Dynamic Simulation of Active Compressor Stability Control for a Gas Turbine Engine

Vili Panov

Siemens Industrial Turbomachinery Ltd
PO Box 1, Waterside South, Lincoln, LN5 7FD, UK

ABSTRACT

The type of fuel, ambient conditions and gas turbine engine variations are among many variables, which determine the amount of metered fuel required for efficient and reliable gas turbine start-up with minimized thermal stresses and without compressor instabilities. Hence, scheduling of fuel for gas turbine engine start-up without dynamic adjustments for unpredictable influencing variables cannot assure reliable start-up. To address this variability, implementation of the active compressor stability control for the start-up scheduling was considered in this study.

Generic active control method has been proposed, and this method can be applied on both open and closed start-up scheduling loops. Active control philosophy is based on the control of stability margin using corrective action to avoid or recover engine from compressor instabilities.

Active control method is designed to sense incipient stall using stall detection method and subsequently to initiate engine control system corrective action to avoid compressor instabilities by adjusting the fuel flow schedule. Centre casing dynamic pressure signal is used for synthesis of fast and reliable measure of compressor destabilization. When compressor instability is detected, engine control system initiates adjustment of fuel flow schedule defined by corrective function, which is based on the synthesized measure of compressor destabilization.

To assess applicability of proposed control method, dynamic simulations of engine start-up have been carried out using generic gas turbine simulation tool GasTurboLib. Nonlinear mathematical model of transient compressor dynamics has been developed to describe instability behaviour of axial compressor. During simulation of start-up sequence, compressor instabilities have been induced to study respond of proposed active control method. Description of implemented compressor model and numerical simulation results are given in this paper.

INTRODUCTION

The operating range of axial compressors is limited by the onset of flow instabilities. Compressor aerodynamic instabilities are generally categorized in two distinct classes: rotating stall and surge [1]. Surge is violent instability characterized by one-dimensional fluctuation in mass flow through the compression system.

The occurrence of surge is preceded by the stalling of some compressor blade row elements. Rotating stall is characterized by regions of reduced or reversed flow that rotate around the annulus of the compressor. There are two types of stall – progressive and abrupt. Progressive stall can be recognised as a gradual reduction in total pressure ratio after stall begins. Abrupt stall has a sharp discontinuity in the pressure ratio characteristic with sudden drop in pressure.

Focus in this study was on stability management of compressor component of gas turbine during engine start-up. By minimizing number of unsuccessful start attempts, due to compressor instabilities, damage to different engine components can be avoided or reduced, and hence life of gas turbine engine can be extended.

During start-up when stall occurs, pressure ratio of the compressor initially exceeds some critical value at a given speed, resulting in a subsequent reduction of compressor pressure ratio and airflow delivered to the engine combustor. If such condition is undetected and allowed to continue, the combustor temperatures and vibratory stresses induced in the compressor may become sufficiently high to cause damage to the gas turbine engine.

Different methods for starting engine with stall-free and repeatable engine acceleration had been considered in the past. These methods alleviate stall by increasing the compressor stall margin, using scheduled control actions in one of the following ways: controlling the fuel flow schedule, adjusting compressor variable vanes or compressor bleed.

In this paper active compressor stability management has been considered. Dynamic pressure sensor is used to determine the onset of rotating stall by measuring the level of compressor destabilization. When stall onset is detected, the proposed method initiates control system corrective actions to avoid compressor instabilities by adjusting the fuel flow schedule.

NOMENCLATURE

Variables

A area [m^2]
C specific heat [J/kg k]
h heat transfer coeff [W/Km^2]
L length [m]
$\dot{m}$ mass flow [kg/s]
n rotational speed [rpm]
P pressure [Pa]
q heat flow [J/s]
R specific gas constant [J/kg K]
T temperature [K]
V volume [m^3]
γ specific heat ratio [-]
η efficiency [-]
π pressure ratio [-]
ρ density [kg/m^3]

Indices

a speed of sound
air air
cb combustor
cm compressor
d design point
f frequency
in inlet
m metal
out outlet
p const pressure
pl plenum
ref referent position
s surface
v const volume
vgv variable guide vanes

ACTIVE COMPRESSOR STABILITY CONTROL

Active compressor control encompasses a wide range of technical approaches [2,3]. Initially, this term referred to feedback control, used to extend the stable operating range of gas turbine compression system, i.e. active stabilization. This term also covers variety of methods utilized to change steady state operating point of engine upon the detection of disturbances, e.g. by controlling bleed or restaggering stators. Finally, active compressor control includes feedback control used to enhance compressor robustness to disturbances, i.e. to permit operation closer to the stability limit [4].

