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Analysis of an Industrial Energy Audit for Refrigeration

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ABSTRACT

The investigation focused on the energy consumption of part of the refrigeration equipment of a food supply chain company based in France. The cold rooms concerned are used for fresh and frozen food. The objective of the audit is to verify and analyse the electricity consumption of the equipment and to propose sustainable solutions to improve the whole site. The requested energy audit concerned only two machine rooms (machine room 5 and 7). The study predicted an economic gain of up to €30,000/year from equipment maintenance and a significant reduction in environmental impact at this site. In addition, this energy audit identified additional areas for improvement in machine room 7 that could lead to supplementary energy savings of €16,600 and a potential reduction of 15 tonnes of CO_2 emissions.

Keywords: Energy efficiency, audit, preventive diagnosis, CO_2 emissions, energy saving.

INTRODUCTION

Since 2015 in France, companies that are not part of the Small and Medium Enterprises (SMEs) must carry out a regulatory energy audit of their activity every 4 years. The date of the first audit was set for 5 December 2015. The first audits have therefore expired and the wave of renewals is underway. The companies concerned are all companies with more than 250 employees over their last two closed financial years, or with an annual turnover of more than €50 million and a balance sheet total of more than €43 million. The regulatory audit would concern around 5,500 companies in France. The role of energy efficiency in reducing greenhouse gas emissions and addressing energy security issues is well established [1] and the financial incentive policies in place in France encourage and facilitate the adoption of the approach. It should be noted that companies that have introduced an energy management system certified according to ISO 50001 are exempt from the legally required energy audit

LITERATURE REVIEW

Conducted by an engineering firm specialising in energy, an energy audit, based on a detailed analysis of the data of a building and its equipment, makes it possible to identify areas for energy savings (heating system, insulation, ventilation, lighting, etc.), to draw up a costed proposal and a work programme (investment and expected gains) [1]. The actions to be carried out depend on the energy consumption drifts measured.

Thus, the measurement of energy consumption is the mandatory step to manage energy efficiency actions. General metering and sub-metering of electricity sub-metering, which refine energy consumption over smaller areas, provide energy referents with quantitative and impartial data to determine the energy uses that have a significant impact on consumption. The measurement also provides information on how consumption evolves over time and according to influencing factors [2].

Since "you only control what you measure", a precise knowledge of measuring and monitoring energy performance is necessary [2]. Thus, the technicians in charge of the audit must have a good expertise in energy performance measurement and monitoring plans, including knowledge of metrology, sensors, energy meters, standard reference systems and measurement protocols for energy diagnostics.

This case study is based on the equipment of a logistics company for packaged food products. The Group, which specialises in the transport and logistics of fresh and frozen products, decided to draw up an assessment of its energy requirements. The main objective of this study is to evaluate the energy performance of the two machine rooms No.6 and No.7 (which will be noted MR6 and MR7) in order to determine whether a mutualisation is possible. The secondary objective is to optimize engine room 7 with a view to improving its energy performance.

In order to implement this energy audit, it is necessary to determine, for the site, the main energy consumption items and to carry out an analysis by comparison or calculation with the consumption that would be obtained by implementing more efficient solutions, while establishing the real energy needs of the site.

In the end, the recommendations of the diagnosis will relate to the quantification of the refrigeration needs provided by machine room 6 and the estimation of the energy potential of machine room 7. The action plan will consist of analysing these available cooling requirements and then proposing modifications to improve energy performance.

MATERIALS & METHODS

Presentation of the site

It is a company specialising in cold transport and logistics for all food and heat-sensitive fresh and frozen products. The company is located in the west of France. The photo (figure 1) shows the extent of the structure. The opening time of the company is 52 weeks/year, the number of employees varies from 200 to 299. The site has positive and negative cold rooms (CR) and some administrative offices. The study focused on the equipments in machine rooms 6 and 7 and on the cold rooms they are connected to. The site operates on a just-in-time basis from Sunday from 7 pm to Saturday at 5 pm. Figure 2 shows the relative weight of the various energy consumptions on the site. It should be noted that the cold batch absorbs 44% of the total consumption, 25% of which is only for the screw compressors, which constitutes a significant share. The items included in "Other utilities" correspond to all other energy consumption (56%) not included in the audit.

Figure 1: Overview of the diagnosed site (Source Google Maps)

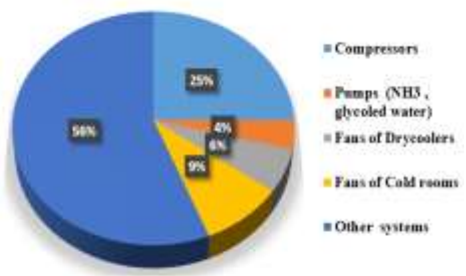

Figure 2: Breakdown of electricity consumption on the site

Audited scope

The perimeter audited includes one positive cold room (CR6) comprising two evaporators connected to the condensing unit of engine room No. 6 and four negative cold rooms (CR) conditioned with 14 evaporators fed by engine room No. 7 (Table 1).

