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ANN APPLICATION TO MODELLING AND CONTROL OF SMALL ABSORPTION CHILLERS

Jerko Labus[1], Ivan Korolija[2], Ljiljana Marjanovic-Halburd[3], Yi Zhang[2], Alberto Coronas[4]
[1]Faculty of Mechanical Engineering, University of Belgrade, Belgrade, Serbia
[2]Institute of Energy and Sustainable Development, De Montfort University, Leicester, UK
[3]Bartelett School of Graduate Studies, University College London, London, UK
[4]CREVER-Group of Applied Thermal Energy, Universitat Rovira i Virgili, Tarragona, Spain

ABSTRACT

The main aim of this paper is to demonstrate the application of Artificial Neural Networks (ANN) in small absorption chillers modelling and their control optimisation. The Genetic Algorithms (GA) optimisation method was coupled to the ANN model in order to solve the optimal operation problem where the objective function was the minimal primary energy consumption. This paper analyses the impact of control strategy on energy performance of small capacity absorption chillers, while emphasizing the usability of ANN model, and comparing this strategy to conventional operation strategies.

INTRODUCTION

Absorption Chillers

An absorption chiller is a cooling machine which uses a heat source to provide the energy needed to drive the cooling system, instead of electricity used by conventional compression machines. The absorption chiller is based on sorption process where a liquid or solid sorbent absorbs refrigerant molecules and the resulting mixture starts to change physically and chemically during the process that follows. The working principle is based on different boiling temperatures of the refrigerant and the absorbent. The single-effect NH_3-H_2O absorption chiller, shown in Figure 1, consists of a generator with rectifier (G), a condenser (C), an evaporator (E), an absorber (A), solution heat exchanger (SHX), solution pump (P) and two throttling valves. External heat input to the generator causes the refrigerant to boil out of a solution. Once in the vapour state it is compressed at higher pressure while the concentrated absorbent stays liquid. Rectifier is an additional device which provides the purity of the refrigerant vapour before entering to the condenser. Inside the condenser, refrigerant vapour condenses by transferring the heat through the external heat carrier (usually water) circuit. The external heat carrier circuit is typically connected to a cooling tower or a dry cooler enabling the heat rejection to the surrounding. The condensed liquid refrigerant flows through an expansion device into the evaporator. Inside the evaporator, the refrigerant receives the heat from another external heat carrier circuit and evaporates, producing the cooling effect on the circulating water. The low pressure refrigerant vapour then proceeds to the absorber where it is absorbed by the strong absorbent solution. The absorbed refrigerant vapour condenses to a liquid, releasing the heat received in the evaporator to the external heat carrier circuit. This circuit is usually designed to be common for absorber and condenser (cooling water circuit). The weak absorbent solution is then pumped through the solution heat exchanger to the generator where heat is used to separate the refrigerant again.

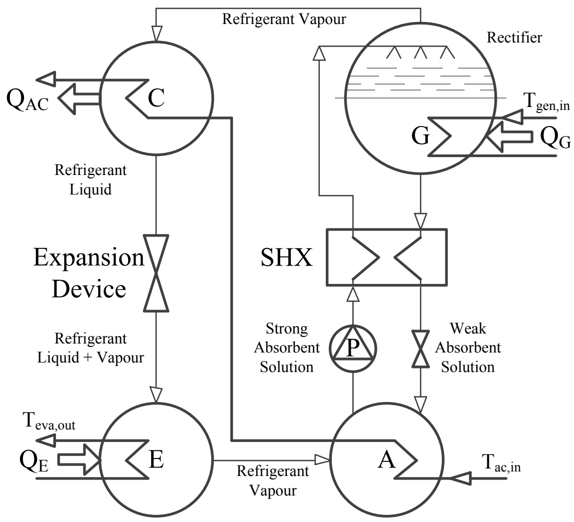

Figure 1 Absorption chiller

Absorption chiller control strategies review

The control of absorption chillers can be internal and external. The internal control is usually defined by the manufacturer's design with the main function to maintain the chiller’s internal parameters, such as concentrations, flow rates, pressures and temperatures. Internal control enables smooth operation and prevents damage within operational limits defined by manufacturer. The external control of an absorption chiller deals with parameters of external heat carrier circuits in order to satisfy cooling demands. These two types of control are interrelated. Changing any of the exterior circuits' parameters has the direct influence on the internal states of the chiller, where the role of internal control is to adjust and keep them within acceptable limits which will not jeopardize the normal operation. An absorption chiller is attached to three external circuits: hot medium circuit (connected to the generator); cooling medium circuit (passing through the absorber and condenser); and chilled medium circuit (connected to the evaporator). Water is the most commonly used medium in all three circuits. The external control parameters are hot medium inlet temperature and mass flow rate, cooling medium inlet temperature and mass flow rate and chilled medium outlet temperature and mass flow rate.

