



GLOBAL JOURNAL OF RESEARCHES IN ENGINEERING: A
MECHANICAL AND MECHANICS ENGINEERING
Volume 23 Issue 4 Version 1.0 Year 2023
Type: Double Blind Peer Reviewed International Research Journal
Publisher: Global Journals
Online ISSN: 2249-4596 & Print ISSN: 0975-5861

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By Victor V. Shishov & Maxim S. Talyzin

Bauman Moscow State Technical University Moscow

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The paper presents results of analysis existed refrigeration plants used for retail applications.

The ESMA was applied to investigate some operating refrigeration plants equipped with monitoring system and define elements for further optimization. The initial date for analysis were date taking from monitoring system (pressures and temperatures in working points). The description of calculation method was given.

The degree of thermodynamic efficiency was calculated.

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The prospect of the ESMA application to analyse operation and optimization of existing refrigerating systems as well as to design new refrigerating systems consists in that it allows refusing expensive equipment for improvement refrigeration plant efficiency.

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GJRE- A Classification: LCC: TJ163.5



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I. INTRODUCTION

Power consumption of refrigeration plants makes 48% to 60 % from total power consumption of retail store.

Because of ozone depletion and global warming risks, we ought to use new ecological friendly refrigerants, which require improving refrigeration components, processes and cycle architecture to increase energy efficiency.

We need to use methods that could help to improve energy efficiency based on scientific methods.

There are some methods that used for thermodynamic analysis – energy method, exergy method and entropic and statistical method.

Energy method of analysis based on first law of thermodynamic and does not allow getting information about energy losses by system components. Exergy and entropic and statistical method based both on first and second laws of thermodynamic, but usage of

entropic and statistical methods for system analysis used for cold generation is preferable.

In this paper introduced usage of entropic and statistical method of analysis for refrigeration plants operation.

II. ENTROPIC AND STATISTICAL ANALYSIS OF REFRIGERATION PLANTS

a) Entropic and Statistical Method of Analysis

Real processes in refrigeration plants are irreversible and nonequilibrium.

The reason of irreversibility is finite difference of mass streams potentials (temperature difference, pressure difference etc.) due to processes nonequilibrium.

The measure of irreversibility is entropy production. According to second law of thermodynamics:

$$dS \geq \frac{dQ}{T} \quad \text{Eq. (1)}$$

The entropy has property of additivity e.g. total entropy production of the system equal to the sum of entropy change of each sub-system.:

$$\Delta S = \sum_i \Delta S_i \quad \text{Eq. (2)}$$

where $\sum_i \Delta S_i$ - entropy production in sub-systems, i – sub-systems quantity.

According to Gyui-Stodola's theorem, for it is necessary to perform work for entropy production compensation. This work will be transferred to environment as heat. The total work spent for refrigeration plant operation will equal:

$$L_{tot} = L_{min} + \Delta L \quad \text{Eq. (3)}$$

L_{min} – minimum work for ideal refrigeration cycle with total reversible processes.

Distribution of energy expenses for entropy production compensation for one stage refrigeration cycle is shown on Figure 1.

Ideal Carnot cycle (c1-c2-c3-c4) is totally reversible, it means that work, which is spent for cold generation, will be minimum and is expressed by the area c1-c2-c3-c4. Real cycle is expressed by the area 1-2-3-4-5-6. Work, which is necessary for entropy production compensation in compressor, is expressed

Author ^α: Bauman Moscow State Technical University Moscow, 105005, Russian Federation. e-mail: shishov-1941@mail.ru

Author ^σ: International Academy of Refrigeration Moscow, 105005, Russian Federation. e-mail: talyzin_maxim@mail.ru

by area d-e-e3-e2. Area e3-2-3-4-e1 or equal to it in size e-f-e4-e3 expresses work, which is necessary for entropy production compensation in condenser. Work,

which is necessary for entropy production compensation in evaporator, is expressed by area c-d-e2-c2, in throttling devices – by area a-b-c3-e1.

