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MODULAR APPROACH TO OFF-DESIGN GAS TURBINES SIMULATION: NEW PROSPECT FOR REHEAT APPLICATIONS

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ABSTRACT

This paper proposes the application of a modular approach to off-design prediction of multi- and single-shaft gas turbines. The performance effects of some variations of plant configurations are easy to predict. A solution adopted for module matching is shown, and the instability phenomena in some elementary units (i.e. stalling and surge in the compressor) are studied.

The partial load behavior study of aeroderivative gas turbines is presented and performance comparisons are made with a particular heavy-duty advanced application. An analysis example shows some reheat effects in combined power plant applications.

This work enabled the authors to make interesting observations on control techniques and gas-turbine configurations for advanced, combined power-plants. It also shows the wide possibilities of code applications.

NOMENCLATURE

A = reference component inlet area
c = absolute cascade velocity
m = mass flowrate
p = pressure
r = mean radius
R = degree of reaction
Rg = gas constant
T = temperature
u = transport velocity
W = power
β = compression ratio
$\beta'_{\Phi,0}$ = finite partial difference of β with respect to Φ in the reference condition
$\beta'_{\Omega,0}$ = finite partial difference of β with respect to Ω in the reference condition.
Δ = finite difference
∇ = gradient vector
Φ = flow coefficient
η = global efficiency
Ω = angular speed

subscript and superscript

0 = reference condition
i1 = main inlet
o1 = main outlet
amb = ambient conditions
c = coolant
CC = combined cycle
exh = exhaust
GT = gas turbine
HP = high pressure
LP = low pressure
nom = nominal condition
SF = Supplementary fire

INTRODUCTION

The continuing goal of design and research efforts has been to increase the performance (i.e. efficiency and specific work) in gas turbine power plants. So, cycle characteristics such as compression ratio or the firing temperature have achieved very high values. On the other hand, some modifications for the Joule cycle have been introduced (steam injection, regeneration, intercooling, the combined power plants and so on) which obtained a more efficient power cycles.

There have been numerous research contributions in the field of new gas turbine design and partial-load behavior studies.

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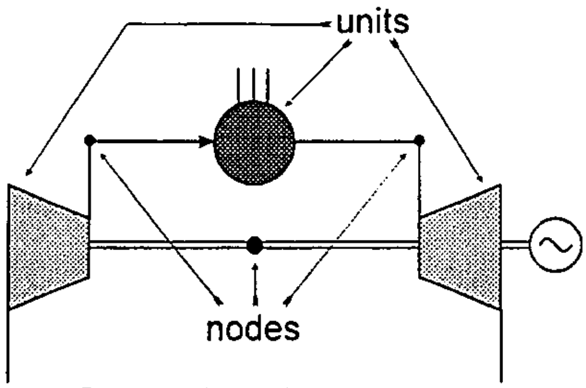

Figure 1a: Example of a Power Plant.

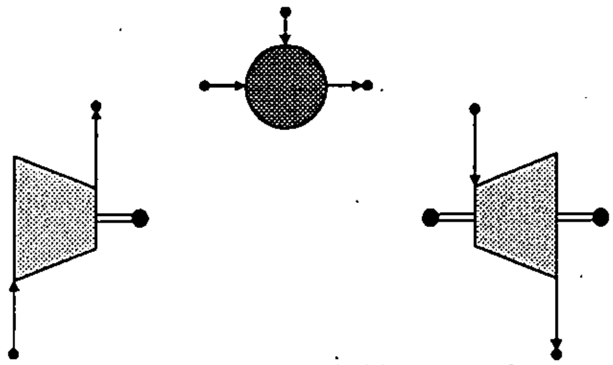

Figure 1b: Power Plant Divided in Elementary Components

Certainly today, the plants' complexity requires adequate calculation tools for simulation behavior and performance predictions and particular attention must be given to the partial load study. The use of the modular approach for thermodynamic analysis of energy conversion systems has been already presented by Carcasci and Facchini (1995); in this work the same approach is applied to off-design prediction.

Generally, power plant simulations are solved by codes with fixed input data or with limited variations (dedicated approach). In contrast, global generalization of power plant analysis can be ensured by a modular approach. Therefore, a modular code, in the most general of cases, must be able to:

- create a power plant configuration, without creating a new program source;
- handle any combination of input data, provided that there is a sufficient and consistent number of parameters for the plant solution;
- determine the characteristic parameters of the elementary components, when an increased number of boundary input data are fixed.

The development of a modular code has been studied by various researchers with different approach. (Sonnenschein, 1992; Somerton et al., 1987; Erber et al., 1989; Cioli and Desideri, 1991; Perz, 1991; Carapellucci and Cau, 1992; Distelmans and Ruyck, 1992; Perz, 1993; Benvenuti et al., 1993; Bettocchi et al., 1994; Simtech, 1994; Enter, 1995a,b,c; Thermoflow, 1995a,b,c,d). The comparison between the proposed method and other approaches has been shown in the previous work.

The method presented in this work is based on a full implicit linear approach, where the code reduces the non-linear equation system to a linear system with variable coefficients, then all equations are simultaneously solved with a classic matrix method. The powerplant definition is obtained with a system of elementary components, where there are mass and energy flows which undergo chemical, energetic and thermodynamic transformations. In the mathematical model, the power plant components will be referred to as a unit (Fig. 1a.). The units are connected by:

- Streams, which are connections allowing the transport of mass between two units.
- Mechanical energy links, which transport mechanical energy and are related to particular angular speed (an example is a shaft).

The points connecting two units will be referred to as nodes (Fig. 1b.). A method to provide sufficient information to describe the state of flow in each node of some plant and in every operational condition must be determined. Each unit is characterized by some typical parameters; i.e. its efficiency. So a unit will be compared to a black-box, which is able to simulate a particular energetic transformation. Obviously, certain internal characteristics of the components and certain properties of the flow in some nodes will be known (data of the power plant system). However, none of these is considered essential and in this respect, this modular approach is more general than other semiparallel or sequential methods. For a general and flexible solution method, the use of a linear system is advantageous. Therefore, all the equations, which define a powerplant, are linearized and their coefficients updated in the course of the calculation. The solution of the system is obtained after a reduction of the matrix dimension for the calculation time optimization (Carcasci and Facchini; 1995).

