Sizing Of A Small Capacity Refrigeration Installation (7.10kw) For The Conservation Of Fish At The Thinwgul Market In Mamou In The Republic Of Guinea

Abstract: The objective of this research is to size a refrigeration installation adaptable to the site of the Thinwgul market 1.5 km from downtown Mamou with an area of 4269m2 in the Republic of Guinea for the conservation of fish. During this study, we determined the main parameters essential for the proper functioning of the cold room for conservation, which are among others: compressor (3kW), condenser (9.135kW and 9.67 m2); evaporator 59 m2; regulator (0.38.10-4m3/h and 8.7bar); pipes: suction (21,090.10-4 m2), discharge (2,603.10-4 m2); cooling performance coefficient: real (3.15 and 4.06), ideal (4.71 and 4.89). Then, according to the analysis of these main results obtained during this study, namely mainly the coefficients of real and ideal refrigeration performance, this suggests a proposal for a refrigeration system with a power of 7.10kW. The refrigerant used in this study is Hydrochlorodifluorocarbon (CHF2Cl) or refrigerant 22 (R22)

Keywords: Energy, Electricity, Optimizing, Storage, Machine.

INTRODUCTION

On a global scale, the production of cold for comfort, conservation and refrigeration needs appear to be a major energy issue. It has many applications in a wide variety of fields (food industries, medicine, comfort in the home, etc.). The fish processing industry is in the aspect of energy efficiency very attractive due to high energy consumption. According to the Norwegian state organization Enova, the Norwegian fishing industry used around 1.1 TWh of energy in 2007. The energy use pattern varies as fish are frozen quickly with a load of high refrigeration for relatively short times, then stored at very low temperatures for long periods. This means that if there was an opportunity to use cheap energy at high loads, significant savings could be made. This could be achieved by using more electricity when it is cheap and stores energy in low temperature thermal energy storage, and offloading it and thus reducing electricity consumption when the price is higher (Björk, A. D. A. M., & Kongstad, C. S. 2016; Niering, E. 2010; & Mendoza-Serrano, D. L., & Chmielewski, D. J. 2014). In addition, in the field of conservation of fishery products, the production of cold occupies a predominant place where it makes it possible to limit losses linked to the conservation of fishery products (Bah, M. et al., 2016; & Rapport inspection de la Direction. 2009). The cooling needs of a tool or a product, cold storage or deep freezing are specific to each industry. Whether it is an agri-food, pharmaceutical or petrochemical industry, optimizing an industrial refrigeration installation is today a major challenge for companies in order to sustain the production tool and minimize the carbon footprint (Affaf, O. A., & Affaf, Y. 2019). In sub-Saharan countries, the deterioration of food products is made rapid because of climatic conditions (temperature, relative humidity) which are favorable to the proliferation of bacteria, yeasts and molds.

The conservation of agro-food products: poisons, meat, vegetables, fruits in Mamou pose enormous financial losses to the community because of the lack of cold storage, in general and especially at the level of the markets in particular. A study for the installation of a cold room at the Thinwgul market in Mamou in the Republic of Guinea would extend the shelf life of products which will improve trade, income for producers and commercial agents. Thus, this article concerns the study of a project for a refrigeration installation of the order of 7.10 kW.

MATERIAL AND METHOD

Material

Description of the study area

The Urban Commune of Mamou is located 270 km from Conakry; it covers an area of 8,000 km2 with a population of 222,000 inhabitants. The average annual rainfall is 1651mm, the average annual temperature is 25 ° C (Dimensiomement de l’installation photovolakique du centre de santé de Dounet. 2008). In the Urban Commune of Mamou, there are only three (Mendoza-Serrano, D. L., & Chmielewski, D. J. 2014) conservation points for low-capacity fish and animal products. These points are all located in the city center (Poudrière district).

Thinwgul market built in 2014 is located in Thinwgul district 1.5 km from downtown Mamou. It is surrounded by four large districts: Boulibinet, Abattoire, Tambassa and Loppet. This market has an area of 4269m2; it includes three blocks subdivided into: 20 stores, 343 points of sale for women sellers of perishable products and 10 latrines.
Theoretical study of the refrigeration system

This study is based on certain essential parameters which are among others:

Refrigeration cycle

The enthalpy diagram depends on the refrigerant used. It allows the study and sizing of refrigeration machines with great precision. In practice, this cycle is plotted on the following bases: isentropic compression, isenthalpic expansion, in design superheating (SH = 5 °C); and the subcooling (SR) varies between 5 °C to 10 °C (http://jltimin.free.diagramme_2015).

A - Cooling capacity

The cooling capacity (Q_c) for a cold room is given by relation (1):

\[ Q_c = \frac{Q_{th}}{t} \]  

Or: \( Q_{th} \) is the daily thermal load of the cold room in [kJ] and \( t \) the operating time of the installation.