Table 1: Technical equipments to be audited

Relevant machine room	Cold rooms involved	Number of evaporators	Evaporator brand	Installed cooling capacity
MR6	Negative Cold room N°6 (CR6)	2	Rafael	159 kW
Total				159 kW
MR7	Negative reception station	4	Günter	140 kW
	CR N°1	4	Günter	260 kW
	CR N°5	3	Günter	258 kW
	CR N°7	3	Günter	345 kW
Total				1003 kW

This table shows the "theoretical" energy requirements according to the sizing carried out by the Design Office. These data are displayed in the Executed Works Files of each cold room.

Description of the refrigerating machines in machine room n°6 (MR6)

Negative cold production (-34°C) is provided by a refrigeration system operating in double stage NH3 (superfeed) with 2 Sabroe 163L1556 BZ screw compressors, one of which is in continuous operation and the other in standby. These compressors are rated CP11 and CP12. This installation is used for the refrigeration of the cold room n°6 with a volume of 10770 m3. The set point of this cold room is -18°C. The main construction characteristics of this plant are listed in table 2.

Table 2: Compressor Characteristics for MR6

Brand Mark	Type	Year of manufacture	Rated regime (With Ammonia)	Electric Motor (kW)	Cooling capacity (kW)	Theoretical COP
CP11 -34°C York	TDSH 163L	1995	(-34/-16.5/40°C)	160	254	2.12
CP12 -34°C York	TDSH 163L	1992	(-34/-16.5/40°C)	158	254	2.12

The table shows the nominal data calculated by the manufacturer according to the operating speeds.

The values -34/-16.5/+40°C correspond to the ammonia evaporation temperature at -34°C, the “super feed” operating temperature at -16.5°C and the condensing temperature at +40°C. These temperatures are ammonia saturating vapour temperatures.

2.4- Description of the MR7 refrigeration equipment

The production of negative cold (at -34°C evaporation) is ensured by a refrigeration plant operating in double stage NH3 (in super feed). This production (Figure 3) supplies 4 negative cold rooms, all of which are controlled at a temperature set point of -18°C. The total volume to be refrigerated for the rooms is 68534 m^3.

Figure 3: Front view of one of the three screw compressors

The characteristics of the 3 compressors are listed in Table 3. The 3 compressors have a total available cooling capacity of 615kW.

Table 2: Compressor Characteristics for MR7 (Manufacturer's data)

Brand Mark	Type	Year of manufacture	Rated regime (Ammonia)	Electric Motor (kW)	Cooling capacity (kW)	Theoretical COP
CP1 Sabroe	TDSH 193L	2008	(-34/-18.5/40°C)	350	205	1.7
CP2 Sabroe	TDSH 193L	2008	(-34/-18.5/40°C)	350	205	1.7
CP3 Sabroe	TDSH 193L	2008	(-34/-18.5/40°C)	350	205	1.7

METHODS

The expertise of these refrigerating installations is mainly based on the measurements to be carried out and their analysis. The essential steps of the approach can be summarized as follows:

- The installation of additional measuring devices (temperature probes, electrical measuring devices)
- Extraction of the data recorded by the centralised technical management system (CTM) which allows the electrical consumption of each to be collected of the machine rooms and that of the related technical equipment, the measurements of temperatures, flow rates and operating pressures, etc.).
- The analysis of the measurements and the implementation of an action plan

In the following, the measurements made on MR6 are presented and analysed as well as those of MR7.

RESULT AND DISCUSSION

Analysis of the operating regime of the MR6 and its equipment

Figure 4 shows the evolution of the pressures (High Pressure (HP), Medium Pressure (MP) and Low Pressure (LP)) during 35 hours of compressor operation. It shows a large number of start-ups induced by the ammonia pressure build-up in the LP liquid separator receiver. It can therefore be noticed a phenomenon of instability, ‘pumping’ of the installation to maintain a satisfactory pressure in the LP liquid separator tank which feeds the flooded evaporators. The salient values of the pressure variations are listed in Table 4, together with the corresponding saturating vapour temperatures.

Figure 5 shows compressor starts and stops, as well as the variation in power consumption over the course of a day. In this configuration, compressor CP11 was stopped and only CP12 was running intermittently as required.

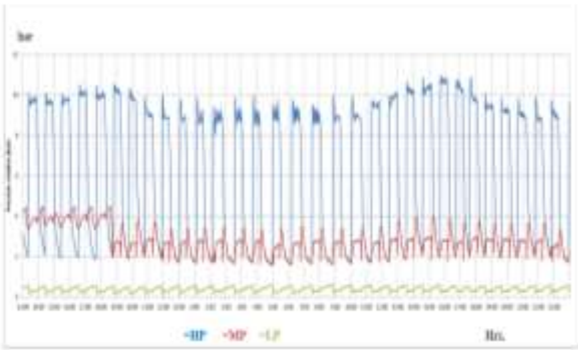

Figure 4: Evolution of operating pressures of MR6 compressors

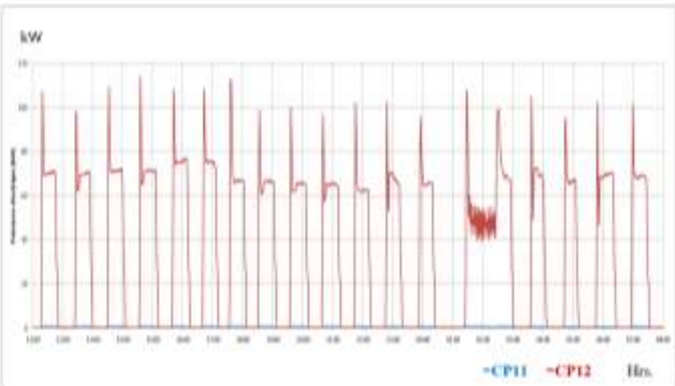

Figure 5: Evolution of the electrical power of MR6 compressors

Table 4: Summary of operating pressures and their correspondences in saturating vapour temperatures

