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Transient Analysis of Gas Turbine Power Plants, Using the Huntorf Compressed Air Storage Plant as an Example

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ABSTRACT

The design of modern gas turbines requires the predetermination of their dynamic behavior during transients of various kinds. This is especially true for air storage and closed cycle gas turbine plants. The present paper is an introduction to a computational method which permits an accurate simulation of any gas turbine system. Starting with the conservation equations of aero/thermodynamics, the modular computer program COTRAN was developed, which calculates the transient behavior of individual components as well as of entire gas turbine systems. For example, it contains modules for compressors, turbines, combustion chambers, pipes etc. To demonstrate the effectiveness of COTRAN the shut-down tests of the air storage gas turbine plant Huntorf were simulated and results compared with experimental data. The agreement was found to be very good.

INTRODUCTION

During the operation of a gas turbine power plant, load points are frequently encountered which are far away from the stationary design conditions. The reason for this to occur may be plant internal or external, such as start-ups, general load changes or rapid shutdowns. In any case, the gas turbine is subject to a transient event during which the entire thermodynamic and aerodynamic process and therefore also the mechanical conditions change. If it is possible to accurately predict these transients, then the results of such calculations can be considered already in the early stages of the development and corresponding precautions can be taken. This possibility can be created by establishing a computational method which considers the transient process within the individual components of the gas turbine system.

The necessity to develop such a computational method had been recognized for a long time. Within the German-Swiss HHT-Project (high temperature gas-cooled reactor with helium turbine of large output) EIR (Swiss Federal Institute of Reactor Research) and BBC cooperated in writing a computer progam which simulates nuclear gas turbine systems [1]. Later, the computer program COTRAN was developed by BBC to analyze the transients within conventional gas turbogroups.

The present paper will describe the physical basis of this computer program. As an example the results of a simulation of the compressed air energy storage plant "Huntorf" (FRG), which was built by BBC, will be shown and compared with measurements.

NOMENCLATURE

c_f	friction coefficient
c_p	specific heat capacity at constant pressure
c_v	specific heat capacity at constant volume
D_h	hydraulic diameter
$D = e_i e_j D_{ij}$	deformation tensor
h	specific static enthalpy
H	specific total enthalpy
I	moment of inertia
$\hat{K} = \frac{1}{2} V^2$	kinetic energy per unit mass
M	mass
$\dot{m}$	mass flow
p	static pressure
P	total pressure, power
$\dot{\vec{q}}$	energy flux vector
$\dot{Q}$	rate of energy flow
R	gas constant
S	cross-sectional area
t	time
T	absolute static temperature

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T^*	absolute total temperature
$\mathbf{T} = e_i e_j T_{ij}$	shear stress tensor
v	volume
$\vec{V} = e_i V_i$	velocity vector
x_i	rectangular coordinates
$\kappa = \frac{c_p}{c_v}$	ratio of specific heat capacities
ρ	density
$\Pi = e_i e_j \Pi_{ij}$	stress tensor
ω	angular velocity

STRUCTURE OF COTRAN

In COTRAN each important component of a gas turbine power plant is mathematically modelled by a set of partial differential and algebraic equations which describe its transient behavior. The components are treated as modules which are combined into a complete system to represent any gas turbine power plant. The program is therefore independent of a specific power plant configuration and permits the simulation of single components as well as complex systems.

THEORETICAL DESCRIPTION OF TRANSIENT PROCESSES

The transient processes which occur within a component are in general described by the conservation equations of aero-/thermodynamics [2, 3, 4, 5]. They are:

Continuity equation:

$$\frac{\partial \rho}{\partial t} = -\nabla.(\rho\vec{V}) \tag{1}$$

Equation of motion:

$$\frac{\partial(\rho\vec{V})}{\partial t} = -\nabla.(\rho\vec{V}\vec{V}) - \nabla p + \nabla.\mathbf{T} \tag{2}$$

Mechanical energy equation:

$$\frac{D}{Dt}\left(\frac{V^2}{2}\right) = -\nabla.(p\vec{V}) + p\nabla.\vec{V} + \nabla.(\vec{V}.\mathbf{T}) - \mathbf{T}:\mathbf{D} \tag{3}$$

Thermal energy equation:

$$\frac{\rho}{\kappa}\frac{Dh}{Dt} = -\nabla.\vec{q} - p\nabla.\vec{V} + \mathbf{T}:\mathbf{D} \tag{4}$$

Combining the two energy equations (3) and (4) yields the total energy equation. For the total enthalpy it reads:

$$\frac{\partial H}{\partial t} = -\vec{V}.\nabla H - (\kappa-1)\left[\frac{1}{\rho}\nabla.(\rho\vec{V})(H+\hat{K}) + \frac{\vec{V}}{\rho}.\frac{\partial(\rho\vec{V})}{\partial t}\right] + \kappa\left[-\frac{\nabla.\vec{q}}{\rho} + \frac{1}{\rho}\nabla.(\vec{V}.\mathbf{T})\right] \tag{5}$$

