# IMPROVEMENT OF THE SYSTEM FOR FREEZING VEGETABLES. A CASE STUDY: FROZEN VEGETABLE PLANT IN THE REGION OF LA RIOJA (SPAIN)

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**Abstract:** This paper consists of the study and optimising of the cold cycle in a frozen vegetable plant in La Rioja (Spain). Various investigations have been undertaken to reduce consumption and improve the operability of the equipment used in the process.
The paper describes the procedure for determining the operating parameters of the cold cycle, the data obtained and a series of improvement proposals, applied to an actual case study in a frozen vegetable plant in La Rioja (Spain).
The study reveals the fragility of the production system insofar as it depends on atmospheric conditions to condense the ammonia in the cold cycle. The analysis of the existing production processes and the products and sub-products generated revealed a heat sink that could be used, based on treated waste water. The study also provides an analysis procedure specific to this type of industry [1].

**Keywords:** Freezing, thermodynamic cycle, efficiency, rotary compressor, cooling tower, freezing tunnel.

## 1. INTRODUCTION

The study started by determining the sources of energy used in the plant, the uses to which they were put, the losses that occurred and the waste produced in those uses. This required basic knowledge of the production process to distinguish the uses of energy and its origins, and of the wastes generated in each production phase.

The first problem was due to the discontinuous nature of the production cycle. Normally, these companies do not produce continuously but work with wide seasonal variations. This means that there are production peaks in the months in which the main products are harvested and dips in production when they make secondary products. Thus the full installed power capacity of the plant is used only for short periods of the year [2].

Also noteworthy is the difficulty of studying the equipment, which consists of groups of devices that in practice are not accessible to measurement equipment without stopping or modifying the production system, which limited the degree of detail of the study.

It was therefore necessary to set reasonable objectives and design the research suitably to reach the target level of precision [3].

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This work involved a more extensive investigation than that presented in this paper. The work undertaken included the study of the steam generators, the electrical installation, the lighting systems and the plant's cold cycle, although only this last process is discussed here.

## 2. GENERAL PROCESS DATA

### 2.1. Information gathering

This study focuses on the cold cycle needed for the plant's operation. The purpose is to determine the functioning of the equipment in the various stages of the cycle during the production period that is, to learn the functioning of the compressors, cooling towers and freezing tunnels, determine their output and design systems to improve them.

Initially, it was necessary to obtain a general idea of the plant in order to inter-relate the processes. Information was therefore collected on the product cycle and an inventory was made of the installations with logs of consumption and functioning. It was necessary (and it proved possible) to establish interconnections between all of them to learn the flows involved in each item of equipment.

### 2.2. Method

Initially, the operating data of the various items of equipment were obtained throughout the year. This point was important to discover which items of equipment were used in which processes and for which products since these data were necessary to estimate the installation's annual behaviour and to calculate the technical and financial viability of the measures proposed as solutions.

Each item of equipment was then audited, measuring its typical parameters, the energy consumed and the energy given off. The energy consumed is in the form of electricity while that given off is heat.

It was also necessary to measure the environmental parameters – temperature and humidity of the air and temperature of the mains water.

Once these data were obtained, a time series was created for statistical treatment in order to predict the installation's behaviour, using the Holt – Winter [4] method to smooth the series.

## 3. CASE STUDY: COLD CYCLE

As mentioned, this study is focused on the cold process, which causes the greatest production problems, mainly due to the condensation capacity of the ammonia cycle.

Condensation takes place in a bank of cooling towers fed by a manifold to which the ammonia is channelled from the various compressors regardless of its use. The cooling towers use a water circuit which is cooled by a flow of air driven by powerful fans. This means that random properties of the air used can significantly change the towers' cooling capacity.

To analyse this problem suitably, the functioning of each item of equipment was studied to obtain the cycle undertaken and to conclude how it worked and how it could be improved.

### 3.1. Compressors

The functioning and interconnections of each compressor were determined [5] and the operating hours of each unit throughout a production cycle were logged. The results can be seen in Figures 1 and Figure 2.