Pressure	Maximum Value	Mean value	Minimum value
HP	10.94 bar / 30,8 °C	9.55 bar / 26,7 °C	8 bar / 21.5°C
MP	4.49 bar / 6.8 °C	3.4 bar / 0.7 °C	1,8 bar / -10.9 °C
LP	0.61 bar / -23.7°C	0.34 bar / -27.6°C	0.27 bar / -28.7 °C

Figure 6 shows the electrical consumption of all the elements that have a significant impact on engine room No. 6. It mainly shows the consumptions of compressor, which accounts for 80% of the consumption, and the utilities such as the fans for the air coolers, pumps, etc.

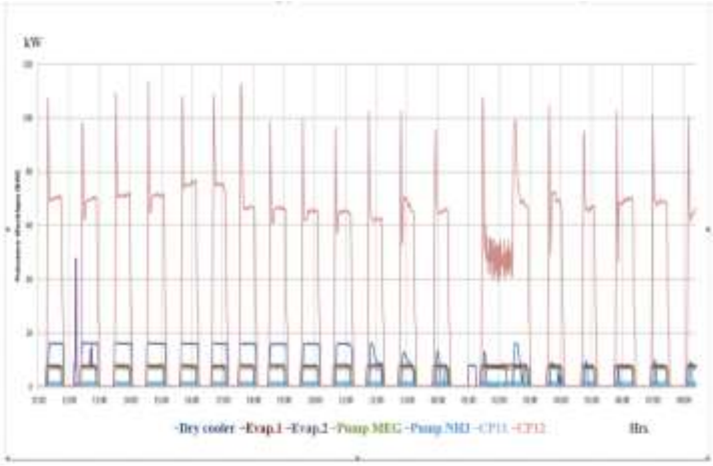

Figure 6: Evolution of Electrical Power in MR6

The plate condenser is equipped with PT 100 probes upstream and downstream (to measure the temperature difference between the inlet and outlet of the condenser), as well as an ultrasonic flow meter measuring the volume flow rate of glycoled water through the heat exchanger. This fluid flow contains 20% of Mono Ethylene Glycol (MEG). The recording of the aforementioned data by the acquisition system makes it possible to calculate and represent (Figure 7) the evolution of the heat output discharged by the condenser ($P_{condenser}$).

Oil cooling is very important on screw compressors in order to ensure the correct operation and durability of the installation. In fact, lubrication reduces friction and limits the temperature rise of the discharged gases. The cooling capacity of the oil can be approximated by the expression [5]:

$$P_{oil} = (0.1827 * P_{condenser}) + 11 \quad (1)$$

P_{oil}: oil discharge power in kW.

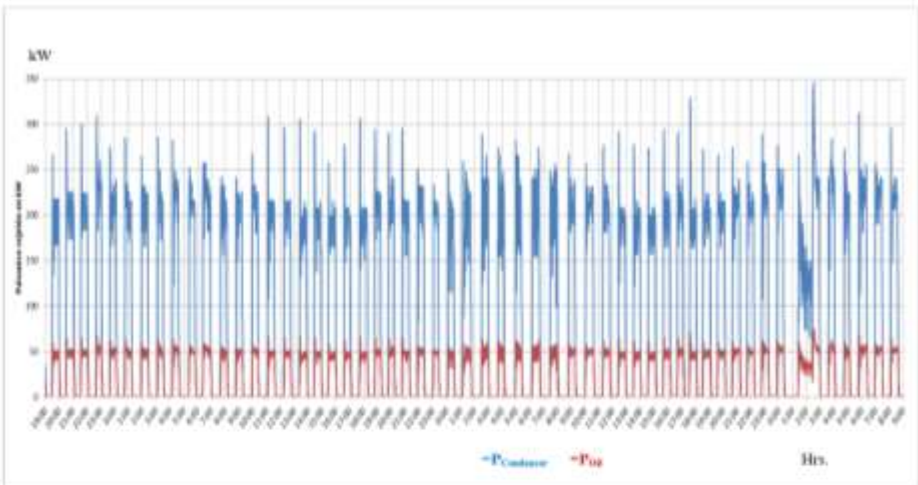

Figure 7: Evolution of heat fluxes released from MR6.

The cooling requirements (P_{CC}) for MR6 were then calculated and plotted (Figure 8):

$P_{CC} = P_{Condenser} + P_{Oil} - P_{Compressor}$ (2)

This capacity reflects the refrigeration needs of cold room N°6 (CR6).