Originally new approach to suppress compressor instabilities by using dynamic feedback had been proposed two decades ago [5]. Since this concept was proposed several researchers demonstrated this idea [6-8]. These demonstrations were using compressor mass injectors [6] or high bandwidth actuators for IGV modulation [7,8], to achieve active compressor stabilization. Alternative approaches to active compressor control in recent implementations also considered following concepts: compressor recirculation [9], compressor exit bleed modulation [10] and aeromechanical feedback control [11].

The concept of enhanced compressor robustness has been considered in this study, and generic active control method has been proposed. This method can be applied on both, open and closed start-up scheduling loops, and is based on the control of stability margin using corrective actions to avoid or recover engine from compressor instabilities.

Signal from centre casing dynamic pressure transducer is utilized to detect compressor instabilities. Detection method based on centre casing dynamic pressure monitoring is capable of sensing high frequency pressure oscillations and hence it can be used for control of both, stall and surge instabilities. Control of stability margin is achieved, by synthesizing a measure of compressor destabilization, and using this measure in generation of corrective action.

Start-up Scheduling

We can recognize two start-up philosophies: closed loop and open loop start-up scheduling. Open loop strategy is based on a predetermined fuel schedule, where the rate at which the fuel increases is controlled by a set of parameters and breakpoints. Fuel flow is usually prescheduled as a function of gas generator speed. Such fuel control schedules are not absolutely accurate as they do not account for variations in performance of different engines of a given engine model, engine performance deterioration, external ambient conditions such as temperature and pressure of the ambient airflow, temperature of the fuel flow being delivered to the engine and etc.

Depending on the engine speed, the acceptable range in amounts of metered fuel available during gas turbine engine start-up can be quite limited. If insufficient fuel is metered and burned, necessary torque will not be applied to the engine for achieving normal idle rotational speeds.

The opposite situation results with excess-metered fuel, which raises temperature of gas turbine components too rapidly and to excessive levels. Excessive heat and rate of temperature increase can cause compressor instabilities. In addition to this, a rate of temperature increase and excessive quantities of heat can cause exorbitant thermal stresses, which can reduce engine component lifetime.

Engine closed loop start-up scheduling is achieved by monitoring given engine requirements and closing the loop on the monitored parameter to provide required start-up scheduling by adjusting fuel flow. In this arrangement fuel flow is variable delivered to the engine with a governor controlled feedback loop arrangement, maintaining fixed gas generator acceleration or torque profile.

Compressor Stability Monitoring

Compressor instabilities can be recognised as oscillation of centre casing pressure, where the frequencies of surge oscillations are typically over an order of magnitude less then those associated with rotating stall.

Centre-casing dynamic pressure transducer signal is utilized to detect compressor instabilities and to generate appropriate corrective control actions. It has been concluded that controlling gas generator acceleration rate it is possible to control compressor destabilization process. This has been used as a basis of proposed control method, where level of centre casing dynamic pressure amplitude is used for synthesis of measure of compressor destabilization.

Details of two monitoring techniques are given in this section. Compressor stability monitoring is based on moving standard deviation (MSTD) and moving exponential weighted average (MEWA) of pressure transducer signal. Increased value of moving standard deviation of rate of change of pressure signal (MSTD) indicates presence of compressor instabilities, and level of moving exponential weighted average of absolute value of rate of change of pressure signal (MEWA), represents severity of pressure fluctuations.

Monitoring of moving standard deviation (MSTD) of pressure rate of change is given by following expression:

$$MSTD\left(\frac{\Delta P}{\Delta t}\right)_i = \sqrt{\frac{\sum_{j=i}^{i-n}\left[\left(\frac{\Delta P}{\Delta t}\right)_j - \sum_{k=i-n}^{i}\frac{\left(\frac{\Delta P}{\Delta t}\right)_k}{n+1}\right]^2}{(n+1)}} \quad (1)$$

In above equation Eq. (1), moving average MAVE is calculated as:

$$MAVE\left(\frac{\Delta P}{\Delta t}\right)_i = \frac{\sum_{k=i-n}^{i}\left(\frac{\Delta P}{\Delta t}\right)_k}{(n+1)} \quad (2)$$

where n is number of previous data points, i.e. time steps. Rate of change of pressure is calculated using following numerical scheme:

$$\left(\frac{\Delta P}{\Delta t}\right)_i = \frac{P_i - P_{i-1}}{t_i - t_{i-1}} \quad (3)$$

where subscripts i and $i-1$ represent current and previous time step respectively.