Evaporating temperature

The evaporation temperature (T_e) is given by relation (2).

\[ T_e = T_{c1} + T_{SC} \]  

Or: \( T_{c1} \) is the temperature in the coldroom, \( T_{SC} \) is the superheat temperature of the vapor in the evaporator given by the manufacturer.

b) Condensing temperature

The condensing temperature (T_c) is given by relation (4).

\[ T_c = T_{ev} - T_{SR} \]  

Where: \( T_{ev} \) is the temperature of the external environment and \( T_{SR} \) : Subcooling temperature in the condenser, given by the manufacturer.

c) Temperature at the compressor inlet

The temperature at the compressor inlet (T_i) is given by relation (5).

\[ T_i = T_e + T_{SC} \]  

d) End of compression temperature

The compression being polytropic, then the temperature at the end of compression is given by equation (6).

\[ T_2 = T_i(\frac{\tau}{\tau - 1}) \]  

Where: \( \tau = \frac{P_2}{P_i} \) : Compression ratio; \( P_i \) : Evaporating pressure in bar and \( P_2 \) : Condensation pressure.

METHOD

The method consists of dimensioning the main components of the refrigeration system.

Compressor

The choice of compressor depends on certain quantities which are: the mass flow rate of the refrigerant to be moved \( (q_{m_1} \text{[kg/s]}) \), the volume flow rate that the compressor must aspirate \( (q_{vth} \text{[m}^3/\text{s}]\)), the theoretical volumetric flow of the compressor or hourly volume swept by the pistons \( (q_{vth}\text{[m}^3/\text{s}]\)) (http://herve.silve.pagesperso.. 2015). These different sizes are calculated by the following formulas:

a) Mass flow rate to be displaced of the refrigerant \( (q_{m_1}) \)

\[ q_{m} = \frac{Q_{th}}{h_{c1} - h_{c2}} \]  

b) Volume flow to be aspirated by the compressor \( (q_{vth}) \)

\[ q_{vth} = q_{m} \cdot V_1 \]  

c) Theoretical compressor flow (or hourly volume swept by the pistons)

\[ q_{vth} = \frac{q_{m}}{\eta_v} \]  

Where: \( \eta_v \) : Mechanical efficiency of the compressor; \( \eta_v \) : Volumetric efficiency.

Condenser

The heat output \( (\phi_k) \) and the exchange surface \( (S_k) \) of the condenser are determined respectively by relations (13) and (14):

\[ \phi_k = q_{m}(h_2 - h_3) \]  
\[ S_k = \frac{\phi_k}{h_5} \]  

Where; \( K_c \) : Global heat exchange coefficient of the condenser in [W/m²K]; \( h_5 \) : Enthalpy at the outlet of the condenser in [kJ/kg]; \( h_3 \) : Enthalpy at the outlet of the evaporator in [kJ/kg]; \( h_2 \) : Enthalpy at the outlet of the regulator in [kJ/kg].

Regulator

The selection of the expansion valve is made at the manufacturer according to the cooling capacity \( (Q_c) \), the liquid volume flow \( (V_{liq}) \) and the pressure drop.

\[ V_{liq} = \frac{q_{m} \cdot V_1}{h_{liq}} \]  

The pressure drop that the regulator must create is \( \Delta_p \):

\[ \Delta_p = P_k - P_o \]  

Evaporator

The exchange surface \( (S_{ev}) \) of the evaporator is determined by the relation (17):

\[ S_{ev} = \frac{Q_{th}}{K_{ev} \cdot dT} \]  

Where; \( K_{ev} \) : Overall coefficient of thermal transmission of the evaporator in W/m²K; \( dT \) : Temperature difference between the temperature of the medium to be cooled and the evaporation temperature of the refrigerant.
Calculation of pipes

The internal section $S_i [m^2]$ of the conduits is given by the relation (18):

$$S_i = \frac{q_m \cdot V_m}{u_t}$$  \hspace{1cm} (18)

Where: $q_m$ : Mass flow rate of the fluid in the pipe [kg/s]; $V_m$ : Mass volume of the fluid in the pipe in $[m^3/kg]$; $u_t$ : Fluid velocity in the pipe in [m/s].

Refrigeration system performance

The performance of the refrigeration system is evaluated by calculating the various coefficients.

A- Coefficient of cooling and calorific performance of the real machine

The coefficients of cooling and heating performance of the real machine are determined by relations (19) and (20).