In an analogous way

$$\frac{\partial P}{\partial t} = -(\kappa-1)[\nabla.(\rho\vec{V}H) + \nabla.\vec{q} - \nabla.(\vec{V}.\mathbf{T})] - (\kappa-2)\frac{\partial\left(\frac{\rho V^2}{2}\right)}{\partial t} \tag{6}$$

is obtained with

$H = h + \hat{K}$, $P = p + \rho\hat{K}$ and $\hat{K} = \frac{1}{2}V^2$.

If there is no heat exchanged with the surroundings, then $\nabla \cdot \vec{q} = 0$. For viscous flow the energy of the shear stresses $\nabla \cdot (\vec{V} \cdot \mathbf{T})$ is always different from zero. It is dissipated as heat and therefore causes total pressure losses.

Cartesian coordinates are chosen for the corresponding decomposition into components. In tensor notation the conservation equations become:

$$\frac{\partial\rho}{\partial t} = -\frac{\partial}{\partial x_i}(\rho V_i) \tag{7}$$

$$\frac{\partial(\rho V_i)}{\partial t} = \frac{\partial}{\partial x_j}(\rho V_i V_j) - \frac{\partial p}{\partial x_i} + \frac{\partial T_{ij}}{\partial x_i} \tag{8}$$

$$\frac{\partial H}{\partial t} = -\kappa V_i\frac{\partial H}{\partial x_i} - (\kappa-1)\left[\frac{1}{\rho}\frac{\partial}{\partial x_i}(\rho V_i)(H+\hat{K}) + \frac{V_i\partial(\rho V_i)}{\rho\partial t}\right] + \frac{\kappa}{\rho}\left[-\frac{\partial q_i}{\partial x_i} + \frac{\partial}{\partial x_i}(V_j T_{ij})\right] \tag{9}$$

PREPARATION OF THE GOVERNING EQUATIONS FOR PRACTICAL APPLICATION

The governing equations derived in the previous section constitute the general basis for the analysis of transient processes. However, due to limited capacity of computer core storage a three-dimensional solution is hardly possible. Even a two-dimensional treatment requires considerable computation time and cost. For systems of engineering interest a detailed spatial analysis of the transient events is in any case not of great practical significance. A sufficiently accurate solution of the governing equations can be obtained by means of a one-dimensional approximation. This simplification and introduction of the total temperature into the energy equation lead to:

$$\frac{\partial\rho}{\partial t} = -\frac{\partial}{\partial x_1}\left(\frac{\dot{m}}{S}\right) \tag{10}$$

$$\frac{\partial\dot{m}}{\partial t} = -\frac{\partial}{\partial x_1}(\dot{m}V_1) - \frac{\partial}{\partial x_1}(pS) + (\dot{m}V_1 + pS)\frac{1}{S}\frac{\partial S}{\partial x_1} - c\frac{\dot{m}|V_1|}{f2D_h} \tag{11}$$

$$\frac{\partial T^*}{\partial t} = -\frac{\kappa}{c_p}\frac{\dot{m}}{\rho S}\frac{\partial(c_p T^*)}{\partial x_1} - \frac{(\kappa-1)}{c_p}\left\{\frac{1}{\rho}\frac{\partial}{\partial x_1}\left(\frac{\dot{m}}{S}\right)\left[c_p T^* + \frac{1}{2}\left(\frac{\dot{m}}{\rho S}\right)^2\right] + \frac{\dot{m}}{\rho^2 S^2}\frac{\partial\dot{m}}{\partial t}\right\} - \frac{\kappa}{\rho c_p}\Delta\dot{Q} \tag{12}$$

With these differential equations and the known thermodynamic properties, as well as the equation of state of the fluid medium, the transient behavior of a component is completely determined.

COMPONENTS OF A GAS TURBINE POWER PLANT

A gas turbine plant consists of many components which can be divided into the following categories:

Category 1 encompasses those components where no heat or mechanical energy exchange occurs with the surroundings. Examples are connecting ducts, inlet and exit elements. They serve to transport mass. Their dynamic behavior is described by equation (11) in case of continuous transient processes. During discontinuous transients, such as shocks, all three conservation equations are used.

Category 2 comprises those components within which heat exchange or heat production processes take place such as recuperators and combustion chambers. Their dynamic behavior on the hot and the cold side is represented by the system of differential equations (10) - (12). The heat conduction equation serves to couple the two sets of equations.

