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GASTURBOLIB – SIMULINK LIBRARY FOR GAS TURBINE ENGINE MODELLING

V. Panov
vili.panov@siemens.com

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

ABSTRACT

A new Simulink library, called GasTurboLib, containing blocks specialized for gas turbine modelling has been developed. Different engine configurations can be generated using GasTurboLib components and these models can be used for steady state and transient performance analysis.

This paper describes the newly developed generic gas turbine simulation tool and presents experiences with modelling and simulation of single and twin shaft gas turbine engines. This library enables 0-D modelling, which is the simplest level of modelling but the most widely used in industry. This component-based modelling environment can be used to simulate start-up sequence, load change, control system design, power-system stability studies and real-time modelling.

Traditionally, control method improvements are developed and validated through engine testing. The goal was to develop a functional engine model, which can be started, operated and shut down by a governor model, for the purposes of development of control methods and protection algorithms, thus providing considerable cost savings, as well as enabling better project progress through independence from the availability of test beds.

It has been demonstrated that rapid model generation and reusability of components along with user-friendly graphical user interface make this simulation environment a valuable tool for gas turbine system performance analysis.

INTRODUCTION

Simulation models for gas turbine engines are extensively used today, because they reduce the time, cost and risk of product development. There are a number of proprietary gas turbine specific simulation tools used by industry and research institutions [1-3], and several commercially available programs such as GasTurb [4], GSP [5], etc. Usually commercially available tools suffer from drawbacks like predefined gas turbine configurations and restricted availability of non-standard engine components. To exploit open and extensible architecture, an option for developing an engine simulation model is to use generic simulation tools such as ACSL, Simulink [6-8], EcosimPro [9,10], and Modelica [11], where the user can create engine components and then combine them either programmatically or visually. All these packages have a component-oriented architecture where an engine model could be composed of modular blocks representing individual components. In this work the commercial package Matlab with Simulink, for simulation of dynamic systems and processes, has been chosen as a developing platform and a library called GasTurboLib, which contains framework for gas turbine modelling, has been developed.

NOMENCLATURE

Variables

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

Indices

air	air
amb	ambient
av	air valve
comb	combustor
comp	compressor
d	design point
fuel	gas fuel
fv	fuel valve
gas	hot gas
in	inlet
inlet	inlet duct
m	metal
out	outlet
p	const pressure
pte	power turb exit
s	surface
t	turbine
tc	thermocouple
v	const volume
vgv	variable guide vanes

GASTURBOLIB LIBRARY

The objective of this work was development of component based modelling tool with flexible architecture, to enable dynamic simulation of different gas turbine engine configurations. For that purpose a new software library was developed to be used with Simulink user-friendly drag & drop simulation environment. Developed GasTurboLib library is presented in figure Fig. 1. The main purpose of this tool is to study gas turbine engine transient responses under different working conditions.

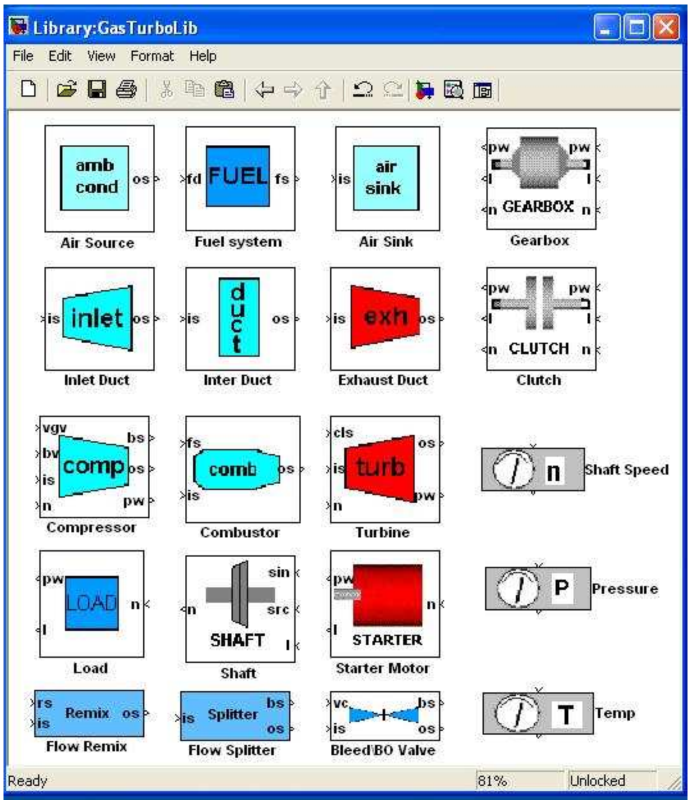

Fig. 1 GasTurboLib Library

A modular software architecture was adopted, because this structure provides modelling flexibility and secures reusability of developed component models. This modular approach, where each module represents typical component, also offers significant benefits in terms of software development and maintenance effort. Simulink's component-oriented architecture enhances code maintainability and readability by concentrating both the routine code and data code of a particular component in a single library block (Fig. 2).

Another important advantage of generic simulation tools like Matlab-Simulink and similar environments is that they usually employ "auto-solvers", where user only needs to specify the required accuracy of numerical methods. Usually several integration algorithms are available, and the most appropriate for the particular dynamic model, can be chosen.

Highly flexible architecture of this simulation tool supports rapid 0-D modelling and enables following:
- development and improvement of current control laws,
- analysis of engine transient performance responses,
- interpretation of faults in operation ,
- plant response studies, etc.

As other 0-D simulation tools [12], GasTurboLib offers building gas turbine models, where averaged fluid characteristics of engine components are computed at discrete positions inside the engine - at the inlet and the outlet of each engine component. Library blocks could be considered a set of black boxes, one for each component of the engine, and they do not require a detailed description of the engine component geometry.

