowable time step was very small, so that both the total time of transient flow studied and the frequency of fluctuations were unrealistically fast. This paper: presents a new set of improved models for the propagation of transients through centrifugal impellers; couples them to the other transient component models to form the regenerative gas-turbine cycle; presents typical time-dependent results with realistic transients that can be run in reasonable times (because the new transient centrifugal turbomachinery models handle larger-amplitude disturbances and run significantly faster than the previous ones); and explains the mechanisms of generation and propagation of transients in the components.

**CYCLE CONFIGURATION**

Electrical power for the Space Station is partially provided by a photovoltaic module, and partially by two solar-powered closed regenerative Brayton-cycle engines. Simplified component arrangements, approximate dimensions, and other information on this concept are included in many NASA and NASA-contractor reports, such as the two reports by Rocketdyne (1986, 1989), and in our previous paper (Korakianitis, Hochstein and Zou, 1993). The approximate temperature-entropy diagram for the cycle is shown in Fig. 1. The working fluid is a mixture of Helium and Xenon (He-Xe) gases. The centrifugal compressor, radial turbine, and alternator rotate as a single unit at (approximately) constant speed. The system rotates in space in and out of the shadow of the earth with a period of approximately 90 minutes, implying transients for the receiver and cooler. The receiver phase-change salt is used to provide energy at almost-constant temperature to the cycle. The compressor-turbine-alternator unit rotates at about 32,000 rpm. At the design point the compressor and turbine pressure ratios are just below 2:1, and the mass-flow rate is approximately 0.8 kg/s. The compressor and turbine are small radial impellers. Although the following component models are applied to this system, the principles are general and they can be used in different gas-turbine engines with different components. In the following the working fluid is modeled as a perfect gas with constant \( c_p = 520.4 \text{ J/(kg \cdot K)} \) and \( c_v = 312.2 \text{ J/(kg \cdot K)} \).

**TURBOMACHINERY MODELS**

The important inputs to predict the transient performance of gas turbines are the type and size of engine components, details of their geometric construction, working fluid, and inputs driving the transient. Our previous turbomachinery models considered flow properties at the inlet, middle, and outlet of the impeller, requiring only a cursory description of the centrifugal geometry. The present models for the prediction of steady and unsteady performance of the impellers (Korakianitis and Vlachopoulos, 1994a) are one-dimensional finite-difference models through the impeller passage, with many nodes, requiring a more-elaborate description of the impeller geometry. The flow through general centrifugal turbomachinery passages (Fig. 2) is assumed compressible, adiabatic, viscous (or inviscid), unsteady, one-dimensional (along \( \xi \)) with area change \( A(\xi) \). The flow is considered along the rotating curvilinear frame of reference \((\xi, \eta, \zeta)\) in the middle of the blade passage.

The governing equations are derived from the time-dependent form of the conservation of mass, momentum and energy applied to a simple control volume shown in Fig. 3.

\[
\begin{align*}
\frac{dm^i}{dt} & - (\dot{m}^i - \dot{m}^o) \\
\frac{d(I^i)}{dt} & - (I^o - I^i) \\
\frac{d(m^o e^j)}{dt} & - (\dot{m}^o h^o - \dot{m}^i h^o) \\
\end{align*}
\]

Introducing the one-dimensional variations of properties and geometry along \( \xi \) as shown in Fig. 3, the governing equations are obtained in a conservative form in the rotating frame of reference. The resulting partial differential equations non-dimensionalised with the total properties at the inlet, the inlet flow area, and the passage length, are:

\[
\frac{\partial q}{\partial t} + \frac{\partial F}{\partial \xi} = b
\]
where

\[ q = \begin{cases} \rho A & \\
\rho A w & \\
\rho A e_R & \end{cases} \]

\[ F = \begin{cases} \rho A w & \\
A(pw^2 + p) & \end{cases} \]

\[ b = \begin{cases} p (dA/d\xi) + \rho A [(\sin \beta (dt/d\xi) + \Omega^2 \sin \beta \cos \beta)] + f_{fr}(A_{fr},t) & \\
-\rho A (\Omega r - \omega \sin \beta) (dt/dr) - f_{fr}(A_{fr},t) & \end{cases} \]

In the momentum term of vector \( b \) one can identify from left to right: the area gradient effects; the Coriolis acceleration effects; the centrifugal force effects; and the wall friction effects, for all faces of the element.

