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# AXIAL TURBOMACHINE MODELLING WITH A QUASI 2D APPROACH. APPLICATION TO GAS COOLED NUCLEAR REACTOR

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## ABSTRACT

This work concerns the design and safety analysis of Gas Cooled Reactors. The CATHARE code is used to test the design and safety of two different concepts, a High Temperature Gas Reactor concept (HTGR) and a Gas Fast Reactor concept (GFR). Transient simulations are performed and described in this paper: 10 inch cold duct break for both HTGR and GFR. The objective is to study decay heat removal situations after primary loop depressurisation. The turbomachine contribution is discussed and can offer an help or an alternative to "external" heat extraction systems. Furthermore turbomachine modelling is questioned. A new approach is developed : it consists in the solving of axisymmetric Navier-Stokes equations. An important effort has been developed to generate correlations in low mass flow and negative mass flow regimes, based on experimental data. The model is tested on open literature cases of the gas turbine aircraft community.

## KEYWORDS

Gas cooled reactor, Direct cycle, Safety analysis, Turbomachine modelling

## 1 INTRODUCTION

In the frame of the International Forum Generation IV, CEA is developing an innovative concept of high temperature gas cooled reactor with a fast neutron spectrum, a robust refractory fuel and a direct cycle conversion, as a promising milestone towards sustainable nuclear energy (Carré *et al.*, 2001). This concept of Gas cooled Fast Reactor (GFR) makes use of technologies derived from the High Temperature Gas cooled Reactor (HTGR) concept whenever possible, especially for the main primary system. Making use of gas turbines, which gave the fossil fuel plants a decisive gain in efficiency and competitiveness, is intended to make future nuclear systems also benefit from the same advanced conversion technologies. The design simplicity permitted by the direct Brayton cycle conversion constitutes an attractive economic advantage.

Thermal hydraulic is a key issue for the core design, the evaluation of the thermal stresses on the structures and the efficiency of decay heat removal systems (Tenchine *et al.*, 2002) : for the decay heat removal situations, the thermal hydraulic performance has to be demonstrated especially after a de-pressurisation accident. This de-pressurisation accident is a real challenge for the GFR, especially for passive heat removal systems which require a natural convection process. For the safety analysis, modelling of the global thermal hydraulic behaviour of the whole reactor is needed. It must include the primary circuit and the heat removal systems.

The second section of this paper is devoted to bibliography; then the paper describes the development of modelling (section 3) and simulation results using the CATHARE code for both concepts of gas reactors (section 4 are devoted to HTGR, section 6 to GFR). A transient simulation is performed for an HTGR concept (section 5). These transient simulation are carried out to study decay heat removal

situations after primary loop depressurisation of GFR (section 7). They mainly involve the power conversion unit. As a consequence turbomachine modelling is questioned. A new approach is developed (section 8). An important effort has been developed to generate correlations for compressors in low mass flow and negative mass flow regimes, based on experimental data (section 9).

## 2 BIBLIOGRAPHY OF DYNAMIC MODELLING OF DIRECT CYCLE GAS COOLED REACTORS

The intensive efforts made in Europe and in the United States to develop the helium HTR during 1960's, 1970's and early 1980's conducted to develop models of dynamic behaviour of closed cycle gas turbine plants. For the HHT project, a model has been built by Hannover Institute for Turbomachinery (Bammert and Grey, 1971) with a special emphasis on turbomachinery modelling and validated on fossil fired plant data (Oberhausen I: (Bammert and Posentrup, 1980)). For the American HTGR-GT project, General Atomics Company has developed a complete system code for helium cooled reactors with one or many (possible) power conversion units (REALY2 code: (General Atomics, 1976)). Each component of the circuit (core, secondary sides) are described dynamically. In 1990 Yan developed a new tool (GT-SIM: (Yan, 1990)) to study more precisely the dynamic control of the whole plant. The recent development of new projects of helium direct cycle HTGR have permitted to develop or adapt new models: South-African Potchefstroomse University built a model for PBMR and HTR project with a fine description of the pipe, core and heat exchangers dynamics (FLOWNET code (Greyvenstein *et al.*, 2002)). Dutch NRG Institute has developed a complete simulation system platform for HTR calculation based on coupling of multi-disciplinary and multi-scale codes ((Verkerk, 2000), (Kikstra, 2001)). Among them RELAP5 code (Relap, 1995) was used for the thermal-hydraulic transient simulation (mainly monodimensionnal using point kinetics or coupling to a neutronic code, 2D or 3D in the core). This platform was applied to different projects (ACACIA, PBMR). The RELAP5 base was also used at INEEL to perform calculations on recent (GTMHR: (Chan *et al.*, 1994)) or actual projects (PBMR) (RELAP5/ATHENA code (Moore *et al.*, 2002)). The Russian Institute OKBM has developed a system code GT-HTR (Bolnov *et al.*, 2003) based on similar assumptions (mainly 1D, 2D or 3D in the core). A platform for coupling multi-physics and multi-scale codes HTRSIMU has been developed to model HTR-10 program (Gao and Shi, 2002).