Category 3 consists of those components where heat and mechanical energy exchange occurs, as for instance in turbines and compressors. Besides the production and consumption of mechanical energy there is also heat exchange between the fluid medium and the material such as blading. If this heat exchange is neglected, then the operating behavior can be determined either from global or from stage characteristics. In this case, only algebraic equations have to be dealt with, which increases the computation speed. If, however, the heat exchange between the blading material and the fluid medium should be taken into account, then the differential equations (10) - (12) must be used. $\Delta\dot{Q}$ contains now not only the heat exchanged but also the mechanical energy consumption or production.

Category 4 comprises all control devices such as controller, bypass valves etc. whose characteristics must be known.

Category 5 is the electrical machine (generator/motor) which together with the turbine and the compressor determines the rotational speed behavior of the entire turbomachinery train according to

$$\frac{d\omega}{dt} = \frac{1}{I\omega}\left(P_T - P_C - P_F - P_G\right) . \tag{13}$$

P_T, P_C, P_F and P_G stand for the power of the turbine, the compressor, bearing friction and the generator.

COUPLING OF COMPONENTS

Two successive components (one outlet c. and one inlet c.) are coupled by a so-called plenum whose volume is made up of half of the volume of each of the two components.

The temperature and pressure transients taking place in the plenum are obtained from equations (5) and (6) neglecting, however, the contribution of the kinetic energy and the time change in mass flow relative to the other terms. This yields the following simplified relationships:

$$\frac{\partial H}{\partial t} = -\frac{\kappa}{\rho}\nabla.(\rho\vec{V}H) + \frac{H}{\rho}\nabla.(\rho\vec{V}) \tag{14}$$

$$\frac{\partial P}{\partial t} = -(\kappa-1)\nabla.(\rho\vec{V}H) \tag{15}$$

Coupling of m outlet (index O) components and n inlet (index I) components and assuming one-dimensional flow results in the two equations:

$$\frac{\partial T^*}{\partial t} = \frac{1}{\rho v}\left[\sum_{i=1}^{n} \dot{m}_{I_i}\left(\frac{c_{pI_i}}{c_p} T^*_{I_i} - T^*\right) - (\kappa-1)\sum_{j=1}^{m} \dot{m}_{O_j} T^*\right] \tag{16}$$

$$\frac{\partial P}{\partial t} = \frac{\kappa R}{v}\left[\sum_{i=1}^{n} \dot{m}_{I_i}\frac{c_{pI_i}}{c_p} T^*_{I_i} - \sum_{j=1}^{m} \dot{m}_{O_j} T^*\right] \tag{17}$$

The non-indexed variables describe the plenum.

NUMERICAL TREATMENT

The mathematical modelling of a gas turbine power plant consisting of any number of the components mentioned above leads to a system of partial differential equations. This is reduced to a system of ordinary differential equations in case of the one-dimensional treatment. Since this system is stiff, the numerical integration method affects both the numerical stability and the computation speed. A one-step implicit method based on trapezoidal rule was found to be very effective [6].

The corresponding computer code COTRAN has been written in FORTRAN and has a modular structure. The computing time depends strongly on the type of transient to be analyzed. For rapid transients the computing time to real time ratio may be as high as 10, while for slow transients this ratio may be much below 1 (IBM 370).

APPLICATION, EXAMPLES

With the aid of the computational method described above it is possible to simulate any gas turbine power plant and therefore predict its transient behavior.

A simulation is especially appropriate when a transient event could result in unallowable temperature and pressure peaks. They could lead to increased thermal and mechanical loads and consequently to a reduction in useful life of individual plant elements. If done early in the development it is relatively easy to take the results of such a simulation into account in the design of the respective elements. A simulation is also necessary when the gas turbine power plant is subject to frequent load changes. It is then possible to meet potential risks by specific countermeasures.

In the course of such simulations the behavior of the compressor is obviously also analyzed. If the calculations indicated that the surge limit might be exceeded, steps could be taken in time for effective surge prevention.

Another example is the examination of the interaction between the gas turbine and the control system. In this case the simulation offers a simple and economical test of the entire plant.

In all of the above examples the simulation also yields the response of the rotor speed. As will become apparent below, this is especially significant for compressed air storage gas turbines in case of a rapid shut-down during the generation mode. The reason is that, unlike in conventional open cycle gas turbogroups, the rotor is not effectively decelerated by the compressor train. An additional reason is the greater mass of high pressure gas in the combustion chambers of air storage gas turbines.

As indicated by the above paragraph, the transient analysis of an air storage gas turbine system is technically important and illustrative of the capability of COTRAN. Here it will be applied to the simulation of a generator-loss-of-load followed by a rapid shut-down of the BBC-built Huntorf air storage plant.

