Analyzing the Efficiency of Using an Installed Refrigeration Capacity of Ambient Air Conditioning System

Andrii Radchenko[1(✉)], Serhiy Forduy[2], Serhiy Kantor[3], Veniamin Tkachenko[3], and Felix Tsaran[1]

[1] Admiral Makarov National University of Shipbuilding, 9 Heroes of Ukraine Avenue, Mykolayiv 54025, Ukraine
[2] PepsiCo, Inc., Kyiv, Ukraine
[3] PJSC "Zavod" "Ekvator", Mykolaiv, Ukraine

Abstract. One of the most reasonable reserves to improve the efficiency of ambient air conditioning systems is to enable the operation of refrigeration compressors in close to nominal modes by selecting a rational design refrigeration capacity and its distribution in response to the current thermal load according to the actual variable climatic conditions to provide a closed to the maximum annual refrigeration production and matching current air conditioning duties at the same time. The approach to improve the efficiency of using the installed air conditioning systems refrigeration capacity is based on comparing the excess of potentially possible refrigeration production (based on the installed refrigerating capacity) for a certain period with its value needed for precooling the ambient air to a certain threshold (intermediate) temperature, that provides a relatively stable thermal load within subsequent deep air subcooling to a target temperature. The range of a stable thermal load can be provided with operation of a conventional compressor in a mode closed to a nominal value, whereas precooling the ambient air with significant fluctuations in thermal load requires regulation of the refrigeration capacity by application of a variable speed compressor or by using excessive cooling capacity accumulated at reduced thermal load.

Keywords: Ambient air conditioning · Design refrigeration capacity · Current thermal load

1 Introduction

Energy consumption for the processing of ambient air in air conditioning systems (ACS) depends on the ambient air temperature t_a and relative humidity φ_a, which changes significantly during the day. It is obvious that the production of refrigeration by refrigeration machine (RM) according to air conditioning duties over a certain period of time, for example a year (annual refrigeration production) $\sum(Q_0 \cdot \tau)$, [kWh], where: Q_0 – is the current refrigeration capacity of RM (thermal load on the ACS), [kW]; τ – is the duration of the ACS operation in hours, is chosen as a criterion to estimate the efficiency of

M. Nechyporuk et al. (Eds.): ICTM 2021, LNNS 367, pp. 520–531, 2022.
https://doi.org/10.1007/978-3-030-94259-5_44

using the installed refrigeration capacities of the RM. Since the amount of refrigeration capacity spent on ambient air processing in the ACS depends on the current ambient air parameters and the duration of the ACS operation, the efficiency of using the installed refrigeration capacity of ACS can be estimated by comparing its value spent for air conditioning over a certain period of the operation and potentially possible refrigeration production due to operation of RM on full, nominal, load.

2 Literature Review

Many publications are devoted to improving air processing in ACS by intensification of heat transfer processes in air coolers [1, 2], evaporators [3, 4] and condensers [5], application of various refrigerant circulation contours [6, 7], alternate safe refrigerants [8, 9], waste heat recovery technics, including combined cooling, heating and power generation [10, 11], modelling [12, 13], optimization [14, 15], experimental and monitoring [16, 17] methods to match current cooling demands. Some of principal technical innovations and methodological approaches in waste heat recovery refrigeration [2, 18] might be successfully applied for air conditioning [19], in particular, evaporative cooling [20, 21], two-stage air cooling [22, 23].

Numerous researchers have studied the energy efficiency of the Variable Refrigerant Flow (VRF) systems [24, 25] and proposed some practical recommendations. Most of the studies have been conducted on solutions of efficient operation of the VRF system in buildings and control strategies of the systems [26]. A control algorithm of the supply air temperature as a threshold temperature in the outdoor air processing (OAP) unit to run the VRF-OAP system more efficiently for buildings was developed [27]. The control algorithm was conducted to adjust the refrigerant flow supply to the OAP and the indoor unit appropriately through supporting the supply air temperature. Results [28] show that ACS have a great potential for energy saving and the adjustability of VRF.

The VRF system with heat recuperative ventilation [29] and a dedicated outdoor ACS was introduced [30]. The evaluation of indoor thermal-humidity environments and energy consumption of the VRF system [31] with a heat pump was conducted [32].

The author [33] proposes the method to calculate thermal load of building. The VRF systems operate with high part-load efficiency [34], that results into high daily and seasonal energy efficiency, so as ACS typically spend most of their operating hours within the range of 40% to 80% of maximum cooling capacity [35].

The majority of ACS designing methods issue from the assumption to cover the maximum current cooling needs or with some limitation [34–36] or annual maximum cooling consumption [37–40]. Such approach inevitable leads to considerable overestimation of design cooling capacities and systems oversizing. So, the problem to determine a rational design cooling capacity without ACS oversizing needs further solution.

The aim of the study is to develop an approach to analyzing the efficiency of using the installed refrigeration capacity of ACS and method to determine a rational design (installed) refrigeration capacity value and its distribution according to the current climatic conditions to avoid oversizing a refrigeration machine and its enlarged cost.

3 Methodology

The efficiency of ACS and their RM performance depends on their loading and a duration of their yearly operation. Therefore, the annual refrigeration production in response to air conditioning duties is considered as a primary criterion for the choice of a rational design overall thermal load of ACS. For this the current refrigeration capacities, generated by RM at any time period in response to the air conditioning duties for ambient air conditioning down to the target leaving air temperature, have been summarized over the year to determine the rational design overall refrigeration capacity of ACS.

In order to conduct this procedure the authors develop a method of rational designing based on the yearly loading characteristic curve of annual summarized refrigeration production dependence on the design specific refrigeration capacity of the RM to choose its value, that provides closed to maximum annual production of refrigeration.