- The cooling performance coefficient of the real machine is:
  $$\epsilon_f = \frac{Q_i}{P_{cm}}$$  \hspace{1cm} (19)

- The heat coefficient of performance of the real machine is:
  $$\epsilon_c = \frac{P_{cm}}{Q_i}$$  \hspace{1cm} (20)

B- Cooling and heating performance coefficients of the ideal machine

Table 1: Characteristics of R22 at the different points of the cycle

<table>
<thead>
<tr>
<th>Levels</th>
<th>T (°C)</th>
<th>P (bar)</th>
<th>$h$ (kJ/kg)</th>
<th>$v_1$ (l/kg)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>-13</td>
<td>3,3</td>
<td>408</td>
<td>0.752</td>
</tr>
<tr>
<td>2</td>
<td>+37</td>
<td>12,00</td>
<td>453</td>
<td>0.960</td>
</tr>
<tr>
<td>3</td>
<td>+30</td>
<td>12,00</td>
<td>431</td>
<td>0.852</td>
</tr>
<tr>
<td>4</td>
<td>+30</td>
<td>12,00</td>
<td>231</td>
<td>-</td>
</tr>
<tr>
<td>5</td>
<td>+25</td>
<td>3,3</td>
<td>250</td>
<td>0.852</td>
</tr>
<tr>
<td>6</td>
<td>-23</td>
<td>3,3</td>
<td>250</td>
<td>0.733</td>
</tr>
<tr>
<td>7</td>
<td>-23</td>
<td>3,3</td>
<td>401</td>
<td>0.733</td>
</tr>
<tr>
<td>8</td>
<td>-13</td>
<td>3,3</td>
<td>408</td>
<td>0.733</td>
</tr>
</tbody>
</table>

RESULTS AND DISCUSSION

Sizing of refrigeration system components

- Compressor
  Relations (7) to (12) make it possible to calculate the driving power of the compressor motor, with: $h_f = 408$ kJ/kg; $h_s = 250$ kJ/kg; $q_m = 0.045$ kg/s; $v_1 = 0.752$ m$^3$/kg; $q_{va} = 0.0338$ m$^3$/s; $\eta_p = 0.82$; $q_{eth} = 0.0413$ m$^3$/s; $\eta_{im} = 0.90$; $P_{cm} = 2.25$ kW. Thus, the relation (12) gives the power of the compressor ($P_{cm} = 3$ kW).

- Condenser
  We chose an air condenser with an air speed varying between 2 m/s to 4 m/s and with: $K_s = 35$ W/m$^2$K; $h_s = 453$ kJ/kg; $h_s = 250$ kJ/kg; $T_e = 57^\circ C$; $T_k = 30^\circ C$.

  Relations (8) and (9) respectively give the heat output of the condenser ($\Phi_k = 9,135$ kW) and the exchange surface ($S_k = 9,67$ m$^2$).

- Regulator
  Relations (15) and (16) respectively give the flow rate of liquid refrigerant received by the expansion valve ($V_{liq} = 0.38 \times 10^{-4}$ m$^3$/h) and the pressure drop that must be generated the expansion valve ($\Delta P = 8.7$ bar).

- Evaporator
  The chosen evaporator is air and dry expansion with finned tubes, its exchange surface is calculated by relation (12), with $K_{ev} = 24$ W/m$^2$K, $\Delta T = 5^\circ C$; we have : $S_{ev} = 59$ m$^2$.

Calculation of pipes

The internal section of the pipes is given by relation (13). The different characteristics of the pipes are given in table 2.
Table 2: Dimensions of the refrigeration system pipes

<table>
<thead>
<tr>
<th>Pipelines</th>
<th>(q_m) (kg/s)</th>
<th>(v_1) (l/kg)</th>
<th>(u_i) (m/s)</th>
<th>(S_i) (m²)</th>
<th>Matériau</th>
<th>dx de (mm)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Suction</td>
<td>0.045</td>
<td>0.752</td>
<td>8</td>
<td>21,090.10⁻⁴</td>
<td>copper</td>
<td>5,080 x 6,350</td>
</tr>
<tr>
<td>Repression</td>
<td>0.045</td>
<td>0.960</td>
<td>12</td>
<td>2,603.10⁻⁴</td>
<td>copper</td>
<td>5,080 x 6,350</td>
</tr>
<tr>
<td>liquid</td>
<td>0.045</td>
<td>0.852</td>
<td>1</td>
<td>2,510.10⁻⁴</td>
<td>copper</td>
<td>8,000 x 9,525</td>
</tr>
</tbody>
</table>

Refrigeration system performance

Relations (19) and (20) respectively give the cooling performance coefficient \(\varepsilon_f = 3.15\) and heating performance of the real machine \(\varepsilon_c = 4.06\).

Relations (21) and (22) respectively give the coefficient of cooling performance \(\varepsilon_f = 4.71\) and heat of the ideal machine \(\varepsilon_c = 4.89\).

**CONCLUSION**

This study allowed us to use a sizing model to determine the characteristics of the main components of the cold room which are: compressor, condenser, and evaporator, and expansion valve, real and ideal coefficient of refrigeration performance. The performance coefficients obtained show that the refrigeration system that we initially proposed for the conservation of fish would function normally. In view of this study, we intend to expand this research on a cold room system operating on the basis of a hybrid system (Photovoltaïque-Generator).

**REFERENCES**