MODELING AND SIMULATION OF A THERMAL ACCUMUTATION TANK, FOR FROZEN WATER, IN TRANSIENT REGIMEN

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Abstract: *The proposed project presents a study and mathematical model of a thermal accumulation tank, for frozen water, in a transient regimen and its respective simulation for feeding the central air conditioning system of the Passenger Terminal of Afonso Pena International Airport, located in the São Jose dos Pinhais city, Paraná.*
For the modeling of the system, we used equations of mass conservation and energy saving, considering the conditions of the environment, methods of operation of the entire system, constructive characteristics of the tank and data of the inherent equipment in the process. The simulation was accomplished using the FORTRAN 90 language. The model allows us to simulate and obtain, based on operational methodologies, ideal conditions of frozen water generation for storage, in other words, it is possible to estimate with great precision, the usage time of the chillers for the cooling of fluid and thus the reflection in the consumption of energy when making freeze water and when using the tank in the acclimatization system of the airport.

Keywords: *Thermal accumulation, Energy, Modeling, Simulation, Transient Regimen*

1. INTRODUCTION

Since 2002, when the blackout of electric system happened in Brazil, demonstrating all the fragility of the energy system, the supply of electric energy was not completely stabilized, being threatened by possible new blackouts and negative perspectives, such as the ones that energy costs become higher and higher (according to article from the newspaper "O Sul" from July 8, 2003).

Therefore, industries and commercial establishments have been looking for the rationalization of electric energy, been obliged to invest in projects looking forward to reach a reduction in the consume of electric energy.

PAVAN (2004) evidences and his studies that acclimatization system is the main responsible for the increase in the demands of an electric energy in hot days.

AMARIN (2005) verified that 35% of the energetic expenditure in commercial buildings are due to the use of the air-conditioning system.

ELLESSON (1997) evaluated that if the cold thermo-accumulation systems are well protected and installed, they are able to reduce the costs of operation, implementation and the capacity of electric energy substations, increase in the capacity of the existing system (if that's the case), make flexible that installation operation as the whole, also creating, a reserve of cold for the situations of failure in the chillers.

It is it within this context that we propose to mathematically model a thermal accumulation tank by frozen water in transient regiment the respective simulation of it for the feeding of the system of central air-conditioning of the passenger terminal of Afonso Pena International Airport, located in the city of São José dos Pinhais, Paraná state, trying to determinate in the primary way, the viability of a revitalizing and put into operation the existing thermal accumulation tank of the system in the airport, comparing the costs of using the stored frozen water in the peak schedules, with those resulting from the operation of chillers connected to electric energy and also with those from the operation of the chillers connected to diesel generators for the same period. Therefore, in this work will look forward to find:

- The running time, uninterrupted and in maximum load, of chillers to freeze the water from the ambient temperature to the temperature desired for storage.
- The temperature of the stored water at the start of the operation of the system with the thermal accumulator.
- The temperature of the stored water at the end of the operation of the system with the thermal accumulator.
- The necessary time to cool the water until the desired temperature after the use of the system with the stored water;

- To check if the volume of the current tank attends the thermal necessities demanded by the of passenger terminal.

The system of thermal accumulation in question, by conception, has as purpose to accumulate frozen water proceeding from chillers in a tank thermally isolated with rock wool to supply the necessities of air conditioning of the passenger terminal of the Alfonso Pena International Airport in the schedules when the chillers are turned off due to the high cost of electric energy (peak schedules), however, this system is currently deactivated because of structural problems.

This study has as its main objective to numerically simulate the reactivation of the thermal accumulation system of frozen water in an optimized form.

2. ACCLIMATIZATION SYSTEM FUNCTIONING

The central air conditioning system of the Afonso Pena International Airport is from the indirect expansion with air condensation type, made up basically by 4 chillers, being 2 of 270 TR and 2 of 200 TR, 4 primary pumps of frozen water with an 112 m ³ /h outflow each, 4 secondary pumps of frozen water with a 75 m ³ /h outflow each, 54 fancoil type acclimatizers and a thermal accumulation tank as illustrated in Fig. 1. The tank has the following characteristics:

- Capacity of water storage: 600 m ³;
- Dimensions: 7 x 10 x 10 m;
- Constructed in masonry, with massive bricks of 10 cm of thickness and plastering, in the two sides, with thickness of 2 cm each. Ceiling in concrete, with bricks of 8 cm and plastering of 2 cm in both sides;
- Thermally isolated in the superior part and laterals with 8 cm of rock woolen and, on top of this, corrugated aluminum with 0,5 mm of thickness;
- Floor in concrete with 10 cm of thickness, which has, in its intermediate part, a plate with 5 cm of thickness of styrofoam.