Compressor instability is indicated when moving standard deviation is higher then predetermined threshold value:

$$MSTD\left(\frac{\Delta P}{\Delta t}\right)_i \geq MSTD\left(\frac{\Delta P}{\Delta t}\right)_{threshold} \quad (3)$$

Monitoring of moving exponential weighted average (MEWA) of pressure rate of change is given by following definition:

$$MEWA\left(\left|\frac{\Delta P}{\Delta t}\right|\right)_i = W * \left|\frac{\Delta P}{\Delta t}\right|_i + (1-W) * MEWA\left(\left|\frac{\Delta P}{\Delta t}\right|\right)_{i-1} \quad (4)$$

where W is weight factor. Absolute value of rate of change of pressure is defined with following numerical scheme:

$$\left|\frac{\Delta P}{\Delta t}\right|_i = \left|\frac{P_i - P_{i-1}}{t_i - t_{i-1}}\right| \qquad (5)$$

where subscripts i and $i-1$ represent current and previous time step respectively. Recommended values for Δt, correspond to typical control loop update rates, and range from 10 to 100 ms.

Active Control Implementation

Active control philosophy is based on the control of stability margin using corrective actions. When compressor destabilization is detected, engine control system initiates corrective action – adjustment of fuel flow schedule. This adjustment of fuel flow scheduling is defined by corrective function, which generates required correction based on the synthesized measure of compressor destabilization.

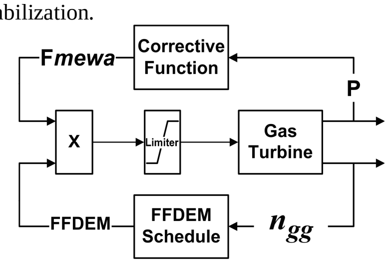

Fig. 1 Open loop implementation of active control

Typically fuel flow logic consists of several control limiting loops which are compared by selectors. A simplified representation of control logic for open and closed loop start-up strategies, with only main fuel flow scheduling loops is presented in this section. Implementation of the active compressor stability control method for the open and closed start-up scheduling loops is presented in Fig. 1 and Fig. 2 respectively. In open loop arrangement, fuel flow demand (FFDEM) is directly adjusted, and in closed loop configuration corrective function performs adjustment of gas generator acceleration profile demand.

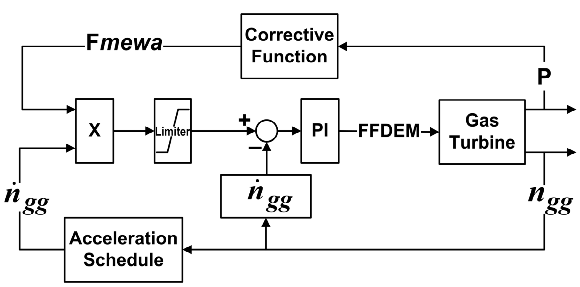

Fig. 2 Closed loop implementation of active control

Measure of Compressor Destabilization

Corrective action is defined with corrective function F_{mewa}, which adjusts fuel flow schedule according to synthesized measure of compressor destabilization. For active control of detected instabilities, evaluation of compressor destabilization is based on monitoring of centre casing dynamic pressure and following form of correction function F_{mewa} has been proposed:

$$F_{mewa} = 1 - \left[\overline{MEWA}\right]^B \qquad (6)$$

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where $\overline{MEWA}$ is compressor instability factor and B correction factor.

Compressor instability factor can be written in the following form:

$$\overline{MEWA} = A \times \Delta MEWA \qquad (7)$$

where A is normalization parameter defined as:

$$A = \frac{1}{MEWA_{full} - MEWA_{inc}} \qquad (8)$$

and $\Delta MEWA$ is measure of compressor destabilization represented with:

$$\Delta MEWA = MEWA - MEWA_{inc} \qquad (9)$$

In above equation Eq. (9), $MEWA$ is severity of pressure fluctuations defined with equation Eq. (4), $MEWA_{inc}$ is first threshold level, which represents incipient instability behaviour, and $MEWA_{full}$ is second threshold level, which indicates fully developed compressor instability.

