Modelling and optimisation of absorption system operation. Application to a real installation

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Key words : modelling of absorption systems, dynamic simulation, optimisation, gas consumption.

Summary :

We present the dynamic modelling of an absorption system and an application to a large air-conditioned system. This study was performed using the ALLAN.Simulation software, a pre- and post- processor enabling the symbolic description of systems. The absorption system is a direct gas double effect absorption chiller or heater, using LiBr/H_2O as working fluid.

This modelling has two aims. The first one is to optimize the water circuit by determining the volume of a storage tank that maximise the energetical and economic operation performances. The second objective is to evaluate by simulation the influence of the previous results on a large air-conditioned system and over given periods.

We describe the modelling approach and the results are compared with measurements monitored during summer 1994. The models created are available for use by engineers for sizing, optimizing or doing economic evaluation of HVAC systems.

Nomenclature :

C : thermal capacity ($C = \dot{m}Cp$), W.K^{-1};

Cp_c : specific heat capacity, J.kg^{-1}.K^{-1};

COP : coefficient of performance;

E : efficiency of a heat exchanger;

k : overall heat transfer coefficient, W.m^{-2};

h : specific enthalpy, J.kg^{-1};

$\dot{m}$: mass flow, kg.s^{-1};

M_c : total mass of a container, kg;

M_{LiBr} : total mass of LiBr in a container, kg;

M_{sol} : total mass of solution in a container, kg;

NTU : number of transfer unit;

P : pressure, Pa;

$\dot{Q}$: heat transfer rate, W;

S : total area of exchange, m^2;

T : temperature, K;

v : specific volume, m^3.kg^{-1};

x : concentration of the solution, $x = \dfrac{M_{LiBr}}{M_{sol}}$.

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Introduction

This study is a part of a project conducted by Gaz de France for the purpose of promoting air-conditioning solutions that integrate natural gas-fired production systems. The work under way includes several reference operations aimed at demonstrating the technical and economical feasibility of those solutions. These pilot operations are monitored to assess more accurately their real energy efficiency.

The technical system in this report is a real system. The intention of the work is to enhance assessment by simulating the system's energy-related behaviour and to optimise its efficiency.

The tools used here will be helpful to validate simplified models developed for the purpose of calculating predictive consumption costs before the conception of new air-conditioned and heated buildings. The future evolutions of the thermal regulation will prescribe these evaluations.

Context of the study

The CLIMGAZ project

This study is a part of the CLIMGAZ project, conducted by Gaz de France for the purpose of promoting air-conditioning systems that incorporate natural gas-fired production systems. The work under way includes several reference operations aiming to demonstrate the technical and economic feasibility of these systems. These pilot operations are being monitored to assess more accurately their real energy efficiency. The technical system in this report is one of those reference operations.

Another aim of this modelling project is to develop a model library of the standard elements which compose air-conditioning systems. The capability of making a modular description of systems will be helpful for research, development or design optimization in continuation of this work.

Modelling method and tool

This study was performed using a modelling approch developed by Gaz de France, to ensure the quality, reusability and most profitable use of validated models. This approach guides design engineers through different steps, in a V-cycle diagram shown by Figure 1.

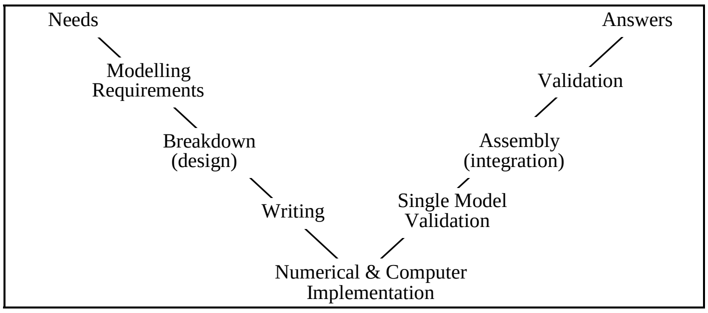

Figure 1 : The modelling approach

All models developped or reused for this study were written and integrated in the ALLAN.™Simulation environment, a program devised by Gaz de France [11]. It's a general purpose tool which uses « boxes and string » representation for modelling and simulating technical systems. Interactive graphic control is provided over the following :

– Description of basic component models, termed "simple models" and defined by equations, parts for discrete events description, and procedural text, external coupling variables on ports and graphical aspects ;
– Assembly of physical or technical system models termed "compound models" ;
– Simulator generation, with the algebraic/differential equation solvers DASSRT or NEPTUNIX ;
– Simulation operation (by batch processing or interactive control) ;
– Interpretation and display of simulation results.

Dynamic sizing of the system

Introduction

The aim of this phase is to enhance assessment by simulating the system's energy-related behaviour and to optimise its efficiency. Thermal models were developed to investigate the behaviour of a gas-fired absorption chiller using LiBr/H_2O. With the developed models, the effect of various parameters were numerically investigated. Dynamic simulation is a powerful tool to analyse the cycle and to determine the time-dependent reaction of the system to various disturbances. The main points of this study are:

– To determine the configuration of the water circuits between the building and the absorption system;
– To optimise the water circuits for use with the machine by determining the optimal volume of a storage tank needed to minimise the running-up time for the primary circuit;
– To determine how to control the chiller in order to minimise system operating costs.
– To determine the influence of cooling capacity sizing, and evaluate the economic impact of chiller over-sizing.