The specific annual production of refrigeration:

$$\sum(q_0 \cdot \tau) = \sum(\xi \cdot c_{ma} \cdot (t_{amb} - t_{a2}) \cdot \tau) \tag{1}$$

where: $\sum(q_0 \cdot \tau)$ – specific annual production of refrigeration [kg·h/kW]; ξ – specific heat ratio; t_a – ambient air temperature [°C]; t_{a2} – air temperature at the air cooler outlet [°C]; c_{ma} –specific heat of moist air [kJ/(kg·K)]; τ – time interval [h].

The specific refrigeration capacity is calculated as:

$$q_0 = \xi \cdot c_{ma} \cdot (t_a - t_{a2}) \tag{2}$$

A rational specific refrigeration capacity $q_{0.rat}$ is determined to exclude unproductive expenses of refrigeration capacity q_0 caused by oversizing RM without obtaining a noticeable effect in increasing the annual production of refrigeration $\sum(q_0 \cdot \tau)$, that was limited by the value of $(0.95\ldots0.97)\sum(q_0 \cdot \tau)$ to choose corresponding rational specific refrigeration capacity $q_{0.rat}$.

Proceeding from a different behavior of current thermal loads, the ambient air treatment in the ACS is considered as a two-stage processing and includes a range of hermal load fluctuation as the first (high-temperature) stage and a range of comparatively stable thermal load as the second (low-temperature) stage. The threshold air temperature is determined to provide a rational distribution of design overall cooling capacity of ACS between two stages with different thermal load behaviors.

Taking into account a relatively stable behavior of the specific thermal (cooling) load on the air cooler (AC) of the ACS within the range $q_{0.10–15} = q_{0.10} - q_{0.15}$ (or $q_{0.10–17} = q_{0.10} - q_{0.17}$) when the air is cooled from the temperature $t_{a2} = 15$ (or 17)°C to $t_{a2} = 10$ °C compared to the ambient air cooling from $t_a$to $t_{a2} = 15$ (or 17)°C the first thermal load range is taken as a basic design (installed) for deep cooling of air from $t_{a2} = 15$ (or 17)°C to $t_{a2} = 10$ °C. Accordingly, a design refrigeration capacity for precooling the ambient air from the current temperature $t_a$to $t_{a2} = 15$ (or 17)°C, as booster component, is determined by the residual principle as the difference between the design specific refrigeration capacity $q_{0.10rat}$ for the entire process of cooling the ambient air from the current temperature $t_a$to $t_{a2} = 10$ °C and its stable component $q_{010–15}$: $q_{0.b10–15} = q_{0.10rat} - q_{010–15}$ (or $q_{0.b10–17} = q_{0.10rat} - q_{010–17}$).

Since the fluctuations of the current refrigeration capacity spent for cooling the ambient air from t_a to $t_{a2} = 10$ °C are caused mainly by its booster part $q_{0.15}$ (or $q_{0.17}$), which corresponds to precooling the ambient air from t_a to $t_{a2} = 15$ (or 17)°C, at elevated current thermal loads $q_{0.15}$ there is some deficit of the installed booster component $q_{0.b10\text{–}15}$ ($q_{0.b10\text{–}17}$) of refrigeration capacity $q_{0.b10\text{–}15}$ ($q_{0.b10\text{–}17}$), calculated by the residual principle, whereas at reduced current thermal loads $q_{0.15}$ (or $q_{0.17}$), on the contrary, its excess $q_{0.bex10\text{–}15} = q_{0.10rat} - q_{0.b10\text{–}15} = q_{0.10rat} - q_{0.15}$ compared to the current specific thermal loads $q_{0.15}$ and in according $q_{0.bex10\text{–}17}$.

How efficiently the installed booster component of the refrigeration capacity of the ACS is spent on precooling the ambient air from its current temperature t_a to $t_{a2} = 15$ °C (or 17 °C) with the change of current thermal loads according to the actual climatic conditions can be judged by comparing the total monthly summarized specific refrigeration consumption to cover the current thermal loads $\sum(q_{0.15}\cdot\tau)$ (or $\sum(q_{0.17}\cdot\tau)$) for cooling the ambient air from the current t_a to $t_{a2} = 15$ °C (or 17 °C) with potentially possible refrigeration production by a booster stage with design specific refrigeration capacity calculated on the residual basis, $\sum(q_{0.b10\text{–}15}\cdot\tau)$ (or $\sum(q_{0.b10\text{–}17}\cdot\tau)$), on the other hand.

4 Results

In order to generalize the results and simplify calculations for any total refrigeration capacities Q_0, it is convenient to present the refrigeration capacity of the RM ACS not in absolute Q_0, but in relative (specific) values per unit air mass flow rate ($G_a = 1$ kg/s) – in the form of specific refrigeration (cooling) capacity, $q_0 = Q_0 / G_a$, kW/(kg/s), or kJ/kg, where Q_0 is the total refrigeration capacity when cooling the air with the flow rate G_a: $Q_0 = (c_a\cdot\xi\cdot\Delta t_a)G_a$, where $\Delta t_a = t_{amb} - t_{a2}$ – decrease in air temperature.

To justify the approach to the analysis of the efficiency of using the installed (design) refrigeration capacities of ACS chillers taking into account the change in thermal loads according to the current climatic conditions, the current values of specific refrigeration capacity q_0 of RM ACS when cooling the ambient air from the current temperature t_a to $t_{a2} = 10$, 15 and 17 °C, respectively $q_{0.10}$, $q_{0.15}$ and $q_{0.17}$ for July of 2017 year, Nikolaev region, Ukraine (Fig. 1) have been considered.

