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Prediction of Thermoacoustic Instabilities with Focus on the Dynamic Flame Behavior for the 3A-Series Gas Turbine of Siemens KWU

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ABSTRACT

Self-induced combustion driven oscillations are a crucial challenge in the design of advanced gas turbine combustors. Lean premixed combustion, typically used in modern gas turbines, has a pronounced tendency to produce these instabilities.

Thus, the prediction of these thermoacoustic instabilities in the design phase of an engine becomes more and more important. A method based on linear acoustic four-pole elements to predict the instabilities of the ring combustor of the Siemens 3A-series gas turbines will be presented in this paper. The complex network includes the entire system starting from both compressor outlet and fuel supply system and ending at the turbine inlet.

Most of the transfer elements can be described by analytical data. Nevertheless, the most important elements, "flame" and "combustion chamber", have to be investigated more in detail due to their complex 3D acoustics.

For the turbulent, premixed and swirled flame, a numerical simulation of the transient behavior after a sudden jump in mass flow at the inlet (step-function approach) is used to obtain the flame frequency response for axial direction as well as circumferential direction. This method has been verified for numerous different flame types *(Krüger et al. (1998), Bohn et al. (1997), Bohn et al. (1996))*. The four-pole element of the annular combustor is derived by an eigenfrequency analysis of the chamber, including a numerical predicted temperature and flow distribution.

The results show the principle possibilities of the instability analysis described. The frequencies predicted correspond well with experience from engine test fields. The importance of several elements for self- induced combustion driven oscillations is pointed out clearly.

NOMENCLATURE

a	speed of sound
F	frequency response
f	frequency
h	unit function or step function response
$\dot{m}$	mass flow rate
$\Delta \dot{m}$	jump in mass flow
p	acoustic pressure
r	reflection coefficient
t	time
T	temperature
T	transfer matrix
u	acoustic particle velocity
z	acoustical impedance (= p / u)
ξ	mass fraction
φ	phase angle
ω	angular frequency

SUBSCRIPTS

ax	axial direction
circ	circumferential direction
Fl	flame
inl	inlet
nozzle	value at burner nozzle
0	steady-state quantities

SUPERSCRIPTS

+	wave travelling in positive direction
-	wave travelling in negative direction

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INTRODUCTION

Environmental compatibility requires low emission burners for gas turbine power plants. In the past, significant progress has been made developing low NO_x and CO burners by introducing lean premixed techniques in combination with annular combustion chambers. Unfortunately, these burners often have a more pronounced tendency to produce combustion driven oscillations than conventional burner designs. The oscillations may be excited to such an extent that strong pulsations may occur; this is associated with a risk of engine failure. Thus, the prediction of these thermoacoustic instabilities in the design phase of an engine becomes more and more important.

Self-induced oscillations are a system issue. They are not determined by the characteristics of only one component of a combustor system, but by the (complex) interaction of all relevant components of the system. So these self-induced oscillations can not be predicted without considering the entire system, including the interaction of all relevant components.

Once any disturbance brought into the system, the oscillation can be maintained and/or excited by an unsteady energy release. For the stability analysis of the system, it does not matter which kind of disturbance this might be (e.g. a jump in mass flow, a change in equivalence ratio ...). Of course, this is only true as long as the disturbance does not effect the dynamical behavior of a component. In order to predict thermoacoustic oscillations it is necessary to analyse whether the complex interaction of the components makes an excitation of the starting oscillation possible or whether the starting oscillation is damped.

Some important contributions to the prediction of combustion-driven oscillations in gas turbines have been made recently, for example *Hsiao et al. (1998), Polifke et al. (1997), Hubbard and Dowling (1998), Keller (1995) or Fleifil et al. (1996).* The common approach of all authors is to represent the entire system by acoustical equations or acoustical transfer elements. One of the most successful approaches is the stability investigation using an acoustical network. Complex systems, as typically encountered in gas turbines, can be described by splitting them off into simple geometrical units of known acoustical behavior. In this approach four-pole elements are used to represent the acoustics. The significance of this approach is that all four-poles are predicted by analytical or numerical (CFD and eigenfrequency analysis) methods. This systematic acoustic modelling of a heavy-duty gas turbine will lead to a deeper understanding of combustion dynamics.

ACOUSTICAL MODEL

TRANSFER MATRICES

The acoustic waves usually encountered in most components of gas turbine combustion chambers are predominantly longitudinal and one-dimensional due to the cross sections of the relevant components being small compared to the wavelength of these low frequency oscillations. Therefore, sound propagation is fully determined by the acoustic pressure (p) and the acoustic particle velocity (u). The time dependence is given by $e^{i\omega t}$, and the well developed four-pole theory of linear one-dimensional wave propagation can be used to describe the acoustics of a combustion system upstream and downstream of the flame. The entire system of interest is divided into simple geometrical units of known acoustical behavior and 2 x 2 - transfer matrices are provided which describe the change of the acoustic quantities p, u within this unit. As a result, the change of p and u over the whole area considered is given by multiplication of the transfer matrices (Figure 1).

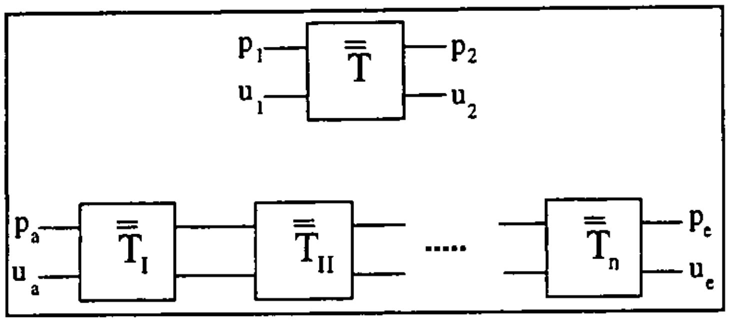

FIG. 1 : Transfer Matrices

Six standard four-poles can be used to build up the entire system for a complex gas turbine:

1a). duct of constant cross section
1b). conical duct (divergent or convergent)

2a). annular duct of constant cross section
2b). divergent or convergent annular duct

3. curved duct (45° or 90°)

4. area discontinuity

5. branch

6a). boundary condition (reflection coefficient)
6b). boundary condition (impedance)

Analytical formulations from literature exist for these elements, for example: *Bohn and Deuker (1993), Meyer & Neumann (1967), Munjal, M.L. (1987).*

The advantage of the network concept is the possibility of incorporating not only the analytical elements described but also additional information derived from either experimental or numerical sources. This is essential for an advanced gas turbine annular combustion system because of the 3D-acoustical field in the ring combustor itself and due to the complex dynamical behavior of the turbulent flame. Therefore, the transfer elements of the two elements "combustion chamber" and "flame" will be discussed separately in detail below.

STABILITY CRITERION

The usual way to determine the resonant frequencies of the system is to solve the eigenvalue problem that arises when boundary conditions are given at all ends of the system. The resulting (complex) eigenvalues are the resonant frequencies ω_{res} of the system, $\omega_{res} = \omega'_{res} + i\,\omega''_{res}$. For $\omega''_{res} > 0$ the oscillation is damped while $\omega''_{res} < 0$ gives an amplified oscillation. (This is easily verified by inserting a complex ω into the expression for the time dependence $e^{i\omega t}$).

If the transfer matrices are calculated from experimental data no complex frequencies can be introduced. The method described above can only be used if analytic solutions for the transfer matrices are given. Thus, a new formulation of a stability criterion developed by *Bohn and Deuker (1993)* and *Faber (1991)* is used that does not make use of complex frequencies. As a result, experimental or numerical data for the transfer matrices can be easily incorporated.