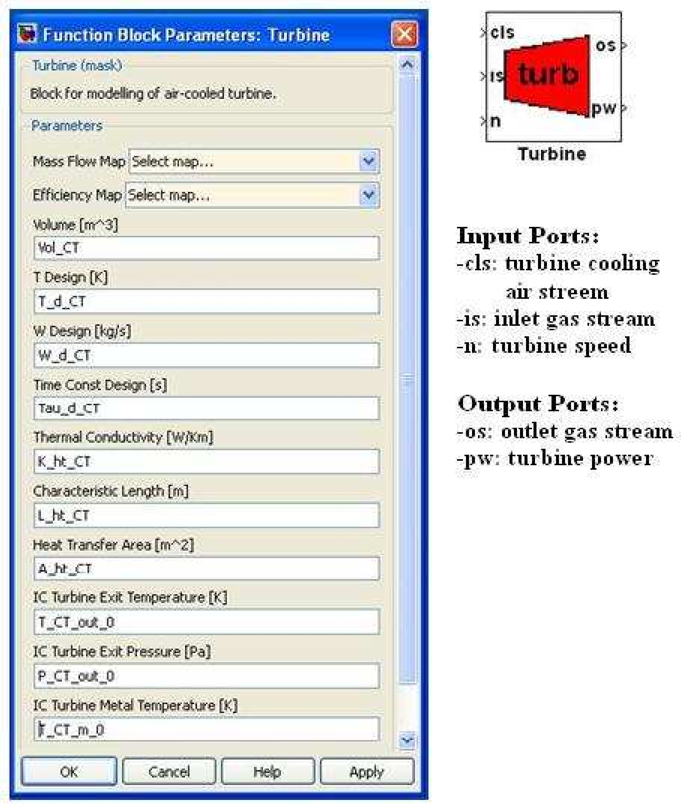

Fig. 2 Turbine component icon and block dialog

This software library has been developed, with the intention that this should be common tool for the model developers, engineering designers, and the gas turbine customers. Therefore, fully open architecture makes code evolution easier and enables implementation of high order mathematical models (e.g. 1-D models), allowing high-fidelity simulation of local phenomena in the engine, together with lower fidelity simulation of the rest of the gas turbine. Using this "zooming" concept, reduced computing power and complexity could be achieved where low order fidelity analysis is sufficient.

GENERIC COMPONENT MODELLING

GasTurboLib library can be used to model the following gas turbine engine systems: mechanical, fuel, control system and gas path components. To ensure simplicity and consistency, a separate set of routines is used to describe thermodynamic properties of working fluid in the engine for the entire simulation. Three gas compositions have been considered, two for calculations prior to the combustion process: air & natural gas and third for calculations of properties of combustion products - hot gas. In each case the gas properties, i.e. specific heat (p=const) and specific heat ratio, are characterised as functions of gas temperature by means of polynomial equations [13]. Ambient conditions module is modeled with ISO Atmosphere with adjustable variations from standard conditions.

GAS PATH COMPONENTS

A gas path component can be seen as an operator, whose purpose is to compute the thermodynamic state of the fluid stream at the outlet of the module. Each of the gas streams linking the gas path components represents a vector of gas properties. Component models calculate values for their output stream elements based on their current state and values in the input stream. Each gas stream has 4 elements: total pressure, temperature, density and mass flow.

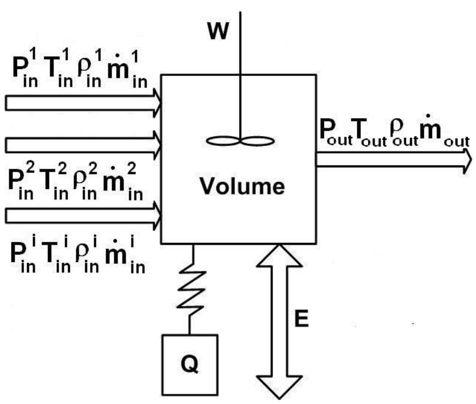

Fig. 3 Generic gas path component

A gas volume with inlet and outlet flows, heat transfer to adjacent metal structure (Q), mechanical work (W) and energy dissipation through component losses (E), are features of all gas path engine components. The generic gas path component is shown in Fig. 3. The component modelling relies on the conservation equations and on empirical information derived from rig tests (e.g. compressor and turbine maps).

In the developed modelling framework each component is represented with control volume and flow model. A flow model is the result of a modelling abstraction, where the volume is neglected. These flow models contain algebraic equations that describe the flow of mass and energy.

To model the dynamic behavior of the individual engine component following generic equations for pressure and temperature have been used:

$$\frac{dP_{out}}{dt} = \left(\frac{\partial P_{out}}{\partial t}\right)_v + \sum_j \frac{\partial P_{out}}{\partial \varsigma_j}\frac{\partial \varsigma_j}{\partial t} \tag{1}$$

$$\frac{dT_{out}}{dt} = \left(\frac{\partial T_{out}}{\partial t}\right)_v + \sum_l \frac{\partial T_{out}}{\partial \xi_l}\frac{\partial \xi_l}{\partial t} \tag{2}$$

where ς_j and ξ_l are internal thermodynamic state and mechanical variables used to model irreversible or dissipative processes. The first term in above equations Eq. (1) and Eq. (2), 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.

Gas dynamics is mathematically modelled by sets of differential equations used to describe accumulation of the gas energy and mass within major volumes of the engine [14]:

- Conservation balance of total mass:

$$\frac{\partial m}{\partial t} = \sum_i \dot{m}_{in}^i - \dot{m}_{out} \tag{3}$$

- Conservation balance, which was derived from ideal gas equation ($p = mRT\ /V$):

$$\frac{\partial P_{out}}{\partial t} = \underbrace{\frac{RT_{out}}{V}\left(\sum_i \dot{m}_{in}^i - \dot{m}_{out}\right)}_{a} + \underbrace{\frac{P_{out}}{T_{out}}\frac{\partial T_{out}}{\partial t}}_{b} \tag{4}$$

First term in Eq. (4) represents the change in pressure caused by the increased mass in the volume, and second term is the increase in pressure caused by the change in volume temperature. Under steady state conditions both of these terms approach zero (a -> 0 and b -> 0).