As can be seen, when the ambient air is cooled from its current temperatures t_a to t_{a2}, the thermal load fluctuations of $q_{0.10}$, $q_{0.15}$ and $q_{0.17}$ are very significant. The almost equidistant trend lines of the specific thermal load $q_{0.10}$, $q_{0.15}$ and $q_{0.17}$ indicate that these fluctuations are due primarily to changes in the specific thermal load $q_{0.15}$ and $q_{0.17}$ for precooling the ambient air to the temperatures $t_{a2} = 15$, 17 °C, within which there is practically damping of the fluctuations of the current thermal load.

At the same time, with the further cooling of the air from the intermediate temperature $t_{a2} = 15$ °C to $t_{a2} = 10$ °C the fluctuations of the specific thermal loads on the ACS $q_{0.10\text{–}15} = q_{0.10} - q_{0.15}$ are relatively small, without taking into account 3–5 short-term bursts-drops, caused by a decrease in the current values of the ambient air temperature below 15 °C (Fig. 2, *a*).

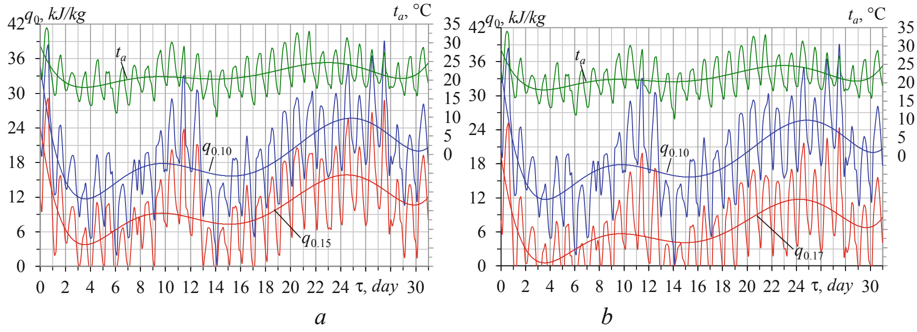

Fig. 1. Current values of ambient air temperature t_a, specific refrigeration capacity $q_{0.10}$, needed for cooling ambient air from t_a to $t_{a2} = 10$ °C, specific refrigeration capacity $q_{0.15}$ and $q_{0.17}$, needed for cooling ambient air from t_a to various intermediate temperatures $t_{a2} = 15$ °C and $t_{a2} = 17$ °C: *a* – $q_{0.15}$ for $t_{a2} = 15$ °C; *b* – $q_{0.17}$ for $t_{a2} = 17$ °C

Obviously, the higher the value of the precooling temperature of the ambient air (intermediate temperature t_{a2}), i. e., the narrower the range of fluctuations of thermal (cooling) loads, the longer the operating life of the refrigeration machine for booster precooling ambient air during the year at current thermal loads.

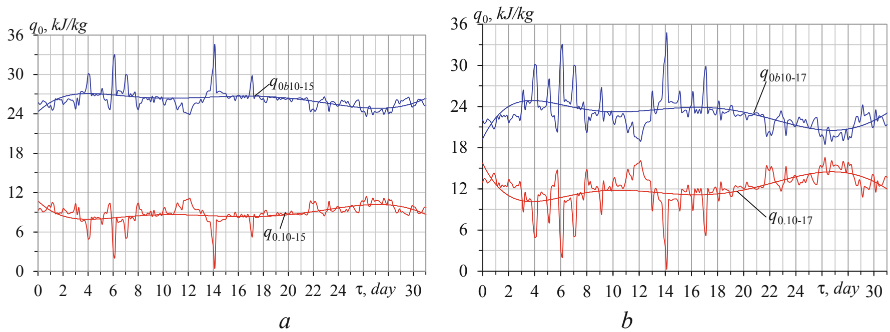

Fig. 2. Current values of booster specific refrigeration capacity $q_{0.b10–15} = q_{0.10rat} - q_{0.10–15}$ with basic specific refrigeration capacity $q_{0.10–15} = q_{0.10} - q_{0.15}$, needed for subcooling air from 15 °C to $t_{a2} = 10$ °C (*a*) and booster specific refrigeration capacity $q_{0.b10–17} = q_{0.10rat} - q_{0.10–17}$ with basic specific refrigeration capacity $q_{0.10–17} = q_{0.10} - q_{0.17}$, needed for subcooling air from 17 °C to $t_{a2} = 10$ °C (*b*) with rational design overall (for two-stage ambient air cooling) specific refrigeration capacity $q_{0.10rat} = 35$ kJ/kg

In order to determine the upper threshold temperature for precooling ambient air, the calculations of the processes of cooling the ambient air from current t_a to a higher intermediate temperature $t_{a2} = 17$ °C were made and the corresponding specific refrigeration capacity for the subsequent subcooling the air $q_{0.10–17} = q_{0.10} - q_{0.17}$ from $t_{a2} = 17$ °C to $t_{a2} = 10$ °C was determined (Fig. 2, *b*).

As can be seen, the specific refrigeration capacity $q_{0.10–17} = q_{0.10} – q_{0.17}$ for subcooling the air from the intermediate temperature t_{a2} = 17 °C to t_{a2} = 10 °C becomes very unstable compared to the lower intermediate temperature t_{a2} = 15 °C (Fig. 2, *a*). This is caused by an earlier (at t_{a2} = 17 °C) fall to zero of the specific refrigeration capacity $q_{0.17}$ for precooling the ambient air due to the narrowing of the temperature range $\Delta t_{17} = t_a$ – 17 °C of cooling the ambient air at an elevated intermediate temperature t_{a2} = 17 °C (compared to $\Delta t_{15} = t_a$ – 15 °C at t_{a2} = 15 °C), that results in replacing the fluctuations of specific refrigeration capacity, previously damped when cooling the ambient air to the temperature t_{a2} = 15 °C, with corresponding increase in specific refrigeration capacity $q_{0.10–17} = q_{0.10} – q_{0.17}$ for further subcooling air from the temperature t_{a2} = 17 °C to t_{a2} = 10 °C.