The off-design behavior of a gas turbine plant depends on the new equilibrium configuration which is determined by setting the control system or the changeable ambient condition.

In this case, the use of dedicated (non-modular) approach can create some problems when the study is applied to multishaft gas-turbine and, especially, when one considers its applications in the more complex combined power plants. The authors have already studied this problems (Facchini, 1993; Facchini and Sguanci 1994, Bettagli and Facchini, 1995a, 1995b) and they have developed a simplified model for off-design simulation of the main elementary components (i.e. compressor, cooled turbine, combustion chamber, multi-pressure heat recovery boiler, steam turbine and so on). Increasing the complexity of the plant, the dedicated study has numerous limitations in calculation time and in code definition. Often, in this case many iterations are necessary to determine the new equilibrium configuration and the set input data is fixed.

Partial load simulations with a modular approach have already been proposed by some authors (Perz, 1995; Erbes et al., 1989) and they are based on the use of characteristic curves furnished by the equipment manufacturer. Therefore the amount of information is often inadequate. On the contrast, a simplified component simulation permits a better description of cycle behavior and a better understanding of manufacturer information.

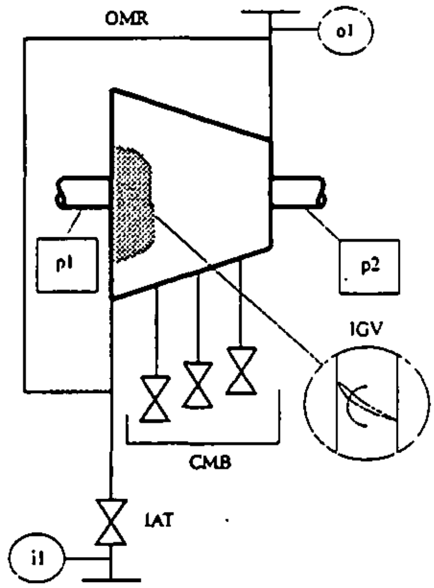

Figure 2: Axial Compressor Module With Control Systems.

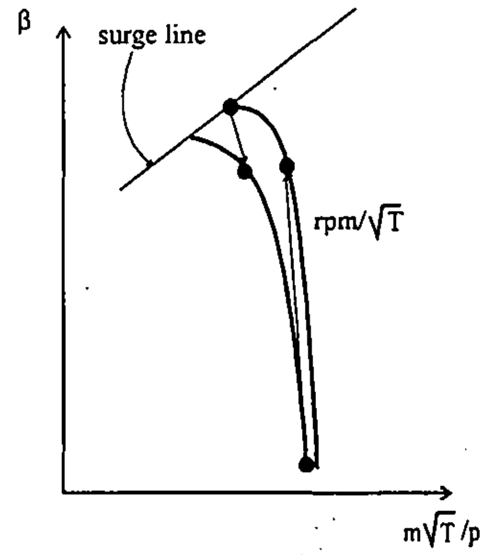

Figure 3: Axial Compressor Characteristic Curves: Instability Control

In this work, the previous simulation code developed for combined cycles off-design performance (Facchini 1993; Facchini and Sguanci 1994; Bettagli and Facchini 1995a, 1995b) has been adapted to a modular approach. A consistent reduction on calculation time is obtained for more complex gas-turbine configurations, without any restrictions on defining plant and boundary conditions.

After the simulation tests (as applied to an aeroderivative gas turbine in the final part of this work), we examine the prospects for using reheat in the gas turbine plant. Reheating is a well-known technique, which is mainly used to increase the power output. It is especially used on propulsion field.

The positive effects of reheat on combined gas steam cycle performance have been already shown by Rice (1980). Additionally, the supplementary firing, at the heat-recovery boiler inlet have been much studied in the last ten years. A negative effect on performance design has been shown, but the off-design performance of combined power plants increases (Chefneux and Mathieu, 1995). All of this shows the necessity of a relatively high turbine exit temperature for the performance optimization of combined power plants.

The new gas turbine GT24 by ABB (1993) is based on the introduction of reheat, named sequential combustion, that permits the strong reduction of NO_x emissions and it obtains a high turbine exit temperature. The very high efficiency levels estimated for the correspondent combined power plant (near 58%), clearly show the relevance of the study in this field.

OFF-DESIGN MODULE DEFINITION

For off-design-performance evaluation, the unit definition becomes more complex and it requires a more detailed design approach. In fact, the design study must give a description of the component (Facchini 1993, Bettagli and Facchini 1995a), which allows the parameter definition for the use of typical off-design correlations (i.e., the triangle of velocity at mean radius and other cascade parameters for the compressor or turbine). For this parameter definition, knowledge of a some of the constructor's data can be important. So, when one leads an off-design study, there is a reduction in the number of input data, because some of these are derived through design study. These design and off-design procedures require a redefinition of unit equations (Carcasci and Facchini, 1995) which is necessary for the modular approach in the ESMS code.

Compressor module

For a greater understanding of a model description of a general unit module, one can examine the axial compressor unit, because particular problems of instability behavior are present, too.

The design and off-design simulations (Facchini, 1993) are based on 0-dimensional approach (mean-line analysis and repeating axial stage, see Cumpsty, 1989; Dixon, 1975)

The control systems preview for the compressor are (Fig. 2.):

- IGV: Inlet Guide Vane (the stator vane setting is extended to all stages)
- IAT: Inlet Air Throttling
- OMR: Outlet Mass Recirculation
- CMB: Compressor Mass Bleeding.

These control systems permit a large range of possibilities in partial load simulations and their effects have been discussed in the previous works (Facchini, 1993; Bettagli and Facchini 1995a, 1995b).