## 3 THE CATHARE CODE

The CATHARE system code has been developed and validated in collaboration between CEA, EDF, IRSN and FRAMATOME-ANP for the French Pressurized Water Reactors (Bestion *et al.*, 2002), but it has also been used for VVER, BWR, RBMK and experimental reactors applications. CATHARE has been adapted to the helium cooled reactor designs for steady-state and transient simulations (Tauveron *et al.*, 2003). A specific turbomachine module has been developed. It is fully described here after. In these reactors applications turbine and compressors are defined with their dynamic characteristics in a 0D approach:

- pressure drop and heat transfer are given by solving modified momentum transfer and energy equations,
- second member terms representing dynamic characteristics of each component are added.

The turbine, compressors and the generator components are coupled to a single shaft. The turbine, compressors and the generator components are coupled to a single shaft. The transient behaviour (turbine speed,  $w$ ) of the single shaft requires solving 1D shaft equation at each time step (equation 1). The generator is represented and connected with a single shaft structure. The power of the generator is a function of a swing angle  $\Phi$  (Kiameh, 2003). This angle is given by the computation of a differential equation (equation 2).

In this model, the following equations are solved to determine the turbomachine shaft acceleration and speed:

$$\frac{\partial w}{\partial t} = \frac{\sum \Gamma_{hydr} - \Gamma_{mech} - \Gamma_{elec}}{I} \quad (1)$$

$$\frac{\partial \Phi}{\partial t} = w - w_s \quad \text{where : } I = I_T + I_C + I_G \quad (2)$$

denotes the moment of inertia due to the turbine,  $I_T$ , the compressors,  $I_C$  and the generator,  $I_G$ .

The electrical torque,  $\Gamma_{elec}$ , is a function of the swing angle and nominal speed,  $w_{ref}$  :

$$\Gamma_{elec} = C_0 \sin \Phi \frac{w_{ref}}{w}$$

The hydraulic torque  $\Gamma_{hydr}$ , is the sum of three hydraulic torques for turbine, LP and HP compressors.

The mechanical friction torque  $\Gamma_{mech}$ , is the sum of friction torques :

$$\Gamma_{mech} = C_{f\_turbine} + C_{f\_LPC} + C_{f\_HPC} + C_{f\_alternator}$$

## 4 HTGR DESCRIPTION AND THERMAL-HYDRAULIC MODELLING

HTGR reactor unit consists of two integrated units: modular high temperature and power conversion system with direct gas-turbine cycle. The modular helium reactor is based on the use of a graphite core, fuel in form of microspheres with multi-layer ceramic coating and helium as the coolant. Helium temperature at reactor outlet is about 850°C and provides high efficiency of electric power generation in direct gas-turbine cycle. The power conversion system including closed gas-turbine cycle is completely arranged in a Power Conversion System vessel (PCS). The turbomachine consists of a generator, a gas-turbine and two compressors disposed in a single vertical shaft structure completely suspended on electromagnets. PCS includes also three compact heat exchangers: high efficiency recuperator, water-cooled pre-cooler and inter-cooler.

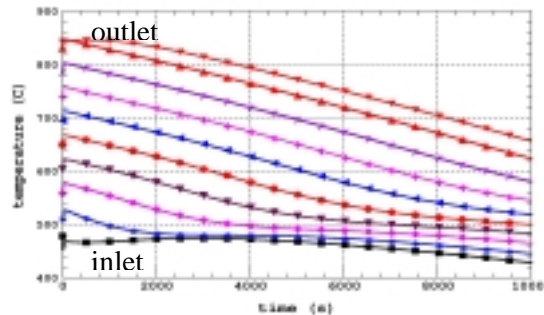
The model consists of a closed-loop primary system with helium as gaseous coolant and two open-loop secondary coolant system, pre-cooler and inter-cooler (helium/water). Pressure drop and heat transfer calculations through each component are based on classical correlations. The core is modelled by parallel 1D elements which can exchange heat through their walls. One element corresponds to the main flow through the fuel graphite blocks. The internal and external bypasses are modelled by two different elements. The recuperator is modelled as a gas to gas counter flow heat exchanger. The modelling of the pre-cooler and inter-cooler, with primary and secondary circuits, is similar to the recuperator one, but there is a gas to water heat exchanger. The volume module is used for the upper and lower plena in the core vessel, the Power Conversion System volume and some inlet and outlet regions of the main components (compressors, recuperator, pre-cooler, inter-cooler). The bypass valve is represented as a control-valve type (the opening rate is given as a function of the turbine over speed) and is located on the bypass pipe. The bypass pipe links high pressure compressor outlet and turbine outlet.

