NUMERICAL MODEL OF A REFRIGERATION LOOP FOR A FOOD STORAGE APPLICATIONS

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ABSTRACT

The abrupt escalating of energy prices makes the search for energy-efficient strategies a pressing issue, even in such market sectors as warehouse food storage, where the main concerns were usually associated with keeping food quality compliant to standards and regulations. Simulations appear as the only tool which can reasonably suggest changes in plant conduction without running the risk of damaging the wares. This paper describes a numerical model of the main refrigeration loop of a food storage facility located in north-eastern Italy. The lumped and distributed parameter components are detailed and modelling issues discussed. Simulation in 'piecewise-steady' state are presented and compared with the on-site data collected during plant operation. It is found that the model, even in its simplified form, can predict the power demand of the compressor within ±15% and the cooling load with even narrower accuracy.

1 INTRODUCTION

Vapour compression systems (VCS) are broadly used in refrigeration, especially when processing or storing food. Despite the simplicity of its base design, the actual realisations can be very complex, so that the control strategies become somewhat more complicated. Yet, operation of the system is driven by the saturation pressures in the heat transfer equipment and by the refrigerant flow rate through the compressor and expansion valve, which are the monitored quantities during plant operation. In this context, mathematical modelling is an attractive way to analyse operation and define the optimal control strategy for a given facility, with the ultimate goal to minimize electric power consumption. In the past decades several works addressed the problem of dynamic modelling of vapour compression systems, [1, 2]. One of the key issues is the way the components of the system are modelled. In some instances, operation maps provided by the manufactures may be used, which allow the model to be true to known performance, but prone to extrapolation errors when used outside of the tested range. On the other hand, a model could be devised from physical principles, this approach allows deeper understanding of the phenomena involved, but is more time-consuming, both in the set-up and validation phases, and can become prohibitively expensive in terms of computational cost. Finally, if a sufficient amount of operating data is available, a so-called black-box model can be obtained e.g. by means of regression analysis. The latter approach is faster to set up and more accurate in the range where data are available, but can quickly lose in accuracy when boundary conditions change, which makes it unsuitable. In contrast with the physical-based approach, the data-driven models is faster to set up and hopefully more accurate in the range of data availability; on the other hand, this kind of models are exposed to the risk of rapidly losing of precision when the operating conditions change out of the range of available data, so they are unsuitable to make predictions useful for the control strategies definition. Also, the correct definition of the time scale of interest is important, [2] in order to decide between a dynamic or steady-state approach. However, in order to develop a control-oriented tool, a 'quasi-steady state' model can be chosen: if the dominant time scale of the inputs to the system differ by order of magnitude from those of the model, the dynamics of the latter can be described as a sequence of steady states, even while subjected to time varying conditions. Moreover, as Bendapudi and Braun, [2], pointed out, the dominant time scale of a VCS depends on the heat transfer in the evaporators and in the condensers. Also for this reason, the major task in developing a VCS model relates to heat transfer in those pieces of equipment. Components can further

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be modelled with a distributed parameter or with a lumped parameter approach. The latter is computationally simpler, but spatial details are lost by the process of averaging over the domain. The former method is richer in spatial details, since the governing equations are applied locally, but is more cost-intensive in terms of computations, and may become unsuitable for control design. The aim of this work is developing a useful mathematical model to simulate operation of a VCS serving a food storage facility located in north-eastern Italy, with the ultimate goal of defining the optimal control strategies that could minimize the electric power consumption of the plant. This paper focuses on the primary refrigeration loop, witch involves the compression and the condensation sections, neglecting, at present, the mathematical description of the secondary loop that includes all the evaporators operating in the single storage areas; in the model these are replaced by the corresponding cooling loads. This choice is consistent with the main purpose of the model, that is to establish the best pressure levels for this circuit, in terms of lowest energy consumption, under several operating conditions, such as cooling loads and environment air conditions. In developing the model a study of the single components and of their interactions has been carried out, with particular attention to the control issues which allow correct handling of the set points for the operating pressures. More detailed analysis is requested by the evaporative condensers, because of their key role in the definition of the high pressure. Moreover the dynamic behaviour of the model is largely influenced by the heat transfer in the condensers and by refrigerant mass accumulation in the high and low-pressure tanks. The numerical model has been developed in Python, because of the complete control over the code, the free and open source availability, in addition to its flexibility and comparative ease of use.

2 MODEL STRUCTURE

The structure of the model is depicted in Fig.1 as a block diagram and corresponds to a simplified version of the actual main refrigeration loop layout.