Taking into account a relatively stable behavior of the specific thermal load on the air cooler of the ACS within the range $q_{0.10–15} = q_{0.10} – q_{0.15}$ when the air is subcooled from t_{a2} = 15 °C to t_{a2} = 10 °C (Fig. 2,*a*) compared to large fluctuations of refrigeration capacity $q_{0.15}$ within the range of ambient air precooling from t_a to t_{a2} = 15 °C (Fig. 1,*a*), the first thermal load $q_{0.10–15}$ is taken as a design stable component of refrigeration capacity for deep cooling of air from t_{a2} = 15 °C to t_{a2} = 10 °C.

Accordingly, a design refrigeration capacity $q_{0.b10–15}$ for booster precooling the ambient air from the current temperatures t_a to t_{a2} = 15 °C is determined by a residual principle as the difference between a design specific refrigeration capacity $q_{0.10rat}$ for the entire process of cooling the ambient air from the current temperatures t_a to t_{a2} = 10 °C and its stable component $q_{010–15}$: $q_{0.b10–15} = q_{0.10rat} – q_{010–15}$.

With this a rational design refrigeration capacity $q_{0.10rat}$ = 35 kJ/kg for the entire process of cooling the ambient air from the current temperatures t_a to 10 °C is assumed to provide closed to a maximum annual refrigeration production $\sum(q_0 \cdot \tau)$ (Fig. 3).

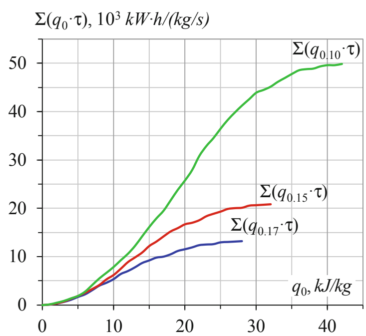

Fig. 3. Specific annual refrigeration production $\sum(q_0 \cdot \tau)$ required for cooling the ambient air to the temperatures t_{a2} = 10, 15 and 17 °C against a design specific refrigeration capacity q_0

As can be seen, for the considered climatic conditions when the air is cooled to the temperature of t_{a2} = 10 °C in the ACS with installed (design) specific (at G_a = 1 kg/s) refrigeration capacity of RM $q_{0.10rat}$ = 35 kJ/kg, which provides close to the maximum annual refrigeration production $\sum(q_0 \cdot \tau) \approx 48 \cdot 10^3$ kWh/(kg/s), while maintaining its increment with a noticeable high rate. Similarly, for cooling ambient air to the temperature of t_{a2} = 15 °C the rational value of specific refrigeration capacity of RM $q_{0.15rat}$ = 25 kJ/kg, and to t_{a2} = 17 °C – the value of $q_{0.17rat}$ = 22·kJ/kg.

Current values of booster specific refrigeration capacity $q_{0.b10–15} = q_{0.10rat} - q_{0.10–15}$ and $q_{0.b10–17} = q_{0.10rat} - q_{0.10–17}$ and booster specific refrigeration capacity excess $q_{0.bex10–15} = q_{0.b10–15} - q_{0.15}$ and $q_{0.bex10–17} = q_{0.b10–17} - q_{0.17}$ with subcooling air from t_{a2} = 15 °C and t_{a2} = 17 °C accordingly to t_{a2} = 10 °C with rational design overall (for two-stage ambient air cooling) specific refrigeration capacity $q_{0.10rat}$ = 35 kJ/kg (according to Fig. 3) are presented in Fig. 4.

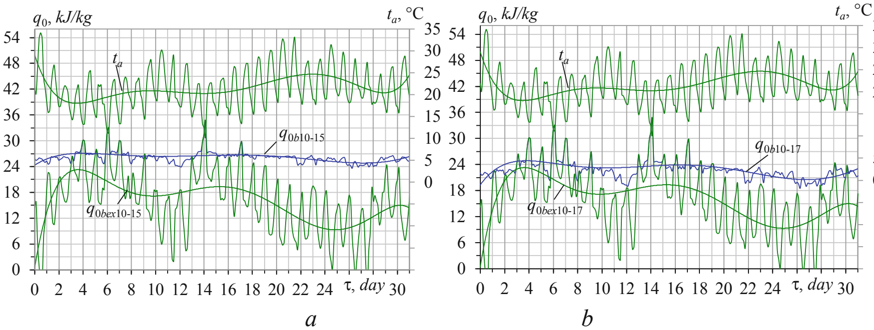

Fig. 4. Current values of ambient air temperature t_a, booster specific refrigeration capacity $q_{0.b10–15} = q_{0.10rat} - q_{0.10–15}$ and booster specific refrigeration capacity excess $q_{0.bex10–15} = q_{0.b10–15} - q_{0.15}$ with subcooling air from t_{a2} = 15 °C to t_{a2} = 10 °C (*a*) and booster specific refrigeration capacity $q_{0.b10–17} = q_{0.10rat} - q_{0.10–17}$ and booster specific refrigeration capacity excess $q_{0.bex10–17} = q_{0.b10–17} - q_{0.17}$ with subcooling air from t_{a2} = 17 °C to t_{a2} = 10 °C (*b*) with rational design overall specific refrigeration capacity $q_{0.10rat}$ = 35 kJ/kg according to Fig. 3

As Fig. 4 shows, the excess of the current values of the specific refrigeration capacity $q_{0.b10–15} = 35 - q_{010–15}$ for precooling the ambient air from its current temperature t_a to t_{a2} = 15 °C above its design value drops from the current specific refrigeration capacity $q_{0.15}$ to zero, and the fact that the excess of the current values of the specific refrigeration capacity $q_{0.b10–15}$ in some days (4, 6, 7, 14 and 17.07.2017) exceeds the design value, is explained by the reduction of refrigeration capacity $q_{0.10–15}$ for deep cooling the air while reducing the current ambient air temperature t_a below 15 °C.