- Conservation balance of energy:

$$\frac{\partial T_{out}}{\partial t} = \frac{1}{C_v \rho V}\left[\begin{array}{c}\underbrace{\left(\sum_i \dot{m}_{in}^i C_p^i T_{in}^i - \dot{m}_{out} C_p T_{out}\right) + q}_{c} \\ \underbrace{- C_v T_{out}\left(\sum_i \dot{m}_{in}^i - \dot{m}_{out}\right)}_{d}\end{array}\right] \tag{5}$$

where C_p and C_v are specific heats, and q is heat flow. Variables with no subscript refer to properties within the gas volume. Variables with subscript X_{out} refer to the outlet stream and subscript X^{i}_{in} refers to inlet streams i=1 to n. The first term in Eq. (5) represents the energy flow entering the volume, and second term, the increase in volume internal energy rate. Under steady state conditions the following conditions are satisfied: c -> 0 and d -> 0.

Generic modeling equations Eq. (1) – Eq. (5), are applied across defined volumes, representing gas pat components. Order of proposed model could be increased by implementation of dynamic momentum balance into modelling framework. With addition of momentum equation into flow model, length term is introduced, and hence the model would be regarded as a 1-D representation.

The energy transfer rate between metal and gas in energy balance equation Eq. (5) is defined as:

$$q = hA_s\left(T_{gas} - T_m\right) \tag{6}$$

Convective heat transfer coefficient $h = N_u \, {}^{k}\!/_{L}$ is a function of non dimensional numbers, Prandtl, Reynolds, etc. In this model simplified approach has been proposed, and heat transfer coefficient has been related to design point gas temperature and mass flow values using following empirical formulation:

$$h = \left(\frac{T_{gas}}{T_{d\_gas}}\right)^{x}\left(\frac{\dot{m}_{gas}}{\dot{m}_{d\_gas}}\right)^{y}\frac{k}{L} \tag{7}$$

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

To model heat transfer effects between the gas and the metal, a dynamic equation of metal temperature is used

$$\frac{dT_m}{dt} = \frac{(T_{gas} - T_m)}{\tau_m} \tag{8}$$

In above equation Eq. (8), T_m is metal temperature, T_{gas} is volume gas temperature, and τ_m is a component thermal time constant defined as:

$$\tau_m = \frac{M_m C_m}{hA_s} \tag{9}$$

where M_m is metal mass and C_m metal specific heat.

Substituting expression for heat transfer coefficient Eq. (7), into above equation Eq. (9), one can obtain relation for component thermal time constant in the following form:

$$\tau_m = \tau_{d\_m}\left(\frac{T_{gas}}{T_{d\_gas}}\right)^{-x}\left(\frac{\dot{m}_{gas}}{\dot{m}_{d\_gas}}\right)^{-y} \tag{10}$$

where $\dot{m}_{gas}$ is the gas mass flow, T_{d_gas} and $\dot{m}_{d\_gas}$ are design point gas temperature and mass flow respectively, and the design point thermal time constant is defined as follows:

$$\tau_{d\_m} = \frac{M_m C_m}{A_s \, {}^{k}\!/_{L}} \tag{11}$$

A brief description of the major gas path engine components included in the library is given below.

A ***compressor*** is modeled as an ideal component with compressor body and exit volume – diffuser, which is represented as a lumped element at the compressor outlet. Heat transfer is included between gas and metal, together with an inter-stage cooling air bleed, and a Variable Guide Vanes (VGV) offset correction. Influence of diffuser is modeled with pressure loss factor. 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.

Compressor temperature rise is given by:

$$T_{out\_comp} = T_{in\_comp}\,\pi_{comp}^{\frac{(\gamma_{air}-1)}{\gamma_{air}\eta_{comp}}} \tag{12}$$

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

Steady state compressor characteristics are described by two bivariate lookup tables, as a function of compressor pressure ratio $\left(\pi_{comp} = P_{out\_comp}/P_{in\_comp}\right)$ and compressor corrected speed $\left(\overline{n}_{comp} = n_{comp}/\sqrt{T_{in\_comp}}\right)$:

- Compressor corrected mass flow map

$$\frac{\dot{m}_{air}\sqrt{T_{in\_comp}}}{P_{in\_comp}} = cmfmap\left(\pi_{comp}, \overline{n}_{comp}\right) \tag{13}$$

- Compressor efficiency map

$$\eta_{comp} = cmemap\left(\pi_{comp}, \overline{n}_{comp}\right) \tag{14}$$

For given compressor temperature rise, the mechanical power input to the compressor is calculated using compressor specific work W_{comp} as follows:

$$PW_{comp} = \dot{m}_{comp}W_{comp} = \dot{m}_{comp}\int_{T_{in\_comp}}^{T_{out\_comp}} C_{p\_air}\,dT \tag{15}$$

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 proposed.