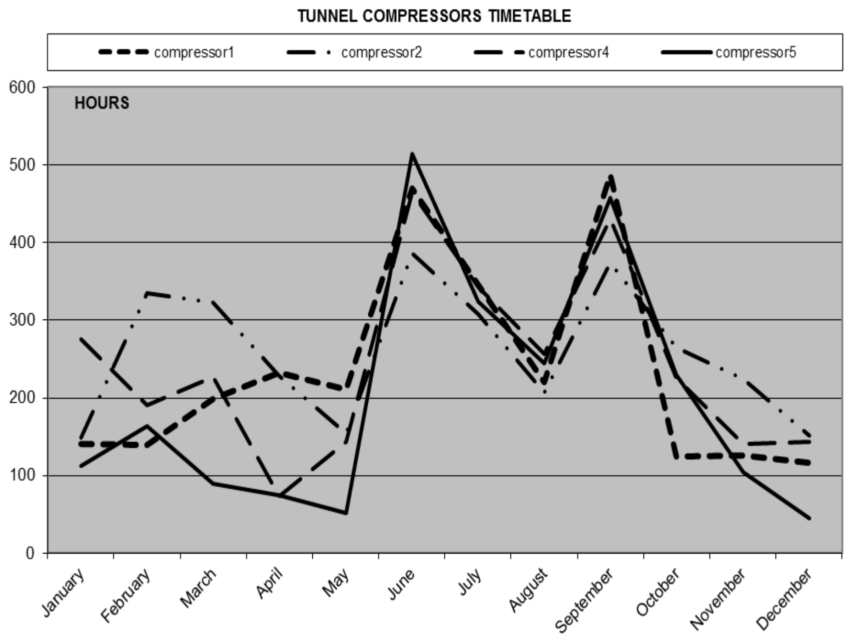


Fig. 1. Operating times of the freezing tunnel compressors.

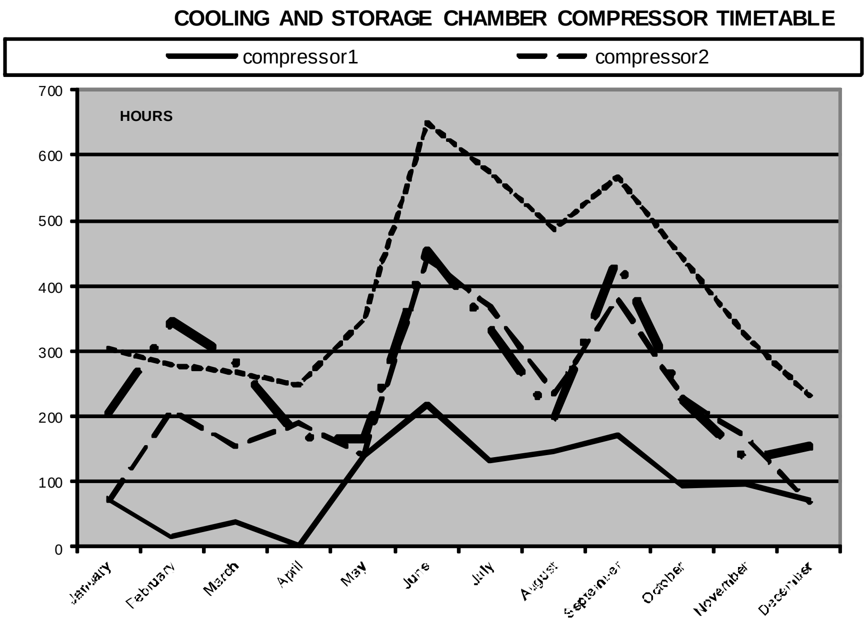


Fig. 2. Operating times of the pre-cooling compressors.

The nominal operating times provided by the manufacturer were obtained for each compressor [6-7]. These figures are for the optimal operating situation in which the cooling towers can condense the ammonia at an ambient temperature of 27 ºC.

However, the evaporation conditions in the cooling towers mean that the compressors do not operate at their optimum design peaks but rather less efficiently. This reduction in efficiency is mainly due to the fact that the compressor must increase its mass flow and this takes it away from its design operating peak.

The mass flows for each element in optimum conditions were also measured to calculate the theoretical output of each item of equipment using the relationship [6]:

$$\dot{Q}_U\left[kW\right]-\dot{Q}_L\left[kW\right]=\dot{m}\left[\frac{kg}{h}\right]\cdot\Delta h\left[\frac{kJ}{kg}\right]\cdot\frac{1}{3600}\left[\frac{h}{s}\right] \tag{1}$$

Once the losses had been calculated, they were compared with those for each load condition at all given times. The average output of the item of equipment throughout the year was determined as a function of the frequency of the data and their statistical behaviour. The environmental conditions for each operating period were also measured by statistically processing data from nearby weather stations [8].