Consider a single pipe including a flame at some specific position (Figure 2): A wave u_2^+ starting downstream of the flame travels to the end of the pipe, is reflected and comes back to the flame (u_2^-), travels through the flame (u_1^-) to the other end, is reflected again, comes back to the flame (u_1^+) and finally passes the flame to come to the point of its beginning. If the phase angle after this cycle is zero and the amplitude has increased, amplification of that wave takes place. The amplitude will grow continuously in time limited only by non-linear effects and strong oscillations will be the result. This formulation of a stability criterion is known as the Nyquist-criterion. The mathematical formulation is:

$$F_0 = |F_0| \cdot e^{i\varphi} = F_{II} \cdot F_{fl}^- \cdot F_I \cdot F_{fl}^+ \begin{cases} |F_0| < 1 \quad \text{for } \varphi = 0 \rightarrow \text{damped oscillation} \\ |F_0| > 1 \quad \text{for } \varphi = 0 \rightarrow \text{amplified oscillation} \end{cases} \tag{1}$$

with the partial frequency responses given by:

$$F_I = \frac{u_1^+}{u_1^-} = \frac{(\rho_0 a_0)_I + z_1}{(\rho_0 a_0)_I - z_1} \qquad F_{fl}^+ = \frac{u_2^+}{u_1^+} = \frac{F_{fl} + \frac{z_1}{(\rho_0 a_0)_{II}}}{1 + \frac{z_1}{(\rho_0 a_0)_I}}$$

$$F_{II} = \frac{u_2^-}{u_2^+} = \frac{(\rho_0 a_0)_{II} - z_2}{(\rho_0 a_0)_{II} + z_2} \qquad F_{fl}^- = \frac{u_1^-}{u_2^-} = \frac{1 - \frac{z_1}{(\rho_0 a_0)_I}}{F_{fl} - \frac{z_1}{(\rho_0 a_0)_{II}}} \tag{2}$$

The derivation of these formulae is straightforward, use is made of the following relations:

$$u_i = u_i^+ + u_i^- \ , \quad p_i = p_i^+ + p_i^- \quad i = 1,2 \tag{3a}$$

$$\frac{p^+}{u^+} = \rho_0 a_0 \ , \quad \frac{p^-}{u^-} = -\rho_0 a_0 \tag{3b}$$

$$F_{fl} = \frac{u_2}{u_1} = \frac{z_1}{z_2} \quad (p_2 = p_1) \tag{3c}$$

Note that the partial frequency responses can also be developed for p instead of u without F_0 being changed.

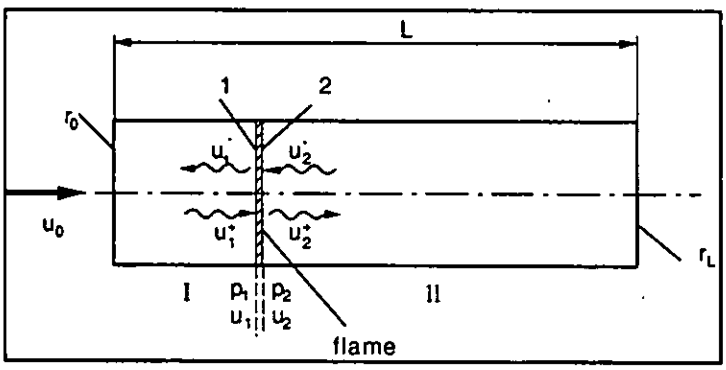

FIG. 2 : Wave Propagation in a Tube Including a Flame

For more complex geometries, the equations above do not change. The acoustical behavior of the part upstream of the flame is represented by z_1, and the downstream part is represented by z_2. The values for z_1, z_2 can either be obtained from numerical or experimental analysis or from analytical equations.

APPLICATION TO 3A-SERIES GAS TURBINE

THE 3A-COMBUSTION CHAMBER

The combustion system of the Siemens Vx.3A-series gas turbine family is characterized by an annular combustion chamber with 24 hybrid burners for gas and oil (Figure 3). The obvious advantages of this open ring design are:

- highly uniform hot-gas flow and temperature fields upstream of the power turbine (low pattern factor)
- reduced combustion chamber surface area, requiring less cooling air, resulting in more compressor discharge air being made available for low-NO_x combustion
- extremely short hot-gas residence time to suppress thermal NO_x formation

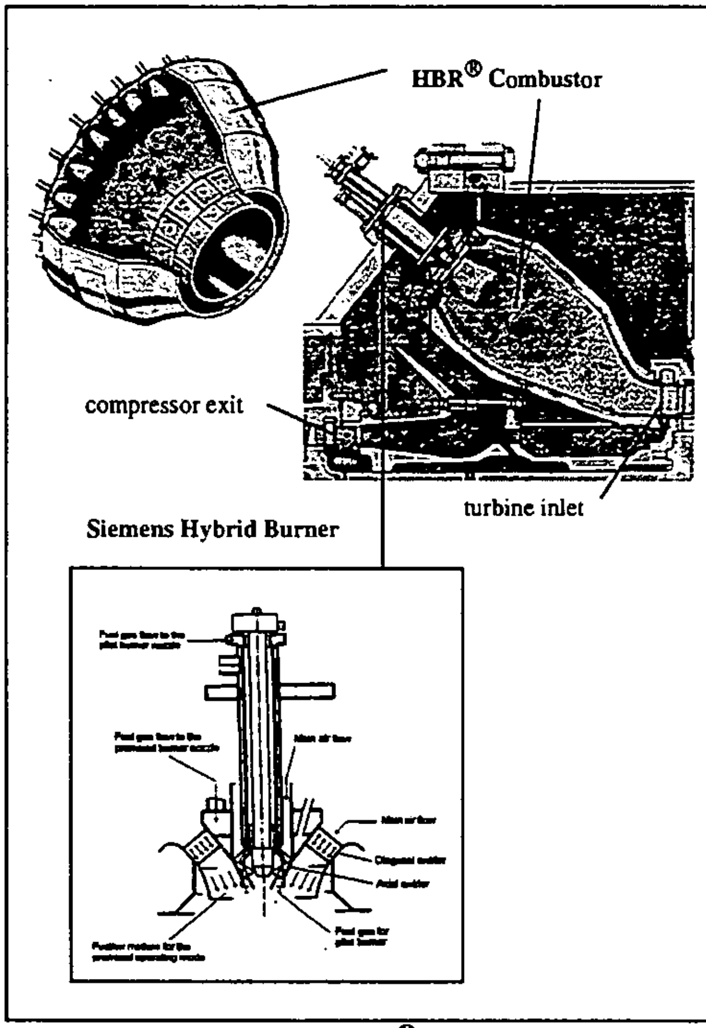

FIG. 3 : Siemens 3A-Series HBR® Gas Turbine Combustor

Some details of the hybrid burner are also shown in Figure 3. The main air flows through the outer, so-called diagonal swirler where the premixing of fuel and air takes place. The central, so-called axial swirler produces a pilot flame to stabilize the main flame. All details of this burner have to be modeled in order to describe its impedance accurately.

ACOUSTICAL NETWORK

The acoustical network using four-pole elements for this kind of gas turbine combustor is shown in Figure 4. As discussed above, combustion driven oscillation is a system issue. Consequently, the entire system, starting at compressor exit, including different types of fuel supply and ending at the turbine inlet, is taken into account for the stability analysis. Figure 4 only gives an overview of the system. This means that most of the elements are modeled in much greater detail as shown in Figure 5 for the premixed fuel supply system as an example.

As will be discussed later, in principle two main oscillation directions have to be considered for this kind of ring combustor: the circumferential as well as the axial direction. Basically, the four-pole elements are one dimensional with the result that the stability analysis has to be split off into the circumferential direction and the axial direction. This procedure affects all transfer elements upstream of the burner, namely the $flame_{ax,circ}$, the combustion chamber ($CC_{ax,circ}$) and the boundary condition ($BC_{ax,circ}$). The most complex network system occurs for the ring combustor itself in circumferential direction as shown in Figure 6. All 24 burner systems are taken into account to represent the annular combustor. The system is closed with a periodic boundary condition. The length of the duct pieces is adjusted to fulfil the requirements of the eigenfrequency analysis of the open ring combustor. This analysis is shown in more detail below. A much simpler network can be used for the axial direction as shown in Figure 7. Again, the four-pole element of the duct is calculated by an eigenfrequency analysis.

The acoustical network for the 3A-series gas turbine described above is used for the stability analysis, which is based on the acoustical model given in this paper.

TRANSFER ELEMENT "COMBUSTION CHAMBER"

STEADY-STATE-COMBUSTION

The dynamic behavior of the combustion chamber depends greatly on the 3D-temperature distribution because the local speed of sound is affected by the temperature, which varies between compressor outlet temperature and combustion temperature. The Ma-number is less than 0.3, with the result that the influence of the flow pattern on the dynamic characteristics is neglected.