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

$$cf_{\pi_{comp}} = \frac{\pi_{comp,vgv}(\bar{n}_{comp})}{\pi_{comp,ref}(\bar{n}_{comp})} = vgvpmap\left(vgv, \bar{n}_{comp}\right) \quad (16)$$

where $cf_{\pi_{comp}} = 1$ for $vgv = ref$, and vgv is VGV offset angle from VGV referent position.
- compressor mass flow correction factor

$$cf_{\dot{m}_{comp}} = \frac{\dot{m}_{comp,vgv}(\bar{n}_{comp})}{\dot{m}_{comp,ref}(\bar{n}_{comp})} = vgvfmap\left(vgv, \bar{n}_{comp}\right) \quad (17)$$

where $cf_{\dot{m}_{comp}} = 1$ for $vgv = ref$.
- compressor efficiency correction factor

$$cf_{\eta_{comp}} = \frac{\eta_{comp,vgv}(\bar{n}_{comp})}{\eta_{comp,ref}(\bar{n}_{comp})} = vgvemap\left(vgv, \bar{n}_{comp}\right) \quad (18)$$

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

▪ Return mapping step, calculates corrected values for compressor efficiency and mass flow :
- compressor mass flow for compressor with ***vgv*** offset is determined using compressor flow characteristic as follows:

$$\frac{\dot{m}_{comp,vgv}\sqrt{T_{in\_comp}}}{P_{in\_comp}} = cf_{\dot{m}_{comp}} \times \frac{\dot{m}_{comp,ref}\sqrt{T_{in\_comp}}}{P_{in\_comp}}$$
$$= cf_{\dot{m}_{comp}} \times cmfmap\left(\frac{\pi_{comp,vgv}}{cf_{\pi_{comp}}}, \bar{n}_{comp}\right) \quad (19)$$

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

$$\eta_{comp,vgv} = cf_{\eta_{comp}} \times \eta_{comp,ref}$$
$$= cf_{\eta_{comp}} * cmemap\left(\frac{\pi_{comp,vgv}}{cf_{\pi_{comp}}}, \bar{n}_{comp}\right) \quad (20)$$

A ***turbine*** is modelled with the turbine body representing the steady state turbine model and interconnecting volumes. The steady performance characteristics are not able to handle transient events, but if they are coupled with volume dynamics, which continuously feed the steady state maps with unsteady data, satisfactory dynamic responses are achievable [15].
Heat transfer between gas in the volume and the metal components is modelled to represent heat soak effects, and mixing of compressor cooling air at turbine inlet is also considered.

The turbine is represented by two steady state maps, which are based on turbine pressure ratio $\left(\pi_t = P_{in\_t}/P_{out\_t}\right)$ and turbine corrected speed $\left(\bar{n}_t = n_t/\sqrt{T_{in\_t}}\right)$:
- turbine corrected mass flow map

$$\frac{\dot{m}_t\sqrt{T_{in\_t}}}{P_{in\_t}} = tfmap\left(\pi_t, \bar{n}_t\right) \quad (21)$$

- turbine normalized efficiency

$$\eta_t = temap\left(\pi_t, \bar{n}_t\right) \quad (22)$$

For given pressure ratio and corrected speed, using turbine steady state characteristics, values for turbine corrected mass flow and efficiency are evaluated through interpolation on digitized maps. Turbine temperature drop is calculated using specific heat ratio evaluated at the mean temperature between the inlet and outlet of the turbine component as follows:

$$T_{out\_t} = T_{in\_t}\pi_t^{\frac{\eta_t(1-\gamma_{hot})}{\gamma_{hot}}} \quad (23)$$

Knowing turbine temperature drop, generated power can be obtained using turbine specific work W_t :

$$PW_t = \dot{m}_t W_t = \dot{m}_t \int_{T_{in\_t}}^{T_{out\_t}} C_{p\_hot} dT \quad (24)$$

The **combustion system** typically consists of multiple combustors, but they can be lumped together and represented by a single combustor having equivalent properties. Therefore, the combustor is modelled as a single volume with a single outlet gas stream, which defines properties of combustion products. This component has two inlet streams, where the first represents fuel and the second air gas stream. A combustor component increases the enthalpy of the working fluid through the combustion of fuel, hence conservation balance of internal energy for combustion chamber takes following form:

$$\frac{\partial T_{out\_comb}}{\partial t} = \frac{1}{C_v \rho V_{comb}} \begin{bmatrix} \dot{m}_{air} C_{p\_air} T_{air} + \dot{m}_{fuel} C_{p\_fuel} T_{fuel} \\ -\dot{m}_{out\_comb} C_{p\_gas} T_{out\_comb} + H_{fuel} \sigma_{comb} \dot{m}_{fuel} \\ +q - C_v T_{out\_comb} \left(\dot{m}_{air} + \dot{m}_{fuel} - \dot{m}_{gas} \right) \end{bmatrix} \quad (25)$$

where H_{fuel} is the lower calorific value of fuel and σ_{comb} is the variable combustor efficiency. The pressure loss factor is considered to account for the friction losses.

Inlet, **exhaust** and **inter-duct** are represented as a volume components including heat soakage. Gas energy dissipation for these modules, is considered through component pressure losses factors. **Gas splitter** and **remix** modules are modeled as volume-less components using conservation constraints for mass and energy.

Bleed and **Blow-off valves** are approximated with an orifice having the effective area using generic valve model. Valve position is calculated in the governor. Variable bleed valve and variable blow-off valves are modelled using valve mass flow equation, which is defined as:

$$\dot{m}_{av} = K_{\dot{m}_{av}} S_{av} \frac{P_{out\_comp}}{\sqrt{T_{out\_comp}}} \quad (26)$$

where $K_{\dot{m}_{av}}$ is valve flow coefficient, S_{av} is valve effective area, T_{out_comp} is fluid temperature and P_{out_comp} is fluid pressure.

MECHANICAL SYSTEM

The mechanical system is described with conservation of mechanical energy through usage of differential equation of the shaft rotation:

$$\frac{dn}{dt} = \frac{\sum_{o} PW_{source} - \sum_{p} PW_{sink}}{n \left(\frac{\pi}{30} \right)^2 \sum_{r} I} \quad (27)$$

where $\sum_{o} PW_{source}$ is sum of power sources, $\sum_{p} PW_{sink}$ sum of power sinks, and $\sum_{r} I$ is total shaft inertia.

Mechanical system of the engine is modelled using following blocks: shaft, gearbox, load, clutch and starter. Vectors of mechanical variables are passed between the components of the model. Vectors contain three components: rotational speed, inertia and shaft power. The power and inertia variables propagate through the mechanical system, with each component adding or subtracting power and inertia, and in the case of the gearbox multiplying power and inertia by appropriate constants. Speed is calculated in the shaft block and it propagates through mechanical system components unchanged, excluding gearbox component where is multiplied by the gear ratio. For each component variable mechanical losses are also considered.