Modelling of the absorption chiller

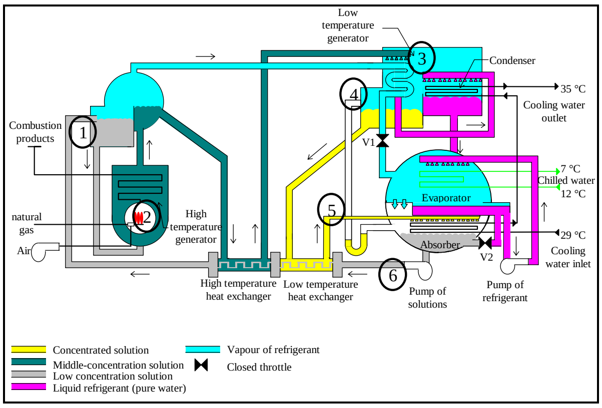

Figure 2 : Schematic diagram of a double-stage absorption chiller

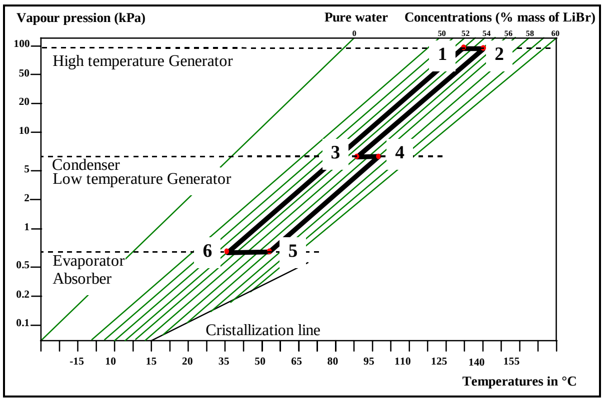

Figure 3 : Oldham's diagram of the solution states in a double-stage absorption chiller

Figure 2 presents a schematic diagram of a double-stage chiller. Figure 3 illustrates the evolution of the state of the LiBr solution during an operating cycle.

The weak solution from the absorber enters the high temperature generator

①-②, heated by a direct gas burner, and produces superheated water vapour which increases the concentration of LiBr solution. The solution flows into a heat exchanger and preheats the weak solution from the absorber ②-③. The solution then enters the low temperature generator. To improve the efficiency of the cycle, the vapour from the high temperature generator is condensed in the low temperature generator, at a lower pressure. More water vapour is produced and the concentration of the solution becomes stronger ③-④. Then the solution flows into a second heat exchanger which preheats the weak solution from the absorber ④-⑤. The cooling effect is obtained in the evaporator, and the vapour is absorbed into the strong solution from the generators. The heat of absorption is given up to the cooling water ⑤-⑥ and the low temperature weak solution flows through the heat exchangers to the high temperature generator ⑥-①.

General assumptions

- The fluids considered are binary solutions. In our case, they are LiBr/H_2O solutions;
- Each technical element is considered as a whole, defining a volume for which we write energy and mass balance equations;
- The fluids are assumed to be at the equilibrium state;
- It is assumed that containers and tubes are in thermal equilibrium with the working fluids;
- The velocity of the fluid and the pressures losses are ignored, but we take into account the heat losses in the generators and the absorber.
- In the mass balances, we ignore the vapour mass: the generators and the absorber are considered as a two-component liquid.

Absorbent/refrigerant properties

The thermodynamic and thermophysical properties of a large number of working fluids are now known and can be easily implemented for use within our models. At the time of printing, the models have been tested for LiBr/H_2O systems (LiBr as absorbent, H_20 as refrigerant). Different references were compared and the data we used for the thermophysical properties of refrigerants (specific heat, thermal conductivity, viscosity) were found in [1]. The material properties of LiBr solutions and H_2O refrigerant such as (P, T, X), (H, T, X), (cp, T, X) relations can be found in [2] and [3].

The Absorber and generator models

As for all models in this study, the governing equations are written in the same form for each element in dynamic state [4].

For the absorber, high-temperature and low-temperature generator models, an equation of mass conservation is written for each species.

For the total mass of solution:

$$\frac{dM_{sol}}{dt} = \sum_k \dot{m}_k \quad \textbf{(1)}$$

For the mass of LiBr, the definition of the concentration of the solutions gives :

$$M_{LiBr} = x.M_{sol} \quad \textbf{(2)}$$

$$\frac{dM_{LiBr}}{dt} = x\frac{dM_{sol}}{dt} + M_{sol}\frac{dx}{dt} = \sum_k \dot{m}_k x_k \quad \textbf{(3)}$$

Ignoring kinetic and potential energies, the energy balance on each fluid flow is reduced to an internal energy balance:

$$h\frac{dM_{sol}}{dt} + M_{sol}\left(\frac{\partial h}{\partial x}\right)_T \frac{dx}{dt} + M_{sol}\left(\frac{\partial h}{\partial T}\right)_x \frac{dT}{dt} + M_c Cp_c \frac{dT}{dt} - M_{sol} v \frac{dp}{dt} = \sum_i \dot{Q}_i + \sum_k \dot{m}_k h_k \quad \textbf{(4)}$$

For each entry and exit point, the pressure-temperature-concentration equilibrium is written in terms of:

$$F(P,T,X) = 0 \quad \textbf{(5)}$$

For the high-temperature generator, the power is supplied by a direct gas burner model. In the low-temperature generator, the power is given by de-superheating and condensation of gaseous refrigerant in a built-in condenser. The absorber is fitted with a heat exchanger model to remove the heat of mixing to the cooling circuit.

Heat exchangers

The absorption machine is fitted with many heat exchangers which are modelled by a simplified NTU method. These are:

- The solution heat exchangers: the fluids are LiBr/H_2O solutions at different concentrations;
- The internal heat exchanger of the absorber: the fluids are a solution of LiBr/H_2O and water from a cooling tower in our case. Note that the LiBr/H_2O solution is considered as a fluid in condensation, because it absorbs water vapour from the evaporator;
- The internal heat exchanger of the low temperature generator: the fluids are a solution of LiBr/H_2O and superheated water vapour from the high-temperature generator. This heat exchanger is considered as a condenser;
- The evaporator in which the cooling effect takes place;
- The condenser where the heat of the process is evacuated to the cooling tower;

All these heat exchangers are of counter-current type, and the efficiency is modelled by [5] :

$$E = \frac{1 - e^{-ntu(1-R)}}{1 - R.e^{-ntu(1-R)}} \quad \textbf{(6)}$$

with $R = \frac{C_{\min}}{C_{\max}}$

In the case of condensation and evaporation, the efficiency is given by :

$$E = 1 - e^{-ntu} \quad \textbf{(7)}$$

Control models

The other elements present in a real system were created, such as throttles, solution pump, burner, control device.

The control device model exactly reproduces the operation of that installed on a commercial system. It can simulate the startups and shutdowns of the chiller, with all the periods of preheating, or the sequence of cooling after a shutdown. These chiller startups or shutdowns are related to the cold water outlet temperature. A commercial unit is typically set to stop below an outlet temperature of 7°C, and to start above a temperature of 12°C. This control is reproduced by the model.