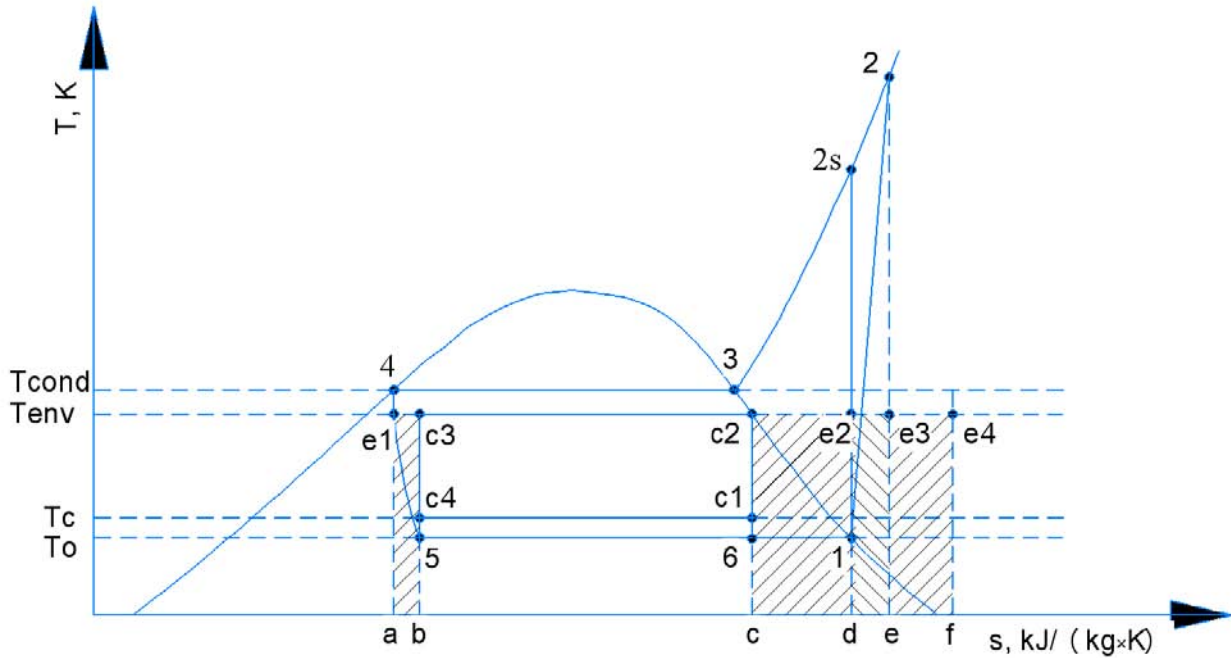


Figure 1: Distribution of Energy Expenses for Entropy Production Compensation by Elements of Single Stage Refrigeration Cycle

Total real work for single stage refrigeration cycle is expressed by the sum of areas.

$$L_{tot} \propto (c1-c2-c3-c4) + (d-e-e3-e2) + (e-f-e4-e3) + (c-d-e2-c2) + (a-b-c3-e1) = (1-2-3-4-5-6) \propto (h_2 - h_1) \quad \text{Eq. (4)}$$

Analysis of the single stage refrigeration cycle analysis will be carried out in the following sequence. Specific mass cooling capacity at evaporation temperature.

$$q_0 = h_1 - h_4 = h_1 - h_5 \quad \text{Eq. (5)}$$

Minimum specific work which is necessary for cold generation.

$$l_{min} = q_0 \times \frac{T_{env} - T_c}{T_c} \quad \text{Eq. (6)}$$

Adiabatic compression work, calculated by using h-lgp diagram:

$$l_s = h_{2s} - h_1 \quad \text{Eq. (7)}$$

Actual specific compression work.

$$l_{comp} = q_{cond} - q_0 = h_2 - h_4 - (h_1 - h_4) = \frac{l_s}{\eta_s} \quad \text{Eq. (8)}$$

Degree of thermodynamic efficiency.

$$\eta_{therm} = \frac{l_{min}}{l_{comp}} \quad \text{Eq. (9)}$$

COP at adiabatic compression.

$$\xi = \frac{q_0}{l_s} \quad \text{Eq. (10)}$$

Actual value of COP.