The axial compressor unit equations are:

1. Continuity equations for: mass flowrate; chemical composition and angular speed.
2. Equations of thermodynamic transformation
3. Energy balance
4. Characteristic equations

The first three equations are analogous for the design and off-design procedures and they are defined in the previous work

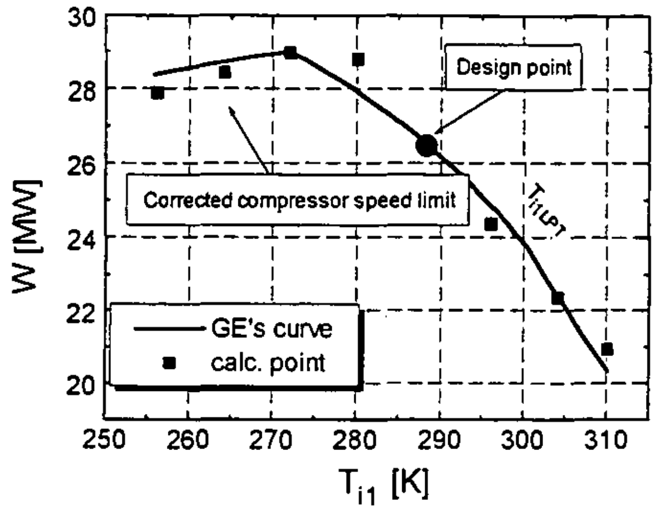

Figure 4: Power Output Vs. Inlet Compressor Temperature. Results and Constructor Curves Comparison.

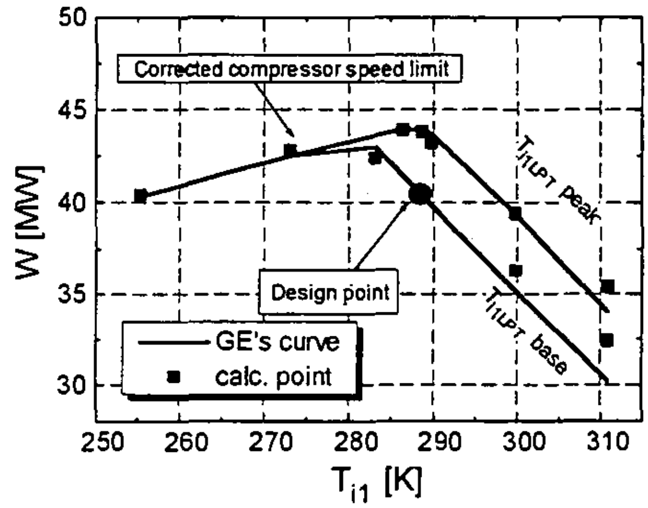

Figure 5: Power Output Vs. Compressor Inlet Temperature. Calculated Data and Constructor Curves Comparison

(Carcasci and Facchini, 1995). The characteristic equation is necessary for the off-design simulation and it represents the partial load performance with an imposed design. In this work, the link equation between compressor ratio, angular speed and flow coefficient has been chosen. Using a linear solutor, a linearized expression of this equation must be defined:

$$\beta - \beta^0 = \beta^1_{\phi,0}\left(\Omega-\Omega^0\right)+\beta^1_{\Omega,0}\left(\Phi-\Phi^0\right) \tag{1}$$

Therefore, the compressor ratio (β) and the flow coefficient (ϕ) are not the basic flow parameters for the modular approach. They must be expressed in another compatible form. Considering the respective basic definitions, one obtains

$$\beta = \frac{p_{i1}}{p_{o1}}$$

$$\Phi = \frac{c_{axial}}{u} = \frac{Rg}{Ar}\left(\frac{mT}{\Omega\, p}\right)$$

where the variables depend to the considered compressor section; with the (β-β^0) and (ϕ-ϕ^0) terms' linearization, one can write:

$$\beta - \beta^0 = \frac{p^0_{o1}}{\left(p^0_{i1}\right)^2}\left(p_{i1}-p^0_{i1}\right)+\frac{1}{p^0_{i1}}\left(p_{o1}-p^0_{o1}\right) \tag{2}$$

$$\Phi-\Phi^0 = \frac{Rg}{Ar}\left[\frac{T^0_{i1}}{p^0_{i1}\,\Omega^0}\left(m_{i1}-m^0_{i1}\right)-\frac{m^0_{i1}}{p^0_{i1}\,\Omega^0}\left(T_{i1}-T^0_{i1}\right)- \right.$$

$$\left. -\frac{m^0_{i1}T^0_{i1}}{p^0_{i1}\left(\Omega^0\right)^2}\left(\Omega-\Omega^0\right)-\frac{m^0_{i1}T^0_{i1}}{\left(p^0_{i1}\right)^2\Omega^0}\left(p_{i1}-p^0_{i1}\right)\right] \tag{3}$$

and introducing the expressions from eq. (2) and eq. (3) into eq. (1), one obtains the characteristic equation. For characteristic equation definition some calculations at various Φ and Ω are carried out and afterward a finite difference formulation is obtained.

Instability behavior.

Instability, stalling or choking in the compressor or in the turbine, can occur during a partial load simulation. When some of these conditions are verified, one cannot determine the necessary parameters for calculation. Therefore, this fact is probably due to the transient calculation and, so, it is not correct to stop the simulations immediately. In this case, an appropriate procedure to overcome the instability condition is necessary and, only after detection of repeated instability, one can conclude that the plant's correct behavior is not possible. Obviously, the evaluation of the listed instability conditions, is simplified and compatible with the simulation model used.

To overcome instability conditions one can see a typical characteristic compressor curve (Fig. 3.), where the possibility of finding new corrected behavior clearly appears if the new performance conditions are imposed. This is done by using a gradient vector $\underline{\Delta}$ with these components:

$$\underline{\Delta} = \left(\Delta m_{i1}, \Delta p_{i1}, \Delta T_{i1}, \Delta\Omega\right) \tag{4}$$

where the variations are referred to inlet unit conditions.

Considering the arbitrary boundary condition settings, some incompatibilities can occur when a particular data input prevents, directly or indirectly, the variations of some vector $\underline{\Delta}$ components. To avoid the consequence of no convergence of the calculation code, these incompatibilities must be checked and the relative vector component must be de-activated.