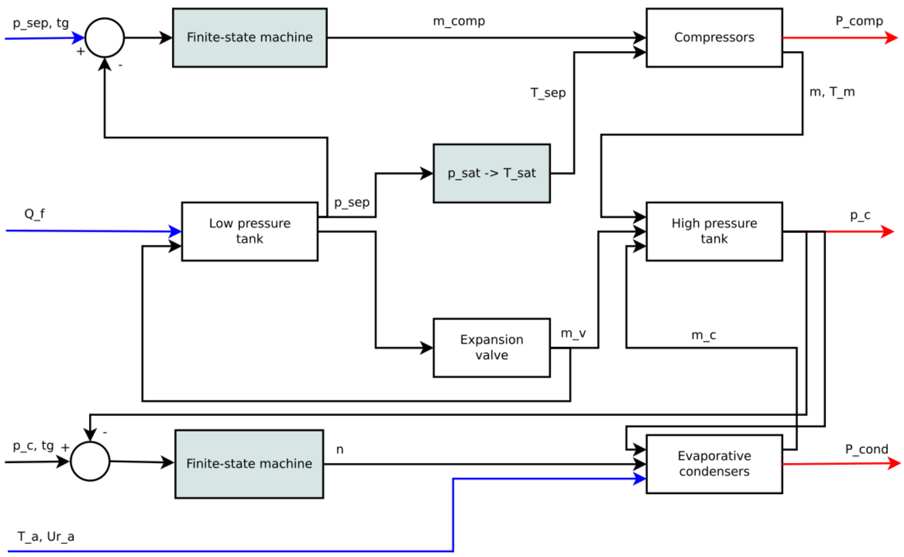

Figure 1: Full primary refrigeration loop model block diagram.

The inputs to the model are ambient air conditions (temperature and relative humidity), the cooling load required by the stored foodstuff and the nominal operating pressures; its outputs are the electric power demand of the compressors, the thermal load dissipated by the evaporative condensers, and the actual condensation pressure, which usually deviates from the nominal value owing to the influence of actual ambient and operating conditions. The loop can be thought of as composed by two main sections, which are defined by the operating pressure: the compression and the condensation section. The former consists of the four low-pressure tanks, their expansion valves, and the five screw compressors. The low-pressure vessels are modelled as a single block, together with their expansion valves, also represented by a single item. The cooling demands of the different storage areas are lumped together and constitute the load at the low-pressure end of the downstream collector. The condensation section is composed of four evaporative condensers and of the high-pressure receiver. The compressors and the evaporative condensers are modelled individually. The mass flow rate from the low pressure section is determined by the operation of the compressors, whilst that from the high pressure section is defined by the block corresponding to the expansion valve. Inside the condensation macro-block the condensed flow rate to the receiver is determined by the operating point of the four condensers; that is further explained in the following. Pressure losses which occurs in the piping and condenser coils are neglected.

3 MODEL COMPONENTS

The existing main refrigeration loop consists of five screw compressors, four low-pressure collectors each with its electronic expansion valve (EEV) controlling the refrigerant flow rate, four evaporative condensers and a high-pressure collector tank.

3.1 Compressors

The five screw compressors installed differ in cooling capacity and in control strategies. All possess a slide valve, to modulate the volumetric flow rate and control the compression ratio independently. Three compressors have their electric motors equipped with an inverter, which controls the volumetric flowrate by changing power supply frequency and thus of the rotational speed of the screw. A lumped parameter model of the compressor is appropriate, since, for control purposes, knowledge of the inlet and outlet values of the process variable and of the power supplied suffices. Compressors are therefore modelled using third-order polynomials yielding their mass flow rate, electric power consumption and outlet refrigerant temperature, by means of the independent variables, that characterize the compression process, namely saturation temperatures corresponding to the inlet and outlet pressures. This methodology is in accordance with recommendations of standards such as UNI-EN 12900. The generic process variable $X = \{\dot{m}, P_e, T_{out}\}$ is calculated through Eq.(1).

$$X = a_1 + a_2 T_e + a_3 T_c + a_4 T_e^2 + a_5 T_e T_c + a_6 T_c^2 + a_7 T_e^3 + a_8 T_e^2 T_c + a_9 T_e T_c^2 + a_{10} T_c^3 \qquad (1)$$

The coefficients $\overline{a} = \{a_1, a_2, a_3, a_4, a_5, a_6, a_7, a_8, a_9, a_{10}\}$ are a function of the rotation speed of the compressor and the level of suction overheating, $\overline{a} = f(n, \Delta T_{sur,a})$, and are often supplied by the compressor manufacturers. In this work a data driven approach is used instead; determination of the polynomials coefficients is obtained by curve fitting of the operating data available from the actual plant.