In order to obtain the temperature distribution, 3D-Navier-Stokes (CFD) calculations have been performed taking one burner and 1/24 segment of the combustion chamber into account. This means for the segment, periodic boundary conditions have been used. The turbulence is modeled using the standard k-ε-model. For the combustion an improved eddy-break-up model is used taking into account the chemical kinetics with an integrated 2-step reaction scheme for methane-air as well as the interaction of turbulence and chemical kinetics. More details about the numerical scheme can be found in *Krüger et al. (1998)*, where a validation of this procedure for premixed turbulent flames is also given.

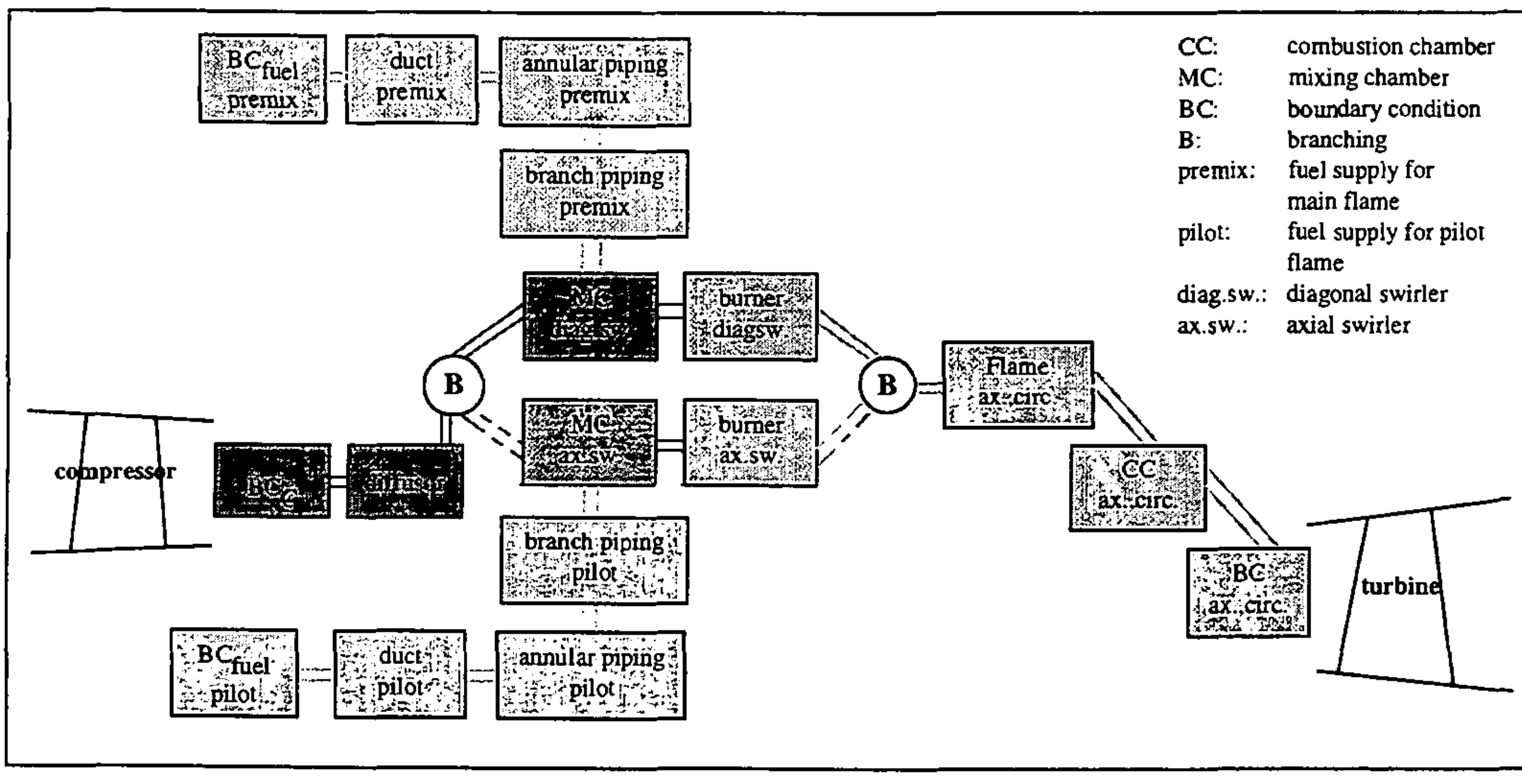

FIG. 4 : Acoustical Network (Overview)