The degree of effectiveness of each item of equipment was determined as a function of the deviation from the theoretical results obtained, calculated on the basis of the equipment manufacturer's information and the measurements taken on site in each situation. Once the relationship between the theoretical prognosis and the actual measurement had been obtained, the possible origins of the losses were analysed in order to assess corrective measures [9].

Thermographs were taken of the compressors [10] to show the temperatures of the fluid inside and the temperature logs and oil changes were analysed. This provided information on the mechanical behaviour of the element, that is, of the energy dissipated in the form of heat in processing, which must be absorbed by the dissipation mechanisms such as lubricating oil and cooling fins.

Finally, with the behaviour of units throughout the year being known, the results were extrapolated to predict the effect of the proposed improvements.

The influence of the operating parameters of the rest of the equipment had to be taken into account, so the diagram of the product's cold cycle was constructed as data were obtained from each station.

### 3.2. Cooling towers

An inventory was made of all the units and their connection to the system and to one another was investigated. This equipment functions on the basis of the forced circulation of an air flow in contact with the exchangers through which the ammonia circulates, sprayed with cold water taken from a well.

The dissipation capacity of the equipment is determined by the weather conditions since the total heat of the air and water dictate the absorption capacity.

This property of the equipment means that in the hottest months the cooling towers are unable to dissipate sufficient heat to condense the ammonia circulating in them under certain production demands. As the demand for cold increases, the compressors increase the flow and heat up, which means that they have to be stopped if the heat cannot be dissipated through the oil, causing significant production losses.

To determine the effectiveness of the cooling towers, electrical mains analysers and temperature probes [11-12] were used to detect the properties of the air and water used in cooling [13]. The amount of flow through the circuit was also measured using ultrasound gauges. With these data, and knowing their functioning through the compressors' chronograms, equation 1 can be used to determine the operation of the installation.

### 3.3. Freezing tunnel

The freezing tunnel is a unit where vegetables are frozen by contact with a turbulent flow of air chilled by the ammonia circuit which feeds it. It was not possible to analyse the freezing tunnel in great detail because this would have required a great deal of time and would have caused production stoppages.

To determine operational failures, thermographs [14] were taken of the points determined as critical and heat bridges were analysed as shown in Figures 3 and Figure 4.

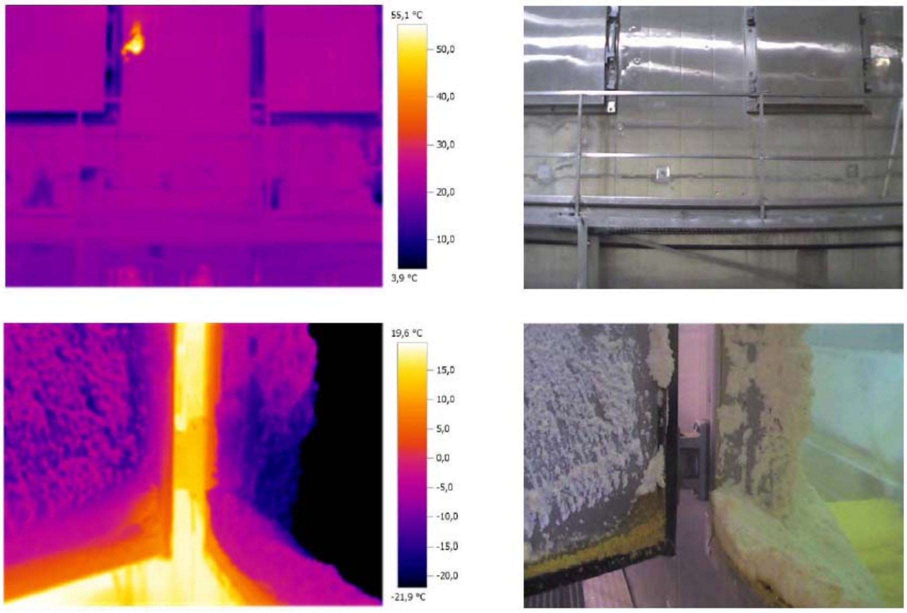


Fig. 3. Freezing tunnel thermographs.