The control of large-scale absorption chillers is usually obtained through the hot medium circuit. When designing a control strategy of an absorption chiller in air-conditioning applications, the primary objective is to control the capacity by maintaining the temperature of the chilling medium leaving the evaporator. The internal control system compares the temperature of the leaving medium to the set point and adjusts the amount of solution supplied to the generator by adjusting heat input to the generator. The combination of solution temperature and concentration determines the temperature at which the refrigerant vaporizes in the evaporator. Thus, in order to control the output of the chiller to meet the changing cooling loads, either the solution temperature or the solution concentration must be varied. A common method to vary the temperature of the solution is to vary the amount of absorbent solution delivered to generator either by using throttling/bypass valve or by using an adjustable frequency drive to vary a speed of the generator pump's motor. In order to vary the solution concentration, absorption chillers vary the heat input to the generator. Nowadays, almost all large-scale absorption chillers are equipped with a microprocessor-based control panel that monitors and controls all operations of the machine. The common way for external control of large capacity absorption chillers is through the modulating energy valve. Two variants of this control are the most common. The first variant is to modulate heat medium (usually steam or hot water) flow from 10-100%. The second variant is to adjust the heat medium inlet temperature. The use of modulating energy valve in conjunction with internal control for variable solution flow rate, especially with adjustable frequency drives, enables faster chiller response time to the changing load and cooling medium conditions.

Conventional absorption chiller control logic was explained in detail by Jenkins (2003). A capacity valve controls heat supply to the absorption chiller by using PI controller, which was defined by a non-linear function. Mann and Stewart (1963) and Anderson (1966) described the control of the absorption systems which employed two separate temperature sensors, one which measures the temperature of the medium entering the evaporator and another which senses the temperature of the medium after it has been chilled. The control was adapted to vary the refrigeration capacity of the system in response to varying load conditions and not in response to changes in internal conditions in the refrigeration system itself. Ogawa et al. (1992) introduced the fuzzy logic rules for a PID controller in a control system for an absorption refrigerator. The heat supplied to the generator was in direct correlation with the outlet temperature of the chilled water. Yeung et al. (1992) described a simple control mechanism for chilled and cooling water temperatures in a solar air-conditioning system in Hong Kong. The control mechanism monitors the chilled water temperature and when it dropped below the set point, the hot water supply to the chiller was cut off. The cooling water temperature was controlled with a differential controller in on-off mode. This control strategy resulted in a fluctuating operation of the chiller with increased thermal losses due to a higher number of start-up and shut-down procedures.

Koeppel et al. (1995) investigated the optimal supervisory control of the equipment in a cooling plant with double effect absorption chiller in order to minimize the total operating cost. They developed a simplified strategy for cooling tower fan control, which involves the determination of a linear relationship between a set point for cooling tower supply water temperature and ambient wet-bulb temperature. Chow et al. (2002) developed a GA control strategy for the optimal use of fuel and electricity for a direct-fired absorption chiller which resulted in significant energy savings. The possibility of supervisory and optimal control strategies of HVAC systems were explained comprehensively by Wang and Ma (2008).

In contrast to large-scale absorption chillers, which have been present at the market for a long time, the interest for small absorption chillers has only been raised recently. With only a few commercial units and with small number of expert groups working in this research field, the control methods are still in the phase of development and testing. The usual control for small capacity absorption chillers is ON-OFF operation (Lazzarin, 1980, 2007a, 2007b). The main problem for this type of control is the thermal inertia. When the chiller is turned off, it cools down; if the following ON period is subsequently delayed by the thermal inertia, the reduction of COP is significant. Lazzarin considered two other control strategies. The first strategy used the hot water inlet temperature to modulate the capacity of the chiller by more than 50%. The second strategy was to modulate the hot water flow rate, which resulted in the capacity reduction of close to 60%. Finally, Lazzarin (2007a, 2007b) considered the option to use both strategies. However, the results showed that the decrease of COP was unavoidable whenever the machine is modulated to meet the cooling demand.