Potentially possible monthly summarised refrigeration production for precooling the ambient air to the intermediate temperature t_{a2} = 15 °C in a booster stage of two-stage air cooler available in accordance with its design specific refrigeration calculated on a residual basis, $\sum(q_{0.b10–15}\cdot\tau)$, its potential excess $\sum(q_{0.bexc10–15}\cdot\tau) = \sum(q_{0.b10–15}\cdot\tau) - \sum(q_{0.15}\cdot\tau)$ over the current values of the specific refrigeration spent for precooling the ambient air to t_{a2} = 15 °C in a booster stage (refrigeration consumption) monthly summarised $\sum(q_{0.15}\cdot\tau)$ and monthly summarised refrigeration consumption $\sum(q_{0.10–15}\cdot\tau)$ for deep subcooling the air from t_{a2} = 15 °C to t_{a2} = 10 °C is plotted in Fig. 5, *a*, as well as their values for precooling the ambient air to an increased intermediate temperature $t_{a2} = 17$ °C are presented in Fig. 5, *b*.

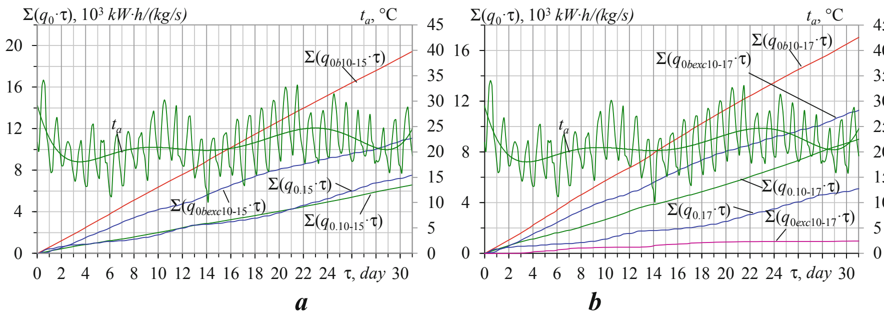

Fig. 5. Current values of total summarized per month specific refrigeration consumption $\sum(q_{0.15}{\cdot}\tau)$ for precooling the ambient air to 15 °C, refrigeration spent for subcooling air from 15 °C to 10 °C, $\sum(q_{0.10-17}{\cdot}\tau)$, potentially possible refrigeration production for precooling ambient air $\sum(q_{0.b10-15}{\cdot}\tau)$ according to a design refrigeration capacity of booster stage, potentially possible excess of refrigeration production for precooling ambient air in a booster stage $\sum(q_{0.bexc10-15}{\cdot}\tau) = \sum(q_{0.b10-15}{\cdot}\tau) - \sum(q_{0.15}{\cdot}\tau)$ (*a*) and total monthly summarized specific refrigeration consumption $\sum(q_{0.17}{\cdot}\tau)$ for precooling the ambient air to the intermediate temperature 17 °C, refrigeration spent for subcooling air from 17 °C to 10 °C, $\sum(q_{0.10-17}{\cdot}\tau)$, potentially possible refrigeration production for precooling ambient air $\sum(q_{0.b10-17}{\cdot}\tau)$ according to a design refrigeration capacity of booster stage, potentially possible excess of refrigeration production for precooling ambient air in a booster stage $\sum(q_{0.bexc10-17}{\cdot}\tau) = \sum(q_{0.b10-17}{\cdot}\tau) - \sum(q_{0.17}{\cdot}\tau)$ and $\sum(q_{0exc10-17}{\cdot}\tau) = \sum(q_{0.10\mathrm{rat}}{\cdot}\tau) - \sum(q_{0.17\mathrm{rat}}{\cdot}\tau) - \sum(q_{0.10-17}{\cdot}\tau)$

An elevated intermediate temperature $t_{a2} = 17$ °C of ambient air precooling does not provide stabilization of thermal loading of subsequent subcooling the air from increased intermediate temperature $t_{a2} = 17$ °C down to the target temperature $t_{a2} = 10$ °C that leads to arising an excess of refrigeration production $\sum(q_{0.bexc10-17}{\cdot}\tau) = \sum(q_{0.b10-17}{\cdot}\tau) - \sum(q_{0.17}{\cdot}\tau)$ compared to its consumption $\sum(q_{0.10-17}{\cdot}\tau)$ $q_{0.10-17} = q_{0.10-17} = q_{0.10} - q_{0.17}$ when cooling the air from $t_{a2} = 17$ °C to $t_{a2} = 10$ °C (Fig. 5, *b*).