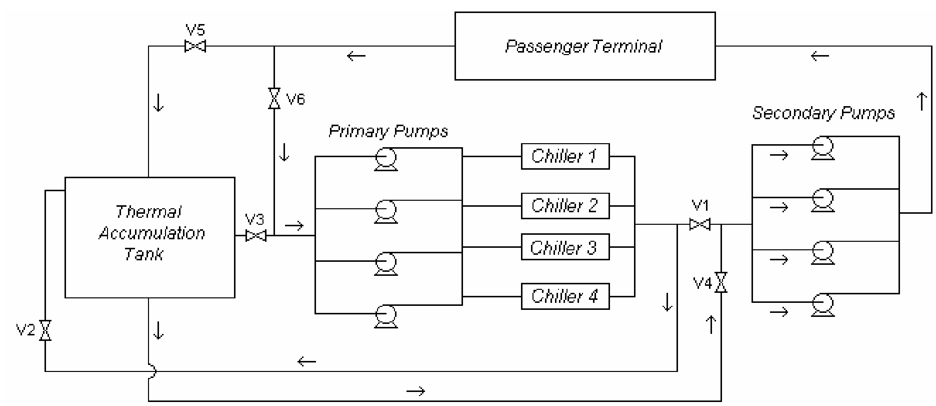

Figure 1. Thermal accumulation system of frozen water

Closing the V2, V3, V4 and V5 valves, the passenger terminal will receive frozen water from chillers. On the other hand, closing the V1, V2, V3 and V6 valves, the acclimatization system starts to operate only with the thermal accumulator. During the period of water production frozen for storage, the V1, V4, V5 and V6 valves must be closed.

Currently, the chillers turned on at 6:00 a.m. and has as exclusive function to generate frozen water for the central air conditioning system of the passenger terminal. The functioning follows in an uninterrupted way, however with adequate power to the thermal load of the terminal, until 6:00 p.m., when, due to the beginning of peak schedule, they are turned off from the electric system and start to operate with the aid of diesel generator groups. At 9:00 p.m., with the ending of the peak schedule, the chillers are again turned on to the electric system to supply the necessities of the passenger terminal, functioning until 1:00 a.m., when then they are turned off until 6:00 a.m.

Due to the profile of thermal demand of the passenger terminal, and aiming at the energy optimization, we propose that the chillers continue to be turned on from 6:00 a.m. to 6:00 p.m., exclusively attending the passenger terminal. At 6:00 p.m. the system of thermal accumulation of frozen water enters in operation, supplying frozen water to the system until 9:00 p.m., when then only the chillers return until 1:00 a.m., and from this time on, the chillers are redirected, through maneuvers of valves, to the tubing of the system of frozen water storage. The frozen water generation is continued until 6:00 a.m., being able to be interrupted before if the temperature of the water in the tank reaches the desired temperature. At 6 a.m. the chillers return to supply the necessities of the passenger terminal. During the operation of chillers, between 6:00 a.m. and 6:00 p.m., and between 9:00 p.m. and 1:00 a.m., the stored water remains in rest.

Figure 2 shows the evolution of the average thermal demand throughout one day of the passenger terminal of the airport in kWh, thermal load in kWh, the energy consumption, in kWh and the power of chillers, in kW throughout one day.