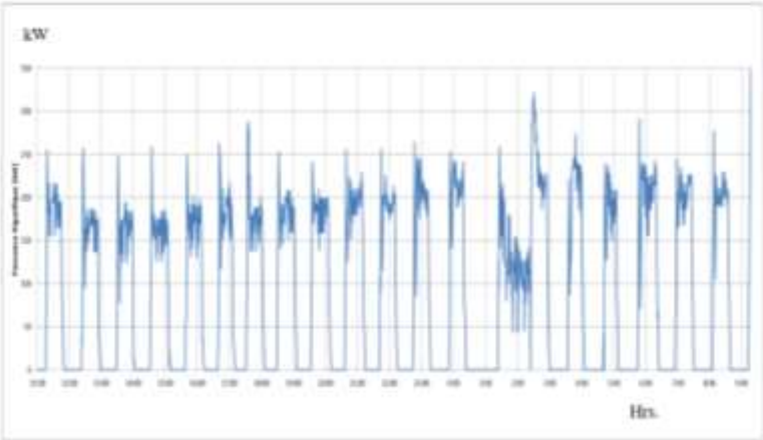

Figure 8: Evolution of the cooling capacity of CR6.

Figure n°9 shows the temperature control in CR6. Two temperature measurements are recorded via two temperature sensors placed in two different areas of the chamber. There is an analogy with figure n°4: the pumping phenomenon that creates an unstable regulation although the temperature in CR6 remains correct with a control between -18°C and -19.5°C).

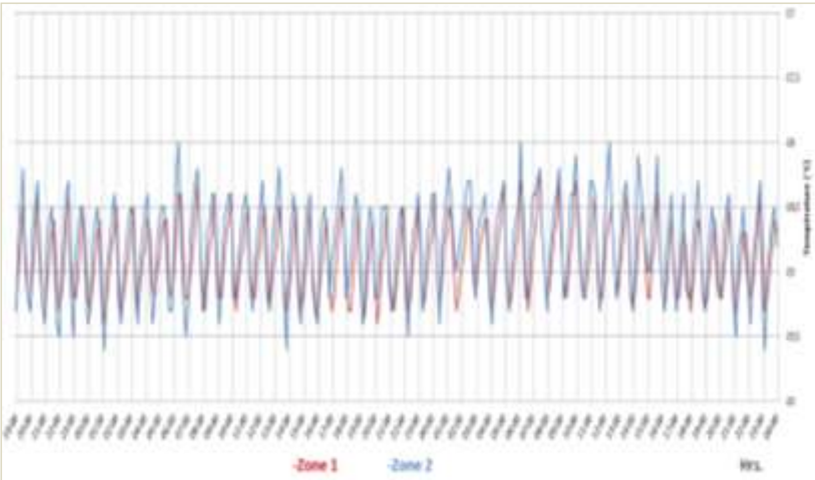

Figure 9: Temperature evolution in CR6

Let's consider now the evolution of the coefficients of performance. Figure 10 shows the evolution of the compressor COP and the engine room COP.

The coefficient of performance or COP is generally the quotient of the useful cooling demand over the energy expenditure to meet that demand. The COP is thus a ratio of energy or power. It reflects the efficiency of a refrigeration system [6]. Two coefficients of performance are defined here:

- The 'compressor COP' is the ratio between the energy consumed by the compressor and the refrigerating energy generated by the system. This coefficient indicates the performance of a compressor, which is the most energy-consuming part of a refrigeration system. It gives a quick approach to the efficiency of the system. Here the compressor COP is 2.7.
- The 'engine room COP' is the ratio of the energy consumed by all the elements

of the engine room (compressors + fans + pumps) to the refrigerating energy generated by the installation. This coefficient indicates the actual performance of a refrigeration system. Here the average actual COP is 2.2.

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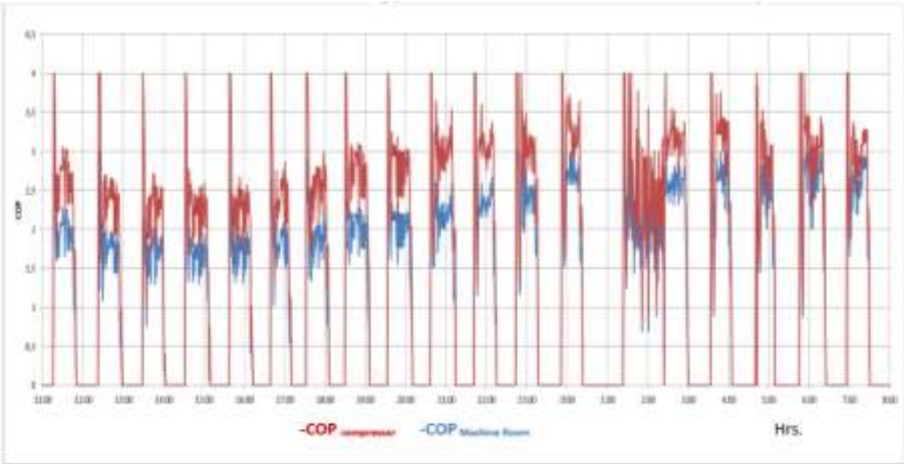

Figure 10: Evolution of the COP of CR6

A brief summary of the powers involved in MR6 is given in Table 5.