In the energy term of vector \( b \) one can identify from left to right: the Coriolis acceleration effects; and the wall friction effects in the tip region (which are zero for shrouded impellers). The viscous forces between the walls and the fluid have been included in the form of frictional forces per unit length. These forces can be approximated using a friction coefficient approach and a corresponding Reynolds number.

In equations 2, \( w \) represents the relative flow velocity along the mean streamline direction \((\xi,\eta)\), and \( e_R \) is the so-called rotating total energy per unit mass (the equivalent of total energy for a stationary frame of reference) defined by:

\[ e_R = \frac{w^2}{\gamma - 1} - \frac{(\Omega r)^2}{2} \]

so that the state equation for the gas takes the form:

\[ p = \rho \cdot (\gamma - 1) \cdot \left( e_R - \frac{w^2}{\gamma - 1} - \frac{(\Omega r)^2}{2} \right) \]

This model requires specification of flow angles (near the operating point these are assumed to be equal to the blade angles \( \beta(\xi) \) along the mean streamline), passage turning angles from axial to radial direction \( \alpha(\xi) \), and radii \( r(\xi) \). The above set of governing equations is valid for the whole range of turbomachinery components, from axial to mixed-flow to radial, and can be used to model viscous and inviscid compressor and turbine passages. These models are more realistic than the simple actuator disk model used in our previous study, and obviate the need to use compressor and turbine performance maps.

An eigenvalue/eigenvector analysis of the spatial operator of the above equations indicates that they are hyperbolic in nature. Any finite difference or finite volume scheme appropriate for hyperbolic equations can be used. For the results shown in this paper a new finite-difference scheme has been used (Vlachopoulos and Korakianitis, 1994). This new scheme is based on the theory of characteristics, and uses the right-eigenvector matrix as a basic operator. The governing equations are projected in a disturbance propagation space (characteristic space) and upwinded. The solution vector is then reconstructed by algebraic manipulations. This new scheme has been validated by comparisons with one and two-dimensional steady and unsteady-flow cases, and it provides faster and more-accurate solutions than other similar schemes (Korakianitis and Vlachopoulos, 1994b).

The compressor and the turbine were discretized in twenty elements each. The distributions of \( r(\xi) \) and \( A(\xi) \) were derived from parabolic distributions passing from the middle, inlet and outlet compressor and turbine dimensions shown in table 1. The \( \alpha(\xi) \) and \( \beta(\xi) \) distributions (shown in nondimensionalized form in Fig. 4), were derived from the typical distributions for centrifugal components published by Krain and Hoffmann (1989).

**Figure 2:** Rotating-passage geometry for centrifugal impellers

**Figure 3:** Control-volume model with 1D variations for the centrifugal impellers

**Figure 4:** Compressor and turbine impeller geometries shown in non-dimensional form

<table>
<thead>
<tr>
<th>Variable</th>
<th>Value</th>
<th>Variable</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>( V_c )</td>
<td>( 60 \times 10^{-6} \text{ m}^3 )</td>
<td>( V_t )</td>
<td>( 100 \times 10^{-6} \text{ m}^3 )</td>
</tr>
<tr>
<td>( r_{cp} )</td>
<td>0.055 m</td>
<td>( r_{tp} )</td>
<td>0.005 m</td>
</tr>
<tr>
<td>( r_{cm} )</td>
<td>0.037 m</td>
<td>( r_{tm} )</td>
<td>0.053 m</td>
</tr>
<tr>
<td>( A_{cm} )</td>
<td>( 3.60 \times 10^{-2} \text{ m}^2 )</td>
<td>( A_{cm} )</td>
<td>( 4.00 \times 10^{-2} \text{ m}^2 )</td>
</tr>
<tr>
<td>( A_{cm} )</td>
<td>( 3.05 \times 10^{-2} \text{ m}^2 )</td>
<td>( A_{tm} )</td>
<td>( 5.06 \times 10^{-2} \text{ m}^2 )</td>
</tr>
<tr>
<td>( A_{et} )</td>
<td>( 2.50 \times 10^{-2} \text{ m}^2 )</td>
<td>( A_{et} )</td>
<td>( 5.65 \times 10^{-2} \text{ m}^2 )</td>
</tr>
</tbody>
</table>