## 5 HTGR TRANSIENT RESULTS AND ANALYSES

The simulated transient concerns a depressurisation of primary circuit due to a cold duct break. It constitutes one of the most investigated event by the HTGR community (for example (Kugeler *et al.*, 2002), (Sugimoto *et al.*, 2002), (Dang *et al.*, 1980)). Furthermore, this simulations are purely “thermal hydraulics” and “thermics” oriented: no coupling with kinetics has been taken into account. For transient involving a rapid reactor trip (i.e. after a break) the assumption is almost justified.

The cold duct break leads to rapid depressurisation of primary loop and causes a decrease in the mass flow rate passing through the core. When the helium mass inventory is lower than 15% of its initial mass, reactor trip starts then the core temperature decreases. This simulation is carried out without turbomachine trip and without decoupling generator component from the grid during depressurisation accident. The transient starts with instantaneous cold duct break and leads to rapid depressurisation of primary side. When the helium mass inventory is lower than 15% of initial mass, the core power decay

is actuated and reduces reactor power, as recommended in (Damian, 2001). Reactor mass inventory decreases rapidly after the break. Turbomachine speed maintains a forced convection flow in primary loop at about  $15 \text{ kg.s}^{-1}$ . Figure 1 illustrates the core temperature profile. The cold duct break causes a short increase of the core outlet temperature before its decreasing. This is due to the core power decay. The large capability of the reactor to evacuate the energy explains the rapid temperature decreases in the core. There is no core overheat (final core temperature outlet at the end of the calculation :  $673^\circ\text{C}$ ).



**Figure 1:** Core temperature profile (transient HTGR-a)

It is a well-known fact that a maintained forced convection is not necessary to cool a HTGR core: radiation<sup>1</sup> and conduction are sufficient to extract the residual heat of the core. However this maintenance of a forced convection without any necessary action gives an idea to for the GFR (section 7).

### Turbomachine behaviour analysis

A main result of the simulation is that the turbomachine speed is constant during all the transient, 3000 rpm. An analysis of stability by linear perturbations is used to confirm these results.

Combination of equations (1) and (2) leads to a second order differential equation (3):

$$\frac{\partial^2 \Phi}{\partial t^2} = \frac{\sum \Gamma_{hydr} - \Gamma_{mech} - \Gamma_{elec}}{I} \quad (3)$$

This system can be described in another form:

$$\frac{\partial^2 X}{\partial t^2} = \sum_i F_i(X, \frac{\partial X}{\partial t}, t)$$

The dynamics of the system around a stabilized point  $(X_0, X'_0)$  is based on the study of the sign of some terms:

$$\frac{\partial}{\partial X_1} \sum_i F_i(X_0, X'_0, t) \text{ and } \frac{\partial}{\partial X_2} \sum_i F_i(X_0, X'_0, t) \text{ where : } F_i = F_i(X_1, X_2, t).$$

The linear stability theory indicates that the stability of the nominal point is given by the study of the sign of two terms 4 and 5:

$$-\frac{\partial \Gamma_{elec}}{\partial \Phi} \quad (4)$$

$$\frac{\partial \sum \Gamma_{hydr} - \Gamma_{mech} - \Gamma_{elec}}{\partial (\frac{\partial \Phi}{\partial t})} \quad (5)$$

In this analysis, some reasonable assumptions are made to complete the analytical resolution:

<sup>1</sup> Radiation between the core barrel to a duct shell, from the duct shell to the vessel and from the vessel to Reactor Cavity Cooling System (RCCS).

- the decrease of turbine and compressors inlet pressure depends only on time :  

$$P_{in} / P_{in,ref}(t) = \Pi(t)$$
- during the transient, variations of inlet temperature and velocity are negligible.

Results are the following:

$$-\frac{\partial \Gamma_{elec}}{\partial \Phi} = -C_0 \cos(\Phi) \frac{w_{ref}}{w} < 0$$

$$\frac{\partial \sum \Gamma_{hydr} - \Gamma_{mech} - \Gamma_{elec}}{\partial \left(\frac{w}{w_{ref}}\right)} = -a\Pi(t) - b, a, b > 0$$

The stability study is in agreement with CATHARE results: instability can not occur because  $a, b > 0$ , whatever  $\Pi(t)$ .