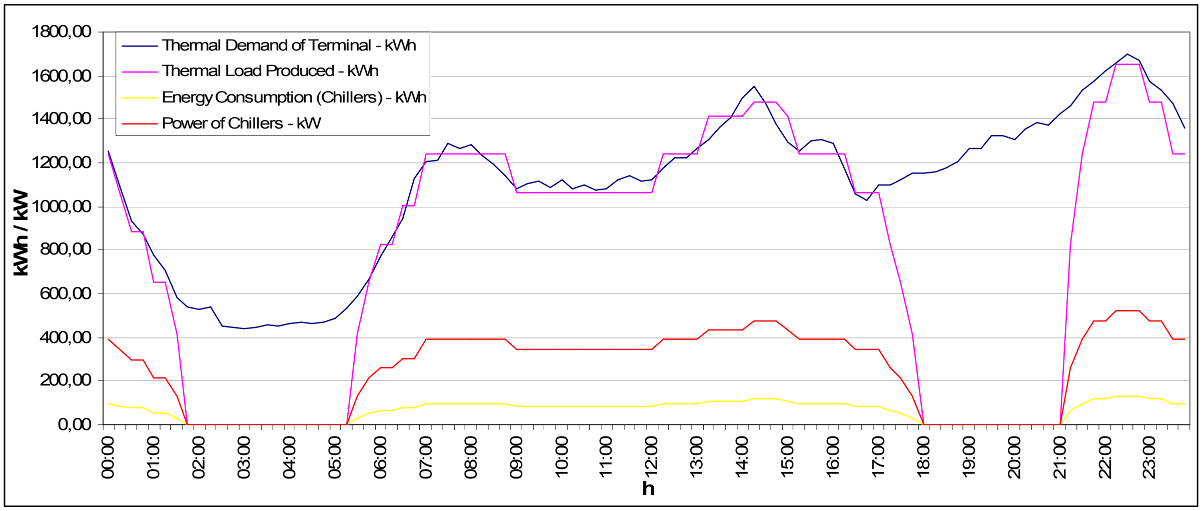

Figure 2. Demanded thermal load x Thermal load generated by the chillers

3. MATHEMATICAL MODEL

Next, the mathematical modeling of the proposed system is presented in way to allow that the objectives of this work are reached and the results obtained are trustworthy, making possible that the decisions referring to the investment in the revitalization of the tank are safe and optimized. The problem was analyzed starting from the following hypotheses:

- Only the frozen water tank will be considered in transient regimen;
- The exchanges of heat with the environment in the tubing and pumps will be rejected;
- The air will be considered as an ideal gas;
- The water level will be considered constant in the tank.

3.1 Modeling for the first stage (tank in loading)

The initial condition, adopted for the resolution of the Ordinary Differential Equation resultant of the modeling, is that the temperature of the water in the initial instant is equal to the temperature of humid bulb of surrounding air, that is, $T(t=0) = T_{Amb}$.

Finally, the only incognito to determine is the temperature "T" of the water in the interior of the tank throughout the time.

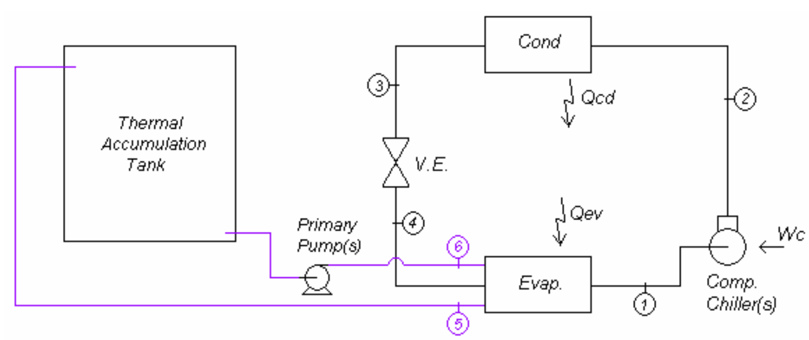

Figure 3. Circuit of generation for the first stage

The circuit of generation and frozen water storage in the tank can be visualized in Fig. 1, where the V1, V4, V5 and V6 valves must be closed. A schematic and simplified drawing of this circuit can be seen in figure 03.

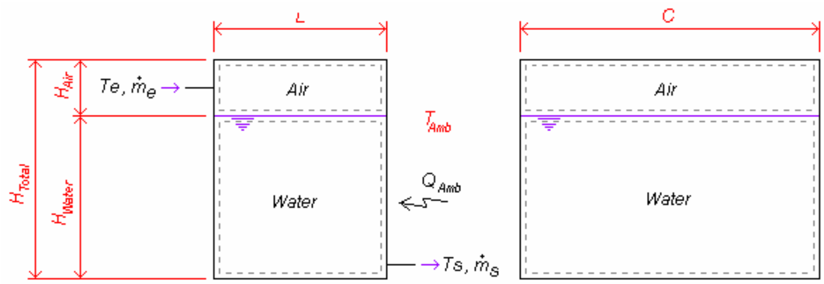

Figure 4. Volume of control for the first stage

All the mathematical modeling for thermal systems have their origin in the equations of mass conservation and conservation of energy, as follows.

