Combined 1D & 3D CFD Approach for GT Ventilation System analysis

The Gas Turbine ventilation system is designed to supply the necessary amount of air for cooling and to prevent the accumulation of hazardous gases in the enclosure by maintaining a slight over-pressure. The classical GE approach to studying ventilation system operating conditions consists of modeling the whole system as a series of discrete losses, where the ASHRAE duct-fitting database provides the corresponding pressure loss coefficients. The system is solved by means of a one-dimensional flow simulation tool (Flowmaster).
The goal of this work was to improve the critical points that affect the above-mentioned procedure, such as modeling of complex fittings and bend interactions. For this purpose, dedicated CFD analyses were performed to characterize the loss coefficient for splitting and bend interactions at different operating conditions (split percentage and inlet flow rate) for two different ventilation systems. The resulting loss coefficient curves have been implemented within the corresponding one-dimensional Flowmaster models. Finally, to characterize off-design conditions, a variable Heat Rejection model (obtained from previous CFD analyses) and real fan curves were used.

This new approach produces more accurate results, as confirmed by the close agreement with experimental measurements. Among the benefits of using this new approach is the ability to characterize the flow behavior of complex fittings. This would be useful in the event of a fitting redesign or for noise reduction analyses.

Current GE approach to studying Gas Turbine Ventilation Systems

A ventilation system must provide a continuous source of cooling air over the entire Gas Turbine operation range in order to:

- maintain a uniform and constant airflow through the flange-to-flange Gas Turbine at all ambient conditions;
- remove heat and maintain the air temperature in the compartment below the operating limit. (The operating limit is set according to the temperature rating of the components located in the compartment);
- eliminate stagnation zones and prevent the accumulation of hazardous gases;
- prevent the ingress of dust and sand in gas turbines located in regions prone to sandstorm conditions by means of proper compartment pressurization.

Specific Design Practices provide a general description, acceptance limits and design criteria that a ventilation system must meet for Oil & Gas applications (e.g., enclosure design temperature ranges, design pressure ranges, purging ranges, etc.).
As mentioned, the current GE approach to studying GT ventilation systems consists of modeling the whole system as a series of "blocks". Each block represents a source of pressure loss (concentrated loss) due to changes in shape (e.g., elbow, transition, etc.), flow direction or the presence of physical obstacles within the system. The ASHRAE duct-fitting database provides the corresponding pressure loss coefficients.
Following the net balancing by means of a one-dimensional flow tool (Flowmaster), the system is characterized in terms of velocities, pressures, and flow rate split.
Critical points for this approach are the modeling of complex fittings and bend interactions. In order to improve the current Ventilation System calculation procedure, dedicated CFD analyses were performed for these critical points. A combined 1D & 3D CFD approach was adopted to study two different GE Ventilation Systems, called for simplicity System A and System B.

Numerical calculations for System A

The current System A Flowmaster network, modeled as a series of discrete losses, is shown in Figure 1. The

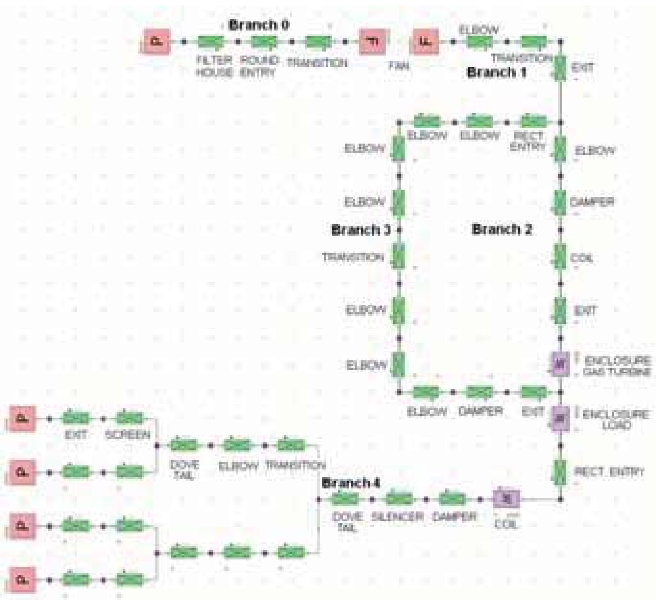

Fig. 1 - System A Flowmaster model based on discrete losses.