Validation with manufacturer's data

A two-stage absorption chiller assembly is presented in Figure 4. An initial validation was performed to compare the steady-state behaviour with manufacturer's measurements. The locations for comparisons are numbered on Figure 4. The excellent results obtained for a steady-state simulation are presented in Table 1.

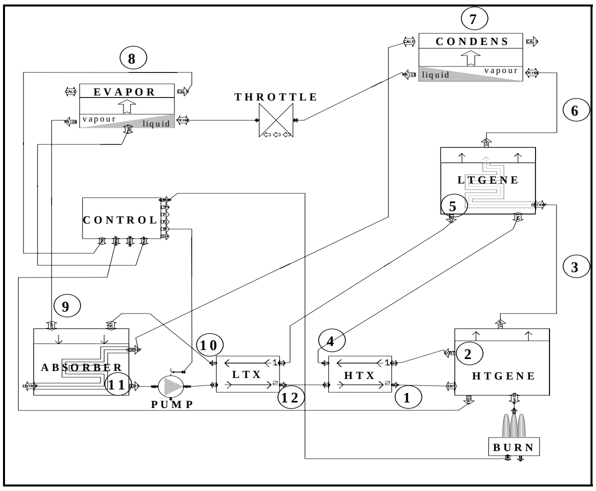

Figure 4 : The compound machine model

	Measurements			Simulation		
Point	T	X	Press.	T	X	Press.
1	135	53		136	53	
2	147	55	88000	145	55.2	95200
3	147	0		145	0	
4	75	55		72	55	
5	78.4	57	5880	77	57.15	5970
9	3.5	0		4.5	0	
10	38.9	57		38.1	57.15	
11	33	53	710	32.9	53	900

Relative power Meas.	Relative power Simul.
1	0.9857

T in °C, X in % mass LiBr, Press. in Pa

Table 1 : Comparison measurements-simulation

Definition of the simulation runs

Simulation of the building

To assess the behaviour of the system over a whole day, a model simulates the cooling demand of a building in the thermal conditions of a summer day. The profile of this load was measured on a real installation. We can calculate a cooling demand from 0% to 100% of maximum power, this maximum power being a simulation parameter we can change to simulate different levels of sizing of the system in comparison with the cooling capacity of the chiller.

A zero load means that no power is dissipated in the building by the office cooling devices. Nevertheless, heat losses on the circuit occur throughout the simulation and are independent of the building load.

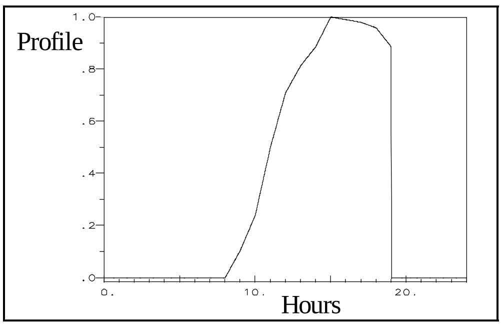

Figure 5 : Profile of the power given to the water circuit over a whole day

The water circuit configurations

Two water circuits are studied. In the « series » configuration, a simple circuit links the absorption machine with the terminal coils which are installed in the building. A storage tank can be installed on the inlet side of the chiller. This kind of circuit needs only one pump to maintain a constant water flow rate to the chiller. This is a standard configuration when only one chiller is connected to the building.

The « parallel » configuration is composed of two simple circuits separated by a expansion vessel. In this case, a pump maintains a constant flow rate in the chiller in the primary circuit. The secondary circuit is connected to the building, and needs a second pump. This circuit is fitted with a three-way valve, to control the temperature of the water to the building. This control is a linear function of the external temperature. For this circuit, it is also possible to install a storage tank in the primary circuit, on the inlet side of the chiller. This configuration is not standard for use with a chiller system, but is often used with heating systems. The advantage of this circuit is that the primary circuit can be preheated (boosting) before building occupancy begins.

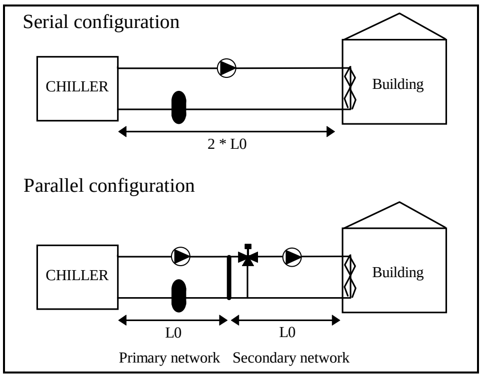

Figure 6 : Configuration of the water circuits

In the simulations, the two circuits are of the same length, and the expansion vessel is positioned in the centre of the parallel circuit. The circuits contain a total water volume of V_0.

The test procedures

In the different simulation runs, the following parameters vary as presented :

The load-rate: this is the ratio of the chiller cooling capacity to the maximum cooling demand of the building. A load-rate of 1.3 means that the system is 30% over-sized (***partial load***), and a load-rate of 1.0 indicates exact sizing of the system (***full load***). In all the simulations, we consider the cooling power of the chiller as a constant (P_0).The varying parameter is the maximum cooling demand of the building: for a load-rate of 1.0, P_{max} equals P_0, and P_{max} equals $P_0/1.3$ for a load-rate of 1.3.

Storage tank volume: the volume is rated to the chiller power. It varies between 0, 7, 11, 15 and 23 l/kW of the cooling capacity.

Chiller control: authorisation for cold water production is either continuous or discontinuous (authorisation from 7 h 00 to 19 h 00).

Boosting : boosting is used only in the parallel configuration. The 3-way valve is closed during pre-cooling, i.e., from 7 h 00 to 8 h 00. From 8 h 00 to 19 h 00, the terminal coils in the building receive cold water if needed.

Primary circuit pump (and secondary if needed): runs constantly.

Cooling water circuit pump: controlled by chiller operation. Start-up of the pump with the start-up of the chiller, and shutdown 3 min after the shutdown of the chiller.