3.2 Evaporative Condensers

The refrigeration plant employs four evaporative condensers, which can operate at temperatures lower than draft cooling towers or water-cooled devices. The evaporative condensers are of two different types, which differ in the relative direction of the air and water flows over the cooling coil and by the presence, or lack thereof, of an additional element (the so-called fill pack) which allows direct heat transfer between the water dripping from the cooling coil and ambient air. In fact, in evaporative condensers, the refrigerant condenses while flowing within the coil, which is wetted on the outside by water sprayed onto it; at the same time an air stream is blown, or drawn, depending on fan arrangement, in co- or counter-flow with respect to the water stream. Water dripping from the coil may be collected directly in a water sump at the bottom, or pass through the fill pack to reduce its temperature and enhance the performance of the condenser. Many approaches to the modelling of evaporative heat transfer have been described in the literature, but only few are suitable for dynamic implementation and control-oriented mode [3]. In this work a distributed-parameter approach has been chosen, using the theoretical description first introduced by Merkel [4] and further discussed by Dreyer [5] and Kröger[6]; a more detailed discussion of the model and of the influence that the choice of the correlations for heat and mass transfer has on its performance can be found in [7].

$$dh_a = \frac{\beta}{m_a}(h_{asw} - h_a)dA_0 \tag{2}$$

$$dT_w = -\frac{1}{m_w c_p w}(m_a dh_a + m_p dh_p) \tag{3}$$

$$dh_p = \frac{U}{m_p}(T_p - T_w)dA_o \tag{4}$$

Equations (2)-(4) constitute the core of the condenser block and are solved by a forward Euler method with fixed step-size discretisation, details are given in [7]. For the condenser operated in counter-flow, a control loop over the enthalpy of moist air at the inlet section has been implemented. Further controls have been introduced to make the model operate in the same way as the actual condenser; therefore, a PI control ensures that all the refrigerant flowrate through the coil exits it as saturated or slightly subcooled fluid. Another control loop is used to simulate the transient allowing the water in the sump to achieve steady-state temperature, starting from that of ambient air. Finally, the fans speed is controlled so that the desired condensation pressure is maintained: this behaviour is forced through a PI control tracking the set point head pressure by imposing a rotational speed to the fans. The evaporative condenser function block is represented in Fig. 2.

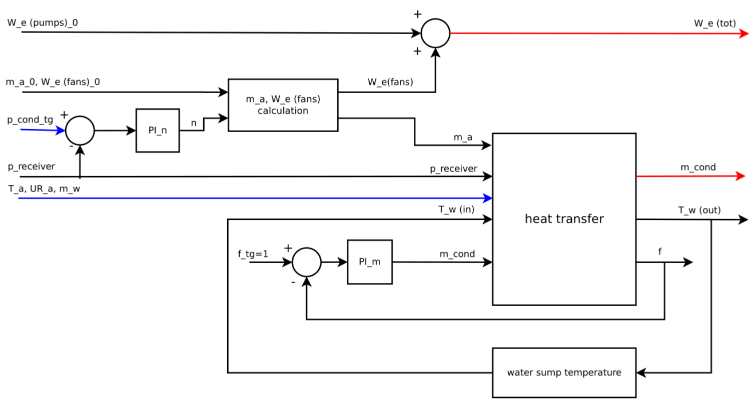

Figure 2: Evaporative condensers function block.

3.3 Collectors

The different low-pressure collectors of the actual plant are considered as a single block, subject to a cooling load resulting from environmental conditions, activities in the warehouse and mass of stored goods. The mass balance and energy balance equations, (5), (6) describing this piece of equipment are shown below:

$$\frac{dM_s}{dt} = \dot{m}_v - \dot{m}_c \tag{5}$$

$$\frac{dU_s}{dt} = \dot{m}_v h_{lam} - \dot{m}_c h_v + \dot{Q}_f \tag{6}$$

Similar, but slightly more complicated, the high-pressure receiver has a superheated vapour flow input from the compression section, a saturated liquid mass flow input from the evaporative condensers, balanced by an equal saturated vapor mass flow output toward the condensers, and saturated liquid mass flow toward the expansion valve. This replicates the actual piping connecting the high-pressure tank with the evaporative condensers to allow regular operation.

3.4 Expansion Valve

Expansion valves are located close to the low-pressure tank and ensure both the correct level of refrigerant in the tank and the desired pressure difference between the collector downstream of the condensers and the separator upstream of the compressor inlet. A PI control in the model strives to maintain the set refrigerant level in the low-pressure collector.