## 6 GFR DESCRIPTION AND THERMAL-HYDRAULIC MODELLING

The second concept studied is the Gas Fast Reactor (GFR). The main differences between the HTGR and the GFR consist in the core set-up whereas the power conversion system remains the same. The fuel is a performing version of “Cercer” delivering 600 MW in thermal power and the core dimensions lead to the very high power density value of 103 MW/m<sup>3</sup> (Garnier *et al.*, 2003). Concerning the modelling, one can refer to section 5. Notice anyway that contrary to the case of the HTGR, the GFR is made of only one core external reflector. Moreover, in order to prevent core wall damage due to excessive temperatures, the concept proposed is to settle a heat exchanger on the top of the reactor. The location of this exchanger is chosen to enhance the natural convection. The heat extraction is performed through a gas-gas (helium-nitrogen) heat exchanger modelled through a 30 heat sinks placed along the heat exchanger to ensure a progressive heat extraction.

## 7 GFR TRANSIENT RESULTS AND ANALYSES

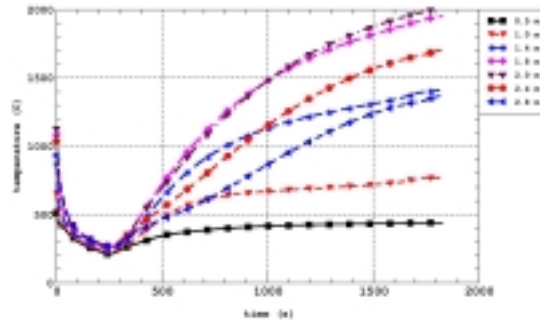
As for the HTGR study, the nominal steady state is first computed. The initial conditions in pressure and temperature used for the nominal state calculation are the same as for the HTGR computation. Concerning the boundary conditions, as for the HTGR simulations. For both transient calculations, the initial and boundary conditions are the same: the 10 inch break is a side oriented horizontal nozzle located in the middle of the cold duct and the fluid outside the circuit is supposed to be helium at a back-up pressure of 15 bar. At time zero, the cold duct break is open and depressurisation of primary loop begins. The core power decay is started as soon as the pressure in the lower plenum reaches 85 % of its initial value, which corresponds to 850 ms after the break opening. The main difference between the two calculations is the time when the turbomachine begins to trip. Indeed, a preliminary calculation without turbomachine trip has pointed out that as long as the turbomachine is running, no extraction from the heat exchanger can be made. The conclusions of this preliminary calculation naturally brought the two simulation cases performed hereafter:

- GFR a), the turbine is let free and the electrical torque control system is activated only when the velocity begins to decrease. As soon as the turbine is stopped, the heat extraction is started.
- GFR b), the turbine trip is forced by activating the electrical torque control system from the beginning of the calculation. When the turbine is stopped, the heat extraction is started.

### 7.1 10 inch cold duct break without forced turbomachine trip and heat extraction

The turbomachine speed begins to decrease 220s after the transient beginning. The generator is then disconnected and the bypass control valve opening rate is tuned to 10% in order to get a realistic speed decay duration (about 200s). The heat extraction is started at 370 s. Notice that a sensitivity test made without extracting the remaining power leads to temperatures in the core as high as 2000°C causing

the computation stop (as can be seen in figure 2, presenting the core wall temperature at different elevations) and highlighting the necessity of such an extraction system.



**Figure 2:** Fuel temperature profile if no extraction is made (case GFR-a)

## 7.2 10 inch cold duct break with forced turbomachine trip and heat extraction

No fundamental change is observed for this transient, except, as was noticed before, for the time when the different actions take place. The generator disconnection and the bypass control valve are activated from the beginning of the transient, leading to a turbine complete stop within 200 s. The fuel temperatures decrease at the beginning of the turbomachine trip, down to 330°C. This value is higher than for transient a) due to the fact that forced convection is lost at the beginning of the transient. After the extraction is started, in the same manner as for transient a), the same behaviour is observed with a fuel temperature increase up to 1150° C and then a slow decay.

## 7.3 Optimisation of turbomachine role for core heat extraction

The comparison between two last simulations (GFR a and GFR b) shows that turbomachine can help to extract heat from the core : maximum temperature in the core is 50°C less due to the machinery and appears around 150 s later. These two results are positive. The open question is thus: can this “passive” system be used during a longer time ? The case HTGR a) has shown that in favourable conditions the turbomachine can continue to operate for a long time even at a long level of pressure.

### 7.3.1 Favourable control system

A most favourable control system for the bypass valve was programmed to prevent machine overspeed and to pursuit turbomachine behaviour (parameters of PID were modified).