$$\xi_{act} = \frac{q_0}{l_{comp}} \quad \text{Eq. (11)}$$

Part of compression work, which is necessary for entropy production compensation in condenser, is the sum of parts of compression works for entropy production compensation in gas cooling process Δl_{sh} and condensation process Δl_{cd} :

$$\Delta l_{cond} = \Delta l_{sh} + \Delta l_{cd} \quad \text{Eq. (12)}$$

where

$$\Delta l_{sh} = (h_{2s} - h_3) - T_{env} \times (s_{2s} - s_3) \quad \text{Eq. (13)}$$

$$\Delta I_{cd} = T_{env} \times (h_3 - h_4) \times \left(\frac{1}{T_{env}} - \frac{1}{T_{cond}} \right) \quad \text{Eq. (14)}$$

Part of compression work, which is necessary for entropy production compensation in throttling process.

$$\Delta I_{thr} = T_{env} \times (s_5 - s_4) \quad \text{Eq. (15)}$$

Part of compression work, which is necessary for compensation of entropy production in heat transferring processes from cooling object to refrigerant (evaporation).

$$\Delta I_{e.evap} = (h_1 - h_5) \times T_{env} \times \frac{T_c - T_o}{T_o \times T_c} \quad \text{Eq. (16)}$$

Part of compression work, which is necessary for compensation of entropy production in heat transferring processes from cooling object to refrigerant (total).

$$\Delta I_{evap} = \Delta I_{e.evap} \quad \text{Eq. (17)}$$

Specific adiabatic compression work is a sum of parts of compression work for compensation of entropy production in all processes of refrigeration cycle.

$$I_{s.calc} = I_{min} + \Delta I_{cond} + \Delta I_{thr} + \Delta I_{evap} \quad \text{Eq. (18)}$$

Energetic losses in compressor.

$$\Delta I_{comp} = I_{comp} - I_{s.calc} \quad \text{Eq. (19)}$$

Rated compression work.

$$I_{comp.calc} = I_{s.calc} + \Delta I_{comp} \quad \text{Eq. (20)}$$

By using analysis results, the chart of distribution losses on system elements can be constructed.

b) Refrigeration Plant Analysis in Case of Condensation Pressure Algorithm Change

One of the goals, which is very important to refrigeration equipment end user, is to evaluate results of implementation of new technical solution (usage of more efficiency compressors, heat exchangers, new control algorithms etc).

The goal of refrigeration plant optimization (it is located in Moscow, Russia) was energy consumption reduction. This plant work with R404A as refrigerant and used for 10 medium temperature consumers. Design date is shown in Table 1.

Table 1: Design Date of Refrigeration Plant

| Cycle | Single Stage |
|-------------------------------|--|
| Refrigerant | R404A |
| Evaporation temperature, °C | -10 |
| Condensation temperature, °C | +45 |
| Suction gas superheat, K | 15 |
| Liquid subcooling, K | 0 |
| Condenser type | Air cooled |
| Cooling capacity, kW | 33,8 |
| Quantity of compressors, pcs. | 2 |
| Compressor type | scroll |
| Model | ZB76KCE-TFD - 1 pcs. ZBD76KCE-TFD – 1 pcs. |
| Consumers quantity, pcs. | 10 |
| Thermal processing | Storage |
| Product type | Egg (1 consumer), milk products (6 consumers), sausages (2 consumers), fish (1 consumer) |

There were two measurement periods – 4.5 days with fixed condensing set point (condensation pressure was keeping on the same value) and 4.5 days with floating condensing set point (condensation pressure was changing depending on ambient temperature).

Initial date (average values for considered period) is given in Table 2, changing of operating parameters is shown on Figs. 2 and 3, where System 1

is refrigeration plant before algorithm change, System 2 is refrigeration plant after algorithm change.

Table 2: Initial date for Analysis

| Temperatures, °C | System 1 | System 2 |
|---|----------|----------|
| Evaporation | -9.0 | -9.07 |
| Condensation | +34.79 | +19.33 |
| Ambient | +8.51 | +8.18 |
| Air in cooling volume | +1.95 | +2.1 |
| Adiabatic efficiency was taken from selection software of compressor manufactures | | |
| Adiabatic efficiency | 0.7117 | 0.6661 |