Table 1: Key dimensions of turbomachinery components
HEAT EXCHANGER AND RADIATOR MODELS

These components and their models are similar to those presented in the previous study (Korakianitis, Hochstein and Zou, 1993). However, their coupling with the turbomachinery and inertia models to create the overall system model is different, as explained in the following section. The steady-state performance of heat exchangers (receiver, regenerator and gas cooler) is evaluated using the design-point effectiveness of the unit and the method of the number of heat transfer units (Kays and London, 1984). For transient performance the radiator and the various counter-flow and cross-counter-flow heat exchangers are modeled using finite difference schemes based on the transient conservation of mass, momentum and energy for each type of heat exchanger. The components are modeled by slices along the flow direction, with elements in each slice modeling the hot and cold working fluids, and the wall, as appropriate.

The purpose of this paper is to couple the new turbomachinery models with the heat exchanger models, and to study fundamental aspects of the propagation of transients in typical closed regenerative cycles. More elaborate heat exchanger and radiator models would contribute to better quantitative results for a specific cycle configuration. Such models are not required for the basic understanding of propagation of transients sought in this study.

COUPLING OF THE MODELS

Given the geometry of the components, the state of the system (cycle condition) at any time t is prescribed by the combination of \( \{ T, p, m \} \) at all points along the flow path. In the heat-exchanger and radiator models one prescribes the flow conditions at the inlet, and the models return the flow conditions along the flow path in the component, and at the outlet. The turbomachinery models for subsonic flow require two boundary conditions at the inlet, and one boundary condition at the outlet of the component. In the turbine \( T_6, p_6 \) and \( \mathbf{m}_6 \) are used. In the compressor \( T_1, \mathbf{m}_1 \) and \( p_1 \) are used.

All calculations start from an initial guess for steady-flow conditions, or from previously-calculated steady-flow conditions. In order to obtain an initial steady-flow condition, the system is run to convergence (with no driving excitation). The models are called successively for each time step, starting from the turbine, along the direction of working-fluid flow. After each iteration around the closed cycle the conditions are updated, and the process is repeated until a steady-flow condition is achieved (the flow properties at all points do not change with time). Because the initial guess is fictitious (it does not satisfy the governing equations), all intermediate solutions until the steady-flow condition is reached are not realistic states of the cycle. The final steady-flow condition gives the correct distribution of flow properties along the flow path, and the correct total working-fluid mass in the cycle. In the sample transients shown in the following, these initial calculations in search of the initial steady-flow condition occur from \( t=0 \) to approximately \( t=15,000 \) seconds, and they are not shown in the time traces. When searching for the steady-flow condition, faster convergence is obtained by starting from an initial guess for the state of the system that is reasonably close to the steady-flow solution. Key parameters of the cycle at the steady-state operating point have the values shown in Table 2. Cycle transients can be driven in a variety of ways, such as: variations in salt temperature \( T_s \); variations in sink temperature \( T_{in} \); loss of coolant pumps; loss of some of the radiator panels; and others.

For constant-speed transients the inertia of the rotating unit is not needed in the models. For variable-speed operation the dynamic equation of the unit is used to evaluate the instantaneous acceleration:

\[
\frac{d}{dt} \left( \frac{J \Omega^2}{2} \right) = W_t - W_{el} = \frac{W_t - W_e - W_{fr}}{J \Omega} = \frac{\Omega}{J} (W_t - W_e - W_{fr} - W_{el})
\]

where for the compressor-alternator-turbine rotating unit \( J = 23.39 \times 10^{-6} \text{ kg} \cdot \text{m}^2 \). The instantaneous acceleration \( \Omega \) is integrated in time to give the rotational speed of the unit at the next time instant. The variable-speed model requires that one prescribes the time-variation of alternator work, which can be arbitrary. In the variable-speed sample cases shown below the alternator work was specified as invariant with time, and equal to the alternator work of the steady-flow condition at the beginning of the transient.