### 7.3.2 Thermal inertia

The turbine inlet temperature is a key point for the turbomachine behaviour. In section 5 we have shown that if inlet temperature of turbine (and compressors inlet temperature : the coolers maintained them at their nominal level) are maintained, the machine will continue to operate. The thermodynamic work done by the turbine is approximately proportional to the inlet temperature (if efficiency and pressure ratio do not vary too much). To our opinion, this is the same reason why turbomachine stops when inlet turbine temperature is too low (case GFR a). The idea is to design a larger wall at PCS inlet. This wall will be used as a thermal inertia and helps turbine inlet temperature to be maintained at a higher temperature than in previous section. The turbomachine is now stopped at 100 s later. The maximum fuel temperature is now 20°C lower and appears at 50 s later. The gain is not so large, but it can be a good way for future research.

### 7.3.3 Role of back-pressure

Back pressure is a very important parameter for GFR design, especially for heat extraction in incidental and accidental situations. At higher back pressure, natural convection is more efficient ((Gaillard *et al.*, 2003), (Bassi, 2003)). Lower back pressure must be tried as an alternative option. A lower back-pressure (10 bar instead of 15 bar) is favourable for turbomachine long behaviour (this is

coherent with case HTGR-a where back pressure was 3 bar): it is a well-known fact that a lower level of pressure generates via a turbo-compressor a lower mass flow rate, and thus a lower level of heat extraction of the core. As a consequence outlet core temperature is higher, so is the inlet turbine temperature, which is favourable to turbomachine pursuit as exposed in section 7.3.2..

### 7.3.4 Summary

Table 1 presents a summary of the results obtained.

	Turbomachine trip time (s)	Maximum fuel temperature (C)	Maximum fuel temperature time (s)
Reference condition (GFR a)	360	1100	900
Forced turbomachine trip (GFR b)	200	1150	750
Favourable control system (7.3.1.)	580	885	1700
Favourable control system and thermal inertia (7.3.2.)	680	865	1750
Favourable control system and lower back-pressure (7.3.3.)	1020	1060	2640

**Table 1** : Summary of different strategies for heat extraction

## 8 AXIAL TURBOMACHINE MODELLING

### 8.1 Introduction

Except for some versions of RELAP5 which represents a turbine as a nozzle (Relap, 1995), main previously cited (in section 2) tools simulate turbomachine behaviour with their dynamical characteristics (i.e. using performance maps) in a 0D fashion. The use of characteristics map itself can be questioned: they can not properly describe heat exchange with walls, and they are not generally fitted for stall or surge behaviour. These reasons have conducted Verkerk and Kikstra (2003) to study the differences between RELAP5 results and a more precise turbomachinery model (ACM) results. On typical transient, the agreement is quite good, except for some off-design points for the turbine. For "extreme" transients like break event a more precise approach can be necessary if we want to have a better confidence in the results obtained in sections 5 and 7. In particular we want to know if our analysis concerning turbomachine behaviour is not too optimistic. Indeed a phenomenon has not been taken into account in turbomachine modelling and can affect the proper turbomachine running: compressor surge phenomenon. windmilling reverse for turbine.

### 8.2 A strong limitation : the large number of stages of helium machines

A rapid review of different concepts of helium cooled turbomachinery can be found in table 2: it presents the number of stages of different turbines and compressors. The main conclusion is that the number of stages is very important, if you compare it to more traditional values for aircraft turboengines application.

Project or designer	Compressor	Turbine	Reference
GT-MHR	16+24	12	(Lecomte, 2001)
PBMR (220 MWth)	9+9	2+2+8	(Liebenberg, 1996)
GTHTR300	20	6	(Yan <i>et al.</i> , 2002)
TIT	?	9	(Kato <i>et al.</i> , 2002)
EVO II	10+15	7+11	(Melese and Katz, 1984)
HHV	8	2	(Melese and Katz, 1984)
SULZER (250MWe)	12+12+12	7+10	(Keller and Schmidt, 1967)
GA HTGR-GT (400MWe)	18	8	(Mc Donald and Smith, 1980)

**Table 2** : Examples of number of stages for helium turbomachine concepts

Notice that the number of stages is almost constant, except for smaller machines (HHV, or larger speed machines: PBMR, GTHTR300 and GA due to 3600 rpm). A rough explanation to this fact can be given. A popular method to pre-design a turbine is to use Smith diagram (Wilson and Korakianitis 1998). This diagram tells the designer that turbines should have a value for  $\psi$  approximately equal to a constant  $\psi_{ref}$ . As a consequence the number of stage for a helium turbine is:

$$N_{stage} = Cp(T_{in}-T_{out})/(\psi_{ref} \cdot R_m^2 \omega^2).$$

However:

- $(T_{in}-T_{out})$  is given by the thermodynamic cycle (around 400°C for considered cases)
- $\psi_{ref}$  is approximately constant due to Smith diagram
- $\omega$  is given by the electrical grid frequency (50 Hz or 60 Hz; this last value gives a lower number of stages, as indicated in table 2)
- $Cp$  is approximately constant for helium. The value is very important compared to air value
- $R_m$  is a radius which the designer can modify within a specified range. To lesser the number of stages the radius must increase. However for technological reasons  $R_m$  can not exceed a maximum value. For helium machines the centrifugal forces on blades is the limiting factor for the radius and therefore for the design.