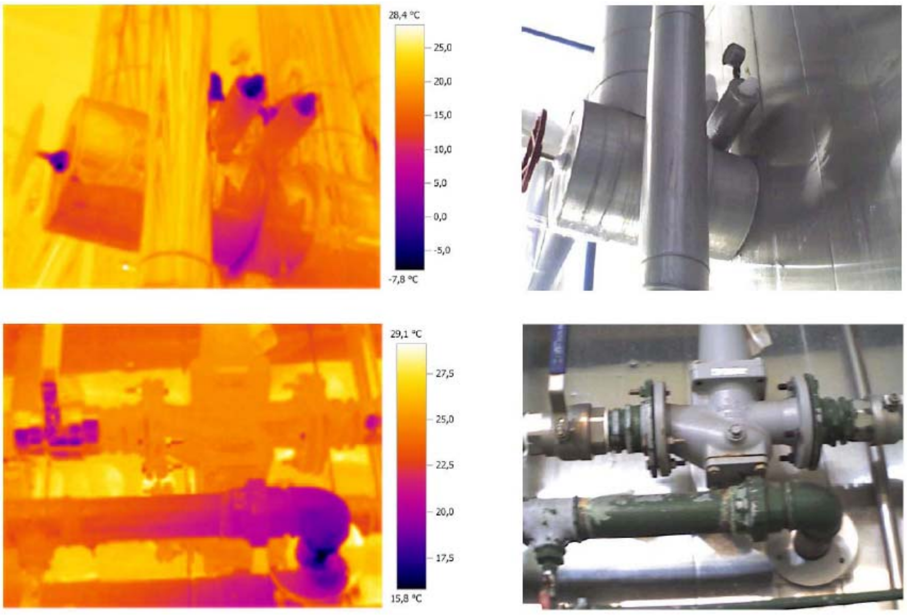


Fig. 4. Thermographs of the freezing tunnel auxiliary installations.

## 4. RESULTS

The calculations shown below were made with Engineering Equation Solver (EES) [15] software. This program contains libraries with thermodynamic state equations for various types of fluids and includes a calculation engine to solve systems of thermodynamic equations.

The program was used to calculate the theoretical cycle that should take place based on the manufacturer's data in the environmental conditions measured. The same environmental conditions and the measured values were used to calculate the actual cycle.

The theoretical and measured thermodynamic cycles for each compressor were studied [16]. In this case, there were various groups of compressors working with the same cycles but with differences in consumption, so the different cycles in the production cycle are shown and the output values obtained are shown in a table below.

The estimated results for the various compressors in the three most standard operating situations are shown in Table 1.

Table 1. Comparison between the theoretical values and the average measured values for usual situations.

| **AVERAGE OUTSIDE AIR CONDITIONS** | **T. AIR. [ºC] 29** | **RH. [%] 30** | **T. AIR. [ºC] 34** | **RH. [%] 20** | **T. AIR. [ºC] 31** | **RH. [%] 43** | **AVERAGE EFFICENCY** |
|---|---|---|---|---|---|---|---|
| | **T. POWER [kW]** | **M. POWER [kW]** | **T. POWER [kW]** | **M. POWER [kW]** | **T. POWER [kW]** | **M. POWER [kW]** | |
| **TUNNEL FREEZER** | | | | | | | |
| COMPRESSOR 1 | 258,00 | 304,91 | 283,8 | 349,04 | 268 | 381,74 | 78,7% |
| COMPRESSOR 2 | 224,00 | 234,03 | 283,8 | 320,15 | 268 | 409,46 | 83,3% |
| COMPRESSOR 3 | 287,69 | 382,44 | 317,88 | 433,43 | 300 | 328,40 | 80,0% |
| COMPRESSOR 4 | 287,96 | 383,82 | 317,88 | 419,16 | 300 | 327,70 | 80,8% |
| **PRE - COOLING** | | | | | | | |
| COMPRESSOR 1 | - | - | - | - | - | - | * |
| COMPRESSOR 2 | 65,30 | 94,57 | 73,2 | 98,03 | 68,4 | 108,08 | 69,0% |
| COMPRESSOR 3 | 65,30 | 91,73 | 73,2 | 92,42 | 68,4 | 97,48 | 73,5% |
| **STORAGE CHAMBER** | | | | | | | |
| COMPRESSOR 1 | 77,30 | 112,58 | 84,5 | 119,51 | 80 | 112,93 | 70,1% |

* OUT OF ORDER DURING THE STUDY

The performance of the tunnel can be considered as identical to that described by the manufacturer [17] since this equipment was installed recently and it's measured functioning parameters match those specified in the technical documentation.