The catalogues of the commercial units (Rotartica, 2007; Chilli, 2008; Yazaki, 2009) show that most of small capacity absorption chillers are controlled either by simple ON-OFF cycling or by modulating valve in the hot water circuit. When absorption chiller is started, it operates automatically and remains in operation as long as there is a cooling demand. The hot water circuit pump or bypass valve is cycled ON and OFF to control the flow or temperature of hot water supply to the generator in response to the chilled water temperature. When the chilled water temperature reaches the required level, the solution dilutes and the pump in the chilled water circuit stops. The chilled water temperature then starts to rise, until it reaches the differential temperature parameter added to the set point the chiller turns on again. The set points of the hot water temperature are adjustable only within the strict range prescribed by the manufacturer.

To achieve best efficiency, small absorption chillers should operate in full load conditions continuously. This is of course impossible in real world applications, where cooling loads as well as parameters of the hot, cooling and chilled water circuits fluctuate with time. Optimal control for chillers thus requires taking the complete heating and cooling energy system into account. The design of cooling system components, e.g. cold and hot storage tanks, solar circuit, back up heater, cooling tower or air condenser and terminal units, may affect the operation strategy of the chiller.

Storkenmaier et al. (2003) used TRNSYS environment to develop a control strategy for a 10 kW small absorption chiller in the solar cooling system of an office building in Berlin, Germany. The control strategy of the system was a combination of an on-off and proportional control in the solar and storage loop and a PI-control for the external chiller circuits. Two continuously variable three-port mixing valves controlled the supply temperature of hot and cooling water of the chiller, thereby providing the necessary control of the cooling capacity. Performance control of the chiller could also be reached via a combination of mass flow and temperature control of the external heat mediums but was limited due to reduced overall heat transfer coefficients of the chiller at lower flow speeds. The TRNSYS type for absorption chiller was developed by using the characteristic equations for absorption chillers as described by Ziegler et al. (1999).

One of the conclusions which can be drawn from the literature review is that a differential on-off operation of the main external circuit pumps should be avoided. Although such a control is simple to install and to operate, the system performance decreases due to frequent on-off operation of the chiller and therefore cannot be justified only with the simplicity. In other words, a mass flow and/or temperature control should be installed in the external circuits.

After studying the control of the cooling water temperature, Kohlenbach (2006) pointed out that if the purpose of a solar cooling system is to provide a defined cooling capacity at preferably low electrical energy consumption, then the cooling water temperature set point should be kept as high as possible. Otherwise, if the purpose of a solar cooling system is to provide maximum cooling capacity and if the electrical power consumption is of secondary importance, then the cooling water temperature set point should be kept as low as possible. Recent studies (Clauß et al., 2007; Albers et al., 2008; Kühn et al., 2008) considered the control strategy based on the characteristic equation model which determines the required cooling water temperature. If the hot water temperature is not high enough for a given cooling load, cooling water temperature is lowered and vice versa, when the driving temperature is high cooling water temperature is increased. The results showed that the new strategy provides stable chilled water outlet temperature and also decreases parasitic electricity consumption of auxiliary equipment with respect to the conventional strategy.

Existing literature shows that small absorption chillers can be controlled in various ways. In addition to the default ON-OFF operation, four strategies are the most promising. The first one is adjusting the hot medium inlet temperature in order to control the outlet chilled water temperature. The second method is to control the inlet cooling water temperature to maintain the chilled water set temperature at required part load operation when hot water temperature is constant. The adjustment can be done by using the frequency inverter or three-port valve, depending on heat dissipation component (cooling tower, dry cooler). The third method, which maybe the optimum method, is adjusting hot and cooling water temperature simultaneously. The fourth method, the control of the absorption chillers by adjusting flow rates, is recommendable only if it is allowed by the chiller's design.

The realization of any of the former three control strategies requires a controller for maintaining the set temperature. In a practical application, the correct parameters of such a controller have to be identified for stable system operation.

This paper is trying to close this gap by investigating the applicability of Artificial Neural Network approach in modelling and control of small absorption chillers. The generalized system taken in this research assumes that this controller would adjust the temperatures in the external circuits through the three-port valves. The ANN model of the small absorption chiller was developed based on the laboratory experiments. In order to compare control strategies, the EnergyPlus model of the small the single-zone retail space located in Spain was developed and the annual cooling load profile was determined. The objective function during the optimisation process was absorption chiller COP. By maximising the chiller's COP value, the primary energy consumption would be kept at minimum.