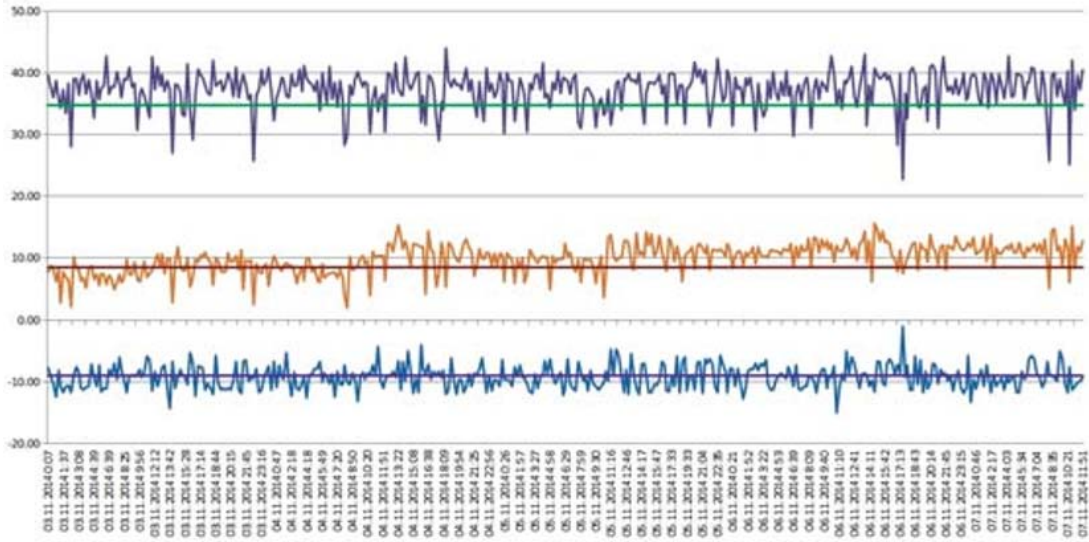


Figure 3: Schedules of Change of Condensation Temperature (Top Schedule), Ambient Temperature (Middle Schedule), Evaporation Temperature (Lower Schedule) and Their Average Values (Horizontal Lines) For System 1

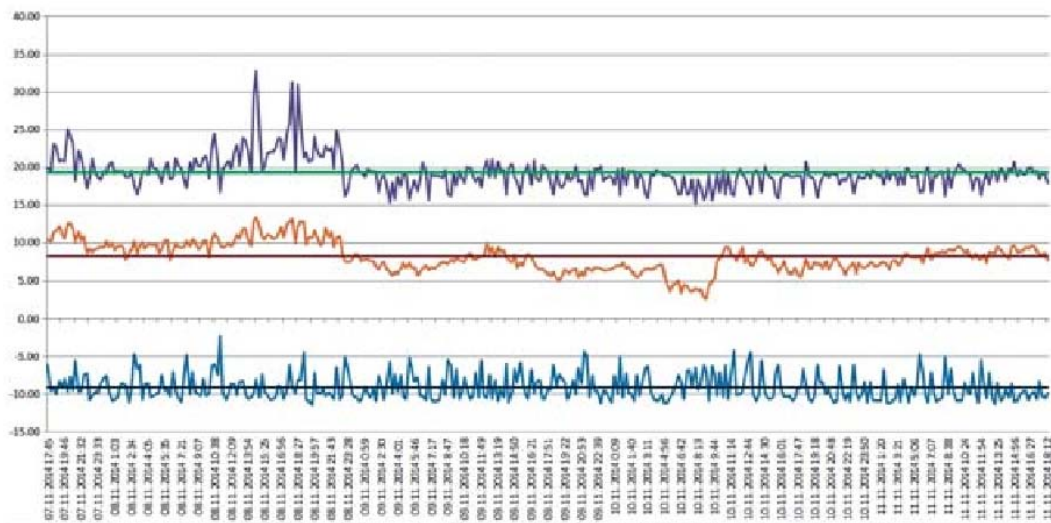


Figure 4: Schedules of Change of Condensation Temperature (Top Schedule), Ambient Temperature (Middle Schedule), Evaporation Temperature (Lower Schedule) and their Average Values (Horizontal Lines) for System 1

Because there was not possibility to measure all necessary values by using regular devices, some values were taken according to statistical information:

- Total superheat on suction line 12 K;
- Calculation of energetic losses in evaporator was done by using average value of cooling air temperature in all cooling room.