Table 5: Summary of MR6 powers

Power	Maximum value	Average value
Cooling capacity (kW)	321	175
Rejected heat flux (kW)	435	239
Power output (kW)	114	64

In conclusion, the compressor control is not optimal, with a pumping phenomenon (untimely stops and starts observed for the CP12 compressor). Moreover, the accuracy of the cold room temperature controls can be improved. Indeed, in the current state, this requires the compressor to start 24 times a day for an operating time of 12 hours/day. A possible remedy could be to run the compressors longer at a lower speed throughout the day.

Analysis of the operating mode of MR7 and its equipment

The low and high pressure saturating vapour temperatures of the compressor plant show a relatively stable operating regime and regulation (Figure 11). The range of variation is greater at the condenser, because the HP pressure is dependent to some extent on the fluctuating temperature of the outside air that cools the dry-cooler. The extreme values as well as the average temperatures of condenser and evaporator are recorded in Table 6.

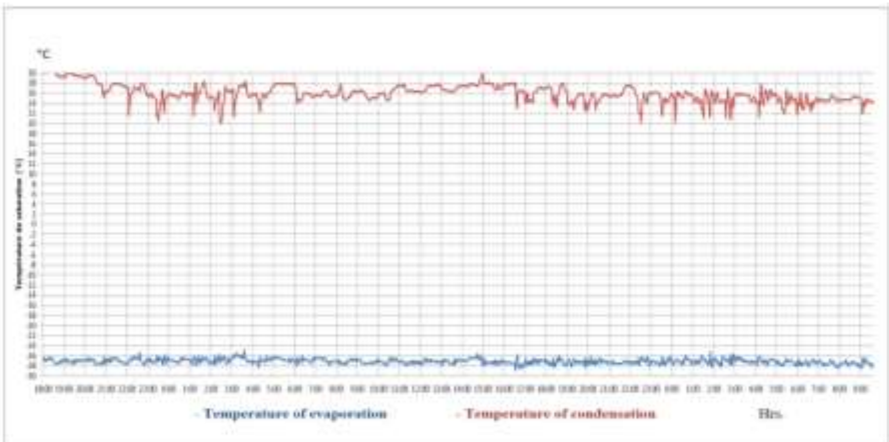

Figure 11: Evolution of the saturation temperatures of the compressor plant in MR7.

Table 6: Summary of saturation temperatures in MR7.

	Maximum	Mean	Minimum
$T_{condensation}$ [°C]	31	25.6	20.1
$T_{evaporation}$ [°C]	-24.9	-27	-28.9

The analysis of compressor power consumption shows that compressor 3 is not running for the period under consideration (Figure 12). Furthermore, when compressor 2 starts up, it is observed that compressor 1, which is in continuous operation, reduces its speed, which is detrimental to the proper functioning of the installation. This is because the total cooling capacity supplied by both compressors decreases instead of increasing when two compressors are operating simultaneously.

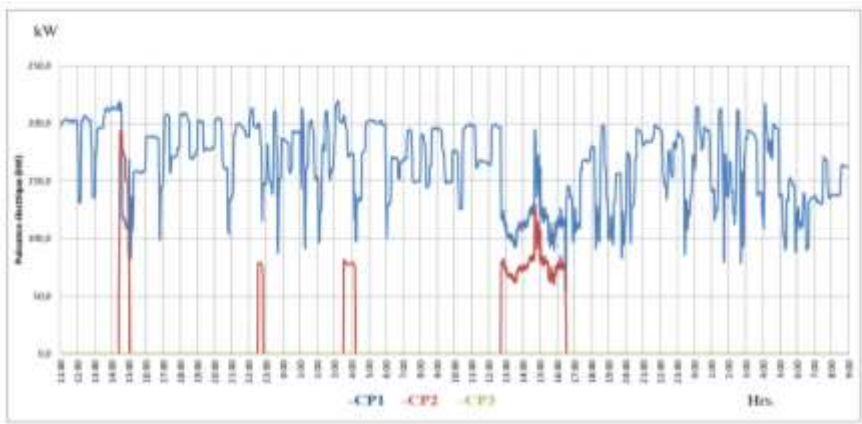

Figure 12: Evolution of the power consumption of the compressors of MR7

The electrical consumption of the compressors (CP1 and CP2) in operation was determined on a typical day (Figure 13). The contribution of CP2 is only 8.5% of the total consumption, as this compressor operates very little.

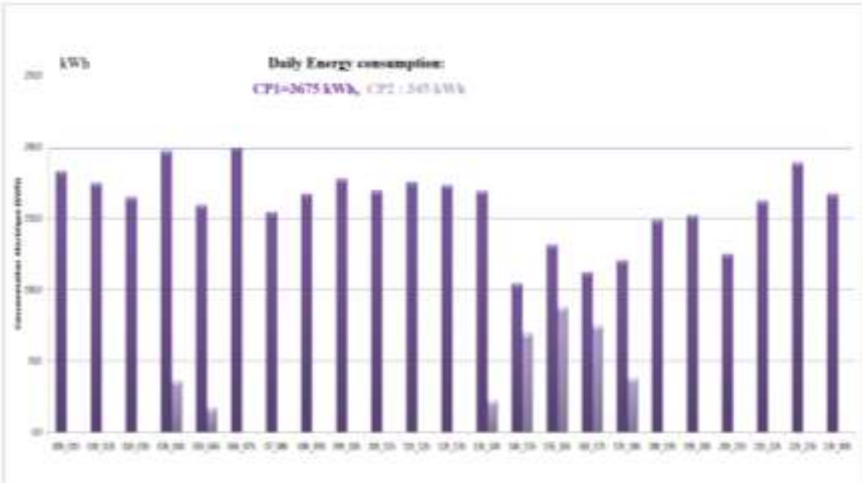

Figure 13: Trend in power consumption of MR7 compressors

As for MR6, the same procedure was adopted to calculate the heat output discharged by the oil and the condenser, and then the refrigeration requirements of MR7. Only the graphs of the evolution of the refrigerating capacity (figure 14) and the refrigerating production requirement of the cold rooms in kWh (figure 15) are reported.