enclosure is modeled as two heaters and the fan as two flow sources with a flow rate of 65000 m^3/h, estimated by using the enthalpy balance equation:

$$\dot{m} = \frac{HR}{Cp(T_{out} - T_{in})}$$

were:
$\dot{m}$ = mass air flow [Kg/s],
HR = enclosure heat rejection [W]
Cp = specifiwec heat at constant pressure [J/Kg °C]
T_{out} = maximum allowable outlet air temperature [°C]
T_{in} = max ambient temperature [°C]

In order to develop a more suitable model (taking into account interactions, 3D characteristics of the fluid, etc.), dedicated ANSYS FLUENT CFD analyses were performed. In particular, a critical point for the discrete losses modeling is the flow split into the Load Compartment and the Gas Turbine Compartment (see Figure 2).

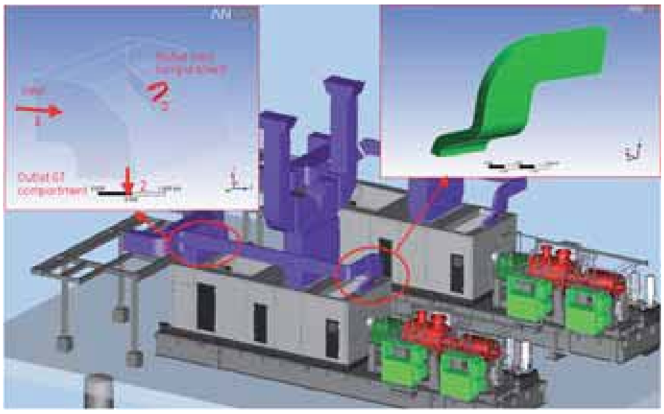

Fig. 2 - Analyzed split (left) and Load Compartment final section (right), System A

It is useful to define the coefficients K12 and K13 as:

$$K_{12} = \frac{P_{01} - P_{02}}{1/2\rho V_2^2} \; ; \quad K_{13} = \frac{P_{01} - P_{03}}{1/2\rho V_3^2}$$

where:
P_{01} = inlet total pressure
P_{02} = GT Compartment total pressure
P_{03} = Load Compartment total pressure
V_2 = GT Compartment mean velocity
V_3 = Load Compartment mean velocity

For the characterization of the flow split at different operating points, two test campaigns were performed. In both cases the inlet flow rate was fixed (65000 m^3/h and 130000 m^3/h, respectively) and, for each of these, a variable split percentage between the GT and Load compartments was used. These analyses provided K_{12} and K_{13}, defined in (2), as a function of the flow rate split (see Figure 3).

	K12 (GT Compartment)	K13 Load Compartment
ASHRAE Database	0.4	0.71
modified ASHRAE model (experience based)	1.15	1.5
CFD	0.4-0.6	2.28

Table 1: Loss coefficients used for standard calculations, System A.

Subsequently, these coefficients were implemented within the corresponding one-dimensional Flowmaster model.

A comparison between the loss coefficients obtained using CFD and those coefficients used for standard calculations is summarized in Table 1.

Finally, to better simulate the ventilation system a bend interaction analysis was performed on the Load Compartment final section, which is highlighted in Figure 2 (for System B the geometry of this section is the same). The total loss coefficient as a function of the inlet velocity is shown in Figure 4. The loss coefficient decreases as the inlet flow velocity increases, and a good agreement with the ASHRAE database value was found for a velocity of about 5m/s. For higher velocity values the difference between the two curves (CFD and ASHRAE) starts to be significant. Again, the loss coefficient curve obtained was implemented within the new model.

The fan, previously modeled as two flow sources, was replaced by the "FAN" element with the corresponding real operating curve.
The final System A Flowmaster model including the main differences from the standard approach is shown in Figure 5.

The results obtained with the new model were compared with the results obtained by the ADV (Air Ducts and Ventilation) department using a model based on the ASHRAE loss coefficient with appropriate corrections based on experience and with the results obtained with a pure ASHRAE model (see Table 2). The reliability of each approach was evaluated through comparison with experimental data.