Introducing expressions for compressor instability factor into definition of correction function, one can obtain following definition:

$$F_{mewa} = 1 - \left[\frac{MEWA - MEWA_{inc}}{MEWA_{full} - MEWA_{inc}}\right]^B \qquad (10)$$

$$\text{for } MEWA > MEWA_{inc}$$

$$F_{mewa} = 1 \qquad \text{for } MEWA \le MEWA_{inc}$$

Factor B is adjustable parameter, which can be tuned for different engine types or different compressor destabilisation mechanisms.

DYNAMIC SIMULATION MODELLING

Generic transient simulation tool GasTurboLib [12] has been enhanced with capability to simulate flow instabilities within compressor component. Developed compressor component block has been used to build single shaft engine model, and start-up sequence has been simulated. During simulation of start-up sequence, compressor instabilities have been induced to assess applicability of proposed active control method. Description of implemented compressor model is given in this section.

Compression System Modelling

A compressor is modelled as an ideal component with compressor body and diffuser, which is represented as a lumped element at the compressor outlet. Model also considers inter-stage cooling air bleed and a Variable Guide Vanes (VGV) offset correction. The inter-stage cooling air stream is assumed to be a constant percentage of compressor inlet flow, and cooling stream properties are calculated assuming a linear temperature and pressure rise along the compressor.

In the adopted modelling framework compressor is represented with volume and flow model. A flow model is the result of a modelling abstraction, where the volume is neglected. This flow model contains algebraic equations that describe the flow of mass and energy. To model the dynamic behaviour of the compressor component following generic equations for pressure and temperature have been used:

$$\frac{dP_{out}}{dt} = \left(\frac{\partial P_{out}}{\partial t}\right)_v + \sum_j \frac{\partial P_{out}}{\partial \varsigma_j}\frac{\partial \varsigma_j}{\partial t} \qquad (11)$$

$$\frac{dT_{out}}{dt}=\left(\frac{\partial T_{out}}{\partial t}\right)_v+\sum_l \frac{\partial T_{out}}{\partial \xi_l}\frac{\partial \xi_l}{\partial t}$$

where ς_j and ξ_l are internal thermodynamic state and mechanical variables used to model dissipative processes.

The first term in above equations Eq. (11), represents volume gas dynamics using ordinary differential equation, and second term describes behaviour of components using nonlinear algebraic equations, which are based on steady state component characteristics.

The flow model is represented by pressure $\prod$ and temperature Q loss coefficients and empirical information derived from rig tests, i.e. compressor component maps.

$$\sum_j \frac{\partial P_{out}}{\partial \varsigma_j}\frac{\partial \varsigma_j}{\partial t}=\pi_{cm}\dot{P}_{in\_cm}+\prod \dot{P}_{in\_cm} \quad (12)$$

$$\sum_l \frac{\partial T_{out}}{\partial \xi_l}\frac{\partial \xi_l}{\partial t}=\pi_{cm}^{\frac{(\gamma_{air}-1)}{\gamma_{air}\eta_{cm}}}\dot{T}_{in\_cm}+Q\dot{T}_{in\_cm}$$

Steady state compressor characteristics are described by two bivariate lookup tables:

- Compressor corrected mass flow map

$$\pi_{cm}=cmfmap\left(\bar{\dot{m}}_{cm},\bar{n}_{cm}\right) \quad (13)$$

- Compressor efficiency map

$$\eta_{cm}=cmemap\left(\pi_{cm},\bar{n}_{cm}\right) \quad (14)$$

where π_{cm} is compressor pressure ratio , $\bar{n}_{cm}$ is corrected speed and $\bar{\dot{m}}_{cm}$ corrected mass flow.

Compressor temperature rise is given by:

$$T_{out\_cm}=T_{in\_cm}\pi_{cm}^{\frac{(\gamma_{air}-1)}{\gamma_{air}\eta_{cm}}} \quad (15)$$

where γ - specific heat ratio varies with temperature, and is determined as the arithmetic average between T_{in_cm} evaluated at the current time step and T_{out_cm} obtained at the previous time step.