DESCRIPTION OF THE AIR STORAGE GAS TURBINE SYSTEM, PROCESS SIMULATION

An air storage gas turbine system consists essentially of a combustion turbine, a generator/motor, a compressor train, two clutches - one between the turbine and the generator/motor, the other between the generator/motor and the compressor train - and an underground air storage facility; see Fig. 1.

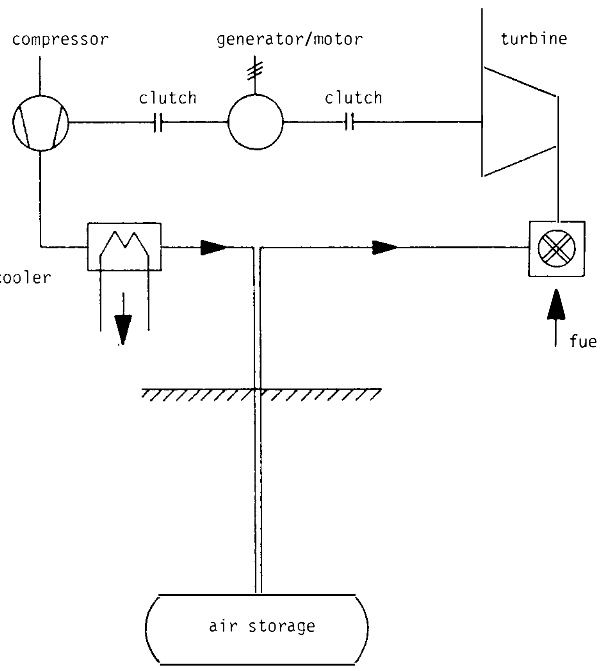

Fig. 1 Schematic diagram of an air storage gas turbine system

There are two modes of operation: the charging mode during periods of low electric energy demand and the generating mode during periods of high electric energy demand. In the charging mode the combustion turbine is decoupled and shut down and the electric machine, operating as a motor, drives the compressor train to pump air into the storage facility. Later, during the generating mode, the compressor train is decoupled and shut down and the electric machine, working as a generator driven by the turbine, delivers electric energy into the grid. The combustion air comes from the storage. Since in most cases the air storage pressure level would be relatively high several stages of compression (with intercooling) and combustion (with reheat) would be the rule. The two main advantages of air storage gas turbine plants are the economic transfer of electric energy from periods of low demand to periods of high demand and the very short start-up time (spinning reserve capability).

The Huntorf station, which will be simulated here, is the first plant of its kind [7]. It has been in commercial operation since December 1978. The turbomachine and the air storage facility have been sized to generate 290 MW for two hours. A complete set of the main design parameters is presented in Table 1.

Table 1. Main design parameters of the Huntorf power station

TECHNICAL DETAILS FOR GAS TURBINE

Rated output	290 MW
Speed	3000 r/min
Air throughput	417 kg/s
Inlet conditions to high pressure turbine	46 bar, 550 C
Inlet conditions to low pressure turbine	11 bar, 825 C
Exhaust temperature leaving low pressure turbine	400 C
Fuel	natural gas

TECHNICAL DETAILS FOR COMPRESSOR

Drive power for the two compressors	60 MW
Design of low pressure compressor	axial
Speed	3000 r/min
Intake conditions	10 C , 1.013 bar
Air throughput	108 kg/s
Design of high pressure compressor	radial
Speed	7626 r/min
State following compressor	50 to 70 bar, 50 C
Number of intercoolers	3
Number of after-coolers	1

The gas turbine/generator-system, Fig. 2, consists mainly of the high pressure turbine HPT, the low pressure turbine LPT and their respective combustion chambers HPC and LPC, a cooling air preheater P and the generator G. During turbine operation cold air from the air storage facility passes through an inlet/quick closing Valve V and enters a plenum where it is divided up into a combustion and a cooling air flow. By adding fuel in HPC the combustion air is heated up to the combustion chamber exit temperature. In front of the high pressure turbine, leakage flow and a portion of the cooling air flow are mixed in so that the gas temperature at the high pressure turbine inlet is below the combustion chamber exit temperature. After the expansion in the high pressure turbine, fuel is added in LPC, again raising the gas temperature. As in front of the high pressure turbine, cooling air and sealing air are mixed in. After expansion in the low pressure turbine the gas gives off some of its heat in the cooling air preheater and leaves then the gas turbine system. A section through the gas turbine is shown in Fig. 3.

shut-down, the blow-off valve BV is opened to limit the maximum rotor speed. For the sake of a clearer schematic presentation the cooling air preheater has been separated into its air and gas side marked by HPP and LPP, respectively. In addition, pipes P 1 - P 6 indicate which plena the individual mass flows are taken from.