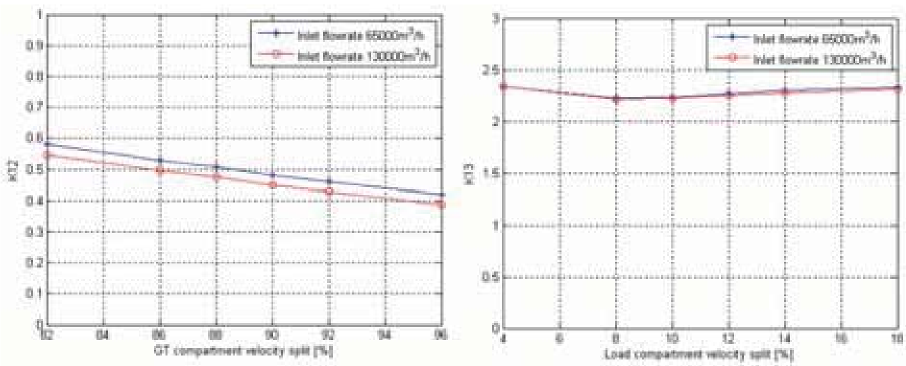

Fig. 3 - K_{12} and K_{13} as a function of flow rate split for two different inlet flow rates, System A.

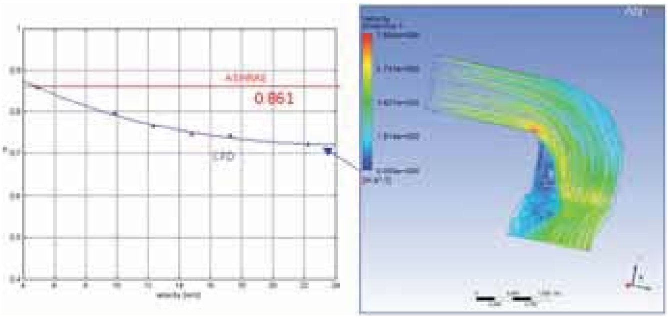

Fig. 4 - Loss coefficient curve for Bend Interaction.

lower than 5%). On the contrary, the pure ASHRAE model produced an error of 18%.

Numerical calculations for System B

Also for the System B split, several tests were performed to determine the split loss coefficients for different operating conditions. Figure 7 shows K_{12} and K_{13} as a function of the split flow rate percentage between the GT and Load compartments for an inlet flow rate equal to 70000 m^3/h (design flow rate). As one can see, both curves follow a linear trend.

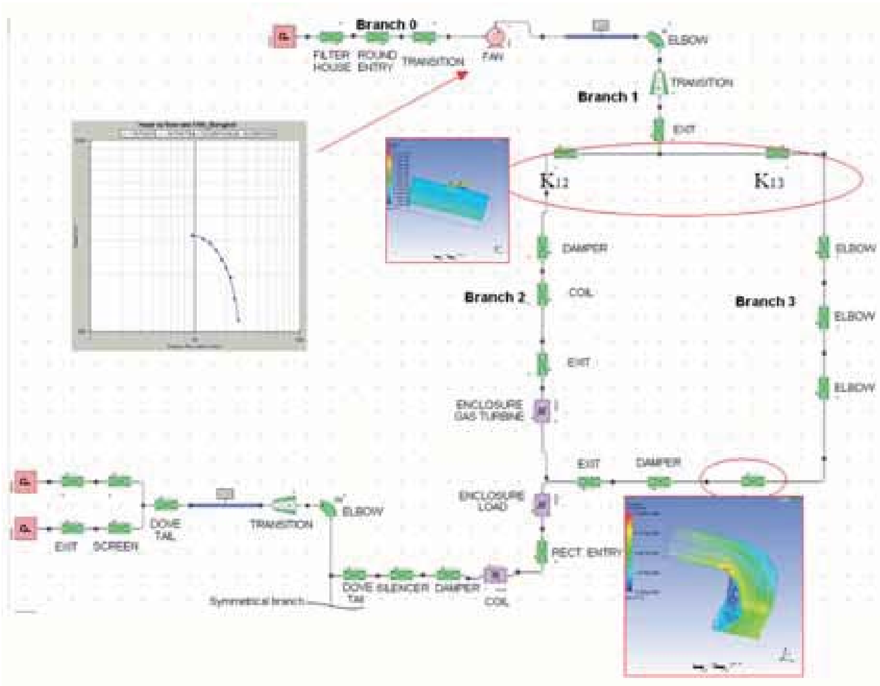

Fig. 5 - New System A Flowmaster model.