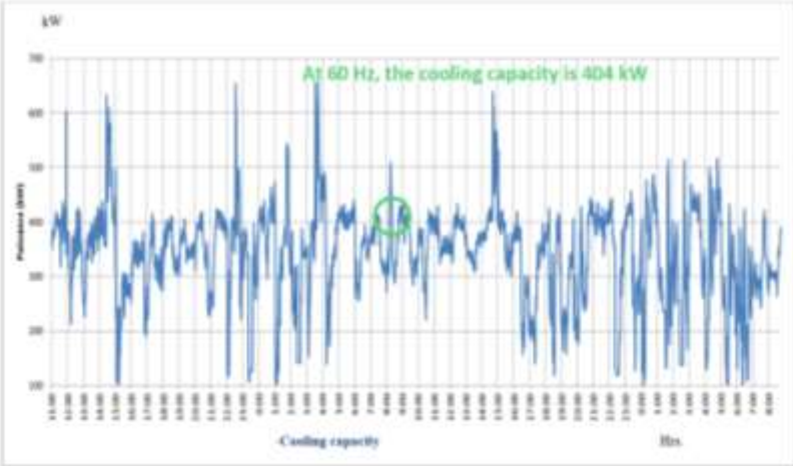

Figure 14: Evolution of the cooling capacity of the MR7

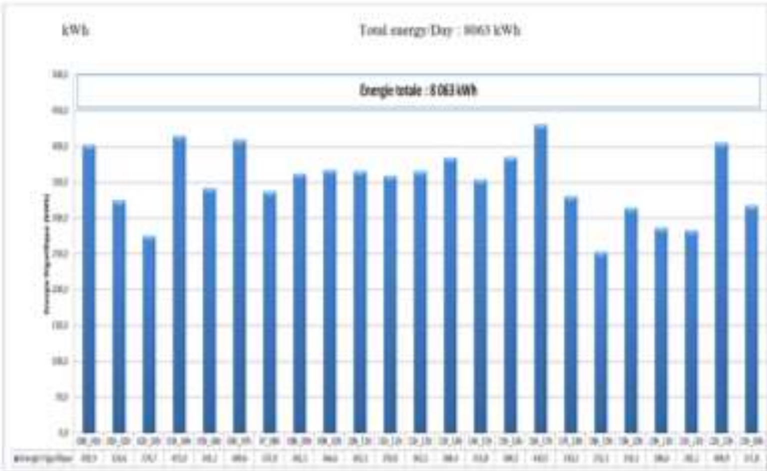

Figure 15: Evolution of MR7 cooling requirements [kWh].

The following 3 figures (Figures 16 to 18) represent the temperatures in the cold rooms. These values do not necessarily reflect the operation of the control because these temperature sensors are not part of the temperature control chain of the systems. However, they do give interesting information on the evolution of the actual temperatures. The values measured are correct and do not show large amplitudes for all the chambers.

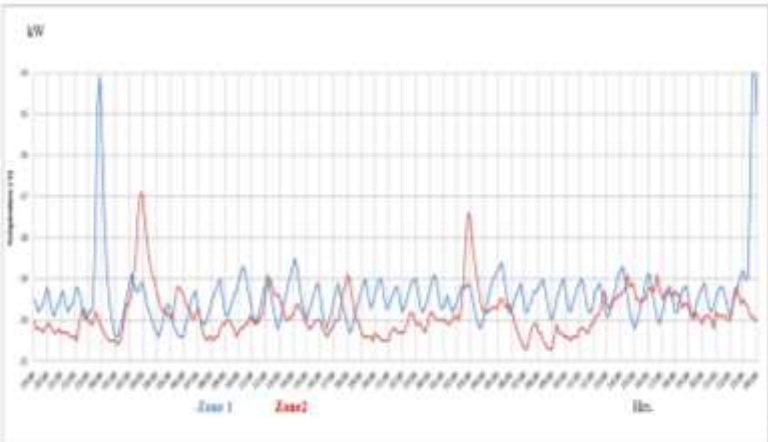

Figure 16: Evolution of CR1 temperatures

Figure 17: Evolution of CR5 and Negative Reception Station temperatures

Figure 18: Evolution of CR7 temperatures

A summary is now presented in the form of the daily power consumption of the various equipment (Tables 7: a) and b) and the operating regime of the plant. The CP1 compressor achieves an electrical power consumption of 200 kW with the compressor running at 100% and 60 Hz. This compressor (CP1) provides a cooling capacity of 404 kW at 100% duty cycle and 60 Hz. The cooling capacity of engine room 7 is therefore 808 kW.