Making an evaluation of mass in the volume of control presented in Fig. 4, the mass water outflow in the volume of control is given by:

$$\frac{dm}{dt} = \dot{m} = \dot{m}_e - \dot{m}_s \tag{1}$$

where $\dot{m}_e$ - mass water outflow that enters the tank, kg/s and $\dot{m}_s$ - mass outflow of water that leaves the tank, kg/s.

However, the water level is constant, as considered in the hypotheses, then, $\dot{m}_e - \dot{m}_s = 0 \text{ e } \dot{m} = \text{constante}$.

Maintaining the volume of control and applying an energy evaluation now, we have, the energy that enters in the system, $\sum Q_{Entra}$ is that proceeding one from the external environment through the walls, floor and ceiling and the energy that leaves the system, $\sum Q_{Sai}$ is that removed by the chillers when cooling the water. So, mathematically writing the cited energies and considering the evolution of the temperature throughout the time, the energy coming from the exterior, Q_{Amb}, it can be quantified by:

$$Q_{Amb} = m \times c \times \frac{dT}{dt} + m_{ar} \times c_{ar} \times \frac{dT}{dt} + Q_E \tag{2}$$

where m - mass of water in the tank, kg; c - specific heat of the water, J/kg K; m_{ar} - air mass in the tank, kg/s; c_{ar} - specific heat of air, J/kg K; and Q_E - heat removed by the evaporator, kJ/s.

Considering the equation of conduction of heat, the energy Q for composed walls, proposed by INCROPERA & WITT (1992).

$$Q = U \times A \times \Delta T \tag{3}$$

where U - global coefficient of heat transference, W/m^2°C; A - the surface area of thermal exchange, m^2; ΔT - difference of temperature, °C.

Applying in the Eq. (3) the referring data to the diverse faces of the thermal accumulator, we have:

$$Q_{Amb} = \left[(U \times A)_{Paredes} + (U \times A)_{Teto} + (U \times A)_{Piso}\right] \times (T_{Amb} - T) \tag{4}$$

The heat absorbed in the evaporator, Q_E can be quantified by:

$$Q_E = \dot{m}_E \times c \times (T - T_5) \tag{5}$$

where $\dot{m}_E$ - mass water outflow in the evaporator, kg/s and T_5 - Temperature of the water in the exit of the evaporator, °C.

Finally, substituting the Eqs. (4) and (5) in the Eq. (2), we obtain the differential equation that allows to simulate the variation of the temperature of the water of the thermal accumulator throughout the time.

$$\frac{dT}{dt}=\frac{\left\{\left[(U\cdot A)_{Paredes}+(U\cdot A)_{Teto}+(U\cdot A)_{Piso}\right]\times(T_{Amb}-T)\right\}-\left[\dot{m}_{E}\times c\ \ \times(T-T_{5})\right]}{m\ \ \times c\ \ +m_{ar}\times c_{ar}}\tag{6}$$

3.2 Modeling for the second stage (tank in rest)

In the second stage, to obtain the water temperature at 06:00 p.m., the mathematical modeling based on Fig. 5 is applied, which shows the volume of control for this condition.

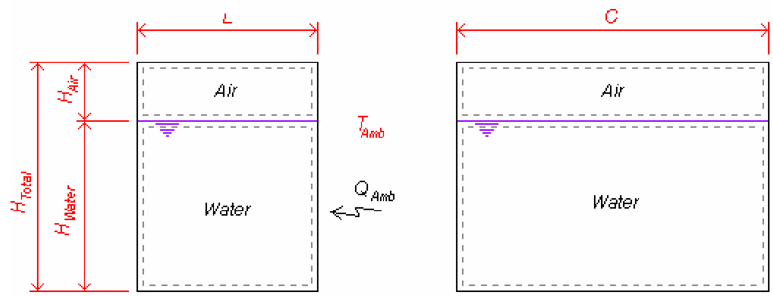

Figure 5. Volume of control for the second stage

Comparing the volume of control indicated in Fig. 5 with that one of Fig. 3, it is verified that the only differences between them are the absences of the mass outflows and the of the temperatures of entrance and exit in the volume of control of figure 4. These data are used, in that case, to determine the heat absorbed by the evaporator. The inexistence of these data was already expected since there will be no cooling of the water, being equal to zero. In this in case, the water will only be subjected to the conditions of the external environment, that is, of Q_{Amb}.