The working implementation of the new scheme used in this paper is explicit, first order accurate in time, and second order accurate in the spatial direction (\( \xi \)). As in most explicit hyperbolic algorithms the time and space differences must satisfy the CFL criterion, \( dt/dx < CFL \), where CFL is usually smaller than unity. The CFL numbers used in this paper were in the range 0.5 < CFL < 0.8. The programs are implemented in standard FORTRAN77. The sample runs presented below have been run on a 40 MHz DECstation 5000/240. Depending on the time step, run times range from 1.8 to 3.5 CPU seconds per physical simulation second.

The previous models (Korakianitis, Hochstein and Zou, 1993) required very small time-step limits and iterations at each time step to maintain the correct total working-fluid mass in the cycle. The present transient-system model is about six orders of magnitude faster than the previous transient one. Two factors contribute to this dramatic reduction in computational time. The first is that the present turbomachinery models are far more stable, and permit much larger time steps. The second is that the new model does not require an iteration for the working-fluid mass, because it maintains by default the correct mass throughout the time integrations. This is achieved automatically due to the upstream-downstream linking of flow properties in the turbomachinery models. The above was verified by monitoring the working-fluid mass in the cycle throughout the following transients.

<table>
<thead>
<tr>
<th>Variable</th>
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<th>Variable</th>
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<tbody>
<tr>
<td>( T_1 )</td>
<td>186.0 K</td>
<td>( T_3 )</td>
<td>823.2 K</td>
</tr>
<tr>
<td>( T_2 )</td>
<td>1042.0 K</td>
<td>( p_3 )</td>
<td>189.3 kPa</td>
</tr>
<tr>
<td>( T_4 )</td>
<td>287.5 K</td>
<td>( T_6 )</td>
<td>415.5 K</td>
</tr>
<tr>
<td>( T_5 )</td>
<td>185.7 kPa</td>
<td>( T_7 )</td>
<td>187.4 kPa</td>
</tr>
<tr>
<td>( T_6 )</td>
<td>363.3 K</td>
<td>( \rho_6 )</td>
<td>0.845 kg/m^3</td>
</tr>
<tr>
<td>( T_7 )</td>
<td>351.6 kPa</td>
<td>( \Omega )</td>
<td>3351 rad/s</td>
</tr>
<tr>
<td>( T_8 )</td>
<td>794.0 K</td>
<td></td>
<td></td>
</tr>
<tr>
<td>( T_9 )</td>
<td>346.2 kPa</td>
<td></td>
<td></td>
</tr>
<tr>
<td>( T_4 )</td>
<td>1007.7 K</td>
<td></td>
<td></td>
</tr>
<tr>
<td>( T_5 )</td>
<td>337.9 kPa</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>
DISCUSSION OF SAMPLE TRANSIENTS

In the following we show constant-speed/variable-work and variable-speed/prescribed-work transients. In the receiver the salt temperature \( T_a \) remains approximately constant during normal operation. However, in the following we are using sinusoidal excitations in \( T_a \) of approximately orbital period (90 minutes), and double-step excitations in \( T_a \) to study system transients. In the latter excitations \( T_a \) is increased by a certain amount at a given time instant, and later \( T_a \) is decreased to its original value, so that the final steady-flow condition is identical to the initial one. Driving excitations in sink temperature, or a combination of such excitations could also be used. Because of the large thermal inertia of the radiator, variations in \( T_a \) have small effects on the overall cycle, and they can not readily be used to show various fundamental aspects of the propagation of transients around the cycle. The driving excitations in \( T_a \) retain the qualitative characteristics of an excitation somewhere in the cycle for Space Station applications (the sinusoidal variation for orbital fluctuations, and the step excitation for abrupt changes, such as a sudden loss of a component). The excitations can be used to clearly show fundamental aspects of the generation and propagation of transients. These aspects are independent of the point of application of the driving excitation. Imposing the driving excitation to a different point in the cycle will result to similar physical behavior and qualitative responses. Quantitative differences will be observed in unsteady amplitudes, but the mechanisms of wave propagation and generation will be those explained below.