Generic equations for modelling compressor volume model take the following form:

$$\frac{\partial P_{out}}{\partial t}=\left[\begin{array}{l}\frac{R_{cm}T_{out\_cm}}{V_{cm}}\left(\sum_i \dot{m}^i_{in\_cm}-\dot{m}_{out\_cm}\right)\\ +\frac{P_{out\_cm}}{T_{out\_cm}}\frac{\partial T_{out\_cm}}{\partial t}\end{array}\right] \quad (16)$$

$$\frac{\partial T_{out}}{\partial t}=\frac{1}{C_{v\_cm}\rho_{cm}V_{cm}}$$

$$\left[\begin{array}{l}\left(\sum_i \dot{m}^i_{in\_cm}C^i_{p\_cm}T^i_{in\_cm}-\dot{m}_{out\_cm}C_{p\_cm}T_{out\_cm}\right)\\ +q-C_{v\_cm}T_{out\_cm}\left(\sum_i \dot{m}^i_{in\_cm}-\dot{m}_{out\_cm}\right)\end{array}\right]$$

where C_v is specific heat and q is heat flow. Variables with subscript x_{out} refer to the outlet stream and subscript x^i_{in} refers to inlet streams i=1 to n.

The energy transfer rate between metal and air in energy balance equation is defined as:

$$q=hA_s\left(T_{air}-T_m\right) \quad (17)$$

Convective heat transfer coefficient $h=N_u \, k/L$ is a function of Nusselt number N_u, but in this model simplified approach has been proposed. Heat transfer coefficient has been related to design point air temperature and mass flow values using following empirical formulation:

$$h=\left(\frac{T_{air}}{T_{d\_air}}\right)^x\left(\frac{\dot{m}_{air}}{\dot{m}_{d\_air}}\right)^y\frac{k}{L} \quad (18)$$

where k is thermal conductivity, L is a characteristic length and A_s is the surface area of metal exposed to air.

Implemented VGV offset correction modifies the compressor maps for small deviations from VGV referent position, i.e. standard schedule. VGV are used to optimize performance and operating stability of compressors. Compressors often operate with VGV positions differing from the nominal schedule, which is consistent with map. For multistage compressors with multiple variable stators, the variable stators are generally ganged together and first row of compressor stator blading (IGV) is used to represent the overall stator movement.

Traditionally compressor component maps used in 0-D performance models address VGV effects by providing separate maps for intermediate stator settings. To support low-computational time and robustness of numerical algorithm, a scaling method for mapping compressor characteristics with VGV offset angle has been implemented.

The proposed compressor scaling method is based on the return mapping algorithm using operator split approach, and is performed in two steps:

Operator split step, performs calculation of three correction factors using the corresponding correction maps:

- compressor pressure ratio correction factor

$$cf_{\pi_{cm}}=\frac{\pi_{cm,vgv}\left(\bar{n}_{cm}\right)}{\pi_{cm,ref}\left(\bar{n}_{cm}\right)}=vgvpmap\left(vgv,\bar{n}_{cm}\right) \quad (19)$$

where $cf_{\pi_{cm}}=1$ for $vgv=ref$, and vgv is VGV offset angle from VGV referent position.

- compressor mass flow correction factor

$$cf_{\dot{m}_{cm}}=\frac{\dot{m}_{cm,vgv}\left(\bar{n}_{cm}\right)}{\dot{m}_{cm,ref}\left(\bar{n}_{cm}\right)}=vgvfmap\left(vgv,\bar{n}_{cm}\right) \quad (20)$$

where $cf_{\dot{m}_{cm}}=1$ for $vgv=ref$.

- compressor efficiency correction factor

$$cf_{\eta_{cm}}=\frac{\eta_{cm,vgv}\left(\bar{n}_{cm}\right)}{\eta_{cm,ref}\left(\bar{n}_{cm}\right)}=vgvemap\left(vgv,\bar{n}_{cm}\right) \quad (21)$$

where $cf_{\eta_{cm}}=1$ for $vgv=ref$.

Return mapping step, calculates corrected values for compressor efficiency and mass flow as follows:

- compressor mass flow for compressor with ***vgv*** offset is determined using compressor flow characteristic:

$$\frac{\dot{m}_{cm,vgv}\sqrt{T_{in\_cm}}}{P_{in\_cm}} = cf_{\dot{m}_{cm}} \times \frac{\dot{m}_{cm,ref}\sqrt{T_{in\_cm}}}{P_{in\_cm}} = cf_{\dot{m}_{cm}} \times cmfmap\left(\frac{\pi_{cm,vgv}}{cf_{\pi_{cm}}}, \bar{n}_{cm}\right) \quad (22)$$

- efficiency of the compressor with ***vgv*** offset is determined using compressor efficiency characteristic:

$$\eta_{cm,vgv} = cf_{\eta_{cm}} \times \eta_{cm,ref} = cf_{\eta_{cm}} * cmemap\left(\frac{\pi_{cm,vgv}}{cf_{\pi_{cm}}}, \bar{n}_{cm}\right) \quad (23)$$

Compressor Instability Modelling

The different approaches to model the compressor non-steady fluid dynamics had been examined in the past [13]. Originally two-dimensional compressor flow-field had been used to describe compressor instabilities and to predict some of the basic characteristics of rotating stall [14-16]. Later further refinements had been introduced to capture the interaction between rotating stall and surge [17,18].