Analysis results are given in Table 3 and on Fig. 5.

Table 3: Analysis Results

| - | System 1 | System 2 |
|-----------------------|----------|----------|
| q_o , kJ/kg | 119.71 | 144.52 |
| l_{min} , kJ/kg | 2.88 | 3.19 |
| $l_{s,calc}$, kJ/kg | 27.96 | 19.32 |
| l_{comp} , kJ/kg | 39.24 | 29.00 |
| η_{therm} | 0.0795 | 0.1184 |
| ϵ_{act} | 3.05 | 4.98 |
| Δl_{cond} , % | 34.45 | 23.98 |
| Δl_{thr} , % | 16.47 | 10.29 |
| Δl_{evap} , % | 12.91 | 21.32 |
| Δl_{comp} , % | 28.83 | 33.4 |

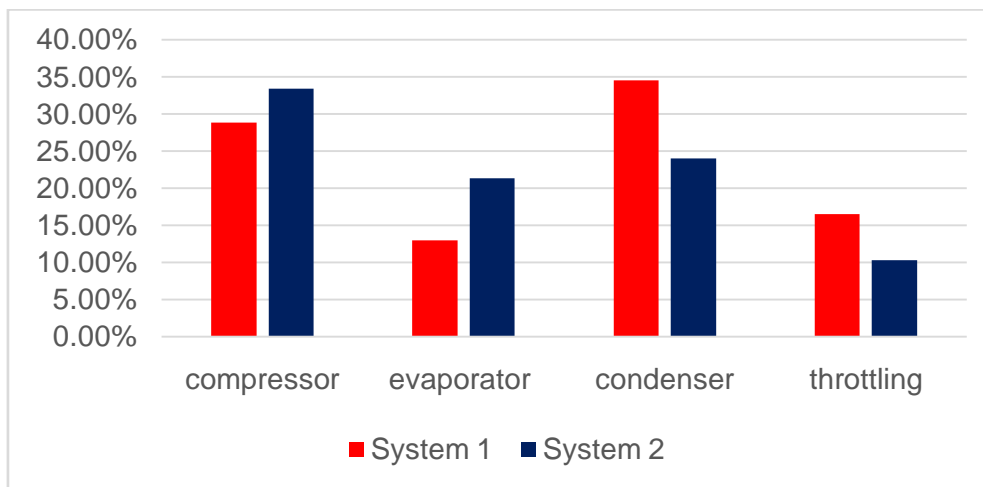


Figure 5: Part of Compression Work (Energetic Losses) which is Necessary for Compensation of Entropy Production in System Elements for Refrigeration Plant before and after Optimization, % from Compression Work

c) Efficiency Analysis at Refrigerant Plant Operation with Different Working Cycles

One more and, perhaps, more relevant task is the analysis of various systems with different working cycles.

Comparison of two systems is given below. System 3 works by using single stage refrigeration cycle, located in Moscow. System 4 works by using refrigeration cycle with economizer, located in Volgsky, Russia. Design date is shown in Table 4.

Table 4: Design Date of Refrigeration Plants

| | System 3 | System 4 |
|------------------------------|--|------------------------|
| Refrigerant | R404A | R404A |
| Refrigeration cycle | Single stage | With economizer |
| Evaporation temperature, °C | -35 | |
| Condensation temperature, °C | +40 | |
| Suction superheat, K | 15 | |
| Liquid subcooling, K | 0 | |
| Condenser type | Air cooled | |
| Cooling capacity, κBТ | 90,2 | 92,0 |
| Compressor type | reciprocating | Scroll with economizer |
| Quantity of compressor, pcs. | 3 | 8 |
| Model | D8DJ-600X with inverter drive – 1 pcs., D8DJ-600X – 2 pcs. | ZF40KVE-TWD – 8 pcs. |
| Quantity of consumers, pcs. | 39 | 26 |
| Thermal processing | Frozen food storage | |

As in the previous case, operation parameters were taken from monitoring system for specified period. Initial average date for analysis is given in Table 5.