The manufacturer's data [18] as a function of the environmental conditions were taken into account for the cooling towers. The results for the electricity consumption of the fans and pumps and the assumed thermal capacity of condensation are compared with the manufacturer's data.

The results are shown in Table 2.

## 5. DISCUSSION OF RESULTS

The output of the compressors is within the values expected for them, except for tunnel compressors 1 and 2.

The freezing tunnel shows no significant operational anomalies.

The cooling towers show low performance in the pumping equipment, indicating problems of loss of head in the hydraulic system.

The biggest problem in the cycle does not appear to be the operation of the individual units of equipment but the insufficient heat absorption of the ammonia at certain periods. When this happens, the compressors start to increase the ammonia flow since the freezing tunnel requires a higher flow rate when the enthalpy of the fluid that it receives is greater than necessary and, therefore, its capacity for evaporation decreases. This means that the compressors operate poorly: when they try to meet the demand, they overheat the oil circuit and stop.

## 6. IMPROVEMENT PROPOSALS

Based on the analysis of these results, it would appear that action is required on the compressors, in the cooling towers and in the ammonia condensation system, where an auxiliary heat sink is needed.

Table 2. Functioning properties of the cooling towers for the periods studied.

| OUTSIDE AIR CONDITIONS | | | T.=34 ºC RH=20% | |
|---|---|---|---|---|
| EQUIPMENT | ELEMENT | MESSED POWER [kW] | THEORETICAL POWER | EFFICIENCY [%] |
| | PUMP | 7,62 | 3,728 | 48,92% |
| | SLOW FAN | - | 9 | - |
| LSCA-510 | FAST FAN | 25,357 | 18,64 | 73,51% |
| | PUMP | 7,2 | 5,59 | 77,64% |
| | FAN 1 | 19,68 | 18,64 | 94,72% |
| LSCB-690 | FAN 2 | 19,88 | 18,64 | 93,76% |
| | PUMP | 7,83 | 5,59 | 71,39% |
| | FAN 1 | 21,06 | 18,64 | 88,51% |
| LSCB-690 | FAN 2 | 19,32 | 18,64 | 96,48% |
| **OUTSIDE AIR CONDITIONS** | | | **T.=31 ºC RH=40%** | |
| EQUIPMENT | ELEMENT | REAL INST. POWER [kW] | MANUF. INST. POWER [kW] | OUTPUT [%] |
| | PUMP | 7,41 | 3,728 | 50,31% |
| | SLOW FAN | - | 9 | - |
| LSCA-510 | FAST FAN | 25,21 | 18,64 | 73,94% |
| | PUMP | 6,928 | 5,59 | 80,69% |
| | FAN 1 | 18,7 | 18,64 | 99,68% |
| LSCB-690 | FAN 2 | 20,85 | 18,64 | 89,40% |
| | PUMPE | 7,41 | 5,59 | 75,44% |
| | FAN 1 | 18,15 | 18,64 | 102,70% |
| LSCB-690 | FAN 2 | 14,27 | 18,64 | 130,62% |
| **OUTSIDE AIR CONDITIONS** | | | **T.=29 ºC RH=30%** | |
| EQUIPMENT | ELEMENT | REAL INST. POWER [kW] | MANUF. INST. POWER [kW] | OUTPUT [%] |
| | PUMP | 7,27 | 3,728 | 51,28% |
| | SLOW FAN | - | 9 | - |
| LSCA-510 | FAST FAN | 25,91 | 18,64 | 71,94% |
| | PUMP | 19,537 | 5,59 | 28,61% |
| | FAN 1 | 18,8447 | 18,64 | 98,91% |
| LSCB-690 | FAN 2 | 19,18 | 18,64 | 97,18% |
| | PUMP | 7 | 5,59 | 79,86% |
| | FAN 1 | 19,18 | 18,64 | 97,18% |
| LSCB-690 | FAN 2 | 19,221 | 18,64 | 96,98% |

### 6.1. Compressors

Initially, it was proposed to replace the heads of some of the ammonia compressors, whose performance is lower than the average of the rest of the compressors.