Fig. 5 illustrates the results of the first sample case. This is a constant-speed/variable-work system response to a 15% sinusoidal transient in \( T_a \) for six periods, starting at \( t=20,000 \) seconds from the steady-state condition. The responses are all periodic, after the initial three or so periods, at the driving period. After the initial disturbance in \( T_a \) there is an initial time lag from \( T_a \) to \( T_4 \) due to the thermal inertia of the receiver. Then there is a second smaller time lag from \( T_4 \) to \( T_a \) as the disturbance travels through the turbine. Similarly, there are consecutive time lags from component to component as the disturbances propagate in the flow path. Changes in \((T_a, p_4, m_4)\) are followed by changes in \((T_4, p_3, m_3)\) a short time later. This disturbance then propagates in the low-pressure side of the recuperator \((6-1)\), and at the same time they affect the high-pressure side of the recuperator, so that disturbances are initiated at \((T_3, p_3, m_3)\) and \((T_2, p_2, m_2)\). These become new, secondary sources of disturbance, starting at points 2 and 3, and affecting the cycle both upstream and downstream. Meanwhile the initial disturbance that had reached point 1, after a short delay passes through the compressor (point 2), and reaches the high-pressure side of the recuperator. This disturbance in \((T_1, p_1, m_1)\) becomes a tertiary source of disturbance that affects the properties at points 3, 5 and 6. Such disturbances are initiated and propagate through the components repeatedly throughout any system transient, resulting in smaller, higher-frequency disturbances affecting every system transient.

As indicated by the insert in the temperature response in Fig. 5, there is a time lag between the responses at compressor and turbine outlet. The turbine responds first, while the thermal disturbance travels through the rest of the components and reaches the compressor outlet approximately 100 seconds later. Expansion disturbances travel at the local fluid velocity minus the local acoustic velocity through the components. Compression disturbances travel faster, at the local fluid velocity plus the local acoustic velocity through the components. Due to the large thermal inertia of the heat exchangers, temperature disturbances are conducted across the hot and cold sides of heat exchangers much slower than disturbances in any property are convected along the working-fluid path.

Although all the responses in \((T, p, m)\) shown in Fig. 5 are periodic after the initial three or so periods, and the period is the same as the sinusoidal driving period in \( T_a \), they contain higher-frequency contents of much smaller amplitudes than the fundamental. The higher-frequency contents are generated by the primary, secondary, tertiary and so on disturbances in flow properties, generated and traveling as explained above.

The shaft work \( W_s \) shown in Fig. 5 is the turbine work \( W_t \) minus the compressor work \( W_c \). The periodic variations in \( W_s \) and \( W_t \) are almost out of phase, thus generating large periodic variations in \( W_s \). The exact relation between the phases in compressor and turbine work is a function of the size of the cycle components, and it depends on the characteristics of the driving excitation. (For example in the variable-speed sample cases shown below the compressor and turbine work are almost in phase).

Fig. 6 illustrates the results of the second sample case. This is a constant-speed/variable-work system response to a 15% double-step transient in \( T_a \), starting at \( t=0,000 \) seconds from the steady-state condition. The 90-minute interval between the two steps corresponds to 5,400 seconds. Approximately 4,000 seconds after the first step (at \( t=10,000 \) seconds) the system approaches a new steady-state. Approximately 4,000 seconds after the end of the second step (at \( t=15,400 \) seconds), the system approaches the initial steady state. The initial high-amplitude responses in flow properties immediately after the steps are followed by smaller-amplitude variations caused by the secondary and tertiary disturbance-propagation effects (explained in the first sample case). Since the amplitude of the driving disturbances in \( T_a \) in the first and second sample cases are the same, the corresponding amplitudes of responses in flow properties in the two cases are about the same. In the first sample case the time derivatives of the driving disturbance were smooth, but in the second sample case they are very large. This introduces higher-amplitude higher-frequency contents in the exciting disturbance and in the system response. This has two effects. First, it enlarges the response due to expansion and compression waves. Second, it enlarges the response due to the secondary and tertiary disturbance amplitudes. The overall effect of the higher-amplitude higher-frequency contents is shown in the work response in Fig. 6.