The component sign depends on particular instability conditions: i.e. the sign choice for overcoming the compressor surge will be

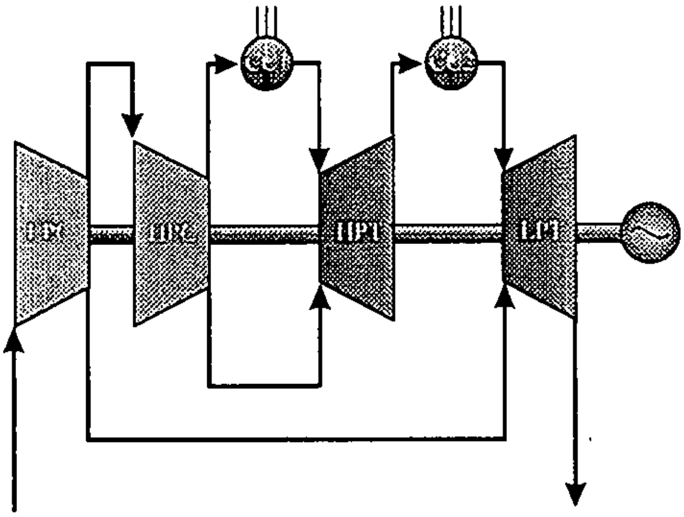

Figure 6: Simulation Model of *GT24* Gas Turbine

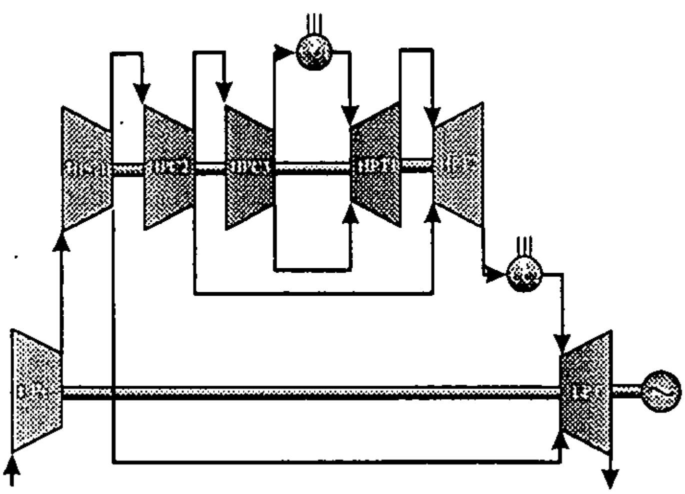

Figure 7: Simulation Model of *LM6000-SF* Gas Turbine

opposed to that of the chok condition(Fig. 3.). Similar evaluations are done on the basis of turbine simulations.

Some program tests

Some present-generation gas-turbine plants are simulated (terrestrial application of aeroderivative type and heavy duty). For these machines, only a few global performance data are known; generally, these are: compression ratio, outlet gas turbine temperature and mass flowrate, gross efficiency, power, ambient standard conditions, shaft configuration, compressors and turbines stages number. Using this set data some general design parameters (i.e: ϕ, ψ, R, stages loss coefficients, turbine blade cooling parameters, pressure losses and efficiency of combustion chamber) are fixed with the goal to obtain the best agreement between the calculated data and these known.

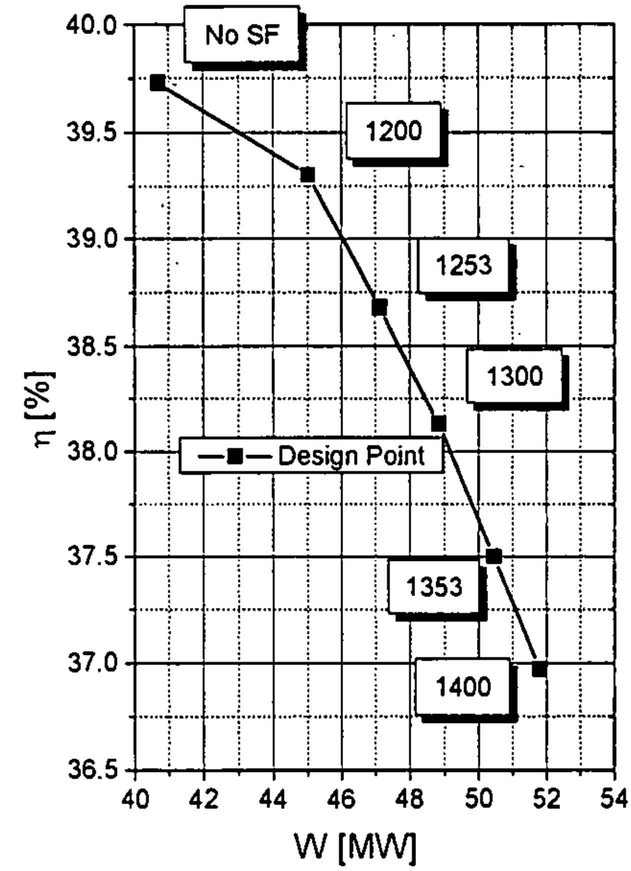

Figure 8: Efficiency/Power Trend at Various Supplementary Fairing Temperature for the *LM6000-SF*

The first simulated plant is LM2500+ (GE, 1994), an aeroderivative twin-shaft gas turbine manufactured by General Electric. It is about a modified version of the well known LM2500 presented in the '70s and is derived from the CF6-50 turbofan; the compressor is supplied by the high pressure turbine and the other turbine is connected only with load. This last version is especially characterized by a new 0-stage at inlet compressor and it has a consistent increase in output power (25%), compression ratio and mass flowrate. In figure 4, the power output trend with ambient temperature variation is shown. The curve refers to a constant low pressure turbine inlet temperature in the right-hand zone On the contrast, when there is a great reduction in compressor inlet temperature (left-hand zone), a reduction in fire temperature is imposed to avoid surge. In other words, a lower compressor speed (free shaft) is obtained. The agreement between the results and the curve furnished by constructor is satisfactory also because the probable IGV setting, in off-design conditions, is not known.

An other interesting aeroderivative multi-shaft turbine is LM6000, manufactured by General Electric, too. Like the previous LM6000 (GE, 1995), it is derived from the CF6-80C2 turbofan. It represents the latest generation gas turbine and this appears to be the most efficient industrial application.