FUEL SYSTEM

The fuel system is represented only with the fuel valve and physical properties of the fuel. The dynamics of the fuel valve itself is described by transport delay and a first-order non-periodic model with a time constant:

$$\tau_{fv} \frac{d\dot{m}_{fuel}}{dt} + \dot{m}_{fuel} = G_{fv} \times FFDEM \quad (28)$$

where FFDEM is control system fuel demand, τ_{fv} and G_{fv} are fuel valve time and gain constants respectively.

CONTROL SYSTEM

The control system is modeled using the governor model representing various control loops and engine sensors. The following sensors have been modelled: thermocouples, speed probes and pressure transducers. All sensors are modelled as first-order models with transport delay. Applied modelling technique is demonstrated using thermocouple sensor, which is represented with dynamic equation of thermocouple temperature:

$$\frac{dT_{tc}}{dt} = \frac{T_{m\_gas} - T_{tc}}{\tau_{tc}} \quad (29)$$

where T_{tc} is thermocouple temperature, τ_{tc} is thermocouple time constant and T_{gas} is gas temperature for the thermocouples which are positioned in engine hot gas path.

NUMERICAL SIMULATIONS

To build the gas turbine model, the blocks presented in the previous sections are simply assembled following the required engine configuration. In this way a nonlinear dynamic model utilizing algebraic and ordinary differential equations for conservation of mass, energy and momentum is created.

The Simulink environment employs an “auto-solver” facility, which during the simulation run provides integration of differential equations using numerical method chosen by user. A fourth order Runge–Kutta explicit one-step integration algorithm has been used for solving system of differential equations in presented simulation examples. A maximum integration step time of 1 ms has been used to secure numerical stability of solution at every time step.

Two transient cases with conceptually different engine configurations were dynamically simulated, and obtained simulation results are compared against engine test data.

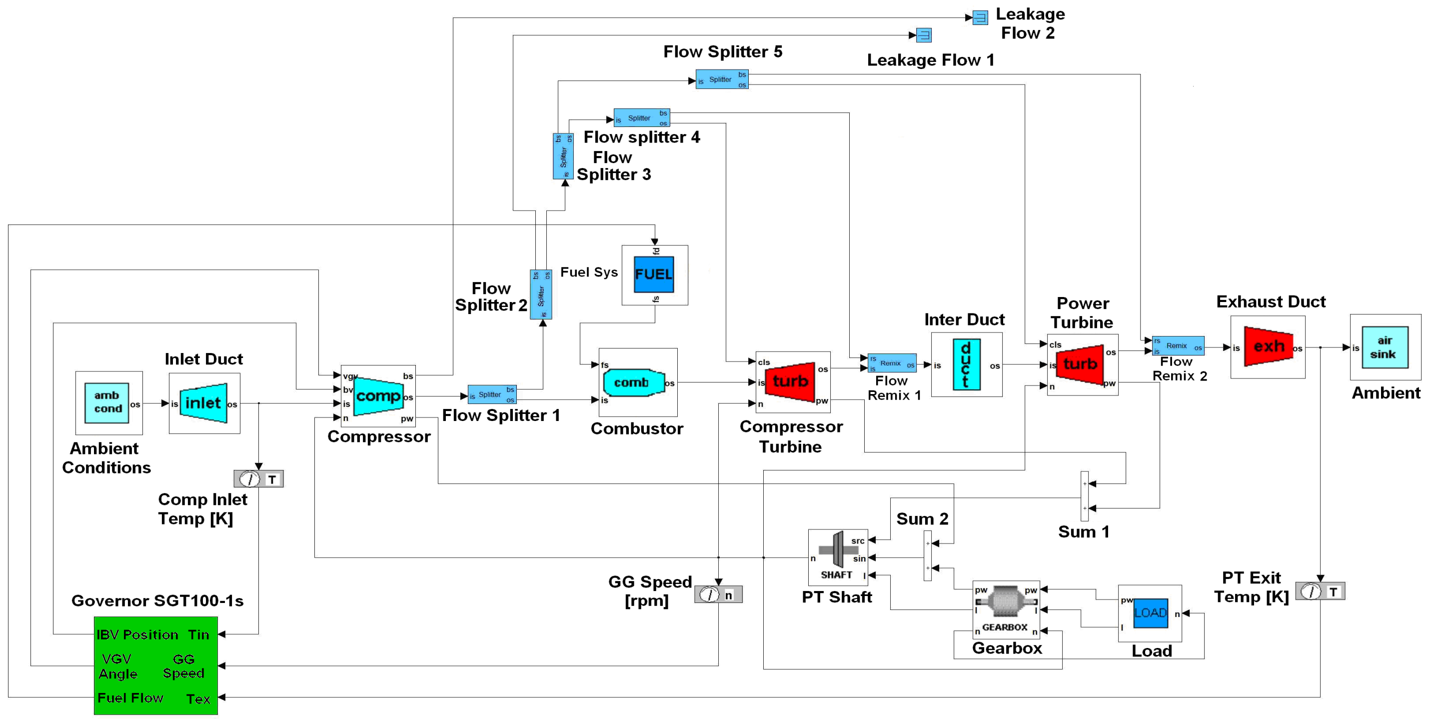

Fig. 4 Single-shaft Simulink model

SINGLE SHAFT MODEL / SGT-100-1S

A simulation model has been built for SGT-100-1s single shaft industrial gas turbine using GasTurboLib components (Fig. 4). Validation has been carried out using full load acceptance engine test. For this purpose the governor model has been integrated into simulation model of the gas turbine.