In all constant-speed/variable-work cases the response in \( T_4 \) has a higher amplitude than the response in \( T_3 \). This is because there are no heat exchangers between the excitation in \( T_a \) and the turbine. The heat exchangers damp the amplitude of the excitation before it reaches the compressor.

In the variable-speed cases the changes in shaft speed are additional sources of disturbances, upstream and downstream for the compressor and the turbine responses. This creates high-frequency contents in the responses of the variable-speed cases in addition to the primary, secondary and tertiary effects explained in the sinusoidal cases.

Fig. 7 illustrates the results of the third sample case. This is a variable-speed/constant-Alternator-work system response to a 0.5% sinusoidal transient in \( T_a \), for six periods starting at \( t=20,000 \) seconds from the steady-state condition. In this case the driving excitation has a small amplitude, the secondary and tertiary effects are much smaller, and the higher-frequency contents are not as evident as in the first two sample cases. The exception to this observation is at the beginning of the first excitation period, for two reasons. The dominant reason is that in the steady-flow condition the shaft speed has no time derivatives, while the beginning of the transient introduces a finite change in the time-derivatives in the shaft speed. This introduces initial large secondary and tertiary disturbance effects, as explained in

| Table 3: Legend for the lines used in transient Figs. 5, 6, 7, 8 and 9 |
|-----------------------------|----------------|
| Variable | Value |
| turbine | compressor | shaft |

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Figure 5: First sample case. System response with constant-speed/variable-work model to 15% sinusoidal variation in $T_\mu$. Table 3 identifies the different line types used in the figure.

Figure 6: Second sample case. System response with constant-speed/variable-work model to 15% double-step variation in $T_\mu$. Table 3 identifies the different line types used in the figure.

Figure 7: Third sample case. System response with variable-speed/constant-alternator-work model to 0.5% sinusoidal variation in $T_\mu$. Table 3 identifies the different line types used in the figure.

Figure 8: Fourth sample case. System response with variable-speed/constant-alternator-work model to 5% sinusoidal variation in $T_\mu$. Table 3 identifies the different line types used in the figure.
the above paragraph. A lesser reason is the effect of the same changes in time derivatives in flow properties at the beginning of the transient. In this case the response in shaft speed is almost sinusoidal, and the higher-frequency contents in shaft-speed are not visible. The almost-sinusoidal changes in shaft speed are such that they put the sinusoidal variations in compressor work almost in phase with the sinusoidal variations in turbine work, so that the resultant sinusoidal variation in shaft work is small.

Fig. 8 illustrates the results of the fourth sample case. This is a variable-speed/constant-alternator-work system response to a 5% sinusoidal transient in \( T_a \), for six periods starting at \( t=20,000 \) seconds from the steady-state condition. The amplitude of the driving excitation is an order of magnitude larger than that of the third case. The larger amplitude of the excitation makes the effects of shaft-speed changes, as well as the primary, secondary and tertiary effects of disturbance propagation much more pronounced than in the third case. The result is a significant high-frequency fluctuation in all flow properties, but the fundamental harmonic of the response is still sinusoidal and at the period of the driving excitation in \( T_a \). Unlike the third case, the variations in shaft speed show the high-frequency effects of disturbance propagation. Similarly to the third case, the fundamental-harmonic changes in shaft speed are such that they put the fundamental-harmonic variations in compressor work almost in phase with the fundamental-harmonic variations in turbine work. The resultant variation in shaft work contains high-frequency contents, but the fundamental-harmonic variation is still small.