Similar to the System A model, the new System B Flowmaster model contains the loss coefficient curves obtained from CFD analyses (including the bend interaction curve) and the real fan operating curve. Finally, in order to better simulate the heat removal, the heat rejection was modeled as a function of the mass flow rate, in accordance with recent studies performed by the SYS-OPT (System Optimization) department, that is:

$$HR = HR_0\left(\frac{\dot{m}}{\dot{m}_0}\right)^n$$

where:

HR = heat rejection
HR_0 = reference heat rejection
$\dot{m}$ = mass flow rate
$\dot{m}_0$ = reference mass flow rate
n = reference exponent

Table 2 summarizes the results obtained for the enclosure pressure. Using the new approach we got a favorable level of approximation with respect to the measured value (error equal to 7%). The other two approaches yielded errors higher than 25%.

Figure 6 shows for each model the load compartment velocity and the corresponding error from the measured value at clean filter house conditions. The measured mean velocity is 12.47 m/s.

Both the new model and the modified ASHRAE model (experience-based) led to a high level of agreement (error

	Enclosure Pressure[mmH2O]	Measured value[mmH2O]	Error[%]
Discrete loss model (ASHRAE)	54.40	43.0	26.5
Discrete loss model (Experience based)	54.86	43.0	27.6
New model (Flowmaster+CFD)	40.00	43.0	-7.0

Table 2: Enclosure pressure, clean filter house conditions, System A.

The final System B Flowmaster model is shown in Figure 8. Figure 9 shows the GT and Load Compartment velocity obtained with the STD model (previous calculations) and the new model for dirty and clean filter house conditions.

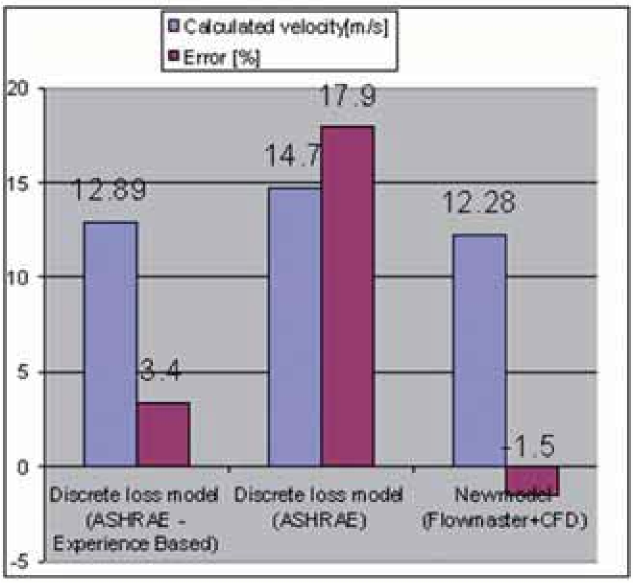

Fig. 6 - Load compartment velocity, clean filter house conditions, System A.

In both cases, the load compartment velocity obtained with the new approach is significantly higher than the old value (+49%). In particular, for the new approach, we got a split of 89-11% compared to a value of 92.7-7.3% obtained from previous calculations. Considering that the target flow rate is 90-10%, the new approach again provides more accurate results.