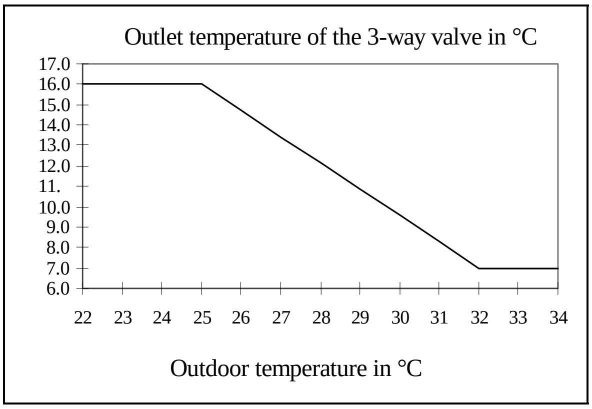

Figure 7 : Regulation law of the three-way valve

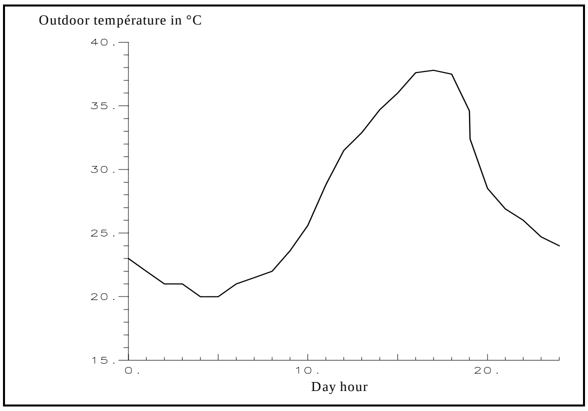

Figure 8 : External temperature in °C

Three-way valve : controlled with a linear function of the external temperature. Note that this valve exist only in the parallel configuration (see Figure 7 and Figure 8).

Operating costs : to make an economic evaluation of system modification, we take into account the energy costs for gas and electricity:

- Gas: 0.15 FRF/kWh (Gaz de France, B2I rate, summer days, 1994);
- Electricity: 0.22 FRF/kWh (Electicité de France, Yellow basic rate for a standard use, summer days, 1994).

Series configuration

Gas consumption and number of startups

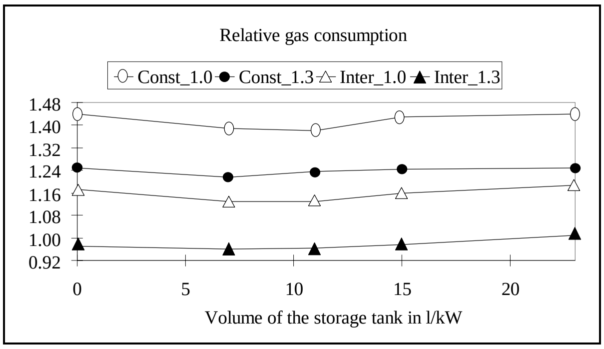

Figure 9 : Gas consumption

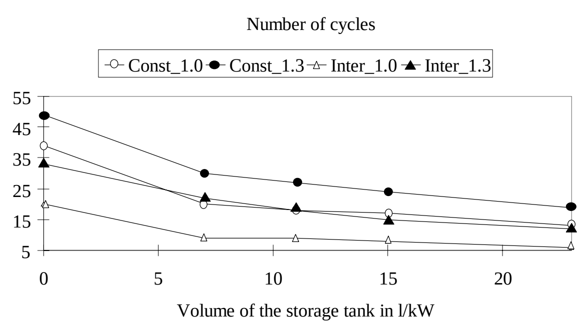

Figure 10 : Number of cycles

Number of startups : the number of startups logically decreases with the increase in storage tank volume, stabilising the running of the chiller (Figure 10). The number of startups drops sharply as the tank capacity rises from the smallest volumes (7 l/kW) then falls more slowly but on a continuous basis as volumes get larger.

Gas consumption: the first interesting result is the effect of the running mode. Gas consumption at partial load but in continuous mode is higher than consumption at full load but in discontinuous mode. This result shows the advantage of a discontinuous running mode compared to the continuous mode.

The second result is that the addition of a small storage tank can reduce gas consumption as shown in Figure 9. From a volume of 7 to 11 l/kW, gas consumption increases. This phenomenon is directly related to the number of chiller startups. A minimum gas consumption exists at about 7 l/kW when the system is working at full load. This minimum is at about 11 l/kW for a system running at partial load.

This shows that incorrect sizing of the chiller can possibly be counteracted by the addition of a storage tank on the circuit. Its size depends on the degree of chiller over-sizing.

Chiller performance

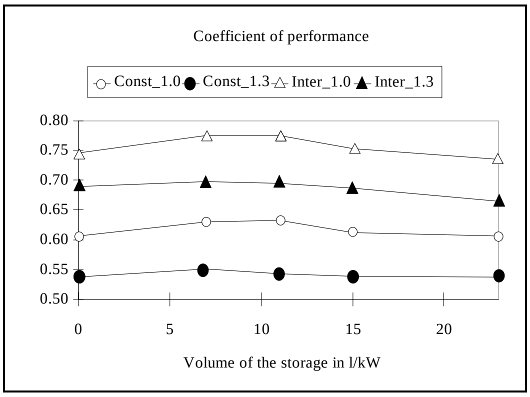

Figure 11 : Coefficient of performance

The Figure 11 presents the COP = $\dfrac{Energy\ produced\ by\ the\ chiller}{Gas\ consumption}$.

Effect of the discontinuous mode : as in the preceding results, the influence of the type of running mode is the most important factor affecting the energy performance of the chiller. In all the tests, the discontinuous mode gives the best operating COP, because of a smaller number of on-off cycles.

Effect of the sizing : chiller over-sizing results, in all cases, in lower performance levels, whatever the type of running mode, due to a high number of startups.

Influence of the storage tank : a maximum COP is observed for every sequence. A volume of about 7 l/kW in continuous mode and 9 l/kW in discontinuous mode maximises the COP.