In this study simplified one-dimensional modelling technique has been used to describe compressor instabilities [19,20]. Nonlinear mathematical model of transient compressor dynamics presented in previous section has been expanded with system of equation to be able to describe instability behaviour of compression system.

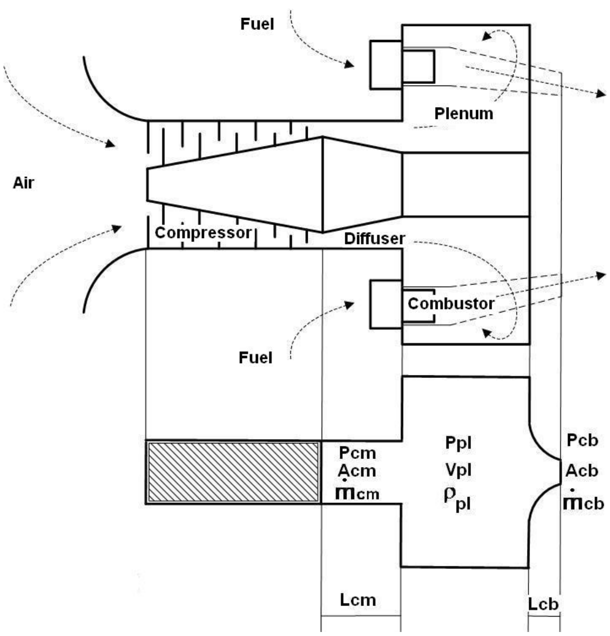

Fig. 3 Compression system scheme

Engine compression system consists of a compressor working in an annular duct, which is connected to an exit plenum of much larger diameter – centre casing (Fig. 3.). Air from the centre casing is discharged into combustion system, which consists of several combustion cans. Fuel and air are mixed and burnt in the combustors, where air is introduced into combustor through area, which is much smaller then equivalent area of the plenum.

In this configuration, oscillations occurring in compression system can be modelled in a manner analogous to those of a Helmholtz resonator. This assumes that all kinetic energy of oscillation is associated with the motion of the fluid in the compressor and combustor ducts, and the potential energy is associated with compression of the gas in the centre casing. The frequency of surge oscillation are typically over an order of magnitude less then those associated with the passage of the rotating stall cells.

A schematic representation of the compression system model is shown in Fig. 4. The compressor and its diffuser are modelled as described in previous section, where diffuser is represented as a length of constant area pipe to account for the dynamics of the fluid in the compressor duct. Similarly the air that enters the combustor through many holes can be represented by pressure drop through single duct of equivalent diameter and length.

Predicting the conditions at which instability will occur in a compressor requires an understanding of the flow process leading to the onset of the instability. The transition from initial disturbance to final stall or surge can usually be divided into three stages: inception, development and final flow pattern. The inception stage is the period when disturbances start to grow, taking them from a few to several hundred rotor revolutions to develop into final stall or surge. When stall occurs, pressure ratio of the compressor initially exceeds some critical value at a given speed, resulting in a subsequent reduction of compressor pressure ratio and airflow delivered to the engine combustor.

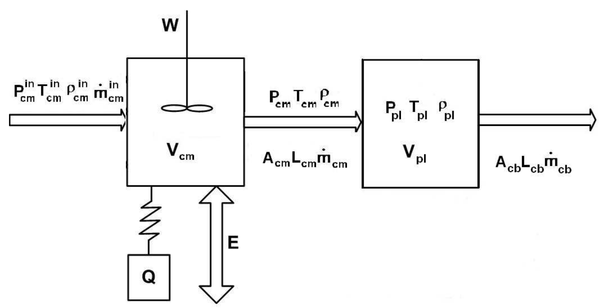

Fig. 4 Compression system modelling

The rate of change of mass flow in compressor duct can be related to the pressure difference across the duct and pressure raise across the compressor:

$$P_{cm} - P_{pl} = \rho_{cm} L_{cm} \frac{d\upsilon_x}{dt} \quad (24)$$

Substituting expression for compressor mass flow into above equation:

$$\frac{d\dot{m}_{cm}}{dt} = \rho_{cm} V_{cm} \frac{d\upsilon_x}{dt} \quad (25)$$

one can write:

$$\frac{d\dot{m}_{cm}}{dt} = \frac{A_{cm}}{L_{cm}}\left(P_{cm} - P_{pl}\right) \quad (26)$$

where υ_x air axial velocity, L_{cm} and A_{cm}, length and area of equivalent compressor duct.