Table 7: Summary of consumption of MR7 equipment

a) Energy consumed in kWh

Compressor 1	3951	65%
Compressor 2	241	4%
Pumps (NH3)	62	1%
Pumps (Glycoled water)	774	13%
Fans of Drycooler	1072	18%
Total	6100	100%

b) Powers of MR7

Capacities	Maximum (kW)	Mean (kW)
Cooling	658,2	341,6
Rejected	960,4	508,7
Power output	406,9	172,9

The CP1 compressor has an electrical power consumption of 200 kW when running at 100% (60 Hz). It provides then a cooling capacity of 404 kW. The cooling potential of engine room 7 is therefore 808 kW for the 2 compressors running at their maximum capacities. The operating regime at the time of the surveys was: -29/-2.7/+28°C in ammonia saturated steam temperatures.

Engine room sharing

Following the analysis previously carried out on the performance of the two engine rooms, the next step is to examine the possibility of pooling the two engine rooms, i.e. to eliminate engine room 6 and to supply refrigerating energy to Cold Room 6 via MR7.

Estimation of the future cooling capacity of MR7

Figure 19 shows the simultaneous cooling capacities of the two plants. This reflects the cooling capacity to be delivered by MR7 after the removal of MR6 and the connection of cold room No. 6.

The graph represents the hourly cooling capacity of the cold rooms. It is a representation of the required future capacity of the engine room.

Here is the detail of the refrigeration requirement on the survey day:

- When the refrigeration needs are ≤400 kWh/h, a compressor is sufficient to supply them.
- When the refrigeration needs are ≥400kWh/h, one compressor is 100% engaged and the second one regulates according to the need.
- For example, from 3pm to 4pm, the cooling requirement was 550 kWh. One compressor provides 400 kWh, and the second one 150 kWh, which corresponds to a compressor operating at a frequency of 25 Hz, its minimum capacity (minimum operating threshold).

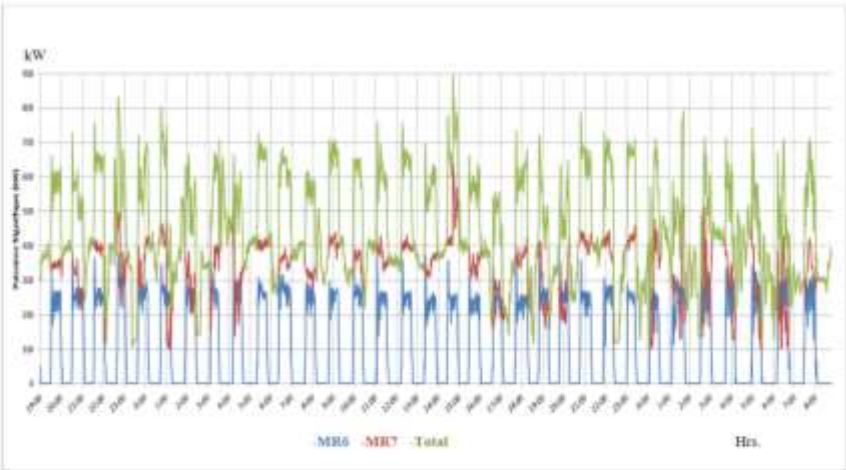

Figure 19: Total simultaneous cooling capacity of the MRs

The daily refrigeration requirements called up by the cold stores are shown on Figure 20. It is a representation of the use of the engine rooms. Cold room No. 6 uses 2,491 kWh of cooling energy in one day. This corresponds to a stable cooling capacity of 103 kW smoothed over 24 hours.

The MR7 engine room has three compressors with a maximum unit cooling capacity of 404 kW (at 60 Hz). This engine room supplies the CR1, CR5, CR7 and the negative receiving station with a compressor that does not run at 100% throughout the day. The maximum energy that a 100% 60 Hz compressor can provide during a day is 9,696 kWh. However, the current energy demand is 8,198 kWh (84% of the power for 24 hours).

A single compressor is sufficient to meet refrigeration needs at certain times of the day. However, the second compressor will regulate when the 100% compressor can no longer meet the refrigeration needs on its own. In addition, the plant already has a third CP3 compressor of the same capacity.

To carry out the above considerations, the following calculation assumptions have been made:

- The operating regime of the engine rooms is at -29/-2.7/+28°C. It is assumed to be constant throughout the day.
- Refrigeration requirements were calculated from the readings we took.
- In summer, in the worst case, a 100% compressor and a 50% compressor will be required (using data transmitted by the costumer)

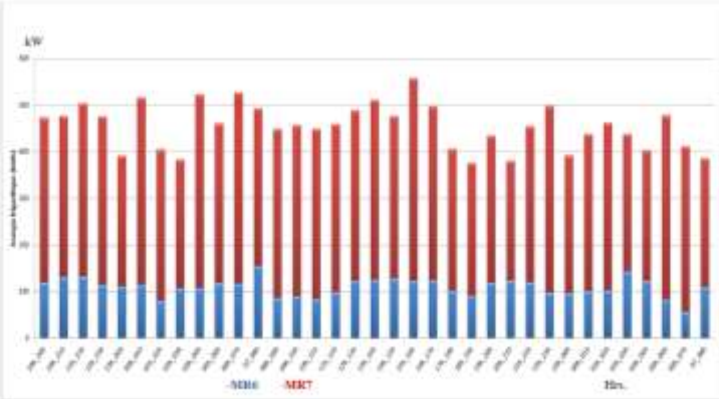

Figure 20: Refrigeration energy of the two engine rooms

Energy optimization of MR7

This section lists some improvements to be made to the system to lead to its optimization