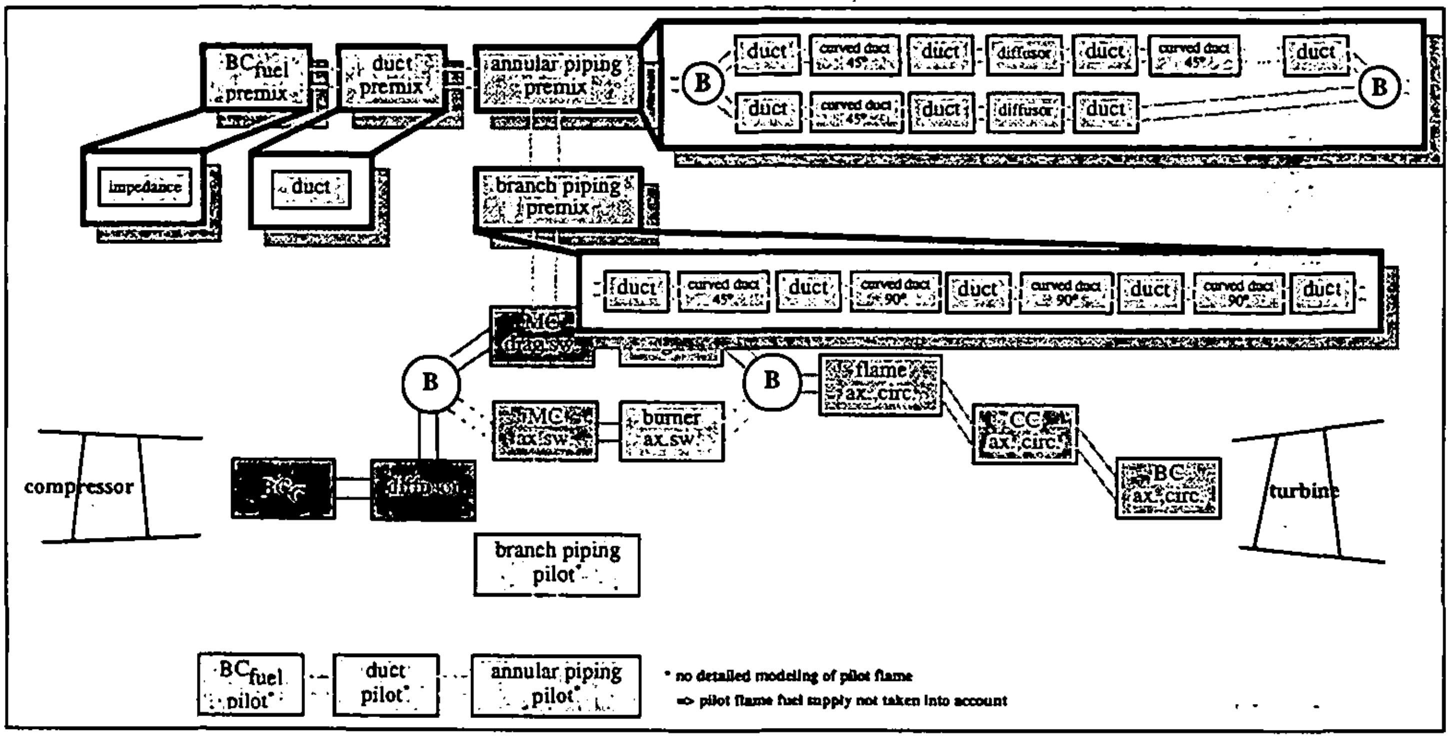

FIG. 5 : Acoustical Network for Premix Fuel Supply in Detail

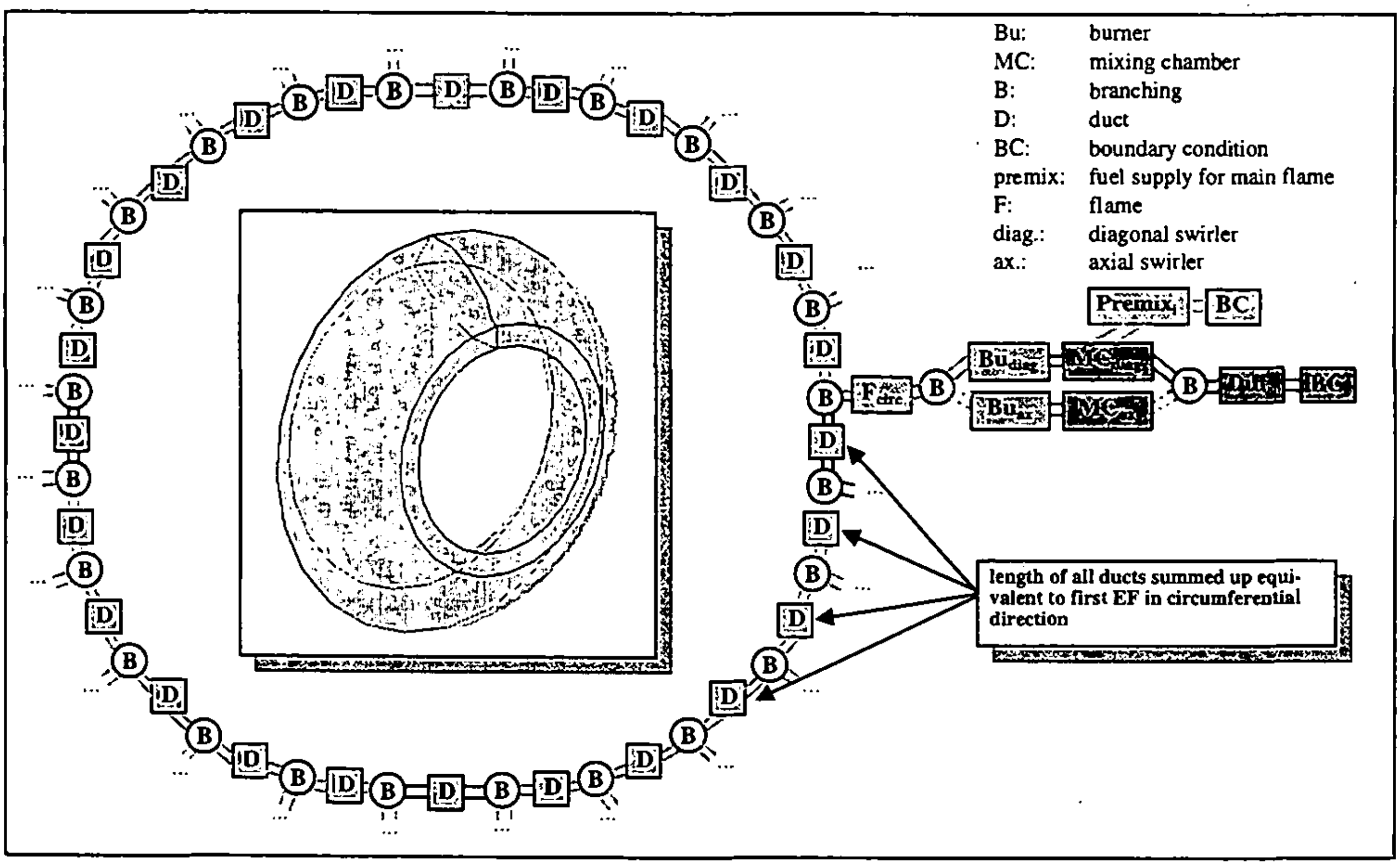

FIG. 6 : Acoustical Network for Combustion Chamber in **Circumferential** Direction in Detail

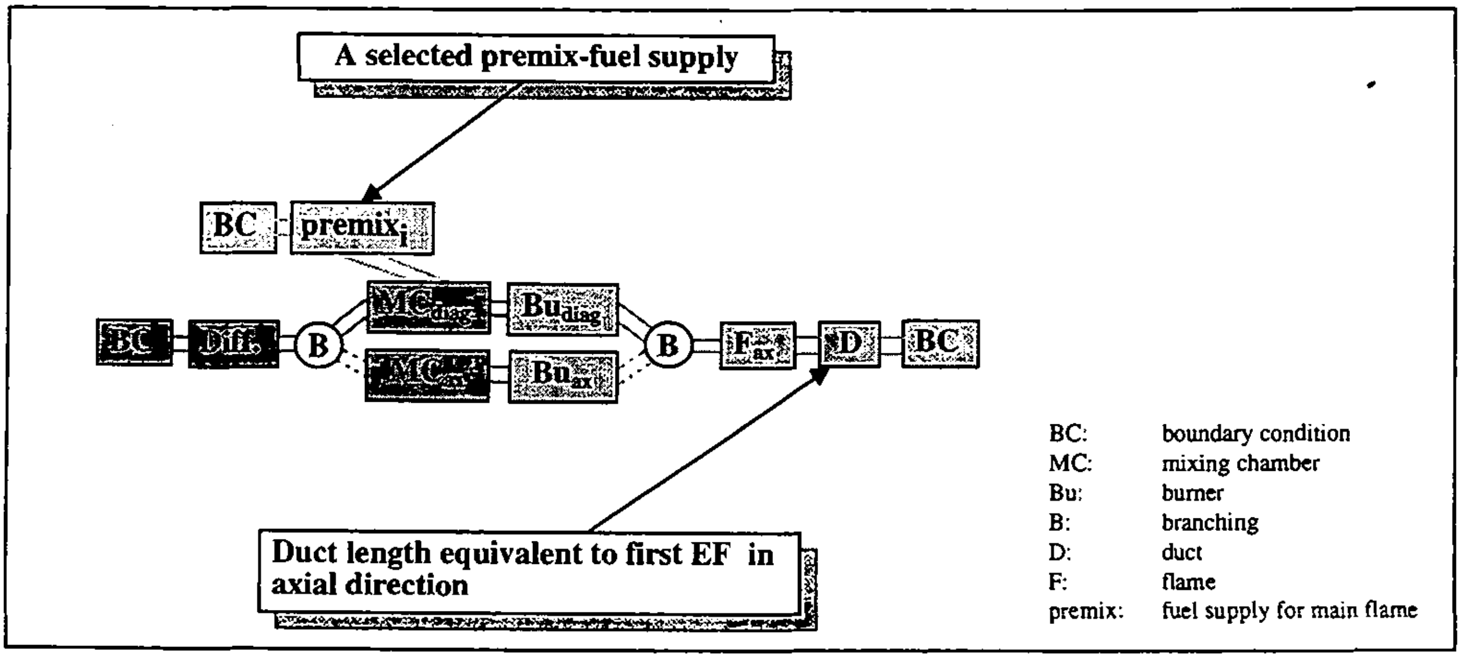

FIG. 7 : Acoustical Network for Combustion Chamber in **Axial** Direction in Detail

The 3D numerical grid - about 320,000 grid nodes - is shown in Figure 8. A homogeneous mixture of air and methane is assumed at the inlet downstream of the diagonal swirler. Temperature (compressor outlet temperature), velocity, swirl component and turbulent quantities k and ε are adjusted at this inlet to fulfil full load conditions of the gas turbine. At the axial swirler, velocity and swirl components are assumed according to the swirler geometry. Temperature and stoichiometric conditions are fixed to represent the heat release of the pilot flame.

The fuel field concentration predicted is shown in Figure 9. The typical V-shape of this kind of flame can be seen clearly. A strong recirculation zone exists due to the highly swirled flow. More details regarding the flame structure and the temperature distribution are given in *Krüger et al. (1999)*. Nevertheless, the temperature and flame structure predicted represent the experience from test engine and other calculation methods with reasonable accuracy.