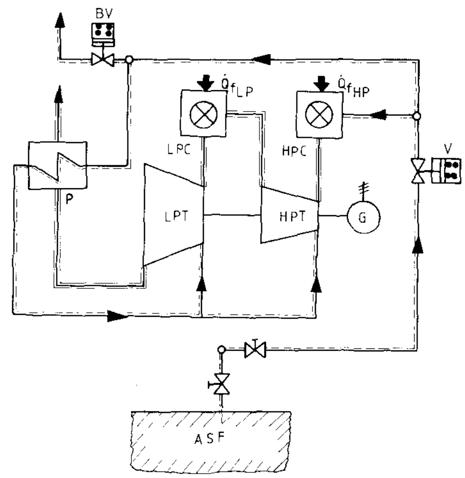

Fig. 2 Scheme of gas turbine system

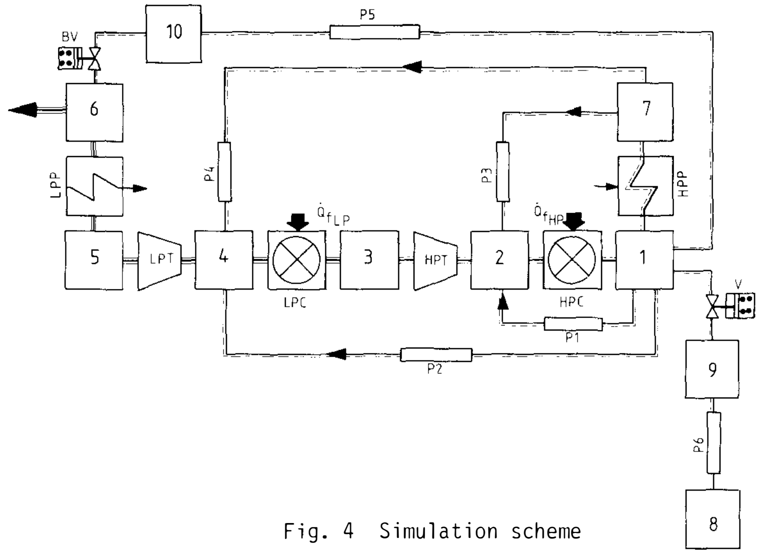

Fig. 4 Simulation scheme

ASF	Air storage facility	$\dot{Q}_{fLP}$	Fuel heat of LPC
G	Generator	P	Preheater
HPC	HP combustion chamber	HPP	Preheater air side
HPT	HP turbine	LPP	Preheater gas side
LPC	LP combustion chamber	V	Inlet shut-down-valve
LPT	LP turbine	BV	Blow-off valve
$\dot{Q}_{fHP}$	Fuel heat of HPC		

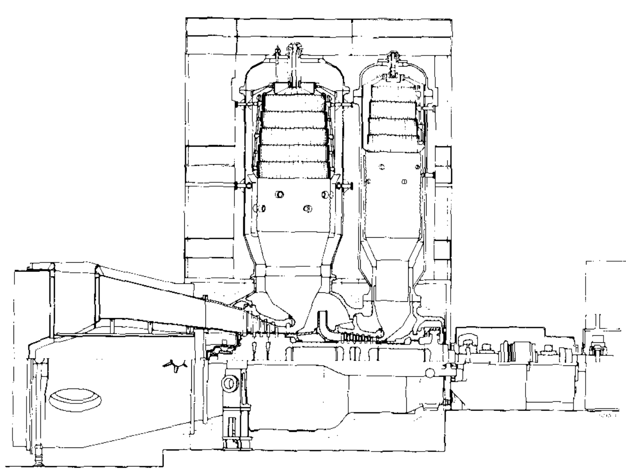

Fig. 3 Huntorf air storage gas turbine

The simulation scheme, Fig. 4, shows the interconnection of the various components. Plenum 8 represents the air storage facility. It is connected via pipe P 6 with the inlet/quick closing valve V. During stationary operation the blow-off valve BV remains closed. In case of an incident that leads to a rapid

DYNAMIC BEHAVIOR OF INDIVIDUAL COMPONENTS, COMPUTATIONAL RESULTS

Starting from stationary operation a loss-of-load incident followed by a rapid shut-down will be simulated below.

Pressure and Temperature Transients in the Plena

Following the loss of load incident the rotor is strongly accelerated due to the full turbine power acting on it. Since a rapid shut-down shall be simulated, closing of the fuel and air valves is initiated only after the speed corresponding to the hydraulic emergency overspeed trip has been reached. Simultaneously the blow-off valve is opened. Part of the energy that is then still contained in the gas masses enclosed in components upstream of the turbine is converted into mechanical energy. During this process pressure and temperature in the system in general drop steadily. As can be seen from Fig. 5, the pressure drop in the high pressure part (curves 1, 2, 7) is initially steeper than in the low pressure part. This means that the head of the high pressure turbine is reduced more quickly than that of the low pressure turbine. Immediately after the blow-off valve has been opened an abrupt pressure drop is observed in plenum 10 which is connected with plenum 1 via pipe P 5 (Fig. 6, curve 10). Thereafter a dynamic pressure equalization between the two plena takes place.