Parallel configuration

Gas consumption and number of startups

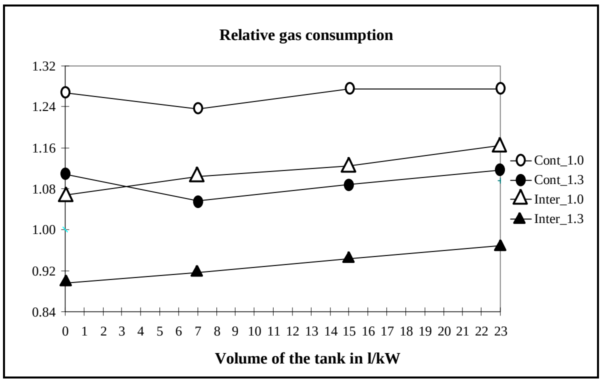

Figure 12 : Gas consumption

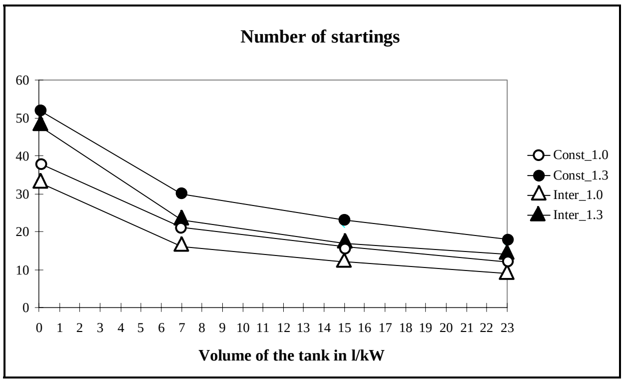

Figure 13 : Number of cycles

Number of cycles

The behaviour in the discontinuous mode is very similar to the behaviour in continuous mode: rapid decrease in the number of startups for small tank volumes (7 l/kW), and a slower decrease for larger volumes.

For a given volume, the difference in the number of cycles varies up to 25 for the same operating time. The larger the volume, the smaller the difference in the number of cycles.

For the test Const_1.3 (continuous mode, partial load), the presence of a storage tank makes a difference of 35 startups over 24 h of operation.

Gas consumption

For a continuous running mode, Figure 13 shows the existence of a minimum gas consumption for a volume of about 7 l/kW.

For a discontinuous mode, consumption increases regularly with volume.

Chiller performance

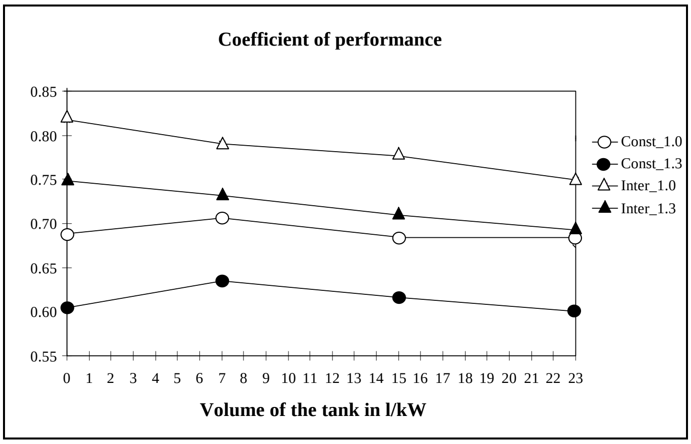

Figure 14 : Coefficient of performance production

The effect of the storage tank is different for the two running modes. In the discontinuous mode, the COP falls when the volume increases. This shows that the energy used to cool the water contained in the tank is higher than the energy lost in the cycles, whose number is low in this mode.

For the continuous mode, the situation is reversed. The very high number of cycles makes the presence of a large volume very advantageous for performance. Maximum performance is obtained for a tank capacity of about 7 l/kW.

Economic impact of the startups

We propose here to evaluate the economic influence of chiller startups. It is accepted that chiller over-sizing will have an impact on the chiller life-time, since the burner and other mechanical elements will be adversely affected by the large number of cycles (dilatation effect, mechanical vibrations...).

The cost of one startup has been estimated. The data given in this part are highly simplified. A correctly sized chiller is assumed to have 15 startups per day. It is assumed that the burner life-time is 10 years. Over a period of 100 days (maximum air-conditioning period), the cost of one startup is estimated at 3 U, based on the real cost of a new burner, where U is a virtual currency.

Results are presented for three examples of costs for one startup:

- C1 : no cost;
- C2 : 2 U/cycle;
- C3 : 4 U/cycle.

The previous tests are used directly, and we take into account all the energy costs plus the cost of startups. The following results present the operating costs of the system for various storage tank volumes.

The results are presented for a full-load sizing of the chiller.

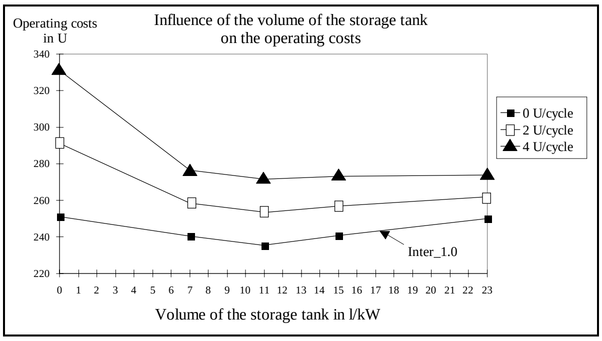

Figure 15 : Operating costs for the series circuit

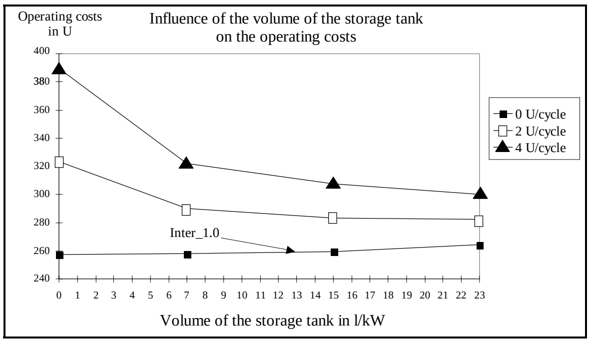

Figure 16 : Operating costs for the parallel circuit

Series configuration

The economic influence in this type of circuit is obvious. For the series configuration, it is very interesting to note that the volume that minimises the operating costs is the same for all three costs. This volume is close to 12 l/kW. The higher the cost of the cycle, the greater the cost gains thanks to the presence of a storage tank.

Parallel configuration

In all the cases, the discontinuous mode gives the lowest operating costs. This mode permits substantial gains (25%) with small storage tank volumes. Above a volume of 15 l/kW, the gains seem to be moderate.

Conclusion

Two cases must be considered, depending on whether a series circuit or a parallel circuit is required.

If the system is fitted with a chiller and a heater on the same circuit, then a parallel circuit is generally recommended. A series configuration is often used if only a chiller is connected to the circuit.