An analogous one-dimensional momentum equation can be written for equivalent combustor duct:

$$\frac{d\dot{m}_{cb}}{dt} = \frac{A_{cb}}{L_{cb}}\left(P_{pl} - P_{cb}\right) \qquad (27)$$

where equivalent combustion duct is represented implicitly with equivalent duct length L_{cb} and area A_{cb}.

The continuity equation for centre casing – plenum takes following form:

$$\dot{m}_{cm} - \dot{m}_{cb} = V_{pl}\frac{d\rho_{pl}}{dt} \qquad (28)$$

Considering polytropic process in the plenum:

$$\frac{d\rho_{pl}}{dt} = \frac{\rho_{pl}}{\gamma P_{pl}}\frac{dP_{pl}}{dt} \qquad (29)$$

the density change in the plenum can be related to the plenum pressure by:

$$\frac{dP_{pl}}{dt} = \frac{\gamma P_{pl}}{\rho_p V_{pl}}\left(\dot{m}_{cm} - \dot{m}_{cb}\right) \qquad (30)$$

This is simple harmonic motion equation with resonant frequency of:

$$f = \frac{a}{2\pi}\sqrt{\frac{A_{cm}}{L_{cm}V_{pl}}} \qquad (31)$$

where speed of sound a is defined as $a = \sqrt{\gamma\frac{P_{pl}}{\rho_{pl}}}$.

Frequency of rotational stall is typically found in the range of 40% to 70% of compressor rotational speed, and instabilities with a quite low frequency generally can be associated with compressor surge.

NUMERICAL SIMULATION RESULTS

Model of single shaft gas turbine engine has been built and used for simulation of start-up sequence. During simulation of start-up, compressor instabilities have been induced to test proposed active control method.

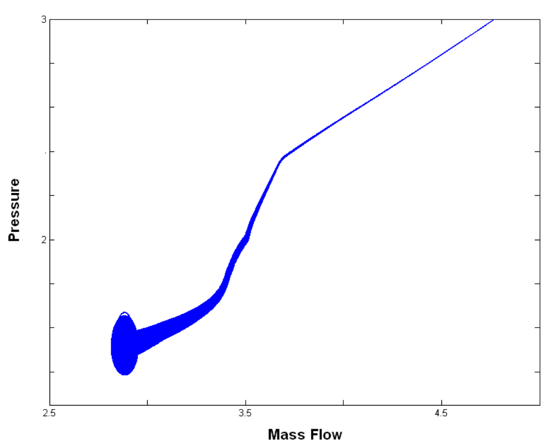

Fig. 5 Compressor instability during start-up

Because implemented model can not predict inception of compressor instabilities, compressor destabilization was induced by introducing sudden discontinuity of pressure at the outlet of compressor. This discontinuity was introduced as an impulse change of pressure loss leading to sharp drop in pressure, simulating formation of stall cells. In this way compressor instabilities are simulated as a sudden onset of pressure oscillations with high frequency, representing abrupt stall instabilities.

Transient response of destabilized compressor is shown in Fig. 5. Destabilization process is clearly observed as a high frequency oscillatory behaviour of compressor pressure and mass flow. Data are displayed in a compressor map type format in which instantaneous pressure change is plotted against the compressor mass flow.

One can observe how typical circular path, which describes excursions of compressor mass flow and pressure, is getting smaller after applying proposed active compressor stabilization method. Implemented start-up scheduling for simulated gas turbine engine was based on open-loop scheduling philosophy. This fuel demand scheduling was implemented into simulation model together with active compressor stability control method.

Dynamic pressure data from plenum have been used for generation of corrective action, simulating measurement of dynamic pressure in centre casing (Fig. 6a). Dynamic pressure signal has been processed using proposed technique to demonstrate applicability of this method in active control philosophy.