3.5 Connection of the Blocks

The components of the primary refrigeration loop described above must be connected in order to form a consistent model able to describe the actual operation of the plant; this has been accomplished by implementing suitable control logics. Compressors operate so as to follow the reference pressure in the low-pressure collector at their suction side. The mass flow rate of the refrigerant (R717, ammonia) through the compressor is determined by its speed of revolution and by the position of the sliding valve. The combined operation of the compressors is determined by a finite-state machine, which has been devised so that at any moment only one at most operates at partial load, while all the other are either off or running at maximum flowrate. The amount of refrigerant in the low-pressure collector is determined by the expansion valve block, in which a PI control dictates the values of liquid flow rate from the high-pressure section, in order to keep a set level of the liquid inside the tank. The high-pressure branch, consisting of four evaporative condensers and the collector tank, is controlled through the value of the reference condensation pressure. Indeed, the actual pressure is determined by the equilibrium reached here between the vapour and the liquid phases of the refrigerant. While the amount of vapour increases thanks to the mass inflow from the compressors, that of the liquid is determined by the heat transfer in the evaporative condensers. Because of this, the high-pressure control is exerted onto the cooling capacity of the condensers, thus mainly adjusting fans speed and, therefore, the mass air flow rate through the coil. The model of the primary refrigeration loop must be able to calculate the effective pressure level reached in the condensation section depending on the ambient conditions, the refrigerant flowrate from the compressors and the settings of the condensers. For this purpose it is assumed that when equilibrium in the collector tank is established, the effective condensation pressure is reached. The control of condensation pressure is obtained modifying the fan speed through another PI loop in the evaporative condenser. In fact, also in this case, simultaneous operation of the evaporative condensers is managed through a finite-state machine, which allows only one condenser to work at partial load at any moment, whilst the others are either non-operative or full load. This mode of operation of the condensers does not represent the actual mode of conduction, which is most often demanded to the personnel on site; yet the aim of this model is also to determine the best conduction strategy which could be fully automated.

4 RESULTS AND DISCUSSION

The numerical model for this work has been completely developed autonomously in Python and can simulate a time-dependent change of the inputs in a 'quasi-steady' mode, that is through a succession of steady state computations in the single blocks. Nonetheless, the validation has been carried out comparing the actual behaviour of the primary refrigeration loop during steady state operations with the results produced by the model under the same input conditions. The model is run for several conditions, i.e. environment air temperature and humidity, evaporation and condensation pressure and refrigeration loads, each representing a stationary operating condition, and yields the electric power demand of the compressors and of the evaporative condensers (to operate the fans) together with the thermal power dissipated at the condensers. Results of the simulations are checked against actual data coming from the real-life plant. For the comparison to be meaningful, a period of 15 minutes in which condensation pressure would not vary more than $\pm, 1\%$ was considered steady state, and the data collected and averaged would supply both the inputs to the model and the benchmark against which to check computed quantities.

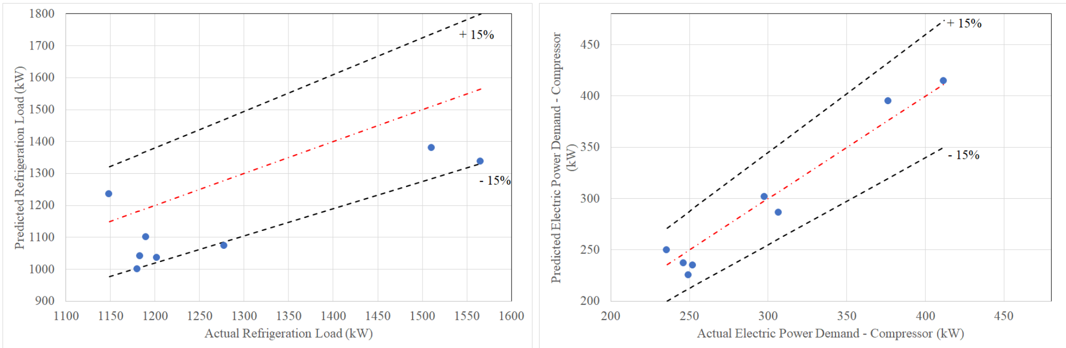

Figure 3: Power demand for the compressors (left) and refrigeration load (right): actual and predicted values.

As shown in Fig. 3, the model, even with its simplified structure, appears able to reproduce the experimental data (which are themselves affected by a degree of uncertainty) within $\pm 15\%$. The model can then be employed to determine the minimum condensation pressure needed according to ambient air conditions. This is particularly important for evaporative condensers, which allow condensation at temperature close to that of wet bulb, but which usually run at much higher temperatures by the food companies, lest the stored goods be spoiled. This tool can therefore be used to determine and demonstrate under which operating conditions the cooling load can be fully covered successfully and what the energy savings with respect to higher condensation pressures may be. It should also be pointed out that the particular choice in modelling the evaporative condenser allows mapping of the enthalpy of moist air and cooling water over the coil and the fill pack (where present), as well as the temperature and quality distribution of the refrigerant, as detailed in [8].

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