The conclusion we wanted to illustrate by design examples and “theoretical” aspects is that all helium machines will have a very large number of stages. This fact is mainly due to the large value of  $Cp$  for this gas.

### 8.3 Basis of our model

Alternative models were developed to simulate long transients for machines in a more precise fashion than machine characteristic maps use. Davis and O'Brien (1991) used stage characteristic maps instead of global machines maps. The interest is that more local aspects can be described with this approach, such as local heat exchanges. However this type of model can not properly describe the effect of axial convection and volume effects. Schobeiri and Aboulkheir (1992) developed an approach based on velocity triangles, turning angle and loss correlations. An alternative step is our contribution. Our intermediary (between 0D and 3D) description of the turbomachinery has been developed based on a quasi-two dimensional approach. Although this type of flow analysis is a simplification of a complex three dimensional system, it is able to describe the behaviour of a compressor or a turbine with a better understanding than the models based on component characteristics. This approach consists in the solving of 1D axisymmetric Navier-Stokes equations. The assumption of quasi-steady behaviour is made: source terms for the lift and drag of each blade cascade of each row are derived from steady correlations. The assumption of quasi-steady behaviour is almost justified in numerous transient phenomena: the transit time of the working fluid through the turbomachines is below 10 ms, which is an insignificant delay in numerous transient phenomena if you consider variations of pressure, temperature, speed of the different components of the system (Hewing and Forster 1977).

## 9 LOW AND NEGATIVE MASS FLOW RATE COMPRESSOR MODELLING. APPLICATION TO SURGE TRANSIENTS.

### 9.1 Introduction

During the break event described earlier in the HTGR as in the GFR, the compressors are submitted to extreme off-design operating conditions. These conditions are associated with significant changes in mass flow, temperature pressure and sometimes rotational speed. For a compressor row the change in mass flow alters the velocity diagram and consequently the row parameters such as the flow angles and the total pressure losses. These parameters determine the efficiency and performance of the different rows and thus the entire compressor performance. A substantial reduction of a compressor mass flow can lead to instability occurrence. This instability can take the shape of low frequency fluctuations of the mass flow in the whole compressor system (surge). Surge can cause critical operating states with particular strong dynamic loading of the blading and can therefore not be tolerated during compressor operation. The objective is to develop a model which can simulate off-

design performance, forecast and describe some aerodynamic flow instabilities. However rotating stall, which is another instability phenomenon, can not be modelled because of the assumptions of circumferentially uniform flow and quasi-steady behaviour.

## 9.2 Low mass flow rate compressor modelling

A simplified model of low mass flow rate compressor behaviour has been developed, based on an extension of a simple stable model (Kroon and Tobiasz, 1971). The initial model for each row has been modified:

- Total pressure losses are calculated with the inlet relative or absolute velocity instead of outlet velocity. This modification is traditional for compressors.
- The downstream jet does not follow the blade geometry. A simple correlation of outlet angle which takes into account inlet off-design angle has been developed.
- A minimal relative amount of relative or absolute losses value has been set. Such value is also correlated to instability occurrence and “zero flow” performance.
- 10 different models were tested to predict this minimal loss value. The approach derived from Day, Greitzer and Cumpsty (1978) gives the best results.

Figure 3 shows the comparison between experimental values and simulated performance with our models, with a good guess of this minimal loss value.

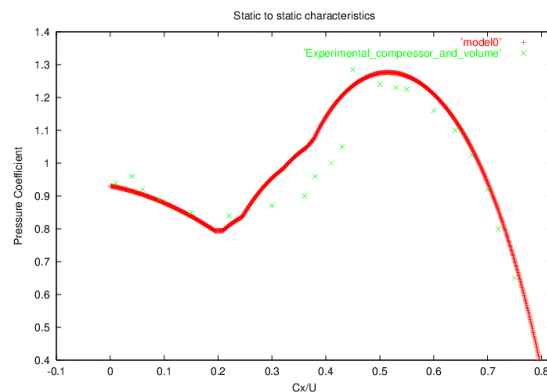


Figure 3: Comparison between simulation and experimental data from (Greitzer, 1976b)

## 9.3 Negative mass flow rate compressor modelling

There is a very little published literature on the subject of reverse flow compressor performance. To our best knowledge Gamache data (Gamache, 1985) are the (unique) reference ones.