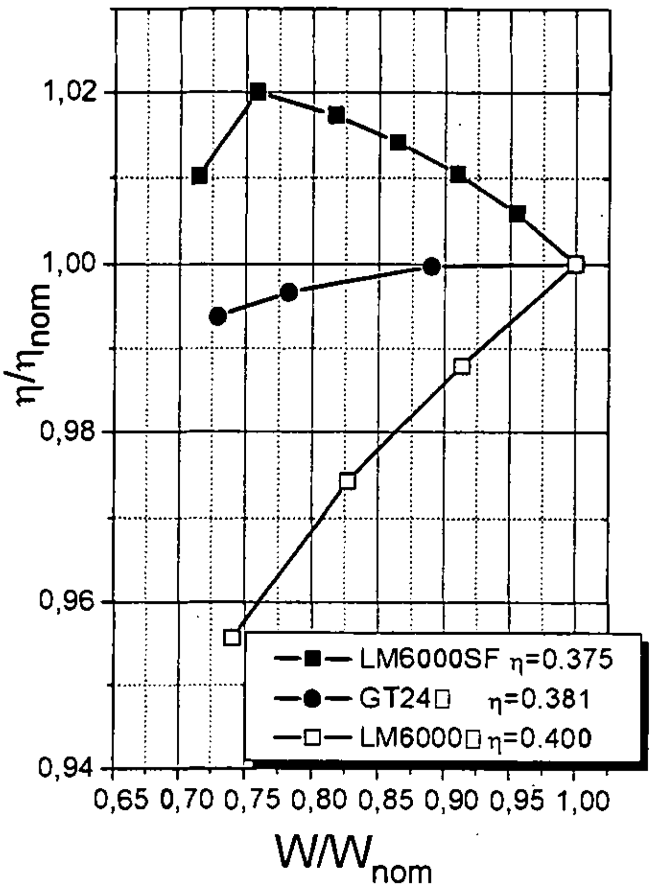

Figure 9: Efficiency/Power Trend.
***LM6000, GT24 and LM6000-SF* Comparison**

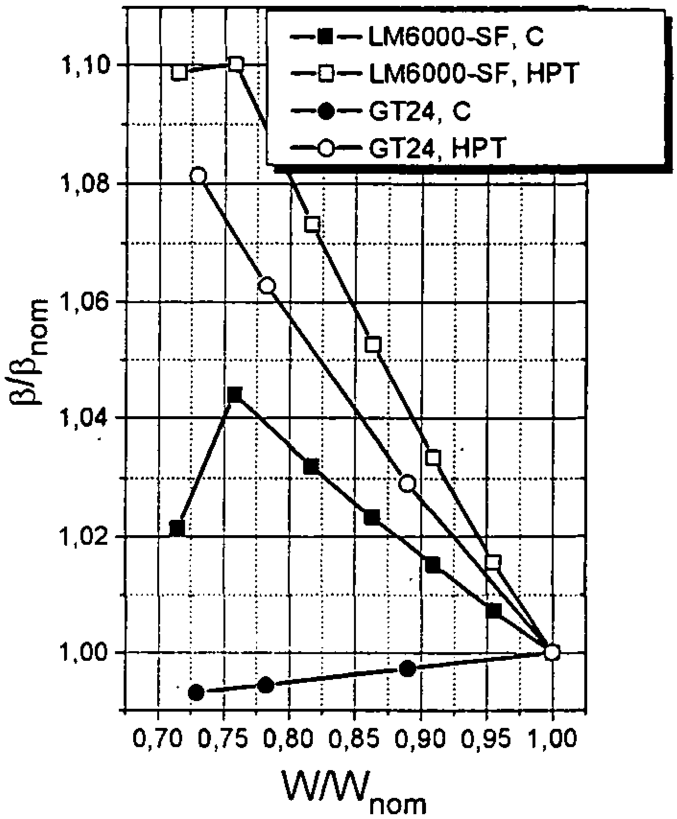

Figure 10: Compression and Expansion Ratio Vs. Power.
***GT24 and LM6000-SF* Comparison**

In table 1., the ISO condition performances and some of the design parameters are reported, two duty conditions are previewed, similar to the one seen in figure 5: the peak load and the base load; where the first is related to a temporary fire temperature increase. The calculations' agreement with the curves published by GE (1993) is satisfactory because the IGV setting has probably been selected, but it is not known in this case either.

New prospects in the reheat

Another simulation test has been related to already-cited GT24, manufactured by ABB. It is a large size (165 MW) gas turbine, and some design parameters and performance levels have been published (ABB, 1993) and are reported in table 1. Figure 6 shows its configure. The simulation model considers two coolant bleeding points on the compressor line, while the real cooling system is probably more complex, but we have not sufficient information for other complications, which, in any case, can be simulated by our code.

Besides, considering the previous LM6000 simulation study, we suppose the introduction of reheat before the power turbine (Fig. 7.), to compare its performance with GT24. Currently, LM6000 application possibilities on combined power plants are limited by very low exhaust flow temperatures and so the higher efficiency of gas turbine cycle can be conditioned in this type of use (table 1).

A consistent increase in the blade cooling of a power turbine is necessary for the proposed version of LM6000 with supplementary fire (SF). to reduce the new coolant mass flowrate requirement, this is derived along the high compressor line and the second fire temperature is imposed as 1353 K, when the same GT24 exhaust gas

	m_{exh} [kg/s]	m_c [kg/s]	T_{exh} [K]	β	W [MW]	η [%]	Ω [rad/s]	T_{SF} [K]
LM2500+	84.0	10.5	784.0	23.1	26.5	35.4	800.-315.	-
GT24	376.0	24.9 (36.0)	883.1	30.0	165.0	38.1	377.	1507.1
LM6000	124.7	22.4	736.6	30.0	40.7	40.0	1000.-377.	-
LM6000-SF	125.3	21.1 (6.5)	882.8	30.0	50.5	37.5	1000.-377.	1353.0

Table 1: Some Global Characteristics of GT24, LM2500+, LM6000 and LM6000-SF.

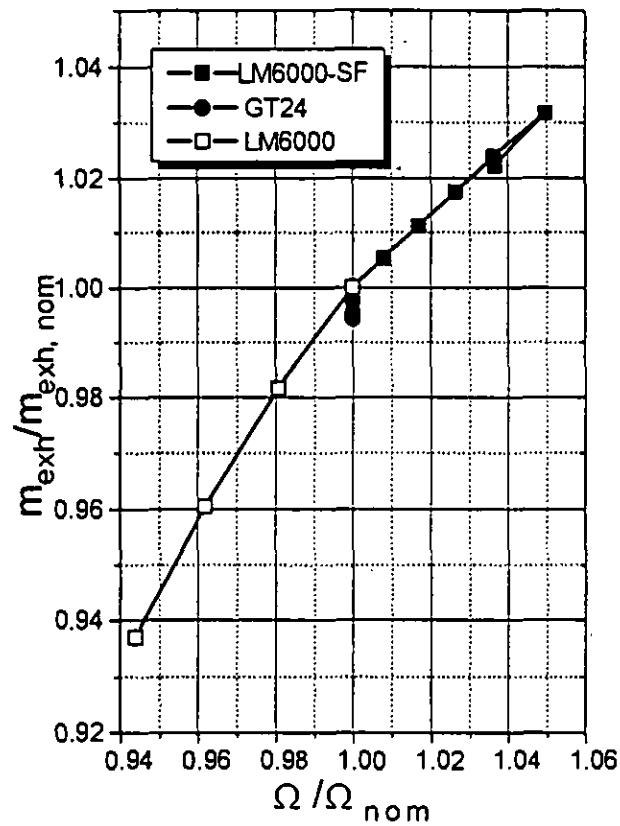

Figure 11: Exhaust Gas Mass Flowrate Vs. Shaft Speed. *LM6000, GT24 and LM6000-SF* Comparison