Table 5: Initial date for Analysis

| | System 3 | System 4 |
|-------------------------------------|----------|----------|
| Temperatures, °C | | |
| evaporation | -33.46 | -32.58 |
| condensation | +36.75 | +31.35 |
| ambient | +10.58 | +15.98 |
| air in cooling volume | -15.14 | -16.25 |
| Average superheat in evaporators, K | 13.67 | 9.73 |
| Adiabatic efficiency | 0.673 | 0.632 |

Analysis results are given in Table 6 and on Fig. 6.

Table 6: Analysis Results

| | System 3 | System 4 |
|-----------------------|----------|----------|
| q_o , kJ/kg | 101.89 | 167.31 |
| l_{min} , kJ/kg | 10.73 | 22.17 |
| $l_{s,calc}$, kJ/kg | 50.67 | 61.91 |
| l_{comp} , kJ/kg | 75.31 | 97.97 |
| η_{therm} | 0.149 | 0.214 |
| ϵ_{act} | 1.35 | 1.71 |
| Δl_{cond} , % | 19.59 | 14.64 |
| Δl_{thr} , % | 22.58 | 9.41 |
| Δl_{evap} , % | 10.88 | 13.2 |
| Δl_{comp} , % | 32.7 | 36.78 |
| Δl_{eco} , % | - | 3.34 |

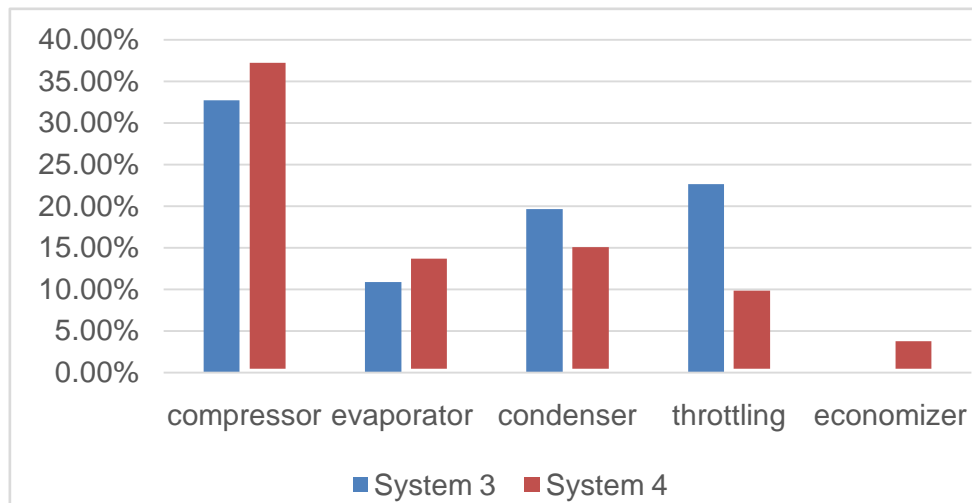


Figure 6: Part of Compression Work (Energetic Losses) which is Necessary for Compensation of Entropy Production in System Elements for Refrigeration Plant before and after Optimization, % from Compression Work.

III. CONCLUSIONS

Application of entropic and statistical method of the analysis during cooling plant operation allows increasing energy efficiency because we can get information about losses in different refrigeration plant components and take measures to increase their operation efficiency. It is possible to use regularly installed sensors.

The most appropriate value for comparison of energy efficiency different cooling plants is the degree of thermodynamic efficiency.

Changing control algorithm of condensation pressure from maintenance of the fixed setting to floating setting allow to increase the degree of thermodynamic efficiency on 32.9% and decrease power consumption on 36.7%. Power consumption was measured by using special sensors.

The degree of thermodynamic efficiency refrigeration plant working with refrigeration cycle with economizer is higher than the degree of thermodynamic

efficiency refrigeration plant working with single stage refrigeration cycle on 30.37%. Data was received during operation of real refrigeration plants.

Nomenclature

| | | | |
|---------------|-----------------------------|----------------|--|
| Q | heat (W) | q_o | specific mass cooling capacity (kJ/kg) |
| s | specific entropy (kJ/kg/K) | l_{min} | minimum specific work which is necessary for cold generation (kJ/kg) |
| h | specific enthalpy (kJ/kg/K) | l | specific compression work (kJ/kg) |
| T | temperature (K) | η_{therm} | the degree of thermodynamic efficiency |
| t | temperature (°C) | Δl | energetic losses (kJ/kg) |
| ε | COP | | |

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