We see undercooling at the outlet of the 5K plate condensers, which is high for ammonia and leads to an extra consumption of almost 10% on the compressors. Such a value of fluid undercooling reflects the presence of air or clogging in the condensers. Indeed, a fluid blockage due to an excess of refrigerant causes a decrease of the condensation zone and an increase of the subcooling zone, as well as the increase of the high pressure which is accompanied by an overconsumption of the compressors [7]. Ideally, as little subcooling as possible should be used with ammonia, in order to optimize the performance of the system . For a decrease in subcooling of 4K, the estimated energy saving is 141.4 MWh/year on the compression work, i.e. a saving of €12,019 (estimates based on the customer's actual annual consumption) and a saving of 11 tons of CO2 not emitted into the atmosphere [8].

A fairly significant pressure drop was observed on the compressor discharge line. Indeed, at an HP working pressure at the oil bottle outlet of 11.2 bar, the condenser inlet pressure was 10.54 bar. This pressure drop of around 0.7 bar corresponds to 2K equivalent saturated vapour pressure on the discharge line, which is high. This pressure drop induces additional and unnecessary consumption on the compressor motors (about 5% too much).

The various elements causing this pressure drop are as follows:

- Oil separator outlet valve (V.03 CP.01): -0.09 bar
- Flap valve (V.02 CP.01): -0.38 bar and upstream constant pressure valve (PCV.01 HP): -0.19 bar

Figures: Two of the valves causing additional pressure drops

These reports were taken on each compressor (CP1 and CP2) and the results are essentially identical.

The installation of a bypass valve in parallel with the constant pressure valve with a lower pressure drop would reduce the pressure drop, as the constant pressure valve is only used during the start-up period. The 0.19bar pressure drop of the pressure valve corresponds to 0.6K. The energy saving potential for an annual consumption of 1,414,282 kWh/year (customer data) is close to 21.2 MWh/year, i.e. a maximum of €1,802 and a gain of 1.65 tons of CO2.

Oil cooling : the installation of solenoid valves on the oil coolers of each compressor would make it possible to isolate them when the associated compressor is not operating. In fact, leaving the 3 oil coolers open creates an additional flow inducing a higher power consumption on the MEG pumps.

Table 8: Determination of MEG overflows

	Oil cooler flow rate (m^3/h)	Condenser flow rate (m^3/h)	Flow rate downstream of pump(s) (m^3/h)	Overflow (m^3/h)
1 Pump running	34	145	262	83
2 Pumps running	110	220	375	45

Installing variable speed drives on the pumps would allow the flow rates to be better adapted to the glycol needs. The annual energy saving with this device has been estimated at 32.7 MWh (i.e., 2780€) with a potential gain in CO_2 of 2.55 tons/year.

In the following, the economic impact of the removal of MR6 is synthesized in regards to maintenance.

Table 9: Economic impact of removing MR6

Preventive Refrigeration Contract	**5 400 €**	**Per year**
Curative intervention on equipments	1 000 €	Per year
NH3 Audit	141 €	Per year
Retrofitting of compressor + valve units	3 600 €	Every 5 years
Routine inspection	900 €	Every 40 months
Implementation of the Refrigeration works	5 000 €	Per year
Mechanical overhaul of compressors	8 000 €	Every 30,000H
Condenser water consumption	294 €	Per year
Overhaul of electric defrost	8 000 €	/
Lowering the contract power	5 600 €	Per year
TOTAL	149 675 €	Every 5 years

CONCLUSION

All the results show the possibility of removing engine room No. 6 and connecting CR6 to MR7, which has a larger refrigeration capacity. The connection would therefore make it possible to eliminate one engine room and have an economic gain on building maintenance close to 30,000€/year, with a significant reduction in the environmental impact of this site. In addition, this energy audit has made it possible to identify areas for improvement in engine room 7.

The following inadequacies were observed in the equipment:

- Significant pressure losses on the discharge line
- Large subcooling at the outlet of condensers
- Unnecessary overflow of MEG circulation pumps due to uninsulated compressor oil cooling when the corresponding compressors are shut down.

The additional savings expected after implementation of the proposed action plans is approximately 16600€ for a saving of 15 tons of CO2.

Altogether, this case study has revealed that there is a large untapped potential for energy efficiency improvements in various industrial sectors. Government industrial energy audit programmes that subsidise companies to carry out an energy audit are the most common policy to try to bridge the energy efficiency gap. 9] These programmes should also be extended to medium and small enterprises with the provision of a common EU standard. This would ensure cleaner production, full site integration and optimisation of energy efficiency in all industrial sectors. The combination of all these concepts is a prerequisite for a comprehensive assessment of energy and environmental issues within a company [10].

Declaration by Authors

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