As can be seen, the specific refrigeration capacity $q_{0.10-17} = q_{0.10} - q_{0.17}$ for subcooling the air from the increased intermediate temperature $t_{a2} = 17$ °C to the target $t_{a2} = 10$ °C becomes very unstable compared to the lower intermediate temperature $t_{a2} = 15$ °C (Fig. 2, *a*). This is caused by an earlier (at $t_{a2} = 17$ °C) fall to zero of the refrigeration capacity $q_{0.17}$ for precooling the ambient air due to the narrowing of the temperature range $\Delta t_{17} = t_a - 17$ °C of cooling the ambient air at an elevated intermediate temperature $t_{a2} = 17$ °C (compared to $\Delta t_{15} = t_a - 15$ °C at $t_{a2} = 15$ °C), that results in replacing the fluctuations of specific refrigeration capacity, previously damped when cooling the ambient air to the temperature $t_{a2} = 15$ °C, with corresponding increase in specific refrigeration capacity $q_{0.10-17} = q_{0.10} - q_{0.17}$ for further subcooling air from the temperature $t_{a2} = 17$ °C to $t_{a2} = 10$ °C.

Comparison of refrigeration capacity of air subcooling $q_{0.10-15} = q_{0.10} - q_{0.15}$ and $q_{0.10-17} = q_{0.10} - q_{0.17}$ shows that the stabilization of thermal load occurs as a result of precooling the ambient air to the lower intermediate temperature $t_{a2} = 15$ °C compared to $t_{a2} = 17$ °C. Therefore, the intermediate temperature $t_{a2} = 15$ °C is assumed as a threshold value and specific refrigeration capacity $q_{0.10–15} = q_{0.10} - q_{0.15}$ is taken as the base, which ensures the effective operation of the RM in the mode close to the nominal.

As can be seen, the total consumption of the specific refrigeration generation per month $\sum(q_{0.15}\cdot\tau)$ for cooling the ambient air to $t_{a2} = 15$ °C is $\sum(q_{0.15}\cdot\tau) \approx$ 7 kW·h/(kg/s), which is less than the excess of the potential refrigeration production for a precooling of the ambient air (to $t_{a2} = 15$ °C) $\sum(q_{0.bexc10–15}\cdot\tau) \approx$ 11 kW·h/(kg/s) and is less 40% of the potential refrigeration output of booster stage $\sum(q_{0.b10–15}\cdot\tau) \approx$ 19 kW·h/(kg/s).

In the first approximation, we can assume that the total per month consumption of the specific refrigeration is $\sum(q_{0.15}\cdot\tau) \approx$ 7 kW·h/(kg/s), which is 40% of the potentially possible refrigeration production of booster stage $\sum(q_{0b10–15}\cdot\tau) \approx$ 19 kW·h/(kg/s), corresponds to the required refrigeration capacity range from 100% to 40% of nominal. Based on this hypothesis, the use of refrigeration compressors with a frequency regulation to 40% could be considered as a way of energy-efficient operation for precooling ambient air to $t_{a2} = 15$ °C.

At the same time, the total per month consumption of specific refrigeration $\sum(q_{0.17}\cdot\tau)$ for cooling the ambient air to $t_{a2} = 17$ °C is about $\sum(q_{0.17}\cdot\tau) \approx$ 5 kW·h/(kg/s), which is much less than the excess of the potential refrigeration production according to a design refrigeration capacity of a booster stage for precooling the ambient air (to $t_{a2} = 17$ °C) $\sum(q_{0bexc10–17}\cdot\tau) \approx$ 11,5 kW·h/(kg/s) over the actual refrigeration needed for precooling the ambient air to $t_{a2} = 17$ °C.

Insufficient damping of the drop in the current thermal load within precooling the ambient air to $t_{a2} = 17$ °C (compared to $t_{a2} = 15$ °C) leads to arising the excess of design refrigeration production $\sum(q_{0exc10–17}\cdot\tau) \approx$ 1 kW·h/(kg/s), though insignificant in comparison with its consumption $\sum(q_{010–17}\cdot\tau) \approx$ 9 kW·h/(kg/s) within further subcooling the air from the temperature $t_{a2} = 17$ °C to $t_{a2} = 10$ °C. Obviously, the use of compressors with frequency regulation of refrigeration capacity can be effective in the range of thermal loads $q_{010–17}$ for subcooling the air to $t_{a2} = 10$ °C, whereas the damping of booster refrigeration capacity $q_{0b10–17}$ for precooling the ambient air to $t_{a2} = 17$ °C their application is possible only in conjunction with the accumulation of excessive refrigeration.

Thus, the proposed approach to analyzing the efficiency of using the installed refrigeration capacity of the ACS can be applied both to choose its rational distribution depending on the character of the change in actual thermal load, and to determine the ranges of thermal loads for the effective application of different methods of managing the ambient air processing in the ACS.

5 Conclusions

A novel method to determine a rational design refrigeration capacity to provide a closed to maximum annual refrigeration production according to conditioning duties at reduced by about 20% design refrigeration capacity compared with traditional designing assumption to cover the maximum current cooling needs has been developed.

Proceeding from a different behavior of current thermal loads, the ambient air processing in ACS is proposed to consider as a two-stage process that includes a range of thermal load fluctuation as the first (high-temperature) stage and a range of comparatively stable thermal load as the second (low-temperature) stage.

The method of rational design refrigeration capacity and its distribution is quite useful for a comfort air conditioning, as well as for cooling air at the inlet of combustion engines to enhance their power output (electricity production) in trigeneration systems. It provides the effective application of energy saving methods, in particular, to cover increased thermal loads by accumulated excessive refrigeration and by application of compressors with variable refrigeration capacity to cover changeable loads.