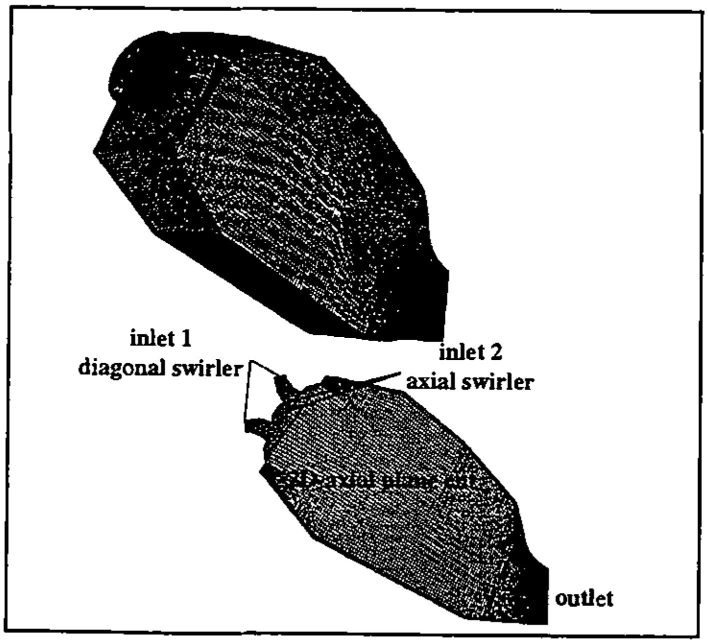

FIG. 8 : Numerical Mesh

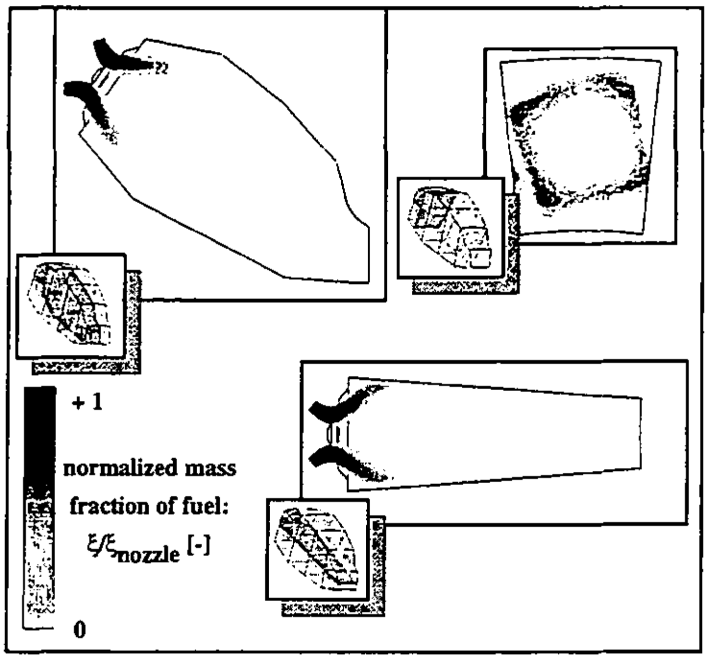

FIG. 9 : Numerical Steady-State Calculation

EIGENFREQUENCY ANALYSIS

This calculated temperature distribution is used to perform a 3D eigenfrequency analysis of the entire burner ring under the simplified assumption of reflecting boundary conditions at all walls. In the critical frequency range (lower than 1000 Hz), two main oscillation structures can be idenüfied: Circumferential modes and axial modes, which are both illustrated in Figure 10. The first four eigenfrequencies f_1 = 166.1 Hz to f_4 = 646.8 Hz are circumferential modes, and the fifth natural frequency f_5 = 731.0 Hz is identified as an axial mode.

Figure 10 shows the normalized acoustic pressure, so the structures of the oscillations can be detected very clearly.

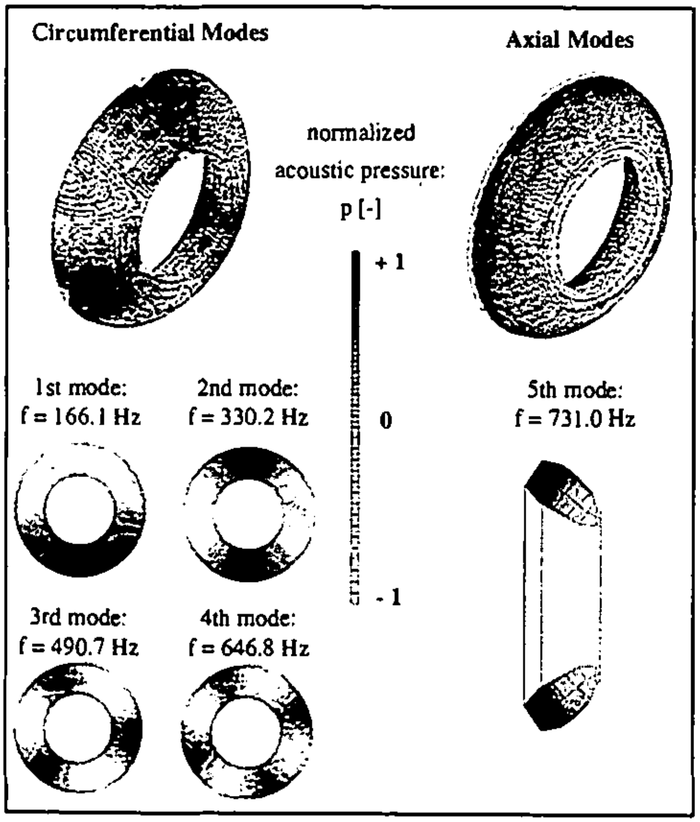

FIG. 10 : Eigenfrequency Analysis of the 3A-Series Annular Combustion Chamber

Using these results, the four-pole element "Combustion Chamber ($CC_{ax,circ}$)" (Figure 4) can be adjusted to fulfil the acoustic characteristics for both axial direction as well as circumferential direction. Simple duct elements are used as shown in Figure 6 for the circumferential and in Figure 7 for the axial direction.

As explained in *Walz et al. (1999)*, of course the results of the eigenfrequency analysis differs significantly with changing boundary conditions. However, the influence especially of the burner impedance on the critical frequencies, is considered by the acoustical network shown here. As a result the predicted frequency of self-induced oscillations does not necessarily correspond to the eigenfrequencies calculated under the assumption of rigid walls. The stability analysis,described later, will show this effect.

TRANSFER ELEMENT "FLAME"

Finally, the transfer function of the flame itself is the only element still required to complete the acoustical network. Only a few idealized analytical predictions e.g. *Hubbard and Dowling (1998)* or experimental data e.g. *Richards and Janus (1997)* are currently available for this type of premixed flame which is characterized by turbulence and swirl. Thus, a numerical prediction scheme is used which predicts the dynamic flame behavior by calculating the step function response. This means that a transient numerical simulation is performed after a sudden jump in mass flow at a time t = 0.0 s at the burner inlet. Only inlet 1 (diagonal swirler) is taken into account because over 90% of the overall mass rate flows through this inlet. The responding transient mass flow rate at different planes is analyzed. Using a Laplace-Transformation, this step function response is transferred into frequency domain with the result of the flame frequency response. More details of this method and validation for different types of flames can be found in *Krüger et al. (1998), Bohn et al. (1997) and Bohn et al. (1996)*.

How does the flame of the Vx4.3A series gas turbine react to a sudden disturbance at the inlet? The transient calculation is performed starting from the steady-state CFD analysis which is described above. At a time t = 0 s, only the mass flow rate at inlet 1 is increased by 10%. All other boundary conditions remain unchanged during the calculation, including the equivalence ratio. The time step is set to a very small value of Δt =0.15 ms in order to obtain a very exact resolution of the step function response. This small time step makes the effort for these calculations very high but helps to understand the dynamic behavior in detail. Due to this high computational effort, the influence of unmixedness as well as the influence of a change in equivalence ratio has not been studied yet but will form part of future work. Effects of different load conditions on the flame frequency response also will belong to the future scope of work.

Figure 11 provides a detailed insight into the movement of the flame front. The isolines of constant CO mass fraction represent the flame shape well. It can be detected clearly that the flame front does not change significantly with time during the first ms, neither in the axial nor in the circumferential direction. Between 1.05 ms and 2.85 ms the flame front grows becoming longer in the x-direction. Between 3 and 6 ms the axial length does not change significantly but the flame grows in the circumferential direction.