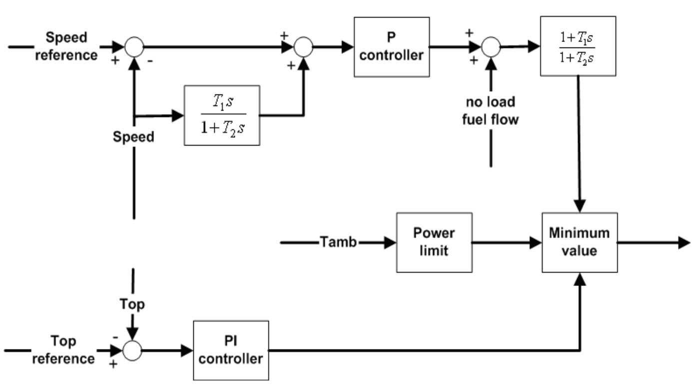

Fig. 5 FFDEM logic

The governor model has been implemented with fuel flow demand (FFDEM) and variable guide vane demand (VGVDEM) logic. The governor model has a performance prediction block, which estimates Turbine Operating Temperature. Performance calculations in control limits are used mainly for prediction of non-measured variables, and for this engine Turbine Operating Temperature is calculated as:

$$T_{OP} = T_{pte} - T_{inlet} + K_{T_{op}} \left(T_{inlet} - T_{amb} \right) \qquad (30)$$

where T_{pte} is average power turbine exit temperature, T_{inlet} is average inlet temperature, T_{amb} ambient reference temperature and K_{Top} is constant dependant on the fuel type.

Control loops in fuel control logic limit engine variables by imposing governing limits (GovLim) on fuel flow demand, and basic FFDEM logic is presented in Fig. 5. The following control limiting loops have been modelled:

- Gas Generator speed droop controller (GovLim 0),
- Turbine Operating Temperature limit controller (GovLim 2),
- Power output limit (GovLim 4),

The outputs of control loops are the input in a minimum value selector, so that the loop, which takes control, is the one, whose output is the lowest.

Droop speed control is the main control loop during normal operating conditions. The input to this control is the speed deviation. Proportional controller is applied on compensated speed error, i.e. speed deviation, which is compensated for the rate of change of speed.

Temperature limit controls the fuel flow to limit the turbine operating temperature. The operating temperature is compared against a reference, which is calculated as an offset to the turbine operating temperature limit for shutdown. A minimum limit is applied to the resulting fuel demand to prevent it falling too low.

Power limit provides a maximum continuous fuel flow limit for the engine for the present ambient temperature. This limit may however be exceeded for a short duration.

Compressor air control logic has been modelled with VGV demand, and simplified VGV control logic is presented in Fig. 6. In this arrangement VGV demand is determined from a least select from the following schedules:

- VGV Speed schedule
- VGV Temperature schedule

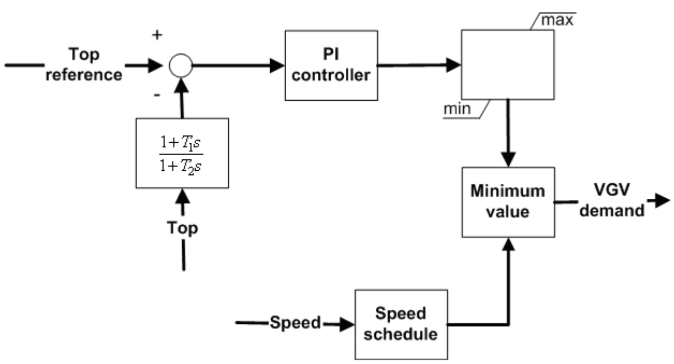

Fig. 6 VGVDEM logic

VGV speed schedule calculates the required position for the VGV's. On a start or a normal shutdown, this is taken from one of two maps against speed.
VGV temperature schedule is based on the Turbine Operating Temperature. VGV's are opened or closed by a proportional plus integral controller to attempt to maintain a constant Top temperature. On fast load changes, however, the VGV's are limited against fuel demand.

Transient behavior of engine during full load acceptance has been simulated. The test data was gained from load acceptance test performed at site, where engine was running an alternator into resistive island load using local load banks. During a load acceptance an initial rapid fall in GG speed can be observed. The action of the governor is to increase the fuel flow to the engine to recover GG speed, and hence subsequently fuel flow demand reaches maximum value defined by power limit control loop. Rapid application of load and increase of fuel demand causes VGV's to move to fully open position.

Simulation results presented in Fig. 7, show that predicted numerical results are in good agreement with real engine data. Simulated governor output variables have been compared with real governor response and accurate prediction for governing limits, fuel flow and variable guide vanes demand was obtained.