Fig. 9 illustrates the results of the fifth sample case. This is a variable-speed/constant-alternator-work system response to a 5\% double-step transient in \( T_a \), starting at \( t=20,000 \) seconds from the steady-state condition. In contrast to the second case, the 90-minute orbital period was not sufficient for the system to reach a new steady state after each step. This delay in reaching a steady state is due to the changes in shaft speed. The time-interval between the two steps was increased to 180 minutes, or 10,800 seconds. Approximately 10,000 seconds after the first step (at \( t = 30,000 \) seconds) the system approaches a new steady state. The second step occurs at \( t = 30,800 \) seconds. Approximately 10,000 seconds after the end of the second step (at \( t = 40,800 \) seconds) the system approaches the initial steady state. The initial high-
amplitude responses in flow properties immediately after the steps are followed by decaying amplitude variations. These are caused by the corresponding variations in shaft speed required to achieve constant alternator work. Secondary and tertiary disturbance-propagation effects of smaller amplitude are evident throughout the transient. Similarly to the third and fourth cases, the changes in shaft speed are such that they put the variations in compressor work almost in phase with the variations in turbine work. The resultant variation in shaft work contains high-frequency contents, but the overall variation is small.

Fig. 10 shows a comparison of a variable-speed/constant alternator-work to a constant-speed/variable-work case. The former is identical to case four. The latter is a system response to a 5% sinusoidal transient in $T_a$, for six periods starting at $t=20,000$ seconds from the steady-state condition. These two cases are driven by the same excitation. The variations in compressor work for the constant-speed case are almost out of phase with the variations in compressor work for the variable-speed case, and the amplitudes are of the same order. The variations in turbine work for the constant-speed case are almost in phase with the variations in turbine work for the variable-speed case, but the amplitudes differ by a factor of five. As a result the fundamental amplitude of the variations in shaft work are much smaller in the variable-speed case than those in the constant-speed case, by an order of magnitude. The response of the variable-speed case has significantly higher-frequency contents than the constant-speed case. This is due to the additional disturbance effects due to the changes in shaft speed, as explained above.

CONCLUSIONS

New steady-state and transient models suitable for predicting the behavior of centrifugal (and mixed-flow and axial) turbomachinery components have been developed. These are coupled to steady-state and transient models suitable for predicting the performance of heat exchangers and radiators. The component models are based on the transient form of conservation of mass, energy and momentum within each component. These component models have been coupled to simulate a closed-cycle regenerative gas-turbine cycle, whose components are similar to those under consideration for the powering unit of the Space Station. Various system transients have been studied. For illustration, transients driven by step and sinusoidal variations in eutectic salt temperature (variations in energy input to the cycle), and under constant and varying shaft speed, have been included. In gas-turbine cycles with heat exchangers the long thermal-response times of the heat exchangers dominate the turbomachinery response.

The numerical models for the transient turbomachinery are inherently stable, permitting fast computations for large-amplitude disturbances for the individual components and for the overall cycle. They also permit a physically-correct coupling with the transient models for the heat exchangers and the radiator. This predicts correctly the upstream and downstream propagation of disturbances from the point where they were introduced. The coupling of the disturbances in the low-pressure and high-pressure streams of the recuperator is a source of secondary and tertiary disturbance generation, which introduces low-amplitude high-frequency content in the overall system response. The higher-frequency content is more pronounced in cases of higher derivatives in the time evolution of the excitation. Variations in shaft speed are an additional source of disturbances. These add higher-frequency content in the overall cycle response, and affect both the phases and the amplitudes of the flow properties at every point in the cycle. As a result the relative phase of the compressor and turbine work is affected by the size of the components in the cycle, and by the relationship between shaft speed and alternator work. Variations in the overall cycle work may be enhanced or damped, depending on these relative phases and amplitudes. In the constant-speed cases shown in this paper the compressor and turbine work were almost out of phase, giving relatively large variations in overall cycle work. In the variable-speed cases shown in this paper the compressor and turbine work were almost in phase, giving smaller variations in overall cycle work.

REFERENCES