As explained above, two directions - the axial and the circumferential - are important for the stability prediction. In order to determine the frequency response of the flame in both directions the step function response also h(t) has to be calculated in both directions. As usual, h(t) is defined as the unsteady mass flow rate divided by the mass flow jump, that is:

$$h(t) = \begin{cases} \dfrac{\left(\dot{m}(t)_{ax} - \dot{m}(t=0)_{ax}\right)/\dot{m}(t=0)_{ax}}{\Delta\dot{m}_{inl}/\dot{m}(t=0)_{inl}} & \text{axial direction} \\ \dfrac{\left(\dot{m}(t)_{circ} - \dot{m}(t=0)_{circ}\right)/\dot{m}(t=0)_{circ}}{\Delta\dot{m}_{inl}/\dot{m}(t=0)_{inl}} & \text{circumf. direction} \end{cases} \quad (4)$$

The unit function responses calculated for axial and circumferential direction are shown in Figure 12. The balance planes for both

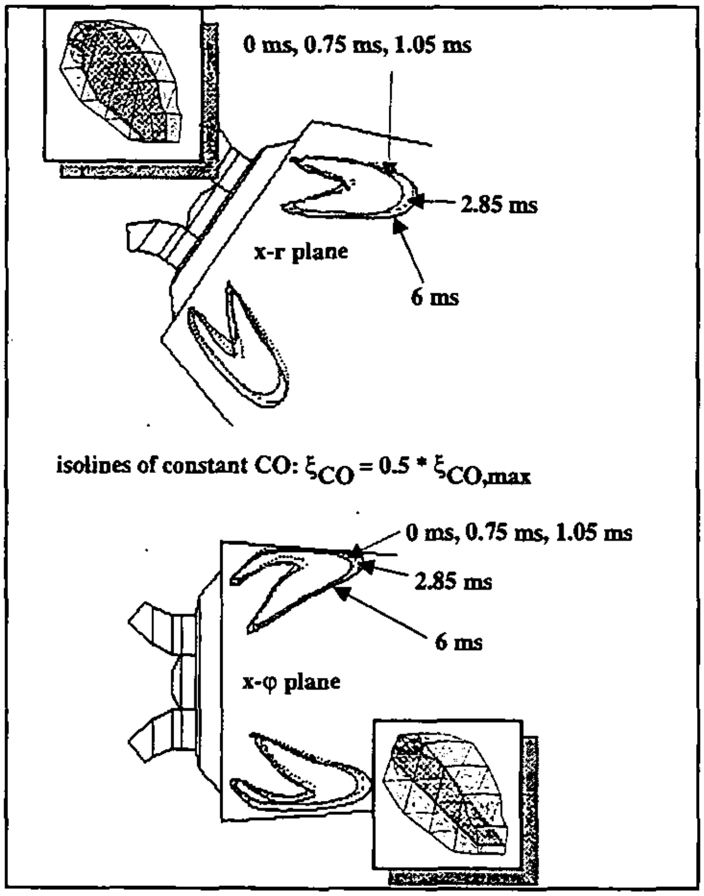

FIG. 11 : Numerical Transient Calculation (3D-CFD) of the Premixed Turbulent Swirled Flame

directions are also illustrated in this figure. They are placed in such a way that the combustion process is finished at this position.

Both step functions show the character of a higher order delay time element. It is remarkable that the characteristic time scales are somewhat different for both directions. While the unit response reaches its maximum at 6 ms for the axial direction, the characteristic time for the circumferential direction is somewhat smaller (4.5 ms). At least after 9 ms the flame system is once again in steady-state. This means that the mass flow rate does not change further with time. The flame is adjusted to the new velocity at the inlet. It is important to understand the different meanings of the unit function responses. Obviously, a 10% jump in mass flow at the inlet increases the mass flow rate at the axial exit to 10% but it also raises the mass flow rate in circumferential direction near the flame to 2.8% of the steady-state rates.

The unit function responses $h(t)_{ax,circ}$ are then transferred into the frequency domain by a Laplace-Transformation in order to obtain the flame frequency responses $F_{fl,ax,circ}$, which are given in Figure 13a and 13b for both directions. These frequency responses show the expected strong phase shift, which is of course different for both directions due to the different characteristic times discussed above. Using equation 3c, the four-pole element for the flame can be directly determined from F_{fl}, again for both directions.

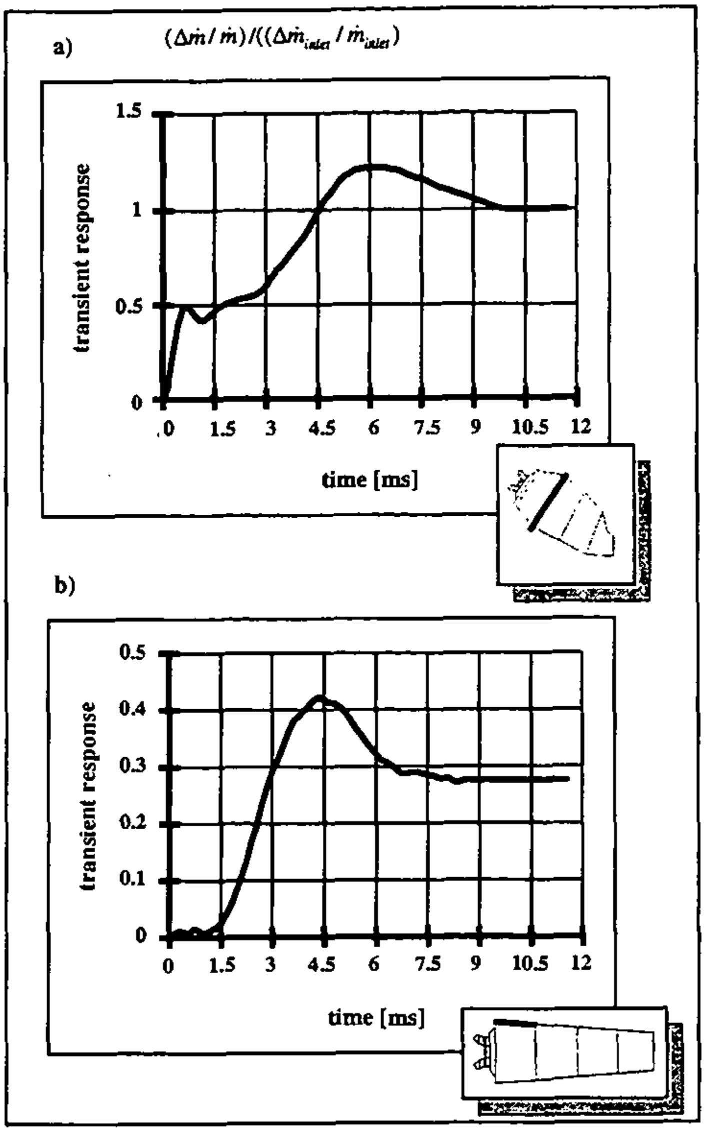

FIG. 12 : Transient Response of Flame
a) in Direction of the Axis b) at Periodic Boundaries

A more detailed discussion of the dynamic flame behavior of this type of flame can be found in *Krüger et al. (1999)*.

STABILITY ANALYSIS

Using all four-pole elements discussed above in detail, a stability analysis using four-pole theory is performed for a Vx4.3A gas turbine combustor. As explained above, both directions (axial and circumferential) have to be discussed separately.