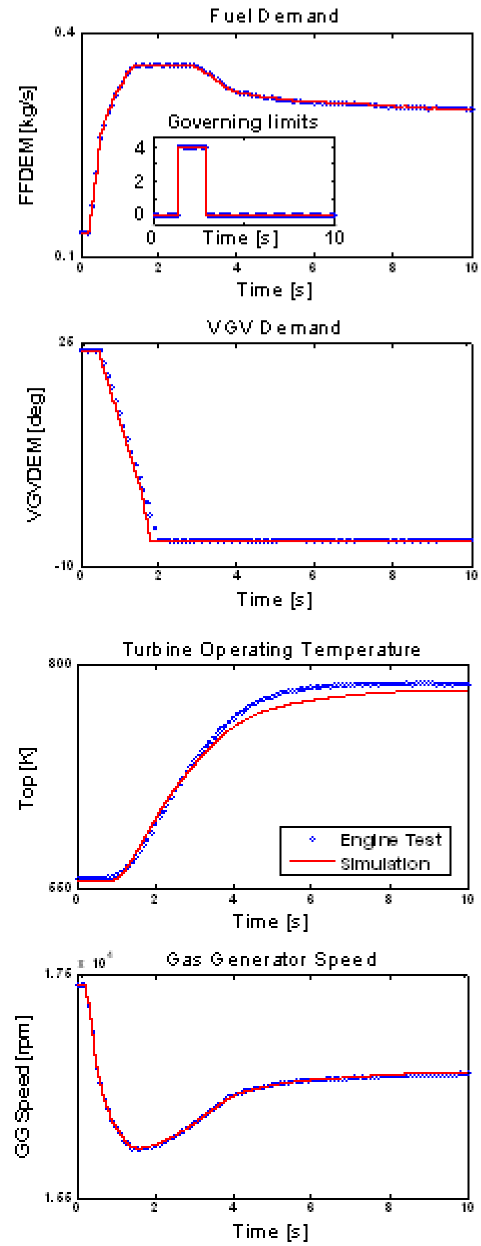

Fig. 7 Load acceptance test

Also the following engine transient responses have been compared with test data: turbine operating temperature as a non-measured variable and Gas Generator (GG) speed as a directly measured engine variable. These two engine parameters are essential for engine control system, and one can observe that numerical simulation results are fairly close to engine test results.

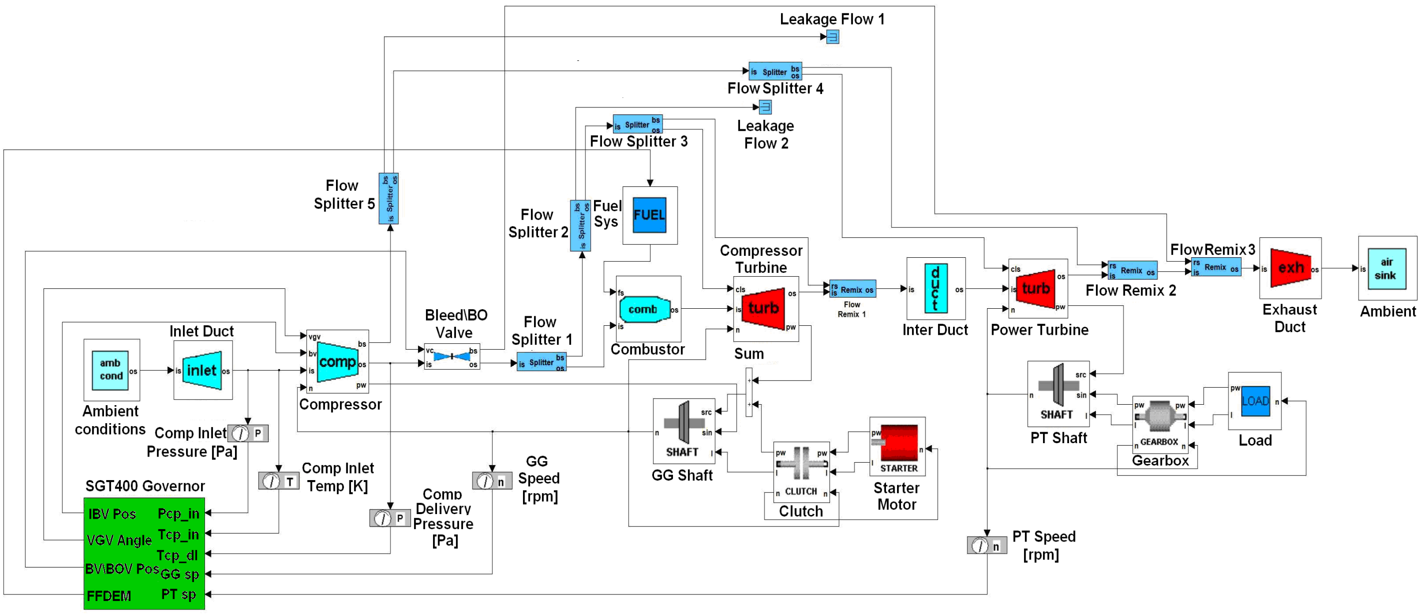

Fig. 8 Twin-shaft Simulink model

TWIN SHAFT MODEL / SGT-400

A twin shaft configuration has been built using the GasTurboLib library and SGT400 gas turbine has been modelled. The model of SGT400 industrial gas turbine engine is presented in Fig. 8.

In this exercise the objective was to develop and validate a model of start-up governor for SGT-400. The governor has been implemented as an additional block into the simulation model. Implemented start up governor model for SGT-400 consists of combustor fuel-flow and compressor air- flow control logic.

A simplified representation of fuel flow control logic is shown in Fig. 9. Typically fuel flow logic consists of following control limiting loops:

- Gas Generator Acceleration Limit (GovLim 1),
- Gas Generator Speed Limit and (GoLim 2),
- Power Turbine Droop Speed Governing (GovLim 3).

Governing limit (GovLim) selection identifies which controlling limit has priority over the others at any given time and sets the fuel demand accordingly. During an engine start the GG speed reference is set to the GG idle speed.

The engine will accelerate from ignition under a closed loop with GG acceleration controller towards idle. Once GG idle speed is achieved the GG speed controller becomes the governing controller. This will maintain the idle speed until GG speed reference starts to ramp from idle up to nominal GG speed at a pre-set rate. At sometime during this ramp, depending on driven shaft loading/inertia, PT speed will reach its min governing point and the gas generator speed loop will be relieved of control.

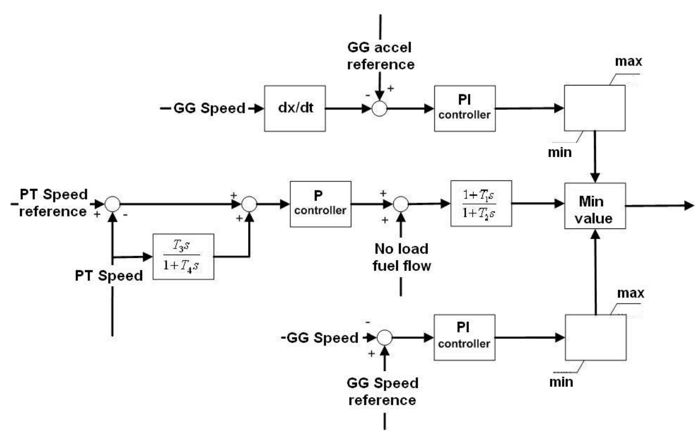

Fig. 9 Start-up FFDEM logic

GG Acceleration control loop governs transition from successful ignition to idle speed. This is a P+I controller controlling to an acceleration limit, which is a function of the corrected gas generator speed. The fuel demand range from the controller has strict dynamic limits applied.