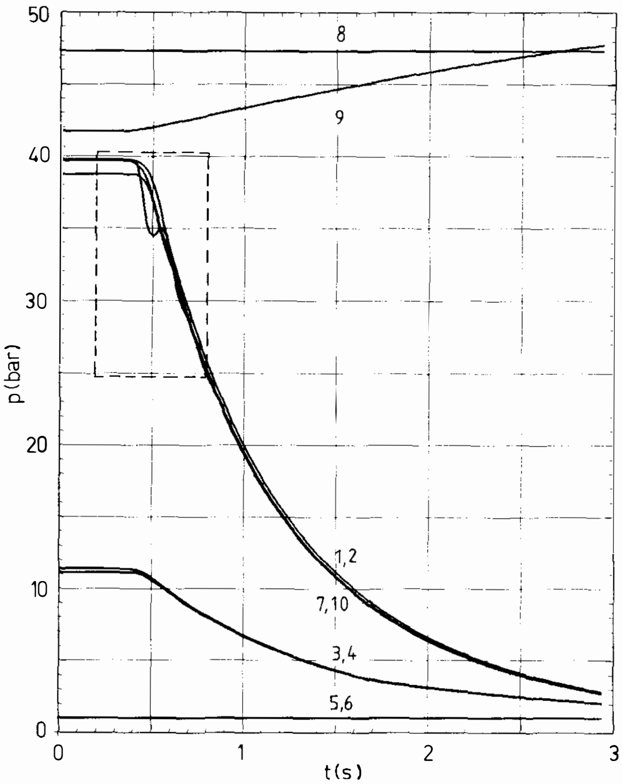

Fig. 5 Plena pressure as functions of time

For greater clarity Fig. 6 is an enlargement of the dashed section in Fig. 5. The plot of the pressure transients in plena 1, 2, 7 and 10 makes the above mentioned situation quite clear. During experiments in the plant [8] the pressure transients in plena 1 and 10 were measured. These results are shown in Fig. 6 as full squares and circles. Calculation and experiment agree well within a wide range.

After the air inlet valve has been closed the pressure in plenum 9 increases steadily and approaches the cavern pressure (Fig. 5, curves 8 and 9). The corresponding temperatures in the various plena follow a similar trend. For this reason they are not shown.

<u>Mass Flow Transients in the Various Components</u>

The behavior of the combustion chamber mass flows is shown in Fig. 7, indices 1 and 2 referring to high and low pressure, respectively. Due to the rapid pressure drop the high pressure combustion chamber mass flow also decreases promptly. The low pressure combustion chamber mass flow, on the other hand, increases at first. This difference in behavior is caused by continuity conditions. The oscillations of curve 1 have to do with the quick closing of the air valve.

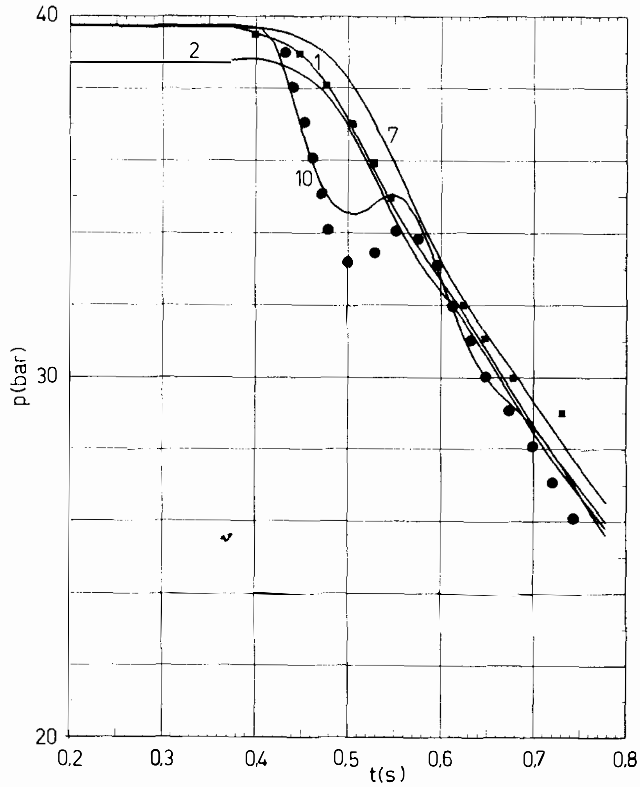

Fig. 6 Comparison of measured (●, ■) and calculated (—) pressure transients in plena 1 and 10