This way the Eq. (2) is summarized into:

$$Q_{Amb}=m\times c\times\frac{dT}{dt}+m_{ar}\times c_{ar}\times\frac{dT}{dt}\tag{7}$$

The heat proceeding from the external environment, Q_{Amb} can be determined by the Eq. (4).

Thus, substituting the Eq. (4) in the Eq. (7) and isolating dT/dt, the temperature "T" of the water can be determined after the time of rest of the tank.

$$\frac{dT}{dt}=\frac{\left\{\left[(U\cdot A)_{Paredes}+(U\cdot A)_{Teto}+(U\cdot A)_{Piso}\right]\times(T_{Amb}-T)\right\}}{m\ \ \times c\ \ +m_{ar}\times c_{ar}}\tag{8}$$

3.3 Modeling for the third stage (tank in consume)

In the third stage, the stored frozen water is used to supply the necessities of passenger terminal, circulating through the system as shown in Fig. 1, in this case the V1 V2, V3 and V6 valves are closed. Figure 6 is a simplified project of Fig. 1 for this condition.

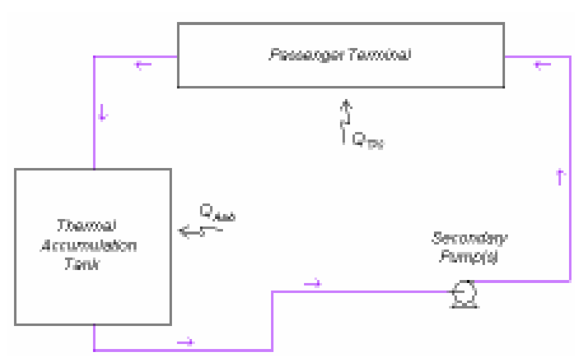

Figure 6. Simplified project of the operation with the use of thermal accumulation

To determine the temperature of the water in the tank after the use of the frozen water in the air conditioners (Fan&Coil) the volume of control shown in Fig. 7 is considered.

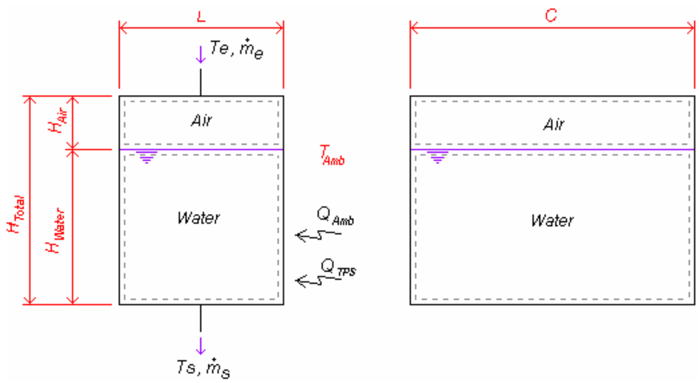

Figure 7 - Volume of control for the third stage

In this case, the incognito to be determined by the modeling and simulation is the temperature (T) of the water throughout the time of operation of the acclimatization system using the stored frozen water. The final temperature will correspond to that registered after 3 hours of operation. The energy that enters in the system is that proceeding from the external environment, coming through the walls, floor and ceiling and from the thermal load of the passenger terminal, Q_{TPS} which will act directly on the energy accumulated in the tank of thermal accumulation. Mathematically writing the cited energies and considering the evolution of the temperature throughout the time, Q_{Amb} for the third stage, can be quantified by:

$$Q_{Amb} = m \times c \times \frac{dT}{dt} + m_{ar} \times c_{ar} \times \frac{dT}{dt} - Q_{TPS} \tag{9}$$

The thermal load from the passenger terminal, Q_{TPS} can be quantified by:

$$Q_{TPS} = \dot{m}_E \times c \quad \times (T_{TPS} - T \quad) \tag{10}$$

where T_{TPS} - Temperature of the terminal of passengers, °C.