Replacing these and monitoring the system load index, assumed to be the optimum designed by the engineer responsible, an increase in performance up to the value indicated by the manufacturer for a new compressor head may be expected.

This increase, as will be seen in the summary table at the end of this section, provides payback times within the period considered by the owner. In calculating payback times stoppages in the production were not taken into account since the modularity of the installation allows the cycle to continue to be supplied from other compressor sets.

### 6.2. Auxiliary geothermal installation as a heat sink

It is proposed to build a geothermal power installation to dissipate heat and help the ammonia condensation system. As with the previous case, the saving has been determined on the basis of the ratio between the electric power used and the heat energy dissipated.

The proposed system consists of various geothermal capture probes 115 m deep, consisting of two pairs of U-shaped tubes. These tubes are inserted vertically all the way into the ground. The ground can be considered capable of dissipating 50 W/m so with 15 probes inserted the dissipation is estimated at approximately 125 thermal kW, taking into account an approximate performance of 70% [19].

This heat exchanger is possible thanks to the use of a thermodynamic cycle of a cooling gas submitted to state changes as shown in Figure 5.

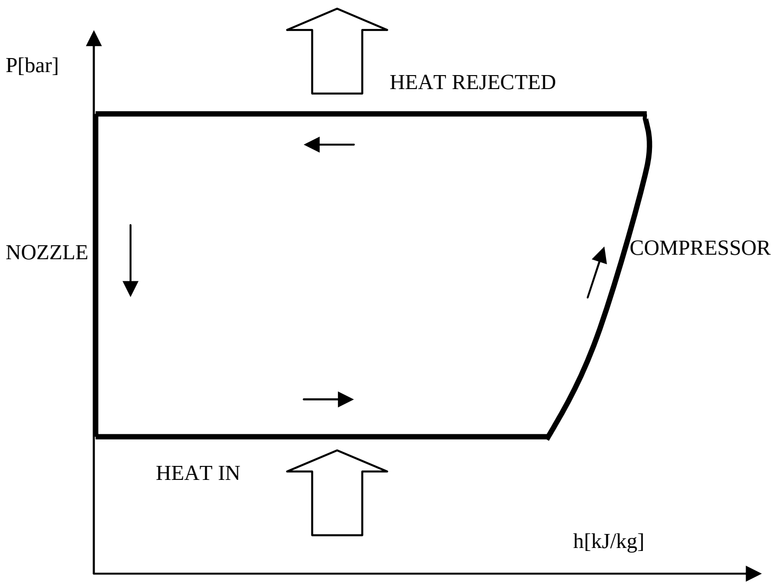


Fig. 5. Heat exchanger.

**6.3. Use of WWTP water as a sink**
A WWTP (waste water treatment plant) is of a system for cleaning the waste water from the various processes in the plant. Various stages provide water clean enough for discharge into rivers.

Because of its flow and temperature, this water can act as a heat sink. The temperatures needed for condensing the ammonia at the installation’s full load from the non-theoretical thermodynamic cycles (previously estimated) are shown in Table 3.

Taking into account the environmental conditions measured, various working situations can be posited for the evaporation condensers and, therefore, for the heat removal capacity for condensing the ammonia.

The maximum amount of heat to be removed in the condensers is shown for each of the proposed systems for the smoothed average values every two months. It can be seen that the averages in the months when the ambient temperature is highest are, on average, low enough to suit the needs of the process.

To try to reduce this effect, it was proposed to exchange the heat from the ammonia with the outlet water from the WWTP. This is possible thanks to the knowledge of the flows and temperatures and the assurance that the outfall standards required by the water authorities will be met [20].

To learn what influence the heat removed by the flow of water from the WWTP on has the cold cycle for each process, functioning periods per product were established and the average maximum and minimum exchanger inlet values were found. Table 4 shows the available mass flows and Table 5 their temperatures.