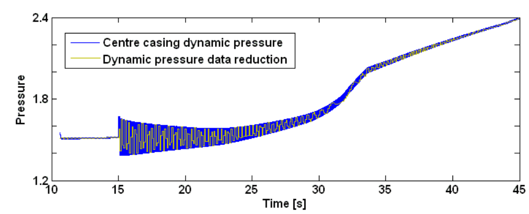

Fig. 6a Centre casing dynamic pressure & pressure data reduction

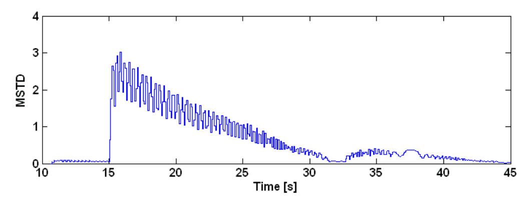

Fig. 6b MSTD dynamic pressure data processing

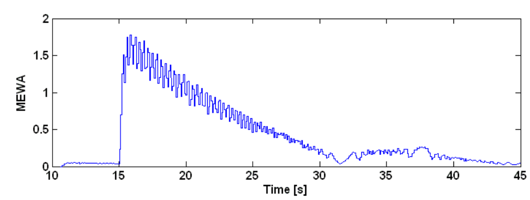

Fig. 6c MEWA dynamic pressure data processing

To simulate typical control system task rate, reduction of dynamic pressure data has been carried out. Dynamic pressure data were sampled every 100 ms to simulate typical measurement sampling rate (Fig. 6a). After applying pressure transducer data reduction, proposed monitoring of moving standard deviation (Fig. 6b) and moving exponential weighted average (Fig. 6c) of pressure transducer signal has been applied. Latter one has been used as a measure of compressor destabilization in corrective function (Fig.

7a), which has been utilized for generation of corrective actions, i.e. modification of fuel flow demand (Fig. 7b).

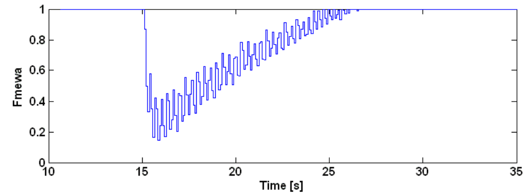

Fig. 7a Corrective function - ***Fmewa***

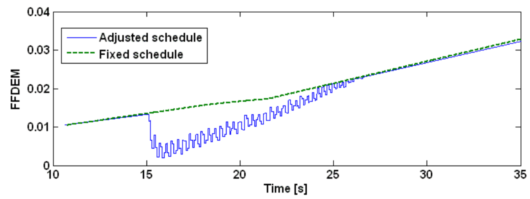

Fig. 7b Corrective action – adjusted FFDEM

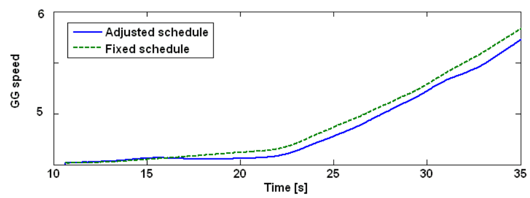

Fig. 7c Adjusted acceleration profile of gas generator

It can be observed that by adjusting the fuel flow schedule, i.e. reducing acceleration rate of gas generator (Fig. 7c) and increasing the compressor stall margin, stabilization process is taking place and leads to recovery from compressor stall instabilities (Fig. 5.).

CONCLUSIONS

The pre-scheduled metering of fuel for engine starting without dynamic adjustments for unpredictable influencing variables cannot assure reliable start-up. To prevent engine trips during starting and secure reliable stall-free start-up, active compressor stability control method has been proposed.

A new stall detection method, capable of detecting stall initiation, has been proposed. Developed stall detection method is used in active control philosophy, utilizing recovery method, which is based on the control of stall margin by adjusting the fuel flow schedule. Proposed generic active control method can be applied on both, open and closed start-up scheduling loops. Numerical simulation of active control method applied on the open start-up scheduling loop is presented in this paper.

Generic transient simulation tool GasTurboLib has been modified with capability to simulate flow instabilities within compressor component. Enhanced compressor component block has been used to build single shaft gas turbine engine model, and start-up sequence has been simulated. During simulation of start-up, compressor instabilities have been induced to test proposed active control method.

It has been demonstrated that proposed method can recover gas turbine from compressor instabilities during engine start-up, by sensing inception of stall instabilities and subsequently initiating control system corrective actions.

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