Finally, substituting the Eqs. (4) and (10) in the Eq. (9), the differential equation that allows to simulate the variation of the temperature of the water of the thermal accumulator throughout the time for the third stage is obtained.

$$\frac{dT}{dt} = \frac{\dot{m}_E \times c \quad \times (T_{TPS} - T) + \{[(U \cdot A)_{Paredes} + (U \cdot A)_{Teto} + (U \cdot A)_{Piso}] \cdot (T_{Amb} - T)\}}{m \quad \cdot c \quad + m_{ar} \cdot c_{ar}} \tag{11}$$

4. RESULTS AND DISCUSSIONS

For the simulation of the stages considered in this work the computational language FORTRAN 90 was used, where the following operational conditions were used:

- Temperature of the water frozen in the exit of the tank: 6ºC, in compliance with the recommended by the ASHRAE Aplication Handbook (1987);
- Running time of the thermal accumulator: 3 hours, between 06:00 p.m. and 09:00 p.m.;
- The time of rest between 10:00 p.m. and 1:00 a.m. will be rejected for the purpose of calculation, once this study will be applied, at first, only to the operation in peak schedule and the caloric need in this period is minimum, in numbers of hundredths of Celsius degrees, what may be rejected;
- Conditions of the air of the external environment, as NBR 6401 (1980) for the city of Curitiba/PR:
    - Temperature of the dry bulb, TBS = 30 ºC;
    - Temperature of the humid bulb, TBU = 23,5 ºC.
- m = 600.000 kg, m_{ar} = 750 kg, c = 4180 J/kgºK, c_{ar} = 720 J/kgºK, $(UA)_{Parede}$ = 67,3 W/ºC, $(UA)_{Teto}$ = 94,7 W/ºC, $(UA)_{Piso}$ = 82,8 W/ºC.

In the first stage, beginning of the process, the water of the tank is at the temperature of humid bulb of air 23,5 ºC, all chillers will work in maximum load for an uninterrupted period, until the water in the interior of the tank reaches the desired temperature of, 6 ºC, therefore, in this first stage, we tried to find the necessary time to cool the water of 23,5 ºC until 6 ºC.

In the first stage, we considered: T(t=0) = 23,5 ºC, T_{Amb} = 30 ºC, final temperature, T_{fl} = 6 ºC, T_s = 5 ºC, $\dot{m}$ = 124,4 kg/s (equivalent to the 4 primary pumps turned on).

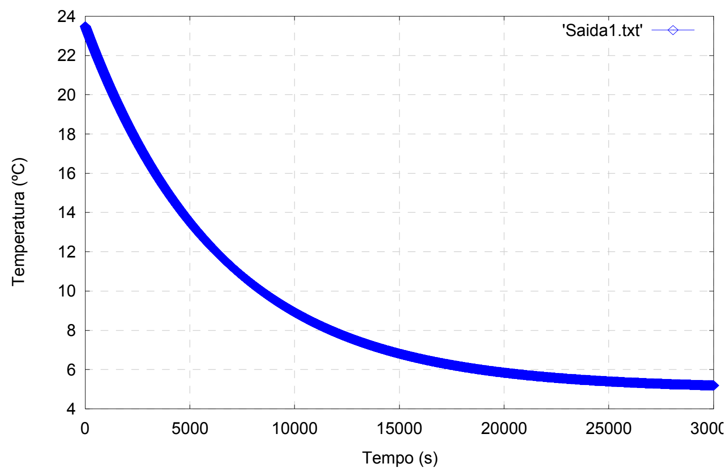

Figure 8. The evolution of the temperature of the water in the tank in the first stage.

In Fig 8 it is observed that for the water to reach 6 ºC, starting from the initial temperature of 23,5 ºC, 14120 seconds, or 3 hours and 55 minutes will be necessary, with the system functioning with 4 primary pumps and all chillers in total load, therefore, the tank enters in the period of rest at 4:55 a.m. and remains that way until 06:00 p.m., totalizing 13 hours and 5 minutes without any activity.

For the second stage, we considered, T(t=0) = 6ºC and the time of rest of the water in the tank, $t_{Repouso}$ = 13:05 hours.

The graph evidences that after 13 hours and 5 minutes of rest, only exposed to the conditions of the external environment, the temperature of the water in the interior of the tank is increased in only 0,11 ºC, therefore, the process of acclimatization of the passenger terminal with the use of the thermal accumulator will be initiated at 06:00 p.m. with the water at 6,11 ºC.