Table 3. Summary of heat in the cycle by product.

| PRODUCT | RICE | | PEAS | | BEANS | |
|---|---|---|---|---|---|---|
| CONDENSATION HEAT [kW] | 1.968,8 | | 4.286,2 | | 4.031,4 | |
| EVAPORATION HEAT [kW] | -1.449,1 | | -3.128,8 | | -2.933,7 | |
| AIR CONDITIONS | | | | | | |
| MIN [ºC] RH [%] | 26,0 | 15,0 | 27,0 | 20,0 | 28,0 | 18,0 |
| MAX [ºC] RH [%] | 28,0 | 30,0 | 32,0 | 50,0 | 38,0 | 40,0 |
| AVERAGE [ºC] RH [%] | 29,0 | 40,0 | 30,0 | 40,0 | 34,0 | 50,0 |
| CONDENSATION CAPACITY | | | | | | |
| MAX [kW] | 7.490,7 | | 6.804,1 | | 6.804,1 | |
| MIN [kW] | 6.584,6 | | **3.292,3** | | **2.280,7** | |
| AVERAGE [kW] | 4.747,0 | | **4.082,5** | | **3.292,3** | |

Table 4. Maximum, minimum and average flows available per month and product.

| | PROCESSED PRODUCT | | | MASS FLOW [kg/h] | | |
|---|---|---|---|---|---|---|
| | RICE | PEAS | BEANS | AVERAGE | MAX | MIN |
| APRIL | ON | OFF | ON | 352,95 | 886,00 | 139,00 |
| MAY | ON | ON | OFF | 370,00 | 696,00 | 11,00 |
| JUNE | OFF | ON | OFF | 579,92 | 1082,00 | 213,00 |
| JULY | OFF | ON | ON | 421,81 | 888,00 | 17,00 |
| AUGUST | ON | OFF | ON | 426,31 | 923,00 | 73,00 |
| SEPTEMBER | OFF | OFF | ON | 400,30 | 733,00 | 13,00 |
| OCTOBER | OFF | OFF | ON | 318,23 | 617,00 | 46,00 |
| NOVEMBER | ON | OFF | ON | 156,00 | 617,00 | 2,00 |
| DECEMBER | ON | OFF | OFF | 187,00 | 590,00 | 1,00 |
| JANUARY | ON | OFF | OFF | 200,83 | 481,00 | 11,00 |
| FEBRUARY | ON | OFF | OFF | 286,63 | 572,00 | 8,00 |
| MARCH | ON | OFF | OFF | 215,76 | 472,00 | 1,00 |
| APRIL | ON | OFF | ON | 304,23 | 820,00 | 42,00 |
| MAY | ON | ON | OFF | 317,17 | 546,00 | 79,00 |

Table 5. Maximum, minimum and average WWTP water temperatures per product and month.

| | PROCESSED PRODUCT | | | TEMPERATURE [ºC] | | |
|---|---|---|---|---|---|---|
| | RICE | PEAS | BEANS | AVERAGE | MAX | MIN |
| APRIL | ON | OFF | ON | 20,04 | 22,5 | 17,2 |
| MAY | ON | ON | OFF | 21,66 | 23,8 | 19,8 |
| JUNE | OFF | ON | OFF | 27,01 | 31 | 24 |
| JULY | OFF | ON | ON | 26,56 | 30,4 | 24,4 |
| AUGUST | ON | OFF | ON | 24,42 | 25,7 | 22,4 |
| SEPTEMBER | OFF | OFF | ON | 21,76 | 23 | 19,8 |
| OCTOBER | OFF | OFF | ON | 19,59 | 23 | 15,5 |
| NOVEMBER | ON | OFF | ON | 13,95 | 17,3 | 11,5 |
| DECEMBER | ON | OFF | OFF | 14,64 | 18,7 | 11 |
| JANUARY | ON | OFF | OFF | 11,67 | 17 | 5,8 |
| FEBRUARY | ON | OFF | OFF | 15,43 | 18,6 | 13,3 |
| MARCH | ON | OFF | OFF | 17,25 | 19,9 | 14,7 |
| APRIL | ON | OFF | ON | 21,41 | 26,1 | 16,2 |
| MAY | ON | ON | OFF | 21,75 | 26,4 | 16,9 |

In each case, assuming a maximum temperature jump of 7 °C and an exchanger efficiency of nearly 95%, a reduction in the heat evacuated by the evaporation condensers was obtained. These values are shown in Table 6.