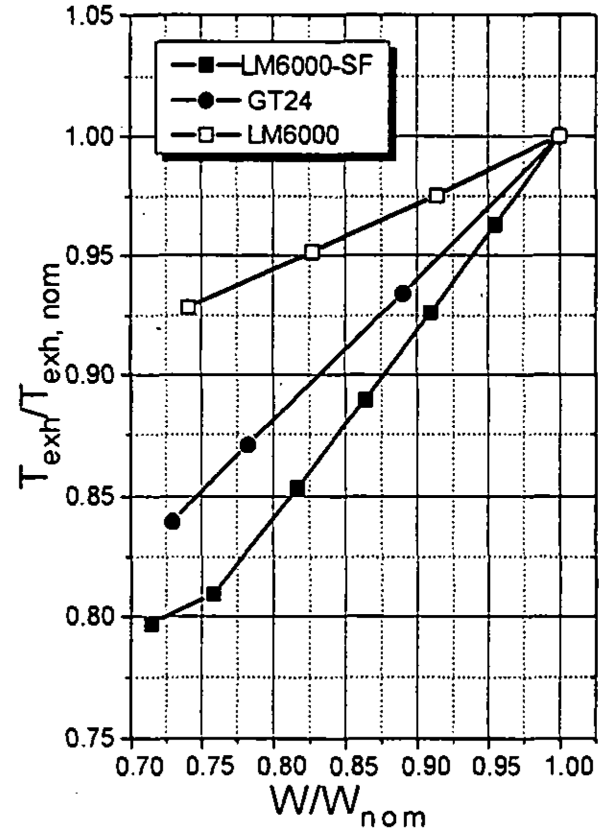

Figure 12: Exhaust Gas Temperature Vs. Power. *LM6000, GT24 and LM6000-SF* Comparison

temperature is obtained. Under these conditions, a 24% increase in power and a 6.2% loss in efficiency are obtained (Fig. 8.). A global performance parameter is also reported in table 1.

The off-design behavior of supplementary fire reduction for GT24 and LM6000-SF is very interesting. The efficiency appears almost constant for GT24 and obviously increases for LM6000-SF (Fig. 9.). In this case, it is important to underline that the curves refer to relative value. When the supplementary fire is turned off, the main fire must be reduced to obtain further load reductions, in this case LM6000-SF performance quickly deteriorates. This effect is less important for GT24 trend, because its fire temperature and pressure levels reheat are higher, and more power control is possible with supplementary fire. Nevertheless, some problems can occur for GT24 in a high pressure turbine, with a consistent reheat reduction. In fact the choked flow condition is favored by the corresponding increase in first turbine expansion ratio (Fig. 10.) due to a reduction of the second turbine inlet temperature and relative expansion ratio. The twin shaft configuration of the LM6000 gas turbine is more cost efficient because with an analogous trend it permits an increase in the speed of the high pressure compressor which determines an higher first turbine inlet pressure, positive to reduce the choked flow danger (Fig. 11.). The corresponding surge problem for a high pressure compressor is limited, because the supplementary fire temperature is not very high and the power control is not greater, and yet the following main fire reduction tends to stabilize the engine behavior. From the exhaust temperature point of view the supplementary fire control obviously involves a reduction and this fact can be not positive for heat recovery in the combined power

	W/W_{nom}=1.0					W/W_{nom}=.90			W/W_{nom}=.80		
	W [MW]	η_{CC}[%] declared	m_{exh} [kg/s]	T_{exh} [K]	η_{GT} [%]	m_{exh} [kg/s]	T_{exh} [K]	η_{GT} [%]	m_{exh} [kg/s]	T_{exh} [K]	η_{GT} [%]
GT24	165.0	57-58	376.0	883.1	38.1	375.2	830.1	38.1	374.4	778.1	37.9
LM6000	41.5	52-53	124.7	736.6	40.0	122.0	715.6	39.4	118.9	695.5	38.7
LM6000-SF	51.5	57-58	125.3	882.8	37.5	126.9	810.5	37.9	128.6	742.0	38.2

TABLE 2: Off-Design Performance Comparison for Combined Cycle Application

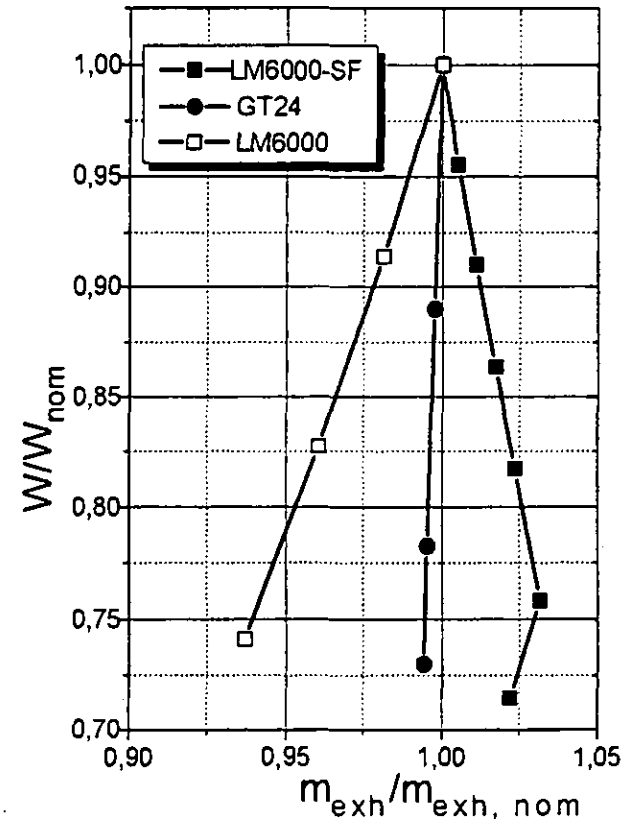

Figure 13: Power Vs. Exhaust Gas Mass Flowrate. ***LM6000, GT24 and LM6000-SF* Comparison**