In order to understand the self-induced oscillation the following pictures will show four different frequency responses:

- frequency response F_0
- partial frequency response F_I (upstream of the flame)
- partial frequency response F_{II} (downstream of the flame)
- flame frequency response F_{fl}

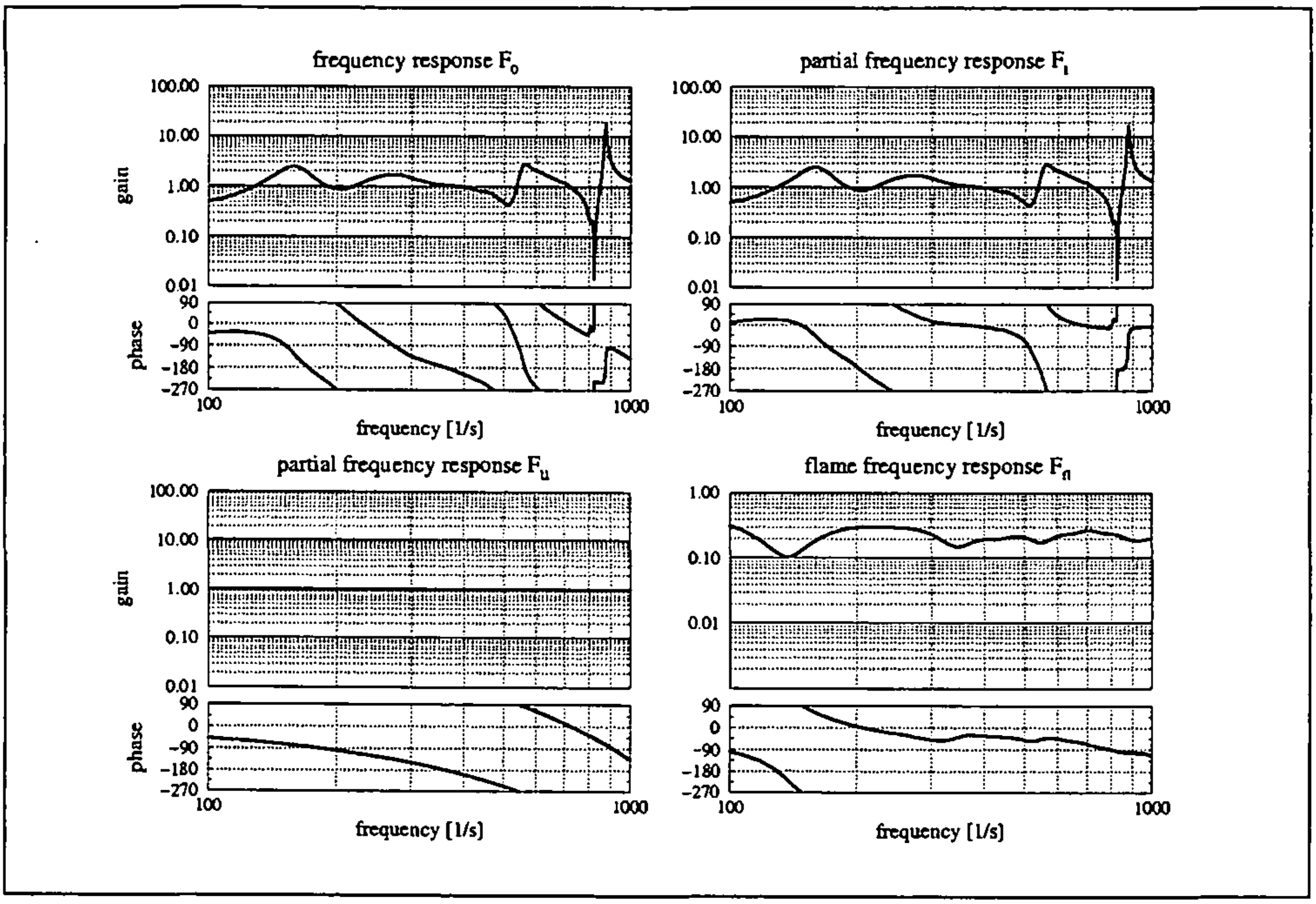

FIG. 13a : Stability Analysis (Axial Direction)

AXIAL DIRECTION

Figure 13a shows the analysis for the axial direction. F_{fl} has been discussed in detail above. The partial frequency response F_u represents the acoustical network downstream of the flame with a reflection coefficient at turbine inlet of 1. The frequency where the phase equals zero (f=730.3 Hz) corresponds to the 5th eigenmode, which is the first axial mode for two "hard" ends. The partial frequency response F_l is influenced by upstream components, especially the impedance at the burner itself.

Using the Nyquist criterion, there are four frequencies in the range between 100 Hz and 1000 Hz where instabilities could possibly occur:

$f_{1,ax}$ = 231,0 Hz with $|F_0| = 1.21 > 1$
$f_{2,ax}$ = 517.7 Hz with $|F_0| = 0.46 < 1$
$f_{3,ax}$ = 721.9 Hz with $|F_0| = 0.99 < 1$
$f_{4,ax}$ = 820.8 Hz with $|F_0| = 0.24 < 1$

The third frequency corresponds clearly to the acoustical behavior of the combustion chamber in axial direction. The analysis shows a gain close to but lower than one, i.e. the oscillation at this frequency is not necessarily amplified, but could. Like the analysis of the circumferential direction will show, there are some other more critical frequencies lower than this one corresponding to the eigenfrequencies in circumferential direction where a excitation is more probable.

The other critical frequencies are clearly not the result of the acoustics of the combustion chamber but can be linked to the impedance of the upstream elements and their interaction with the flame. The analysis shows the possibility of a self-induced oscillation for the first critical frequency at 231 Hz. This result is quite interesting because it was not expected that a frequency so far away from the first axial mode could be critical when only investigating the axial direction. At the second and fourth critical frequency the oscillation is clearly not amplified. So an instability is not to be expected.

A reduction of the the gain of the reflection coefficient at the turbine inlet does not change the critical frequencies because the phase relations remain the same. A part of the oscillation can leave the system, so the overall gain is damped. This could correspond to part-load conditions with an reflection coefficient lower than 1 where it is well known that the tendency to combustion driven oscillations is much less pronounced than for full-load in which conditions close to choking of the turbine fulfil the boundary condition of a high reflection coefficient.

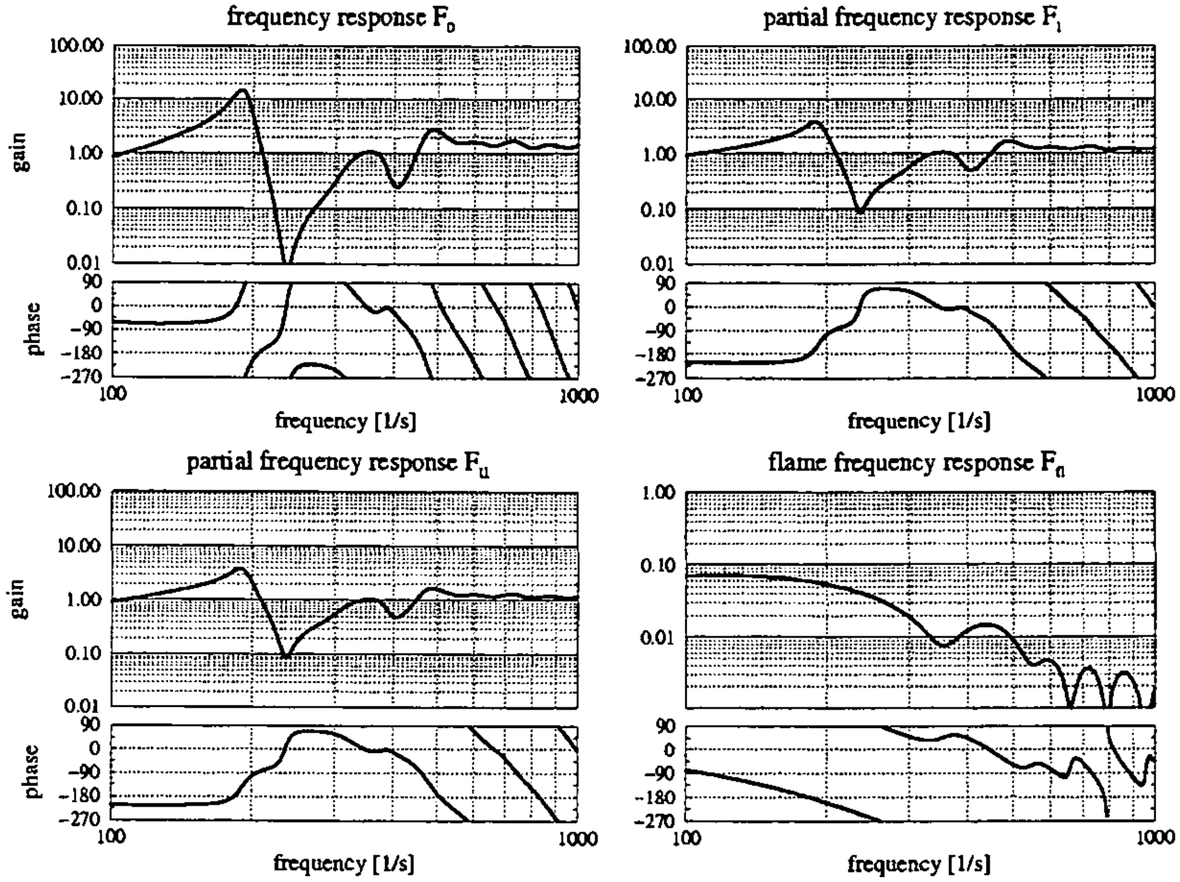

FIG. 13b : Stability Analysis (Circumferential Direction)