If a series circuit is used, the addition of a storage tank of about 11 l/kW is sufficient. The discontinuous mode is the very best solution to optimise the economic performance of the system. The simulation showed that the chiller has no problem in bringing the circuit water temperature down to the level required at the start of building occupancy.

If a parallel configuration is used, the operating costs are higher than in the series configuration. This slight difference is due to the use of two pumps. For this configuration, it is recommended to install a storage tank. The simulations did not show an optimal volume minimising the operating costs. It is nevertheless possible to say that a discontinuous running mode will give the best economic performance. A minimum volume of 15 l/kW will significantly reduce operating costs. The installation of a larger volume must be considered only after a more accurate economic evaluation.

Taking into account the volumes contained in the pipes, the total recommended volumes obtained by simulation that stabilise the system and minimise the operating costs are 14 l/kW for a series configuration, and 18 l/kW for a parallel configuration.

These values, obtained by simulation, are very close to empirical values recommended in technical references [6].

Application to a real system

Description of the system

The system studied is a commercial building located in south-western France, which has a natural gas-fired air-conditioning system [7].

It comprises a two-storey building with 25 offices in an air-conditioned area covering 500m². Hot (60/65°C) and chilled (7/12°C) water are produced by a refrigerating and heating absorption machine. This chiller-heater is a standard double-effect machine using the $LiBr/H_2O$ pair. The machine is cooled by a fluid cooler. A dual-duct air handling unit (AHU) treats the air, with a heat exchanger fitted with a bypass, an air washer, a heating coil and two reaction fans. Terminal coils provide additional heating or cooling in each room. The temperature in each room is controlled by a two-way valve on the supply line to the corresponding coils. The installation is monitored by a management system.

Position of the problem

The results obtained by numerical simulation of a dynamic absorption chiller model have shown that system performances can be improved by using an appropriate control method. Over-sizing of the chiller cooling capacity may have a negative effect on performance due to very unstable operation.

The real system presented above was monitored over a whole year. The measured performances and the operation of the absorption system were closely monitored. The objective of this part of the study is to validate the results obtained in the previous part on a real system. Our particular aims are:

- to optimise the primary water circuit for use with the machine by determining the storage tank volume needed to minimise the running-up time for the primary circuit;
- to evaluate real installation operating costs over a given period.

In addition, Gaz de France wishes to add a library of HVAC models to the existing Thermal Model Library for Buildings [8],[9].

Modelling of the system

The entire system was modelled [10]. The principal elements are :

- a simplified model of the chiller/heater with its cooling tower;
- a 2-stage multizone model of the building;
- a complete dual-duct air handling unit;
- the control and management of the system model;
- a water circuit in parallel configuration. The use of a storage tank is possible if needed.

The simple and compound models were fully validated using the measurements obtained over the year [9]. This brought to light various defects of the real system or differences with respect to initial specifications.

The internal loads due to equipments and human occupancy have been accurately calculated with the real data obtained, and will be used for the next simulations. We performed the calculation of the cooling and heating loads for the system, and we brought out some conclusions about the over-sizing of the cooling and heating capacities of the real absorption machine.

Three simulators were created, in order to compare them with each other.

The first one, named **REAL$_{\mathbf{SIM}}$**, reproduces the behaviour of the real system, with its defects. Information about a number of differences in the control and management of the air-conditioning devices was obtained indirectly with the help of technical references of the system, or directly by monitoring the measurements. The most important defect of the real system is chiller over-sizing, observed from measurements and from its behaviour. This results in frequent burner breakdowns and in very low energy efficiency.

The second one, named **SPEC$_{\mathbf{SIM}}$**, is the model of a system which exactly follows the initial specifications.

A modified system was proposed, **OPTIM$_{\mathbf{SIM}}$**, including optimum chiller sizing and the addition of a storage tank, whose volume is derived from the previous results.

In most cases, the differences between the 3 simulators are concentrated in the control model (temperatures in the building or the chiller/heater, continuous or discontinuous running mode ...). The differences are listed below :

REAL$_{SIM}$ *:* the real system

- Continuous chiller running mode; discontinuous heater running mode;
- Average cooling capacity of PC_{real}, and heating capacity of PH_{real};
- Incorrect chiller startup and shutdown temperatures (shutdown at 3°C rather

than 7°C in a correct setting);

- Incorrect temperature levels in the offices (22°C vs. 19°C in winter, 24°C vs. 23°C in summer);
- No storage tank.

***SPEC$_{SIM}$* :** the specified system

- Discontinuous running mode;
- Initial cooling capacity of PC_{spec}, and heating capacity of PH_{spec} , where $PC_{spec} = 1.59*PC_{real}$, and $PH_{spec} = 2.1*PH_{real}$;
- Correct chiller temperature settings (shutdown temperature of 7°C);
- Correct office temperature settings (19 °C in winter, 23 °C in summer);
- No storage tank.

***OPTIM$_{SIM}$* :** the enhanced system. The same as $SPEC_{SIM}$, with :

- Initial cooling capacity of PC_{opt}, and heating capacity of PH_{opt} , where $PC_{opt} = PC_{real}$, and $PH_{opt} = 1.5*PH_{real}$;
- High temperature levels in the offices in winter (22°C), and low levels in summer (23°C);
- A storage tank with a volume of 13.6 l/kW.

These 3 simulated systems were tested with the same meteorological data. Validation by comparisons with measurements were performed over more than one week for each seasonal running mode. The simulation over a whole year was performed from 04/01/94 to 03/31/95.