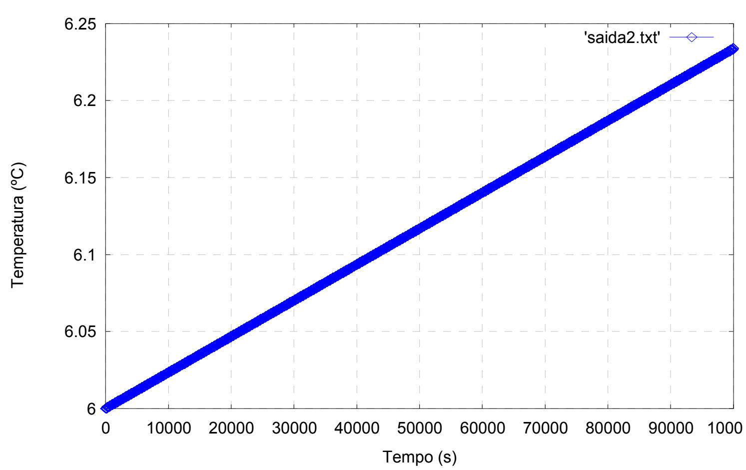

Figure 9. The evolution of the temperature of the water in the tank in the second stage.

Knowing that the temperature of the passenger terminal must be kept at 22 ºC, it is necessary, in this stage of the work, to determine at which temperature the water of the tank of storage will be after 3 hours of operation and in which operational condition will the pumping system act in this period. With this it is possible to conclude if the volume of stored cooled water meets the thermal demand of the passenger terminal.

For the second stage, we considered: T(t=0) = T_{f2} = 6,11 ºC (result obtained from the Eq. 8), T_{TPS} = 22 ºC, $\dot{m}$ = 41,6 kg/s (equivalent to 2 secondary pumps turned on) and the running time of the thermal accumulator = 3 h.

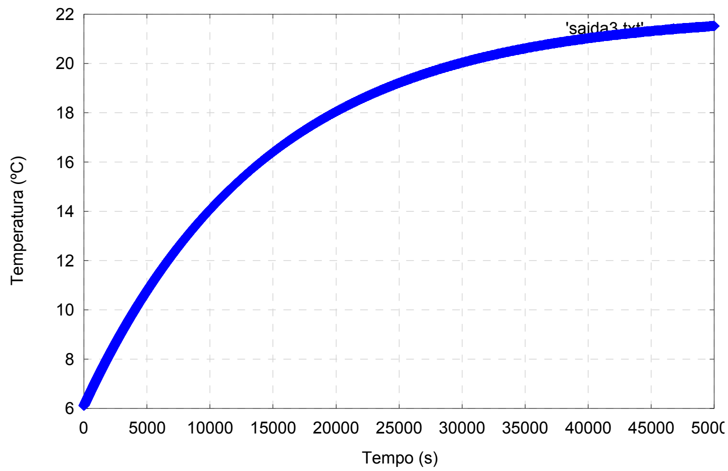

Figure 10. The evolution of the temperature of the water in the tank in the third stage.

Analyzing Fig. 10 it is concluded that after 3 hours of operation, operating with 2 secondary pumps (or 150m³/h), the temperature of the water goes up to 14,86 ºC, this value is in accordance with the ASHRAE Application Handbook (1987). Therefore, it can be affirmed that the volume of the storage tank of water is enough to supply the thermal demand of the terminal.

To complete the operation cycle, it is necessary to know the running time of chillers so that these lower the temperature from 14,86 ºC to 6 ºC.

In this in case that, the same mathematical modeling developed for the first stage is applied, that is, to simulate the time to cool the water of the thermal accumulator, starting from 23,5 ºC until 6 ºC, equation (6), however starting from the temperature of 14,86 ºC.

Using 3 chillers and 3 primary pumps in total load, and applying the same conditions described when the initial temperature was 23,5 ºC, the figure 11 is obtained, where it can be verified that the necessary time will be 17255 seconds, or 4 hours and 48 minutes.

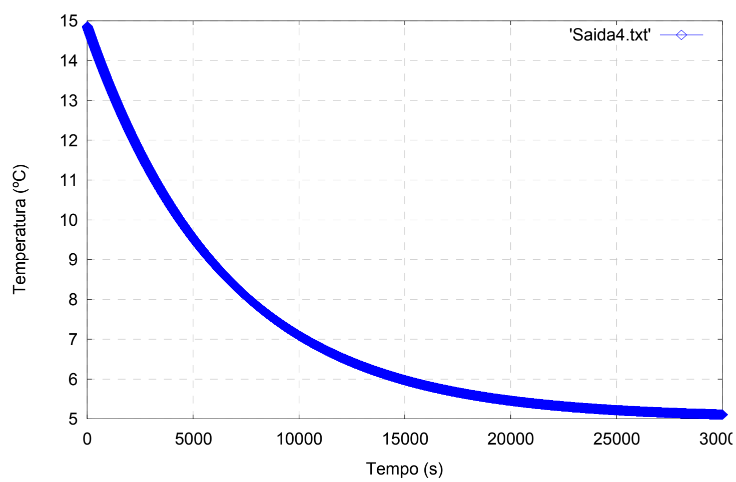

Figure 11. The evolution of the temperature of the water in the tank in the first stage starting from 14,86 ºC

5. CONCLUSIONS

The tariff used for the International Airport Afonso Pena is called "green" horo-sazonal tariff, which establishes the cost of kWh in R$ 0.11 out of the peak schedule and R$ 0,66 in the peak schedule and the cost of the demand in R$ 9,28 at any time.