A simplified model of negative mass flow rate compressor behaviour has been developed, based on an extension of a simple stable model. The initial model for each row has been modified:

- Total pressure losses are calculated with the inlet velocity.
- The downstream jet follows the inlet blade geometry, except for some very off-design stages.
- The downstream sections are reduced.
- The relative or absolute losses are always maximal.

Figure 4 presents the results.

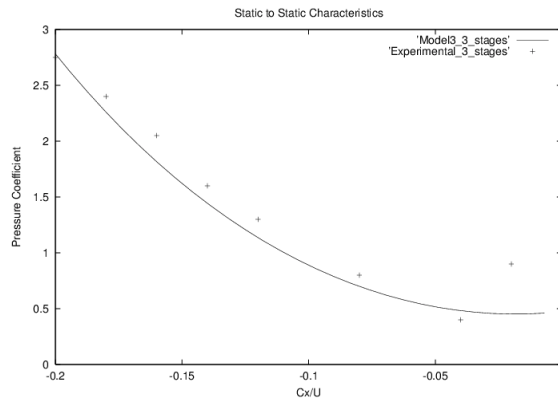


Figure 4: Comparison between simulations and experimental data from (Gamache, 1985)

## 9.4 Surge transients

### 9.4.1 Introduction

The dynamic behaviour of the compressor is tested in a configuration in which the compressor is connected with a plenum and a valve. When the valve is closed, the pressure ratio over the compressor rises. With the decreased flow the “surge-line” is passed and oscillatory behaviour results (see Figure 4). Greitzer (1976b) states that the shape and frequency of the flow oscillation is dependent on a parameter he defines (B) :

$$B = U/2a(V_p/SL)^{1/2}$$

Where: U is the rotational velocity, a is the acoustic velocity, V<sub>p</sub> is the plenum volume, S the cross-section, L the length.

### 9.4.2 Classical surge

Figure 5 exhibits the mass flow oscillations. Similar pressure oscillations are observed. The oscillations become periodical, but the shape is far from a sinusoid. A convenient way to represent the surge phenomenon is to use the time as a parameter (Figure 7 for example): we see the surge cycles. In the same graph we have represented the steady characteristics map.

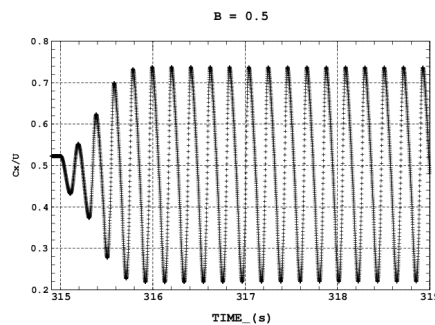
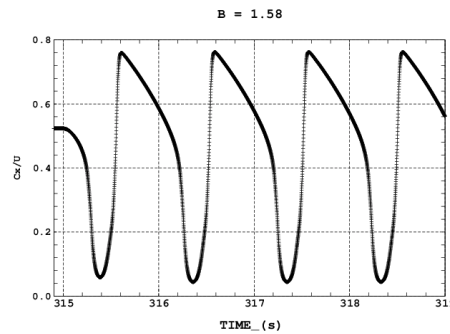


Figure 5: Simulation of surge phenomenon (mass flow oscillations)

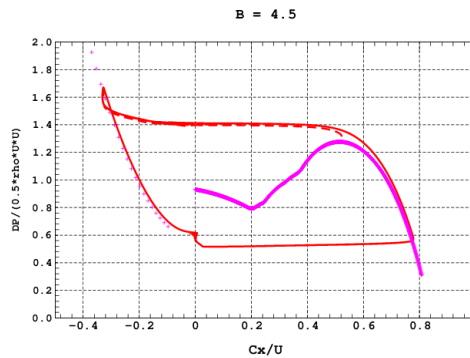
If B is greater, we can see that the cycles are larger and they tend to follow the characteristics map in its stable part (Figure 6).



**Figure 6:** Simulation of surge phenomenon (mass flow oscillations)

### 9.4.3 Deep surge

Oscillations of mass flow and pressure are developed: the mass flow rate becomes negative, the period is very small. The oscillations become periodical, but the shape is far from a sinusoid. We can see that the cycles are large and they tend to follow the characteristics map in its stable parts, at positive and negative mass flow (Figure 7).