CIRCUMFERENTIAL DIRECTION

The conditions in the circumferential direction are more complex due to the high complexity of the acoustical network. Figure 13b illustrates the result of the stability analysis for this direction. Again, the flame frequency response is calculated by a Laplace-Transformation of the step function response as discussed. The partial frequency responses F_I and F_{II} are now equal due to the fact that each represents one half of the cyclic ring (compare to Figure 6). This means 12 branches including all burner and flame acoustics.

Looking to the frequency response F_0, seven critical frequencies now exist in the range between 100 Hz and 1000 Hz:

$f_{1,circ}$ = 184.1 Hz with $|F_0| = 12.63 > 1$
$f_{2,circ}$ = 235.1 Hz with $|F_0| = 0.01 < 1$
$f_{3,circ}$ = 349.5 Hz with $|F_0| = 1.07 > 1$
$f_{4,circ}$ = 508.4 Hz with $|F_0| = 2.39 > 1$
$f_{5,circ}$ = 665.5 Hz with $|F_0| = 1.41 > 1$
$f_{6,circ}$ = 834.0 Hz with $|F_0| = 1.39 > 1$
$f_{7,circ}$ = 997.7 Hz with $|F_0| = 1.41 > 1$

The first, third, fourth, fifth, sixth and seventh critical frequencies correspond with the first and higher order azimuthal eigenfrequencies, but they do not fit them exactly, which can be explained by the burner and flame impedance (compare with chapter eigenfrequency analysis). The second critical frequency does not fit into this sequence, which again can be explained by the interaction between burner impedance and dynamical flame behavior. Anyway, the second critical frequency is strongly damped so that no oscillation is expected. Self-induced oscillations can possibly occur for all azimuthal modes, where the first mode is clearly distinguished by its high value of F_0. The structure of this resonance peak indicates that damping alone would not be sufficient to suppress the oscillation.

Although their gain indicates an instability, oscillations at the fourth and higher critical frequencies are not considered very probable. First, because they are only higher modes of the two basic frequencies and second, the model may underpredict damping at these frequencies.

To sum up the results of the stability analysis, it becomes clear that this kind of annular combustor has a pronounced tendency to produce combustion driven oscillations in a frequency range of about 180 Hz. Indeed experience from engine test show that self-induced oscillations occur in a range between 170 and 180 Hz and approximately 350 Hz for unfavourable operating conditions which are comparable to the boundary conditions used in the CFD predictions and in the the stability analysis. This comparison shows that the prediction of combustion driven oscillation is possible using the method discussed above.

SUMMARY AND CONCLUSION

A stability analysis for the Siemens Vx4.3A gas turbine family has been performed in order to predict combustion driven oscillations. The detailed analysis using an acoustical network of four-pole elements includes the entire combustion system starting at the compressor exit and the fuel supply system and ending at the turbine inlet.

Special attention has been paid to the important elements "combustion chamber" and "flame". Full 3D-analysis has been used to take their acoustical behavior into account. It has been found that in the interesting frequency range between 100 Hz and 1000 Hz two principal directions, the azimuthal as well as the axial, have to be considered. Therefore, all investigations have been split off into these directions. The four-pole element of the annular combustion chamber has been derived by an eigenfrequency analysis. The dynamical flame behavior has been calculated using a full 3D-Navier-Stokes method. The typical higher order delay time character has been found. For the first time, characteristic times for both circumferential as well as axial direction have been calculated for this type of flame. These informations are essential for the success of the stability analysis.

Using a Nyquist-criterion to investigate the stability of the entire system, a clearly pronounced tendency to produce combustion driven oscillations has been found in the frequency range of 180 Hz. This is close to the first azimuthal natural mode of this annular combustor. The shift between the natural mode and the predicted critical frequency can be explained by the influence of burner and flame impedance. The result has been confirmed by engine test data.

By way of conclusion, it can be pointed out that:

- combustion driven oscillation can be predicted for this kind of gas turbine combustor using an acoustical network of four-pole elements.
- the 3D-analysis of the combustion chamber and the flame is the key issue for successful analysis.
- a systematical parameter study very clearly shows the influence of several elements, especially the burner impedance.
- the stability analysis shown here is a powerful engineering tool for developing effective strategies against combustion driven oscillations.

REFERENCES

Bohn, D.; Li, Y.; Matouschek, G. and Krüger, U. "Numerical Prediction of the Dynamic Behavior of Premixed Flames Using Systematically Reduced Multi-Step Reaction Mechanisms", ASME-Paper 97-GT-265, **1997**

Bohn, D.; Deutsch, G. and Krüger, U. "Numerical Prediction of the Dynamic Behavior of Turbulent Diffusion Flames", ASME-Paper 96-GT-133, **1996**

Bohn, D. and Deuker, E. "An Acoustical Model to Predict Combustion Driven Oscillations", 20th International Congress on Combustion Engines (CIMAC), London **1993**

Faber, Ch. " Entwicklung eines Rechenmodells zur Vorausberechnung des Stabilitätsverhaltens von Brennkammersystemen ", Internal report, Inst. of Steam and Gas Turbines, RWTH Aachen (Germany), **1991**

Fleifil, M.; Annaswamy, A. M., Ghoneim, Z. A. and Ghoniem, A. F. "Response of a Laminar Premixed Flame to Flow: A Kinematic Model and Thermoacoustic Instability Results", Combustion and Flame, 106, pp 487-510, **1996**

Hsiao, G.C.; Pandalai, R.P.; Hura, H. S. and Mongia, C. " Combustion Dynamic Modeling for Gas Turbine Engines", AIAA-paper 98-3380, **1998**

Hubbard, S. and Dowling, A.P. "Acoustic Instabilities in Premix Burners", AIAA-paper 98-2272, **1998**

Keller, J.J. "Thermoacoustic Oscillations in Combustion Chambers of Gas Turbines", AIAA-Journal, Vol. 33 No. 12, **1995**

Krüger, U.; Hüren, J.; Hoffmann, S., Krebs, W. and Bohn, D. "Combustion Driven Oscillations: Numerical Prediction of Dynamic Behavior of Gas Turbine Flames, submitted for the 5th AIAA/CEAS Aeroacoustics Conference, **1999**

Krüger, U.; Hoffmann, S.; Krebs, W.; Judith, H.; Bohn, D. and Matouschek, G. "Influence of Turbulence on the Dynamic Behavior of Premixed Flames", ASME-Paper 98-GT-232, **1998**

Meyer , Neumann " Physikalische und technische Akustik ", Vieweg-Verlag, Brauschweig (Germany), **1967**

Munjal, M.L. " Acoustics of Ducts and Mufflers ", John Wiley & Sons, **1987**

Polifke, W.; Paschereit, C.O. and Sattelmayer, T. "A Universally Applicable Stability Criterion for Complex Thermo-Acoustic Systems", VDI Berichte 1313, **1997**

Richards, G.A. and Janus, M.C. "Characterization of Oscillations During Premixed Gas Turbine Combustion", ASME-paper 97-GT-244, **1997**

Walz, G., Krebs, W., Hoffmann, S. and Judith, H. "Detailed Analysis of the Acoustic Mode Shapes of an Annular Combustion Chamber", submitted for the ASME TURBOEXPO, **1999**