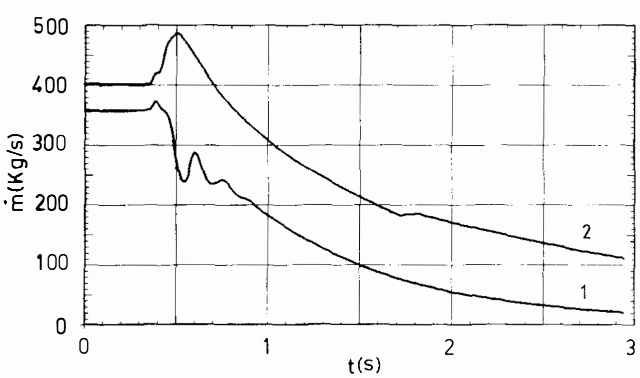

Fig. 7 Mass flows of first and second combustor as function of time

The mass flows in the turbines behave very similarly (Fig. 8).

Fig. 9 depicts the mass flows in the various pipes. The course of the blow-off mass flow, which is represented by curve 5, reflects the sudden opening of the valve. There is a very steep positive ramp followed by a more gradual decrease in mass flow.

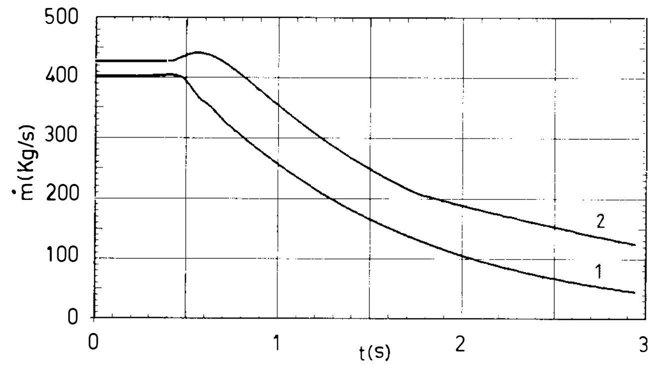

Fig. 8 Mass flows of first and second turbine as functions of time

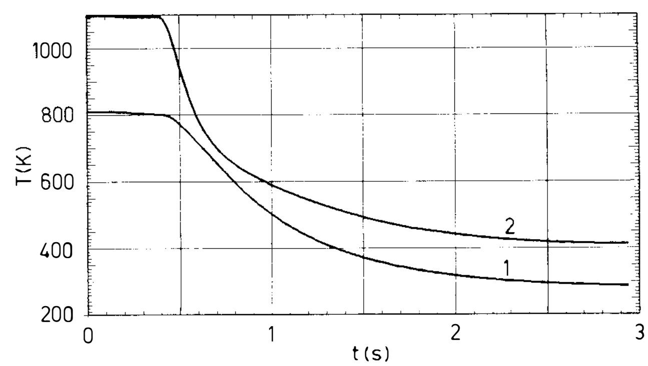

Fig. 10 Inlet Temperatures of first and second turbine as functions of time

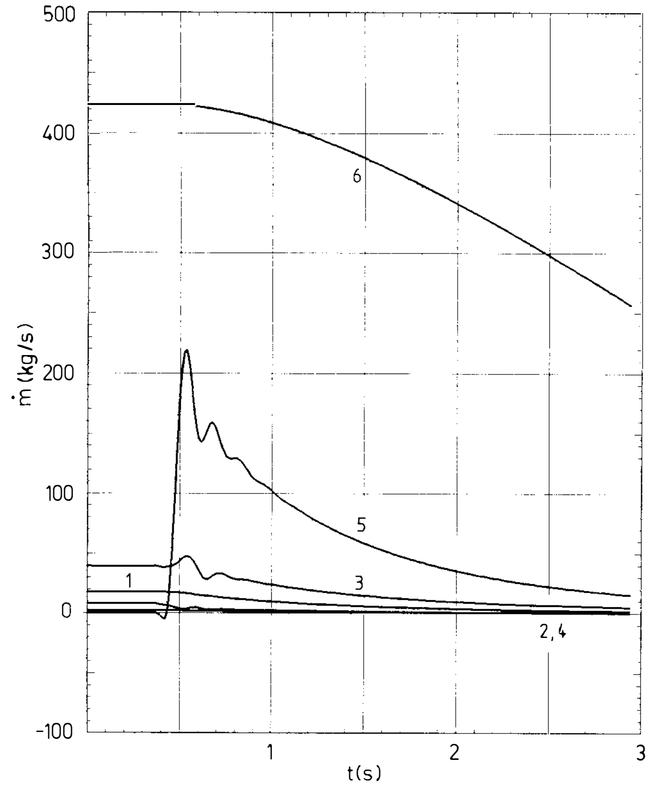

Fig. 9 Mass flows of pipes as functions of time

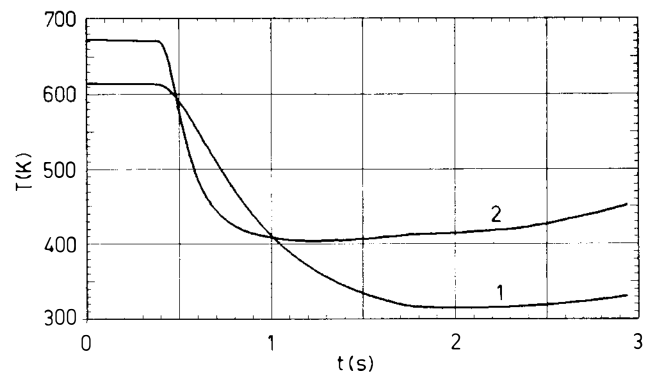

Fig. 11 Outlet temperatures of first and second turbine as functions of time

Dynamic Behavior of the Turbines

The dynamic behavior of both turbines is essentially determined by the temperature, pressure and mass flow transients. Starting with the turbine inlet temperatures (Fig. 10), the pressure ratios and the efficiencies, the corresponding turbine outlet temperatures are determined (Fig. 11). Together with the already known turbine mass flows, these yield the total output of the two turbines (Fig. 12).

Behavior of the Rotor Speed during Loss of Load and Blow-Off

The dynamic behavior of the rotor speed is in general determined by the turbine power acting on the rotor. Before the closing phase of the air and fuel valves the full turbine power is transferred on the rotor. During the closing phase, which in this simulation is initiated by the triggering of the hydraulic emergency