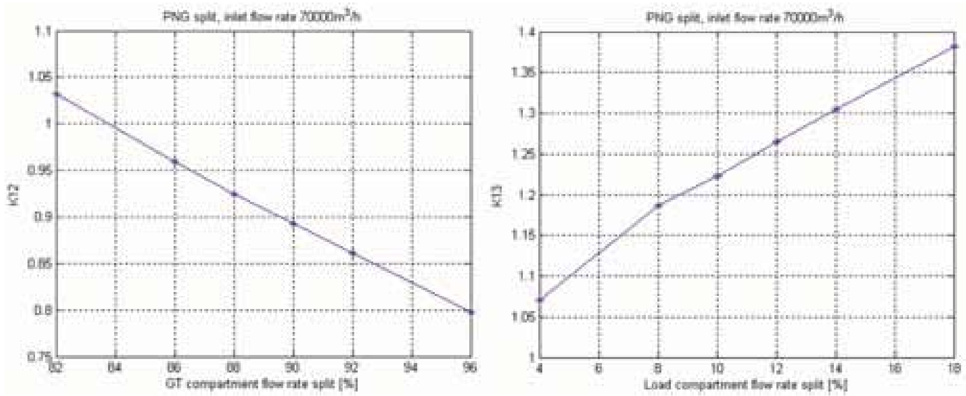

Fig. 7 - K_{12} and K_{12} as a function of flow rate split, System B.

No significant variations between the two approaches in terms of enclosure pressure and temperature were found.

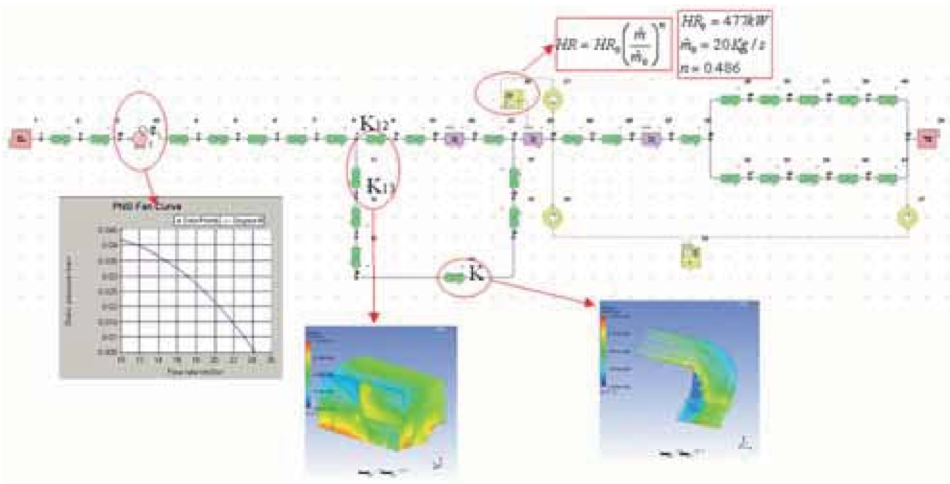

Fig. 8 - New System B Flowmaster model.

Conclusions

In this work, a combined 1D and 3D numerical approach was adopted to study two GE ventilation systems. This approach, compared to the current one-dimensional approach, improves the simulation of the actual operating conditions in terms of inlet flow rate, duct velocity and enclosure pressure, as confirmed by the close agreement with experimental measurements.

Among the benefits of using this new approach is the ability to characterize the flow behavior of complex fittings. This would be useful to support the redesign of fittings or for noise reduction analyses.

References

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L. Barbato, M. Blarasin, S. Rossin
GE Oil & Gas,
Via Felice Matteucci 2, Florence, Italy

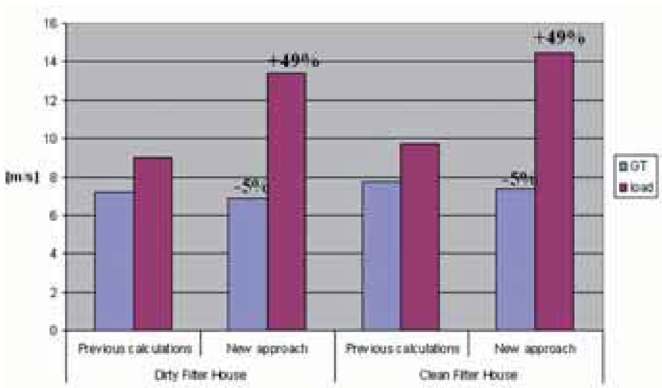

Figure 9 - GT and Load compartment velocity for STD and New approach, System B.