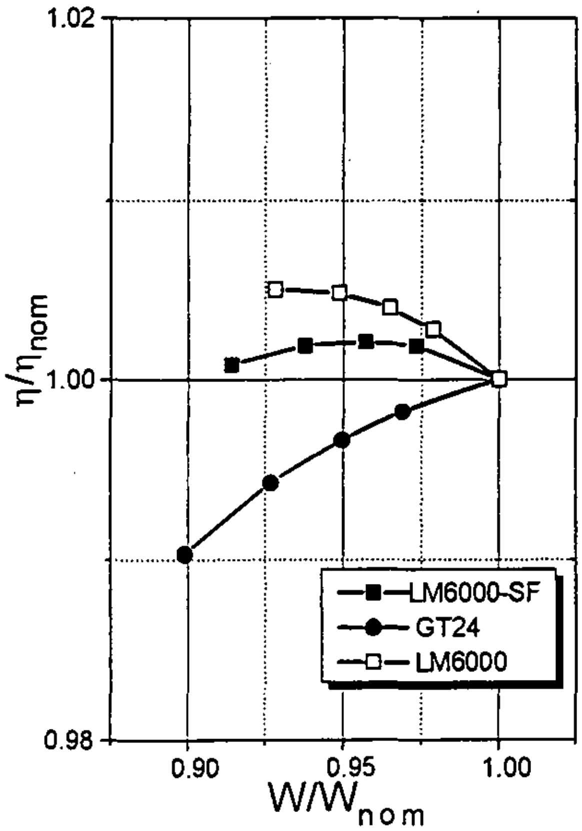

Figure 14: Efficiency/Power Trend with IGVC. ***LM6000, GT24 and LM6000-SF* Comparison**

plants (Fig. 12.). Only with LM6000-SF, a small increase in the exhaust gas mass flowrate occurs and can affect the heat recovery (Fig. 13.), too.

For a positive control of the exhaust temperature and a wider possibility of obtaining the most efficient partial load behavior the setting of IGVC can be very important (Fig. 14.). For LM6000-SF, the consequent high pressure shaft acceleration is in agreement with the control second fire effects and this fact is most evident when the IGVC control is related to a high pressure compressor. So, a composite control procedure is required through a contemporary main fire reduction. The same observations are possible with GT24, where the IGVC control produces a further increase on first turbine expansion ratio and it requires an opportune main fire control.

Other control technique effects can be considered in relation to the previous authors' study, particularly IAT, to improve the heat recovery possibilities in the combined power plants (Bettagli and Facchini, 1995b).

For combined powerplant applications, the global cycle efficiency does not directly reflect the story of a single gas turbine cycle's efficiency. In this case the absolute levels of the exhaust gas temperature and mass flowrate are particularly important. The inevitable efficiency losses, related to reheat introduction, can be balanced by the more efficient design and partial load performance in the bottomer cycle. The comparative study between the three gas turbines considered (see table 2), shows some problems in terms of absolute performance levels and partial load behavior for LM6000. In fact, the very low exhaust gas temperature does not allow, even in design conditions, global efficiency over 52-53% (GE,1995) and these are very dependent on partial load conditions, for other further gas temperature reductions.

The increase in exhaust gas temperature with reheat is very important for heat recovery and, practically, the same characteristics are shown by GT24 and LM6000-SF. The gas turbine efficiency values are very close, too. So, the same combined powerplant global efficiency can be expected. At partial load conditions, the mass flowrate increases and a greater exhaust temperature reduction with LM6000-SF can condition the control of the heat recovery boiler.

CONCLUSION

The modular approach for energy system simulations has been applied to off-design gas turbine analysis single- and multi-shaft. A reduction in calculation time is obtained and some application possibility are shown. Particular attention has been dedicated to the generality of code configuration and data definition; the instability problems in some components are also evaluated.

An applied study has been carried out for some aeroderivative gas turbine. The use of reheat, recently adopted for heavy duty applications, is proposed for aeroderivative gas turbines, too.

A summary comparison between the two solutions has shown the suitability of reheat for the increase in combined plant performance at design and partial load conditions.

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REFERENCES

ABB, 1993; "GT24: Advanced Cycle System", *ABB Power Generation*, Switzerland, Rep. No. PGT210393E.

ABB, 1994; "First GT24 will improve efficiency at Gilbert", *ABB Power Generation*, Switzerland, Rep. No. PGT 2145 94E.

Benvenuti, E., Bettocchi, R., Cantore, G., Negri di Montenegro, Spina, P.R., 1993; "Gas Turbine Cycle Modeling Oriented to Component Performance Evaluation from Limited Design or Test Data", *ASME Cogen Turbo '93*, Bournemouth, U.K., September 21-23, 1993, IGTI-Vol.8, pp.327-337.

Bettagli, N., Facchini, B., 1995a; "Off-Design Performance of Multipressure Heat Recovery Boilers", *ASME Turbo Expo '95*, Houston Texas, June 5-8, 1995.

Bettagli, N., Facchini ,B., 1995b; "Comportamento di un Gruppo Combinato di Media Potenza al Variare del Carico Elettrico e Termico", *VIII Convegno Tecnologie e Sistemi Energetici Complessi "Sergio Stecco"*, Bologna, Italy, June 15-16, 1995; pp. 243-254. *La Termotecnico*, Vol. 9, Sept. 1995, pp.49-55.

Bettocchi, R., Spina, P.R., Alliney, S., 1994; "Resolution Method for Gas Turbine Mathematical Models", *ASME Cogen Turbo '94*, IGTI-Vol.9, pp.361-369.

Carapellucci, R., Cau, G., 1992, "Un Sistema di Simulazione Modulare per la Valutazione delle Prestazioni dei Sistemi Energetici", Atti del VI Convegno Nazionale, *Gruppi Combinati Prospettive Tecniche Ed Economiche*, pp.237-250.