system, the energy input is continuously decreasing and after the valves have been shut the input of energy from the outside is zero. But at this time there is still some gas stored in the system which expands in the turbines. The rotor speed had begun to increase when the generator load was lost and continues to do so until the instantaneous turbine power just balances the friction and ventilation power. After that the rotor speed decreases (Fig. 12 and 13). The maximum rotor speed must not exceed the overspeed test limit. Blowing off air helps to reduce the maximum rotor speed. The immediate effect is twofold: a reduction in turbine mass flow and a more rapid decrease in pressure in the various plena.

Rotor Speed Behavior, Comparison of Computed and Measured Data

The test data have been taken from [8] and are shown as squares in Fig. 13. The comparison with the computed curve shows very good agreement.

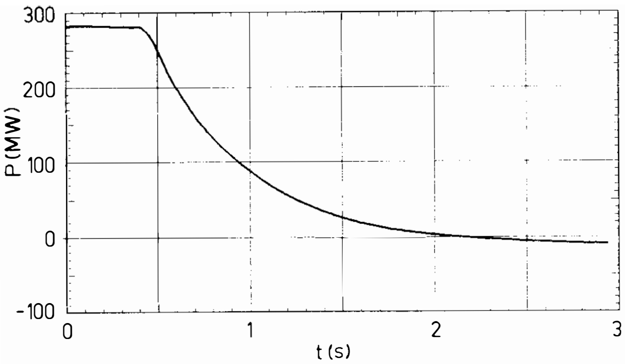

Fig. 12 Net power as function of time

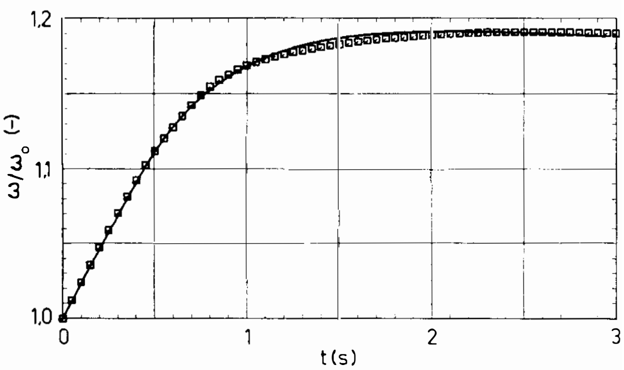

Fig. 13 Relative shaft speed as function of time measured (□) and calculated, ω_o = design angular velocity

CONCLUSION

A brief description of a computational method for the transient analysis of gas turbine power plants was presented. Such analyses are an important element in the design process since they help to ensure safe and realiable plant operation. To demonstrate the effectiveness of this method, the shut-down tests of the Huntorf compressed air storage plant were simulated and the results compared with experimental data. The agreement was found to be very good.

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