MODELLING AND SIMULATION

Absorption chiller ANN model

An ANN model of a small capacity absorption chiller Pink-Chilli PSC12 (Chilli, 2008), schematically presented in Figure 2, was developed from the experimental data obtained in the state-of-the-art test bench at the Rovira i Virgili University in Tarragona, Spain. The experimental dataset includes 138 steady-state measurements covering the chiller's full operating range. The inlet hot water temperature ($T_{gen,in}$) was varied between 80 and 100ºC; the inlet cooling water temperature ($T_{ac,in}$) was varied between 27 and 35ºC; while the outlet chilled water temperature ($T_{eva,out}$) was varied between 5 and 12ºC. The number of data points used for the ANN model development was increased by applying regression technique to the experimental data. This resulted in more detailed and accurate database with around 1260 data patterns for modelling. More details about the functionality of the test bench with the comprehensive explanation about the test procedure as well as the post-processing of the experimental results can be found in Labus (2011) and Labus et al. (2012). The ANN model of the absorption chiller was developed by using MatLab's Neural Network toolbox. By applying trial and error method rule to determine the number of neurons in the hidden layer and the number of hidden layers, the adopted topology for ANN models was 3-7-2, as illustrated in Figure 2. The training of the ANN was based on error back propagation technique using the Levenberg-Marquardt algorithm of optimization. The input parameters were normalized in the [0.2, 0.8] range. To test the robustness and the prediction ability of the model, the experimental dataset was split into three parts: 70% of data was used for the model training, 20% for the model validation and the remaining 10% for the model testing.

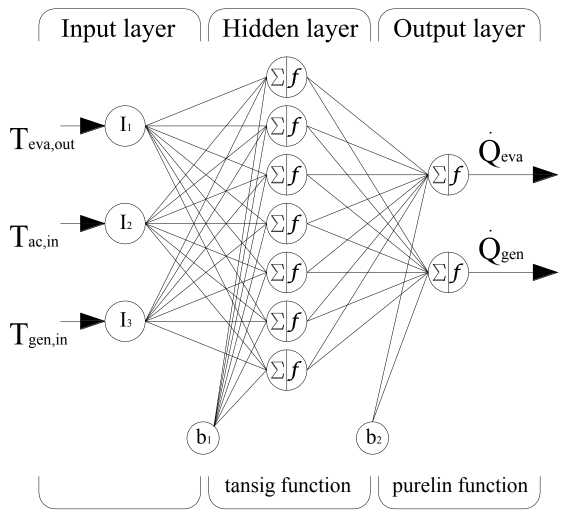

Figure 2 ANN topology

Cooling load profile

Cooling load profile, which was used for testing the chiller optimal operation, was calculated with EnergyPlus. The model of a small single-zone retail space was developed. The retail space is located in ground floor of the building, with the external wall facing west. All other walls are internal and assumed to be adiabatic. The space is 12 meter wide and 8 meter deep, with floor-to-ceiling height of 3.5 meters. The external wall has 50% fenestration area. Construction elements were set to comply with the latest Spanish national standards (MV, 2009). U-values of the ground floor and external wall are 0.85, 1.6 and 1.9 W/m^{2}K respectively. Glazing is composed of two glass panes with argon-filled cavity and has U-value of 2.3 W/m^{2}K. The space is in use between 7am and 9pm everyday, except on Sunday when shops are closed at 3 pm. During the cooling period, the temperature within the retail space was set to 23°C. Internal heat gains, which can have a significant effect on the thermal characteristics and energy consumption of the building, are set to high. Artificial lighting and equipment were set to 30 W/m^2 and 25 W/m^2, respectively. The occupant density was set to 8 m^2/person with a total heat gain of 167 W/person. In addition to internal gains, fresh air requirements were also defined. A minimum of 10 l/s per person was supplied in order to satisfy fresh air requirements. Cooling was provided by a fan-coil system. Fresh air is supplied directly through fan-coil unit where it is mixed with re-circulated air before passing through the cooling coil. Indoor temperature is controlled by a local thermostat which varies water flow rate through cooling coil in response to the cooling demand. Chilled water is supplied directly from the absorption chiller at the constant temperature of 7°C. In the case when there are no cooling needs, the fan-coil fan is switched off.