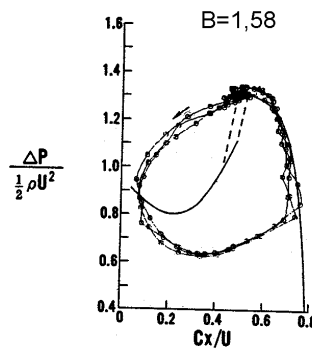


**Figure 7:** Simulation of surge phenomenon (surge cycle)

## 9.5 Elements of qualification

### 9.5.1 Classical surge

The comparison between simulated results (paragraph 9.4.2) and figure 8 is not easy as experimental conditions used in (Greitzer, 1976b) are not totally given. However the principal tendencies are in a good agreement, except for rotating stall (see paragraph 9.1).



**Figure 8:** Experimental investigation of surge phenomenon (B=1,58) from (Greitzer, 1976b)

The theoretical model gives a behaviour in  $1/B$  and a value of 2.71 for the slope. The simulated frequency with CATHARE gives a behaviour in  $1/B$  with a slope of 2.3. The simplified analytical model gives also a behaviour in  $1/B$  and a value of 2.71 for the slope.

### 9.5.2 Deep surge

The comparison between simulated results (paragraph 9.4.3) and experimental results is not easy as experimental and numerical conditions used in (Greitzer, 1976a) and (Greitzer, 1976b) are not extensively described. However the principal tendencies are in a good agreement. The simulated frequency with CATHARE gives a behaviour in  $1/B^2$  with a slope of 2.11. The simplified analytical model gives also a behaviour in  $1/B^2$  and a value of 2.56 for the slope.

## 9.6 Surge risk in case of cold duct break

### 9.6.1 Introduction

As we saw in sections 4 to 7, the compressors, during the break event, seem submitted to extreme off-design operating conditions, especially for the mass flow rate. It was presented in section 9 that mass flow rate decrease could lead to surge oscillating behaviour. The model presented in section 8 has been applied to both compressors (low pressure compressor and high pressure compressor).

### 9.6.2 Results

The model shows that, in the first cases considered, no surge occur during the transient, neither for low pressure compressor, nor for the high pressure compressor. These results confirm the analysis exposed in sections 4 et 7. Another interesting point is that surge risk appear if other location of pipe rupture are considered (bypass duct or recuperator for example).

## 10 CONCLUSIONS

This paper presents gas reactor transient analyses and illustrates the response characteristics of a closed-cycle gas turbine power plant. Two concepts are studied : a High Temperature Gas Reactor and a Gas Fast Reactor concept.

In case of a HTGR, turbomachine provides a forced convection which minimizes the severity of transient excursions during a cold duct break event. The transients analysis points out the major interest of the turbomachine behaviour, the control of its speed during the depressurisation transient : core power can be removed by the forced convection lead by the turbomachine.

Concerning GFR study, the attention is focused on the capability to remove decay heat after primary loop depressurisation consecutive to a cold duct break. Two scenarios are considered: with or without forced turbomachine trip. The results point out the setting of the natural circulation through a heat extraction system as soon as the turbomachine stops. The interest of the turbomachine behaviour is also emphasized to complete these heat removal systems: different strategies are studied.

At this step an important effort to improve turbomachine modelling is necessary as during the transient, the compressors are submitted to extreme off-design operation conditions. A description of the different elements has been developed, based on a 1D axisymmetric approach. This type of flow analysis is able to describe the behaviour of a compressor with a better understanding than the models based on component characteristics. An important effort have been developed to generate correlations in low mass flow and negative mass flow regimes, based on experimental data. The application of these models to HTGR and GFR transients is in progress. First results show that a cold duct break event does not lead to a surge risk, which is not the case for other break locations.

## NOMENCLATURE

$C_{f\_turbine}$	turbine friction torque
$C_{f\_LPC}$	low pressure compressor friction torque
$C_{f\_HPC}$	high pressure compressor friction torque
$C_{f\_alternator}$	alternator friction torque
$C_p$	Specific capacity
$C_x$	Axial velocity
$I$	moment of inertia
$I_t$	turbine inertia

Ic compressor inertia  
Ig generator inertia  
N Number of stages  
P pressure  
Rm Radius at mid-span  
T temperature  
U Shaft velocity  
 $\Gamma_{hydr}$  hydraulic torque  
 $\Gamma_{elec}$  electrical torque  
 $\Gamma_{mech}$  mechanic friction torque  
 $\Phi$  swing angle  
 $\Pi(t)$  time variation of pressure ratio  
 $\psi$  Enthalpy drop per stage  
 $\rho$  Density  
 $\omega$  Shaft frequency  
 $\omega_s$  Synchronous frequency  
in inlet  
out outlet  
ref reference

Pressure coefficient  $(P_{out}-P_{in})/(0.5\rho U^2)$

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