Carcasci, C., Facchini, B., 1995; "A Numerical Method for Power Plant Simulation", *ASME Turbo Expo '95*, Houston Texas, June 5-8, 1995; 95-GT-269 - ASME J. of Energy Resources Technology, March 1996, Vol.118.

Carcasci, C., Facchini, B., Biondi Bartolini, J., 1995; "Simulazione Modulare di Impianti di Potenza e Sistemi Energetici", *VIII Convegno Tecnologie e Sistemi Energetici Complessi "Sergio Stecco"*, Bologna, Italy, June 15-16, 1995; pp. 283-297.

Chefneux, E., Mathieu, Ph., 1995; "Part Load Performance of Fired Combined Cycle Power Plants", *VIII Convegno Nazionale Tecnologie e Sistemi Energetici Complessi "Sergio Stecco"*, Bologna, Italy, June 15-16, 1995; pp.125-136.

Cioli, R., Desideri, U., 1991; "STEMCOD: Un Codice di Calcolo Modulare per Cicli a Vapore", Atti del V Convegno Nazionale, *Gruppi Combinati, Prospettive Tecniche Ed Economiche*, pp.219-230.

Cumpsty, N.A., 1989; "Compressor Aerodynamic", *Longman Scientific & Technical*, New York.

Distelmans, M., De Ruyck, J., 1992; "Labcycle: An Integrated Simulation Package for Energy Systems based on Labview and Oracle", *ECOS'92*, pp.93-99.

Dixon, S.L., 1975; "Fluid Mechanics Thermodynamics of Turbomachinery", *Pergamon Press*, Oxford.

Enter, 1995a; "Efficiency Map Performance Monitoring System", *Enter Software, Inc.*, Houston, TX.

Enter, 1995b; "GateCycle - Heat Balance Analysis Software", *Enter Software, Inc.*, Houston, TX.

Enter, 1995c; "On-Line Plant Optimizer", *Enter Software, Inc.*, Menlo Park, California, USA.

Erbes, M.R., Gay, R.R., Cohn, A., 1989; "GATE: A Simulation Code for Analysis of Gas-Turbine Power Plants", *ASME Turbo Expo '89*, June 4-8, 1989; Toronto, Ontario, Canada.

Facchini, B., 1993; "A Simplified Approach to Off-Design Performance Evaluation of Single Shaft Heavy Duty Gas Turbines" *ASME Cogen Turbo, '93*, Bournemouth, UK, Sept. 1993.

Facchini, B., Sguanci, S. 1994; "RE CYCLE: A System for a Good Efficiency Off-Design Performarnces". *ASME Cogen Turbo 1994*, Portland, Oregon.

Gay, R.R., McCoy, L., Luxton, R., 1995; "Economic Optimization Software Applied to JFK Airport Heating and Cooling Plant", *86th Annual Conference Of The Internotional District Energy Association*, Indianapolis, Indiana USA, June 3-6, 1995.

GE, 1994; " LM2500+ Gas Turbine", *GE Marine & Industrial Engines*, Rep. No. AE-3502 (5/94)

GE, 1995; "LM6000 Gas Turbine", *GE Marine & Industrial Engines*, Cincinnati, OH 45215-6301, Rep. No. AE-3248 (3/95).

Gyarmathy G., 1989; "On Load Control Methods for Combined Cycle Plants", *ASME Cogen Turbo '89*, pp.39-50.

Gyarmathy G., Ortmann P., 1991; "The Off Design of Single- and Dual-Pressure Steam Cycle in Combined Cycle Plants", *ASME Cogen Turbo '91*, IGTI-Vol 6.

Horlock, J.H.; 1966; "Axial Flow Turbine", Butterworths, London.

Horlock, J.H.; "An Analytical Model of Axial Compressor Off-Design Performance", *ASME J. of Engineering for Gas Turbines and Power*, Vol.116, pp.425-434.

Lefebvre, A. H., 1983; "Gas Turbine Combustion", Hemisphere, New York.

Perz, E., 1991; "A Computer Method for Thermal Power Cycle Calculation", *ASME - J. of Engineering for Gas Turbine and Power*, April 1991, Vol. 113, pp.184-189.

Perz, E., 1993; "Computer Aided Analysis of Thermal Power Process", *ASME Cogen-Turbo 1993*, Bournemouth, U.K., September 21-23, 1993, *Vol. 8*, pp.209-212.

Perz, E.W., Riesel, U., Schinagl, H.A., 1995; "A New Approach for Modelling Energy Systems", *ASME Cogen-Turbo Power Conference '95*, Vienna, Austria, August 23-25, 1995, 95-CTP-82.

Rice, I.C., 1980; "The Combined Reheat Gas Turbine/Steam Turbine Cycle", *ASME J. of Engineering for Power*, Vol.102, part 1, pp.35-41, part 2, pp.42-49, 1980

Simtech, 1994; "IPSE Specifications", *SimTech Thermodynamic Simulations*, Graz, Austria.

Somerton, C.W., Brouillette, T., Pourciau, C., Strawn, D., Whitehouse, L., 1987; "RANKINE: a Computer Software Package for the Analysis and Design of Steam Power Generating Units", *ASME J. Engineering for Gas Turbines and Power*, April 1987, Vol.109, pp.222-227.

Sonnenschein, H., 1982; "A Modular Optimizing Calculation Method of Power Station Energy Balance and Plant Efficiency", *ASME J. Engineering for Power*, April 1982, Vol.104, pp.255-259.

Stecco, S., Facchini B., 1989; "A Computer Model for Cooled Expansion in Gas Turbine", 3nd *ASME Cogen Turbo '89*, Nice, France.

Thermoflow, 1995a, "GT Master - Simulation of Combined Cycle and Cogeneration Plants", *Thermoflow Inc.*, Wellesley, MA, USA.

Thermoflow, 1995b, "GT Pro - Software for Combined Cycle and Cogeneration Design", *Thermoflow Inc.*, Wellesley, MA, USA.

Thermoflow, 1995c, "Re-Master:Repowering of Conventional Steam Plants", *Thermoflow Inc.*, Wellesley, MA, USA.

Thermoflow, 1995d, "Steam Pro & Steam Master: Software for Design & Simulation of Conventional Steam Power Plants", *Thermoflow Inc.*, Wellesley, MA, USA.