Thus, assuming an average thermal demand of 1272,91 kWh (362,5 TR) (look at figure 2) in the period from 06:00 p.m. to 09:00 p.m. and that the chillers will have to offer at least, 1239,74 kWh (352,5 TR), the estimated consumption of energy for the cited period is 1170 kWh, what totalizes a cost of R$ 772,00 daily.

However, the power demanded in the period, added to the power of other essential equipment in use at the same time, surpasses the demand contracted for the peak schedule, in this case 800 kW, that is, there will be the incidence of a fine with a value of 3 times the cost of the contracted demand (in this case R$ 9,28), totalizing R$ 27,84 for exceeding kW. Thus, if within the 1170 kWh consumed there is a peak of 50 kW above the contracted demand, a fine of R$ 1,392.00, will be applied on the R$ 772,00 what totalizes R$ R$ 2.164,00.

The consumption of diesel measured in the generators when the chillers are in operation is of 300 liters per hour, 150 liters for each generator, independently of the thermal load. Thus, to operate the system for 3 consecutive hours, the consumption is of 900 liters of diesel. Considering that the cost of the liter of diesel in the city of Curitiba is in average R$ 1,75, the cost, per day of operation, is R$ 1.575,00.

Using the thermal accumulator, the cost will be resulting from the operation of chillers in the schedule between 1:00 am. and 6:00 a.m., period when the contracted demand is of 1200 kW, the energy cost 0,11 R$ kWh and many essential equipments are turned off.

In accordance with what it was previously modeled and simulated, for a configuration using 4 chillers at 100% of power, the system will take 3 hours and 55 minutes to lower the water temperature from 23,5 ºC to 6 ºC, consuming 4,073.33 kWh. However, this is an operation that will only be performed in the "start" of the circuit, once the thermal accumulator is currently off of the system. The cost referring to this operation is R$ 448, 07.

After the period of rest of the thermal accumulator (of the 06:00 a.m. the 06:00 p.m.) has passed and the frozen water accumulated re-circulated by the passenger terminal (from 06:00 p.m. to 09:00 p.m.), the modeling and simulation performed shows that the water temperature will be 14,86 ºC. Therefore, applying the previously described modeling and simulation again, the system will take 4 hours and 48 minutes to lower the water from 14,86 ºC to 6 ºC again, for a configuration of 3 Chillers at 100% of power, totalizing a consumption of 2.894,02 kWh, that is, R$ 318,00 per day, at the cost of 0,11 R$ kWh.

Therefore, of the three methodologies of operation of the system of acclimatization at the peak schedule, the one that uses the tank of thermal accumulation is most economically viable.

According to the objectives presented, it is verified that:

- Starting from the ambient temperature (23,5ºC) and working uninterruptedly with the 4 chillers, the time for the water of the tank to reach 6 ºC is of 14.120 seconds, or 3 hours and 55 minutes;
- In the period of rest of the thermal accumulator, from 04:55 a.m. to 06:00 p.m., the temperature, initially at 6 ºC, gained 0,11 ºC, that is, the temperature of the water at the end of the period is 6,11 ºC;
- The tank initiates the operation of acclimatization of the passenger terminal with the water at 6,11 ºC. At the end of the 3 hours, period where the chillers are off, the temperature of the water in the tank is 14,86 ºC
- The chillers (3 units in maximum load) need 4 hours and 48 minutes to lower the water temperature from 14,86 ºC to 6 ºC;
- The volume of the tank, operating in the conditions described, that is, with only 2 primary pumps functioning, is enough;
- The revitalization of the system of thermal accumulation is viable, due to the very low operational cost of the system when comparing to the other methodologies adopted in the airport, representing only 14,7% in relation to the operation with electric energy and 20,2% in relation to the operational cost with diesel generators.

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