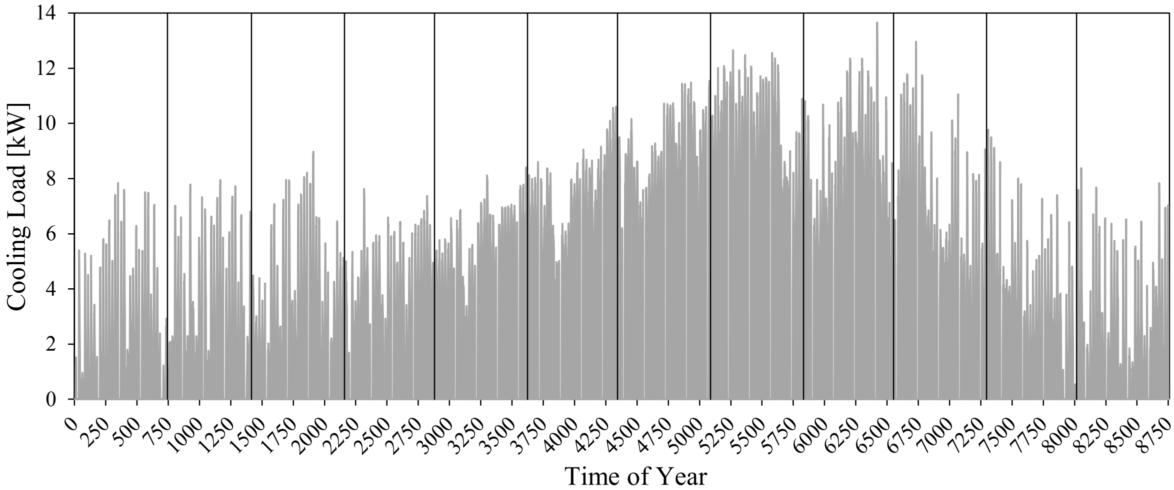

Figure 3 Annual cooling load profile

Cooling load profile presented in Figure 3 is calculated for the Barcelona (Spain) weather conditions. It is clear that cooling demand is present throughout the year. This is mainly due to the high internal gains. The maximum cooling load is slightly above 13 kW. However, the size of the absorption chiller was determined based on the summer design day calculations. The chosen absorption chiller has the nominal cooling capacity of 11.2 kW when operated at nominal temperature conditions ($T_{eva,out}$=7ºC, $T_{ac,in}$=27ºC and $T_{gen,in}$= 90ºC).

Figure 4 shows a histogram of the cooling load. In majority of time, the system operates under part load conditions. In fact, less than 10% of the operating hours (total number of operating hours is 3960) the system has the cooling energy requirements greater than 9 kW, which is equal to 80% of the chiller nominal power. Furthermore, as many as 70% of operating hours the chiller needs to provide less than 7 kW of cooling energy, which means that chiller operates during that period with less than 65% of the nominal power.

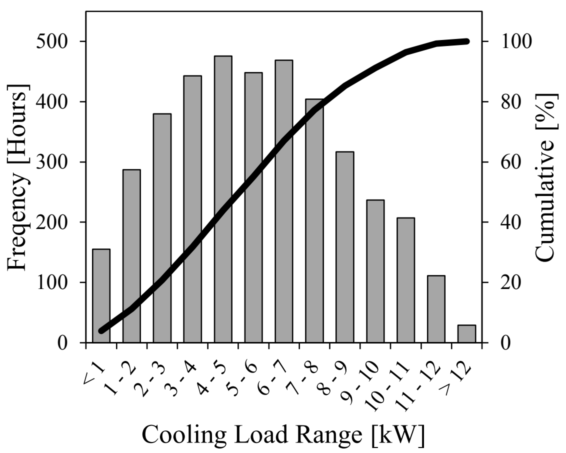

Figure 4 Cooling load distribution diagram

SMALL ABSORPTION CHILLER OPTIMAL OPERATION

The calculated cooling load profile was used to test the absorption chiller optimal operation. The optimal operation parameters of the chiller were calculated with a GA on the ANN model. We optimised the combination of the hot water inlet temperature and the cooling water inlet temperature, giving that the chilled water outlet temperature is fixed to 7°C and all flow rates are set according to manufacturer's specifications. The fitness function for chiller operation is usually energy consumption or cost. In this study, however, the fitness function was to maximize the spot COP of the chiller at each part load steady-state conditions.