References

1. Marques, R.P., Hacon, D., Tessarollo, A., Parise, J.A.R.: Thermodynamic analysis of trigeneration systems taking into account refrigeration, heating and electricity load demands. Energy Build. **42**, 2323–2330 (2010)
2. Eidan, A.A., Alwan, K.J.: Enhancement of the performance characteristics for air-conditioning system by using direct evaporative cooling in hot climates. Energy Proc. **142**, 3998–4003 (2017)
3. Kornienko, V., Radchenko, M., Radchenko, R., Konovalov, D., Andreev, A., Pyrysunko, M.: Improving the efficiency of heat recovery circuits of cogeneration plants with combustion of water-fuel emulsions. Therm. Sci. **25**(1B), 791–800 (2021)
4. Alahmer, A., Alsaqoor, S.: Simulation and optimization of multi-split variable refrigerant flow systems. Ain Shams Eng. J. **9**(4), 1705–1715 (2017)
5. Radchenko, A., Trushliakov, E., Kosowski, K., Mikielewicz, D., Radchenko, M.: Innovative turbine intake air cooling systems and their rational designing. Energies **13**(23), 6201 (2020). https://doi.org/10.3390/en13236201
6. Butrymowicz, D., Gagan, J., Śmierciew, K., Łukaszuk, M., Dudar, A., Pawluczuk, A., Łapiński, A., Kuryłowicz, A.: Investigations of prototype ejection refrigeration system driven by low grade heat. In: Proceedings, HTRSE-2018, E3S Web of Conferences, vol. 70, 03002 (2018). https://doi.org/10.1051/e3sconf/20187003002
7. Konovalov, D., Kobalava, H., Radchenko, M., Scurtu, I.C., Radchenko, R.: Determination of hydraulic resistance of the aerothermopressor for gas turbine cyclic air cooling. In: TE-RE-RD 2020, E3S Web of Conferences, 18074343
8. Kuznetsov, V., Dymo, B., Kuznetsova, S., Bondarenko, M., Voloshyn, A.: Improvement of the cargo fleet vessels power plants ecological indexes by development of the exhaust gas systems. Pol. Maritime Res. **28**(1), 97–104 (2021)
9. Radchenko, R., Pyrysunko, M., Radchenko, A., Andreev, A., Kornienko, V.: Ship engine intake air cooling by ejector chiller using recirculation gas. In: Tonkonogyi, V. et al. (eds.) Advanced Manufacturing Processes II. InterPartner-2020. Lecture Notes in Mechanical Engineering, pp. 734–743. Springer, Cham (2021). https://doi.org/10.1007/978-3-030-68014-5
10. Volintiru, O.N., Scurtu, I.C., Stefănescu1, T.M.: Modeling and optimization of HVAC system for special ships. J. Phys. Conf. Ser. 1122, art no 012004 (2018)
11. Trushliakov, E., Radchenko, A., Radchenko, M., Kantor, S., Zielikov, O.: The efficiency of refrigeration capacity regulation in the ambient air conditioning systems. In: Ivanov V., et al. (eds.) Advances in Design, Simulation and Manufacturing III (DSMIE 2020). Lecture Notes in Mechanical Engineering, pp. 343–353. Springer, Cham (2020). https://doi.org/10.1007/978-3-030-50491-5

12. Cherednichenko, O., Serbin, S., Dzida, M.: Investigation of the combustion processes in the gas turbine module of an fpso operating on associated gas conversion products. Pol. Maritime Res. **26**(4), 149–156 (2020)
13. Radchenko, M., Radchenko, R., Tkachenko, V., Kantor, S., Smolyanoy, E.: Increasing the operation efficiency of railway air conditioning system on the base of its simulation along the route line. In: Nechyporuk, M., et al. (eds.) Integrated Computer Technologies in Mechanical Engineering (ICTM 2019). Advances in Intelligent Systems and Computing (2020), vol. 1113, pp. 461–467. Springer, Cham (2020). https://doi.org/10.1007/978-3-030-37618-5
14. Bohdal, L., Kukiełka, L., Legutko, S., Patyk, R., Radchenko, A.M.: Modeling and experimental analysis of shear-slitting of AA6111-T4 aluminum alloy sheet. Materials **13**(14), 3175 (2020)
15. Radchenko, R., Pyrysunko, M., Kornienko, V., Scurtu, I.-C., Patyk, R.: Improving the ecological and energy efficiency of internal combustion engines by ejector chiller using recirculation gas heat. In: Nechyporuk, M., et al. (eds.) Integrated Computer Technologies in Mechanical Engineering. Advances in Intelligent Systems and Computing, vol. 188, pp. 531–541. Springer, Cham (2021). https://doi.org/10.1007/978-3-030-37618-5
16. Shukla, A.K., Singh, O.: Thermodynamic investigation of parameters affecting the execution of steam injected cooled gas turbine based combined cycle power plant with vapor absorption inlet air cooling. Appl. Therm. Eng. **122**, 380–388 (2017)
17. Sur, A., Das, R.K., Sah, R.P.: Influence of initial bed temperature on bed performance of an adsorption refrigeration system. Therm. Sci. **22**(6A), 2583–2595 (2018)
18. Trushliakov, E., Radchenko, A., Forduy, S., Zubarev, A., Hrych, A.: Increasing the operation efficiency of air conditioning system for integrated power plant on the base of its monitoring. In: Nechyporuk, M., et al. (eds.) Integrated Computer Technologies in Mechanical Engineering (ICTM 2019). Advances in Intelligent Systems and Computing, vol. 1113, pp. 351–360. Springer, Cham (2020). https://doi.org/10.1007/978-3-030-37618-5
19. Radchenko, M., Mikielewicz, D., Tkachenko, V., Klugmann, M., Andreev, A.: Enhancement of the operation efficiency of the transport air conditioning system. In: Ivanov V., et al. (eds.) Advances in Design, Simulation and Manufacturing III (DSMIE 2020). Lecture Notes in Mechanical Engineering, pp. 332–342. Springer, Cham (2020). https://doi.org/10.1007/978-3-030-50491-5
20. Im, P., Mini, M., Munk, J. D., Lee, J.: Cooling season full and part load performance evaluation of variable refrigerant flow (VRF) system using an occupancy simulated research building. In: International Refrigeration and Air Conditioning Conference. Paper 1788 (2016). http://docs.lib.purdue.edu/iracc/1788
21. Konovalov, D., Kobalava, H., Radchenko, M., Scurtu, I.C., Radchenko, R.: Determination of hydraulic resistance of the aerothermopressor for gas turbine cyclic air cooling. In: TE-RE-RD 2020, E3S Web of Conferences 180, 01012, p. 14 (2020)
22. Khatri, R., Joshi, A.: Energy performance comparison of inverter based variable refrigerant flow unitary AC with constant volume unitary AC. Energy Procedia **109**, 18–26 (2017)
23. Konovalov, D., Kobalava, H., Maksymov, V., Radchenko, R., Avdeev, M.: Experimental research of the excessive water injection effect on resistances in the flow part of a low-flow aerothermopressor. In: Ivanov V., et al. (eds.) Advances in Design, Simulation and Manufacturing III (DSMIE 2020). Lecture Notes in Mechanical Engineering, pp. 292–301. Springer, Cham (2020). https://doi.org/10.1007/978-3-030-50491-5
24. Zhang, L., Wang, Y., Meng, X.: Qualitative analysis of the cooling load in the typical room under continuous and intermittent runnings of air-conditioning. Procedia Eng. **205**, 405–409 (2017)
25. Lee, J.H., Yoon, H.J., Im, P., Song, Y.–H.: Verification of energy reduction effect through control optimization of supply air temperature in VRF-OAP system. Energies 11(1) (2018)