Table 6. Heat absorption capacity and percentage of contribution to the total.

| | DATA | FLOW [m3/day] | MASS FLOW [kg/h] | TEMP. IN [ºC] | TEMP. OUT [ºC] | POWER [kW] | PERCENTAGE CONTRIBUTION [%] |
|---|---|---|---|---|---|---|---|
| RICE | AVERAGE | 281,69 | 23,47 | 18,22 | 23,22 | 136,50 | 2,88 |
| | MAX | 660,30 | 55,03 | 14,88 | 19,88 | 319,97 | 6,74 |
| | MIN | 36,70 | 3,06 | 21,60 | 26,60 | 17,78 | 0,37 |
| BEANS | AVERAGE | 339,98 | 14,17 | 21,00 | 26,10 | 82,37 | 2,50 |
| | MAX | 783,43 | 32,64 | 18,40 | 23,40 | 189,82 | 5,77 |
| | MIN | 47,43 | 1,98 | 24,00 | 29,00 | 11,49 | 0,35 |
| PEAS | AVERAGE | 422,23 | 17,59 | 24,24 | 29,24 | 102,30 | 2,51 |
| | MAX | 803,00 | 33,46 | 21,28 | 26,28 | 194,56 | 4,77 |
| | MIN | 80,00 | 3,33 | 27,90 | 31,00 | 12,02 | 0,29 |

The heat dissipated by the WWTP water will result in a reduction of electricity consumption in the evaporation condensers while improving the operation of the compressors in the cycle.

The approximate number of operating hours for the equipment per product was calculated, based on the average annual production supplied by the owner.

Equation 2 was used to calculate the total heat which the evaporation condensers would not need to dissipate per year according to the number of hours and the average saving in heat to be dissipated.

$$Q[kW.h_{TH}/year] = \begin{cases} RICE \\ BEANS \\ PEAS \end{cases} \equiv \begin{cases} 136.5\,[kW] \\ 82.37\,[kW] \\ 102.30\,[kW] \end{cases} \cdot \begin{cases} 1988\,[h/year] \\ 1484\,[h/year] \\ 836\,[h/year] \end{cases} \tag{2}$$

After the approximate condensation needs for the various products and the nominal consumption of the condensers with their respective outputs were estimated, the kW_electrical/kW_thermal ratio was determined and applied to the previously calculated heat to obtain the electricity saving.

### 6.4. Summary of measures

Table 7 shows the most important data for the proposed measures.

Table 7. Summary of viability study for each improvement proposal.

| | INVESTMENT [€] | ENERGY SAVING [kWh/year] | ECONOMIC SAVING [€/year] | INVESTMENT RETURN PERIOD [years] |
|---|---|---|---|---|
| **Heat sink using water from WWTP** | €31,760.20 | 25,556.30 | **€3,066.76** | **10** |
| **Improved tunnel insulation** | €13,072.80 | 5,831.20 | **€699.74** | **-** |
| **Geothermal heat sink** | €108,521.00 | 10,627.80 | **€1,275.34** | **-** |
| **Replacement of tunnel compressor head** | €128,913.90 | 133,193.23 | **€15,983.19** | **8** |
| **Replacement of compressor chamber** | €32,738.15 | 45,254.34 | **€5,430.52** | **6** |

## 7. CONCLUSIONS

This research has shown the close interdependency of all the items of equipment and installations. This relationship, through different types of flow – product, water, air, ammonia, electricity, etc – is highly useful for taking advantage of synergies such as that described for the case of the WWTP.

Likewise, the procedure for undertaking this type of study and the equipment needed to obtain all the data has been established.

As has been seen, environmental conditions are fundamental for the proper operation of the production system. In this case, as with most plants of this type, peak production usually occurs in the hottest months due to agricultural requirements. This poses an important design problem which has not always been considered. Installed power must be sufficient to meet the demands of the process in any situation. However, using this equipment far below its optimal functioning peak reduces efficiency. Efforts must be made to absorb heat in hot periods with high temperature air carrying high humidity. Heat sinks must therefore be found for this heat, in the ground or in water.

## 8. ACKNOWLEDGEMENTS

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