The MatLab GA Toolbox was used to optimise the operation parameters, with the following settings:

- For each part load condition, there are two optimisation variables: hot water temperature and cooling water temperature.
- Fitness function calculates the COP of the chiller, giving the water temperatures and flow rates, using the ANN model.
- Real encoding (Double Vector) was used.
- Population size was set to 40. This large population size is helpful for avoiding false optimum, and is affordable because the calculation of the ANN model is fast.
- Ranking scaling function and Roulette-Wheel selection method was used to apply a weak selection pressure. This, coupled with large population size, ensures good exploration of the solution space.
- Elite count was set to 5.
- All other parameters and options were set to Matlab's default values.

The constraint values of the control parameters were directly determined by the operating range of ANN chiller model: 27°C≤$T_{ac,in}$≤35°C and 80°C≤$T_{gen,in}$≤100°C. The evaporator outlet temperature was constrained to 7°C due to the requirements of fan-coil system. The last constraint was introduced to satisfy cooling requirements of small retail single-zone space at hourly time step by defining nonlinear constraint function (ANN model function): Cooling demand-Q_{eva}=0.

Figure 5 is the result of operation optimisation. Grey shaded area on the diagram indicates the chiller operating zone. Dashed lines show the chiller COPs, while solid lines mark the cooling water temperatures. The bold line represent the conditions ($T_{gen,in}$ and $T_{ac,in}$) for which the chiller operates at the maximum COP for the particular cooling capacity (Q_{eva}). For higher cooling capacities (greater than 8.5 kW) the optimal operation of the chiller is to work with the lowest possible cooling water temperature recommended by the manufacturer (27°C). Changes in the cooling capacity can be achieved by varying the hot water inlet temperature. For example, the nominal cooling capacity of 11.2 kW, obtained at $T_{ac,in}$=27°C and $T_{gen,in}$=90°C, can also be obtained at $T_{ac,in}$=29°C and $T_{gen,in}$=96°C, but that would result in the COP drop from around 0.63 to less than 0.61. To operate with cooling capacities lower than 8.5 kW while maintaining the maximum possible COP, the chiller should operates at lowest hot water inlet temperature which does not affect the stable operation ($T_{gen,in}$=80°C) while the cooling water inlet temperature should be adjusted to meet the required cooling capacity.

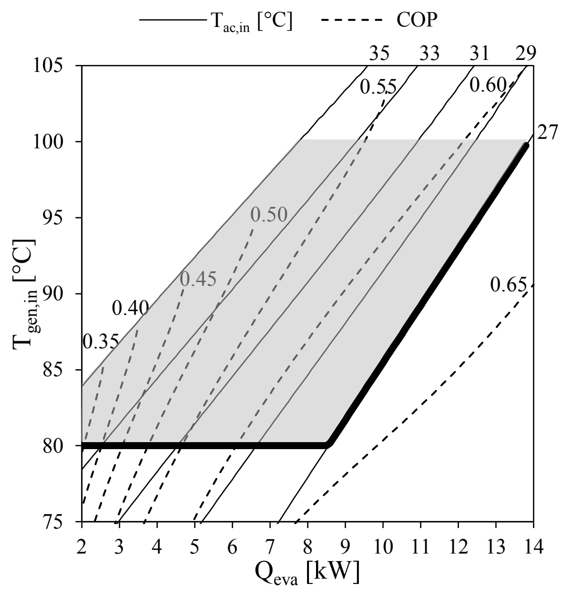

Figure 5 Chiller performance diagram (for $T_{eva,out}$=7°C)

Although in theory reaching very low cooling capacity is possible, in reality the chiller enters the instable zone of operation at around 20% of the nominal capacity (which is 2.2 kW). Due to that, it was assumed that chiller operates in cycling ON-OFF regime with the optimal COP of 0.35 for demands lower than 2.2 kW. In total, that is less than 10% of the chiller working hours (Figure 4). Taking all these into account, it was calculated that the annual cooling requirements of 22,443 kWh/yr can be met by 38,957 kWh/yr of heat using the absorption chiller with optimal operation, which means that the chiller operates at overall COP 0.58.