GG generator speed governing is a maximum fuel limit that restricts the gas generator speed to a safe value. It is implemented as a P+I controller operating on speed error.

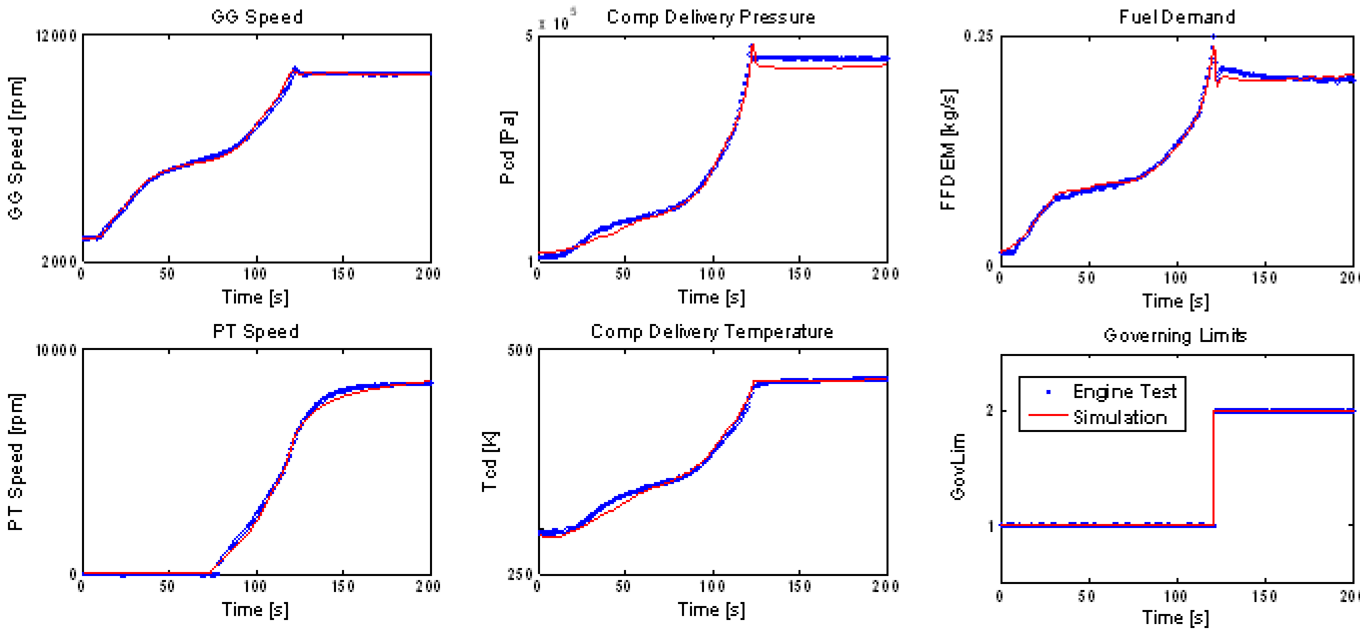

Fig. 10 Twin-shaft start-up test

Droop Power Turbine (PT) speed control loop sets a fuel demand for the turbine based on the difference between the reference speed and the actual speed and also adds a compensation factor to this for the rate of change of speed. This compensated speed error is multiplied by the droop gain and added to an offset for fuel flow at no load. The result is fed through a filter to give the droop mode fuel demand.

Air flow logic consists of the following schedules: Variable Guide Vanes, Blow-off and Interstage\Exit Compressor Bleed schedule. All these schedules have constant values during start up and therefore they have been modelled as constant demand values.

The model of the SGT400 gas turbine with governor has been validated using engine start-up test data. First 200 seconds of engine start have been simulated and validation results are presented in figure Fig. 10. Test data have been obtained form start-up test performed on the test bed using hydro-start system. During starting, gas generator is accelerated to ignition speed, and after light-up detection, hydraulic starter provides acceleration assistance to GG shaft. At some point sufficient torque is generated on free turbine and hence power turbine will breakaway PT shaft from stationary. When gas generator self-sustaining speed is reached, starter motor is disengaged, and GG shaft continues to accelerate according to a pre-set rate towards idle. Simulation has been carried out up to point where GG idle speed is achieved allowing engine to warm-up before PT speed governing takes over control. During simulated start-up sequence, inter-stage bleed valve is at fully open position, and VGV's are fully closed.

Governor model output variables have been compared with real governor response and one can observe good agreement between fuel flow control parameters: governing limits and fuel flow demand. Comparison of simulation and engine test data, for GG speed, PT speed, compressor delivery pressure & temperature, during start are also presented in Fig. 10, and good transient prediction of simulated variables is observed.

CONCLUSIONS

The GasTurboLib Simulink library has been created for generic gas turbine engine modelling. This was demonstrated by building single-shaft and twin-shaft engine configurations from developed GasTurboLib library blocks. The simulation results have shown that, the simulation tool is suitable for gas turbine modelling in component oriented way. Distinctive transient events have been simulated and high degree of agreement between simulation and engine test results has been observed.

Simulation models for SGT100-1s and SGT400 governors have been implemented into simulation models and they have been validated using load acceptance and start-up engine tests. Advantage of engine simulation models, created using a newly developed Simulink library GasTurboLib has been demonstrated. Fully open and extensible architecture of this simulation tool enables rapid & flexible modeling, and secures reusability of developed component models.

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