Simulation in chiller mode

Energy consumption over 9 days

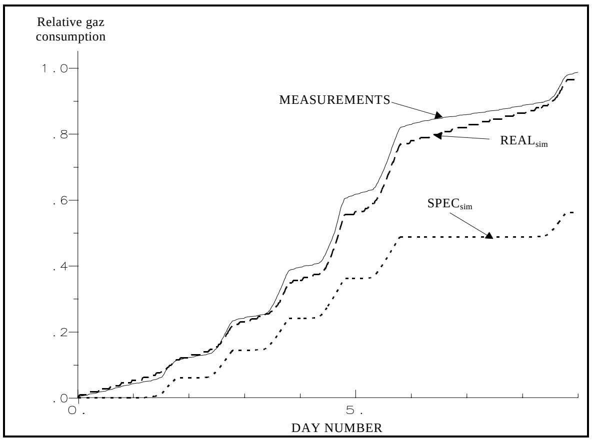

Figure 17 : Gas consumption

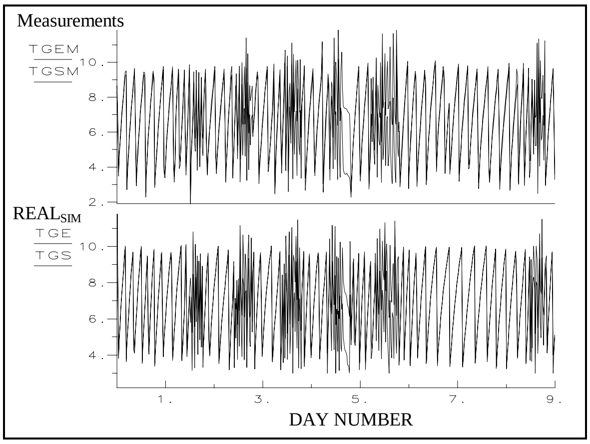

Figure 18 : Chiller inlet and outlet temperatures in °C

Figure 17 presents a comparison of the measured and simulated gas consumptions for the two systems $REAL_{SIM}$ and $SPEC_{SIM}$. At the end of a 9-day period (5 days of occupation and 2 week-ends), we can see the very good agreement between the measurements and the simulation for the same system, $REAL_{SIM}$. Small differences occur in most cases because of the duration of building occupancy.

A comparison between $REAL_{SIM}$ and $SPEC_{SIM}$ over this period shows that a 38% fall in gas consumption is calculated. This difference is significant, particularly if we remember that the temperature levels in the building are more restrictive for $SPEC_{SIM}$ (23°C) than for $REAL_{SIM}$ (24°C). $SPEC_{SIM}$ seems to be more effective, and its gas consumption is lower than those of $REAL_{SIM}$. This phenomenon is largely explained by the discontinuous running mode of $SPEC_{SIM}$, compared to continuous working for $REAL_{SIM}$. The optimum chiller temperature settings have a less pronounced effect.

Figure 18 shows a comparison of the measured and simulated chiller behaviour.

Simulation over the entire summer season

We present here comparisons for the 3 simulated systems over the entire « summer » period, i.e., from 04/01/94 au 10/31/94.

As shown in Figure 20, the working of $SPEC_{SIM}$ (specified system), is characterised by an excessive number of startups : this number may reach 45 per day, against 25 per day for the real system.

In $OPTIM_{SIM}$ simulation, the use of a storage tank has a significant impact in terms of the number of startups: a 50 % decrease may be obtained. The influence of more stable chiller operation is obvious in terms of gas consumption.

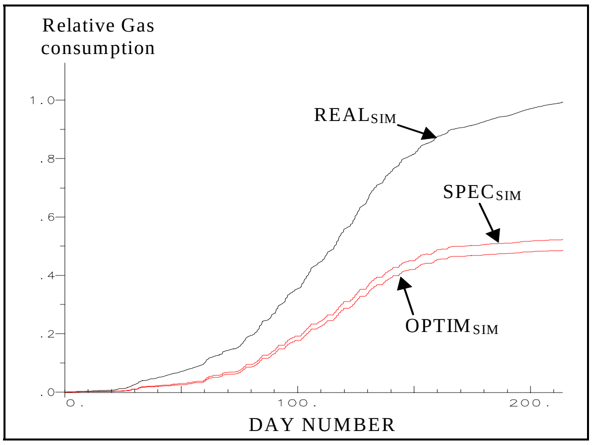

Figure 19 : Cumulated gas consumption

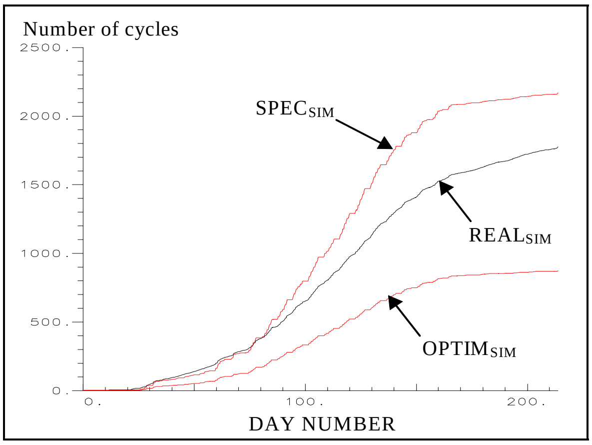

Figure 20 : Number of cycles

The superior performance of the simulated systems $SPEC_{SIM}$ and $OPTIM_{SIM}$ over the entire season is confirmed: total gas consumption decreases by 45% ($REAL_{SIM}$ compared to $SPEC_{SIM}$). An additional storage tank results in a gain of 10 % between $SPEC_{SIM}$ and $OPTIM_{SIM}$. Note that the temperature levels in the building are more restrictive for $OPTIM_{SIM}$, but consumption is lower.

By controlling the chiller efficiently and adding a storage tank, a total gain of 50% is predicted for the system, without adversely affecting comfort.