The next step was to compare the developed small absorption chiller's operational strategy with conventional control methods. Two of the most popular conventional control methods are ON-OFF cycling operation and hot water inlet temperature modulation (with constant chilled and cooling water temperature). However, both of these strategies are excluded from this analysis due to large number of cycling periods which significantly reduces the overall COP (Pérez De Viñaspre et al., 2004; Lazzarin, 2007a). The conventional method used in comparison with optimal operation was by varying both the hot water inlet temperature and the cooling water inlet temperature. For the purpose of this study, the model was built on multiple regression function for wide range of operating points which can be found in manufacturers catalogues. In our case, the model was developed by using the experimental data. A typical absorption chiller model used in many simulation software such as EnergyPlus (2010) and BLAST (1999) was used as the reference model. The operation function of the reference model is based on a cubic equation that determines the generator heat input ratio as a function of part-load ratio (PLR, a ratio between cooling demand and chiller nominal cooling capacity). When operating part-load ratio is lower than 0.2, cyclic control is used, so that the chiller operates only for a fraction of time step. In other words, when cooling demand is lower than the nominal cooling capacity that the chiller generates at minimum power, the chiller will operate at minimum power for a fraction of time step; otherwise, the chiller will be on for the entire time step. By applying this control strategy on the reference model, it was calculated that the chiller annual heat requirements are 43,221 kWh/yr, with overall COP 0.52. The results for this control scenario are presented as grey solid line in Figure 6 against partial load performance curve of the ANN model under optimal operation (grey dash line).

The results clearly indicate that the chiller with optimal operation reduces annual heat requirement by over 10%. This is mainly due to functioning with a higher COP during the almost all working hours (black line in Figure 5) and at the wide range of cooling capacities. Figure 6 shows the chiller COP as a function of the part-load ratio. It can be seen that during the most of the operating times ($0.3 \leq PLR \leq 0.9$), the chiller with optimal operation performs better than the chiller which control is based on the typical (BLAST) absorption chiller model. Even at the low-capacity end, the optimally controlled chiller has higher COP.

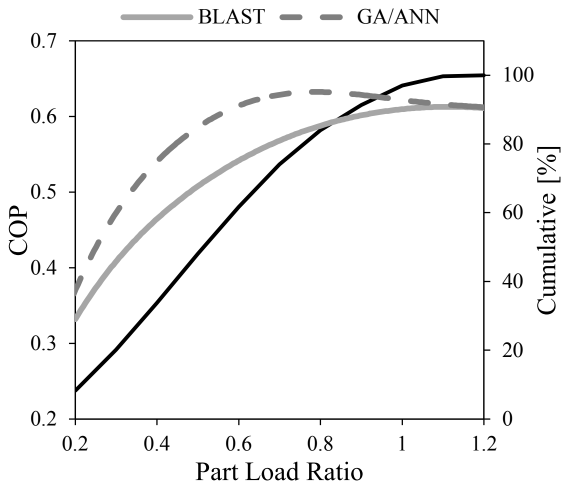

Figure 6 PLR vs. COP for investigated operational scenarios

CONCLUSION

In this paper, some initial findings about optimal operation of small capacity absorption chillers are presented. It was demonstrated how ANN model based on accurate experimental data can be used for development of supervisory control strategy by coupling the ANN model with GA optimization algorithm. To test the control strategy, typical small single zone retail space located in Spain and built according to the latest national building standards was defined. Annual cooling load profile was calculated by EnergyPlus and coupled with the small absorption chiller in purposely developed MatLab models. Two scenarios were tested; first one is the optimal operation and the second one based on typical operation. Developed typical operation was based on EnergyPlus and BLAST models. Both tested control methods vary the chiller performance by modulating the cooling water and hot water inlet temperatures while keeping constant the chilled water outlet temperature. Two the most common types of small absorption chiller controls (cycling ON-OFF and modulating the hot water inlet temperature) were excluded from the analysis due to large number of cycling periods which would significantly reduce the overall COP.

When compared to the typical operation scenario, the GA/ANN approach saved around 10% of heat necessary for chiller's operation on annual basis as the overall COP of the chiller increased to 0.58 from 0.52 for the conventional control. However, there is problem with implementation of GA/ANN control in cooling systems with absorption chillers. The common practice uses local controllers for each operating component with limited information flow between them. This can be overcome by using more advanced controllers.

In this study, we only used a quasi-steady state approach to evaluate the annual cooling performance of an absorption chiller with pre-optimised operation for different operational conditions. This approach is simple to implement and can be an effective way to improve energy performance of existing chillers. However, the transient process of absorption refrigeration cycle has to be taken into consideration in real control applications. As a result, future research should focus on dynamic chiller models coupled with detailed models of the heat source, cooling circuit, and the chilled/hot water user(s).

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