26. Kornienko, V., Radchenko, R., Bohdal, Ł, Kukiełka, L., Legutko, S.: Investigation of condensing heating surfaces with reduced corrosion of boilers with water-fuel emulsion combustion. In: Nechyporuk, M., et al. (eds.) Integrated Computer Technologies in Mechanical Engineering. Advances in Intelligent Systems and Computing, vol. 188, pp. 300–309. Springer, Cham (2021). https://doi.org/10.1007/978-3-030-37618-5
27. Liu, C., Zhao, T., Zhang, J.: Operational electricity consumption analyze of VRF air conditioning system and centralized air conditioning system based on building energy monitoring and management system. Procedia Eng. **121**, 1856–1863 (2015)
28. Fengxia, H., Zhongbin, Z., Hu, H., Zemin, C.: Experimental study on the all-fresh-air handling unit with exhaust air energy recovery. Energy Procedia **152**, 431–437 (2018)
29. Park, D.Y., Yun, G., Kim, K.S.: Experimental evaluation and simulation of a variable refrigerant-flow (VRF) air-conditioning system with outdoor air processing unit. Energy Build. **146**, 122–140 (2017)
30. Kornienko, V., Radchenko, R., Mikielewicz, D., Pyrysunko, M., Andreev, A.: Improvement of characteristics of water-fuel rotary cup atomizer in a boiler. In: Tonkonogyi, V. et al. (eds.) Advanced Manufacturing Processes II. InterPartner 2020. Lecture Notes in Mechanical Engineering, pp. 664–674. Springer, Cham (2021). https://doi.org/10.1007/978-3-030-68014-5
31. Aynur, T.N., Hwang, Y., Radermacher, R.: Integration of variable refrigerant flow and heat pump desiccant systems for the cooling season. Appl. Therm. Eng. **30**, 917–927 (2010)
32. Aynur, T.N., Hwang, Y., Radermacher, R.: Field performance measurements of a heat hump desiccant unit in dehumidification mode. Energy Build. **40**, 2141–2147 (2008)
33. Sait, H.H.: Estimated thermal load and selecting of suitable air-conditioning systems for a three story educational building. Procedia Comput. Sci. **19**, 636–645 (2013)
34. Radchenko, M., Radchenko, R., Tkachenko, V., Kantor, S., Smolyanoy, E.: Increasing the operation efficiency of railway air conditioning system on the base of its simulation along the route line. In: Nechyporuk M. at al. (eds) Integrated Computer Technologies in Mechanical Engineering (ICTM 2019). Advances in Intelligent Systems and Computing (2020), vol. 1113, pp. 461–467. Springer, Cham (2020). https://doi.org/10.1007/978-3-030-37618-5
35. Kalhori, S.B., Rabiei, H., Mansoori, Z.: Mashad trigeneration potential–An opportunity for CO2 abatement in Iran. Energy Conv. Manag. **60**, 106–114 (2012)
36. Cardona, E., Piacentino, A.: A methodology for sizing a trigeneration plant in mediterranean areas. Appl. Thermal Eng. **23**, 15 (2003)
37. Rodriguez-Aumente, P.A., Rodriguez-Hidalgo, M.C., Nogueira, J.I., Lecuona, A., Venegas, M.C.: District heating and cooling for business buildings in Madrid. Appl. Therm. Eng. **50**, 1496–1503 (2013)
38. Freschi, F., Giaccone, L., Lazzeroni, P., Repetto, M.: Economic and environmental analysis of a trigeneration system for food-industry: A case study. Appl. Energy **107**, 157–172 (2013)
39. Oktay, Z., Coskun, C., Dincer, I.: A new approach for predicting cooling degree-hours and energy requirements in buildings. Energy **36**(8), 4855–4863 (2011)
40. Forsyth, J.L.: Gas turbine inlet air chilling for LNG. In: IGT International Liquefied Natural Gas Conference Proceedings, vol. 3, pp. 1763–1778 (2013)