Simulation in heating mode

Energy consumption over 8 days

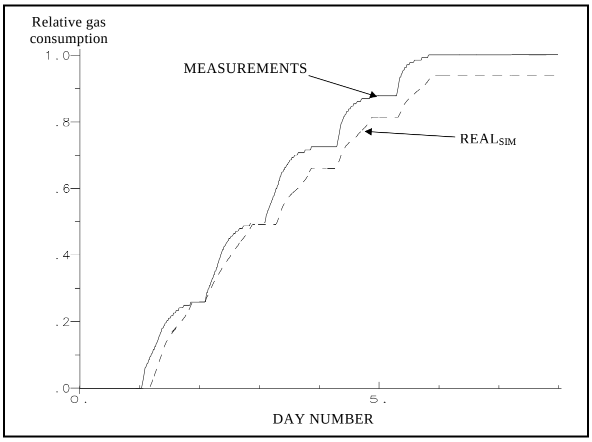

Figure 21 : Gas consumption in kWh

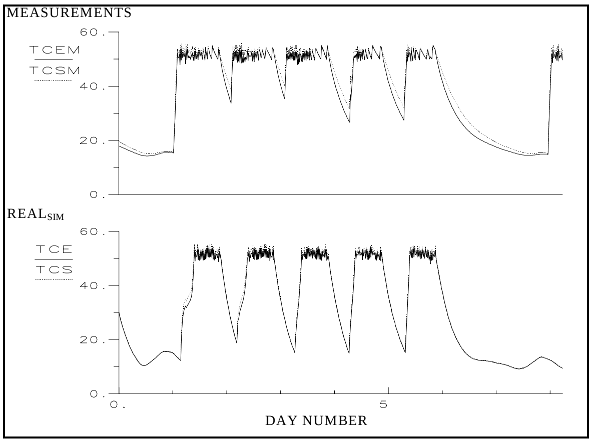

Figure 22 : Heater inlet and outlet temperature

The Figure 21 shows a comparison of the gas consumptions between measurements, $REAL_{SIM}$ and $SPEC_{SIM}$. Part of the difference (10%) between measurements and simulation ($REAL_{SIM}$) is due to an approximation of starting time control for the entire system since the management system manufacturer was unable to give us the exact algorithm to calculate this time.

The consumption difference amounts to 30% between the two simulated systems $SPEC_{SIM}$ and $REAL_{SIM}$. The gain is around 16% between $REAL_{SIM}$ and $SPEC_{SIM}$. The greater part of this gain is due to low temperature settings in the building for $SPEC_{SIM}$ (19 °C for $SPEC_{SIM}$ against 22 °C for the real system $REAL_{SIM}$).

Simulation over the entire winter season

We present here a comparison of the 3 simulated systems over the entire « winter » period, i.e., from 04/01/94 au 10/31/94.

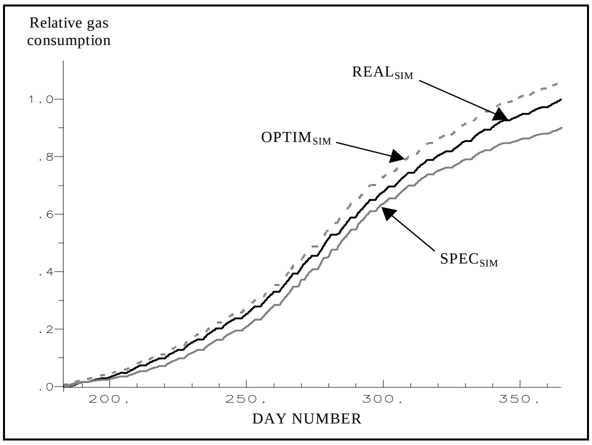

Figure 23 : Gas consumption

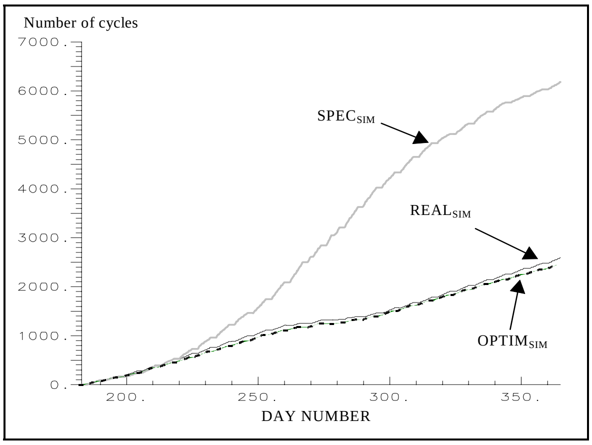

Figure 24 : Number of cycles

In Figure 24, we can see that the behaviour of $SPEC_{SIM}$ is characterised by a very high number of cycles, and that the difference in the number of cycles between $REAL_{SIM}$ and $OPTIM_{SIM}$ is very small, due to the presence of a storage tank, although the heating capacity in $OPTIM_{SIM}$ is higher than for $REAL_{SIM}$.

In terms of gas consumption, the gains for $SPEC_{SIM}$ are moderate (11%). This is in fact due to the temperature levels in the building for $SPEC_{SIM}$, which are lower than for $REAL_{SIM}$. The consumption decrease could be larger, but the over-sizing of $SPEC_{SIM}$ substantially affects its performance.

The higher gas consumptions for $OPTIM_{SIM}$ are explained by the temperature setting in the building (24°C), and the higher heating capacity of the heater (system well sized).

In the general case, the addition of a storage tank seems to have moderate effects in the heating mode, particularly if the system is correctly sized.

Conclusion

The simulation has shown the influence of the sizing of an absorption system. In all cases, the general performances of the system decrease when the power of machine is over-estimated. An excessive number of system startups gives rise to heat losses when the chiller/heater stops or starts operation.

Effect of discontinuous operation of the absorption system

The interesting result obtained in the cooling mode is the fact that discontinuous system operation always give the best performances. Like in the heating mode, a boosting of the chiller is sufficient to prepare the system for the start of building occupancy. Gains of up to 40% may be achieved without losses in terms of comfort.

The use of a storage tank

The addition of a storage tank on the inlet side of the absorption machine results in more stable operation and in a decrease in the number of startups during part-load periods. Resulting gains of 10% can be expected. The addition of a storage tank is highly recommended in the case of a severely over-sized system, and technical problems like burner failures are possible when the number of cycles is excessive. The presence of a storage tank seems to be less important in heating mode than in cooling mode.

Note that the calculated volumes that minimise the influence of the startings of the chiller are in very good agreement with the empirical values found in technical references [6].

Perspectives

The dynamic model of absorption machine is being used with other control solutions, such as multi-stage burner power rates, or continuous power modulation, to test the possibility of adjusting the capacity of the system to the load.

The use of the simulation

The tools used for this study brought to light the feasibility of very large systems simulations. The most important results are related to the sizing of the chiller. The dynamic simulation showed that a classical heat load calculation often over-estimate the cooling and heating capacity. The influence of this problem on the performances is evaluated, and solutions are proposed in order to minimise it.

A large number of models were created for this study and are not presented here, such as classical control models, a multi-zone building model, or the entire control and command device. Finally, the compound model of the entire air conditioned system make it possible simulations on a whole year's for efficiency, cost comparison or optimization of a given system.

Within this study, Gaz de France assessed more experience to promote natural gas air conditionning systems, by proposing efficient solutions for design, monitoring and maintainance.

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