Improvement of the Refrigeration Capacity Utilizing for the Ambient Air Conditioning System

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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 closed to maximum annual cooling production and to match current conditioning duties at the same time. The approach to improve the efficiency of utilizing the refrigeration capacity of the air conditioning system is based on shearing a rational design refrigeration capacity in two ranges concerning current cooling consumption. The first booster range of ambient air precooling to a certain intermediate threshold temperature is characterized by considerable fluctuations in cooling load, whereas the second range of subsequent subcooling air to a target leaving temperature is characterized by the comparatively stable cooling load. The first booster range of cooling load requires regulation of the cooling capacity, for instance, by the application of a variable speed compressor, whereas the second range of subsequent subcooling air can be covered by the operation of a conventional compressor in a mode closed to a nominal value.

Keywords: Annual refrigeration energy production · Design refrigeration capacity · Current cooling load

1 Introduction

Energy consumption for treating the ambient air in air conditioning systems (ACS) depends on the ambient air temperature t_a and relative humidity φ_a, which changes considerably during the day. It is obvious that the production of refrigeration by refrigeration machine (RM) according to ambient air conditioning duties over a certain period of time, for instance a year (annual refrigeration) $\sum(Q_0{\cdot}\tau)$, depends on the current thermal loading Q_0, caused by ambient air processing, as well as on the hour duration τ of the ACS operation. Issuing from this the efficiency of using the refrigeration capacity for ambient air processing in ACS can be estimated by annual refrigeration $\sum(Q_0{\cdot}\tau)$ as a primary criterion in further complex thermal and economical optimization of the whole ACS, including indoor air processing about actual cooling

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loads of building and indoor thermal-humidity environments in targeted climatic conditions.

The existing approaches to improve the efficiency of ambient air processing in ACS are aimed at determining a design refrigeration capacity to cover the maximum yearly cooling load, which leads to oversizing the RM and other equipment and enlarged their cost as a consequence. The application of various controlling systems and variable speed compressors provides lowering energy consumption, but the problem of determining a rational design refrigeration capacity, providing closed to maximum annual effect without overestimating the installed refrigeration capacity, needs further solving.

2 Literature Review

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

Numerous researchers have studied the energy efficiency of the Variable Refrigerant Flow (VRF) systems [21, 22] and proposed some practical recommendations. Mainly the studies have been conducted on solutions of efficient operation of the VRF system in buildings and control strategies of the systems [23]. A control algorithm of the supply air temperature as a set temperature in the outdoor air processing (OAP) unit to run the VRF–OAP system more efficiently for buildings was developed in [24]. The control algorithm was conducted with adjusting the refrigerant flow supply to the OAP and the indoor unit appropriately through supporting the supply air temperature according to the outdoor ambient and indoor temperature and humidity conditions. Results [25] show that ACS have a great potential for energy saving, and the adjustability of the VRF system is better than of centralized ACS.

The VRF system with heat recuperative ventilation [26] and outdoor ACS was introduced [27]. The evaluation of indoor thermal-humidity environments and energy consumption of the VRF system with a heat pump desiccant was conducted [28].

The author [29] proposes the method of calculating the thermal load of the building. The VRF systems operate with high part-load efficiency, 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 [30].

Despite the existence of a lot of control algorithms and devices to minimize the power consumption and provide indoor thermal-humidity environments in all the central ACS, the problem of choosing a rational design cooling capacity to provide annually efficient operation at changeable current loadings and to exclude oversizing RM and cooling equipment and its cost as a consequence needs a solution.

The study aims to develop an approach to improve the efficiency of utilizing the refrigeration capacity of ACS and method to determine a rational design cooling capacity that provides closed to maximum annual effect without oversizing and its distribution concerning current climatic conditions.

3 Research Methodology

The efficiency of ACS and their RM performance depends on their current loading Q_0 and an hour duration τ of yearly operation. The efficiency of using the refrigeration capacity for covering the ambient air processing needs is estimated by annual refrigeration energy production $\sum(Q_0{\cdot}\tau)$ as a primary criterion to determine a rational design refrigeration capacity.

The proposed method to determine the rational design refrigeration capacity is based on the yearly loading characteristic cumulative curve of annual refrigeration production dependence on the design refrigeration capacity of the RM. The current refrigeration, generated by RM at any time period for ambient air cooling down to the target temperature t_{a2} = 10 °C, has been summarized over the year. The rational design refrigeration capacity is selected according to the yearly loading characteristic cumulative curve to provide closed to maximum annual refrigeration production.

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. However, in relative (specific) values per unit airflow rate (G_a = 1 kg/s) – in the form of specific refrigeration 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: $Q_0 = (c_a\, \xi{\cdot}\Delta t_a)G_a$, where $\Delta t_a = t_a - t_{a2}$ – decrease in air temperature.

The specific annual production of refrigeration:

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

where: q_0 – specific refrigeration capacity [kJ/kg] or [kW/(kg/s)]; $\sum(q_0{\cdot}\tau)$ – specific annual production of refrigeration [kJ/(kg/h)] or [kW•h/(kg/s)]; ξ – specific heat ratio of total heat, including sensible and latent heat, to sensible heat of humid air; 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:

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

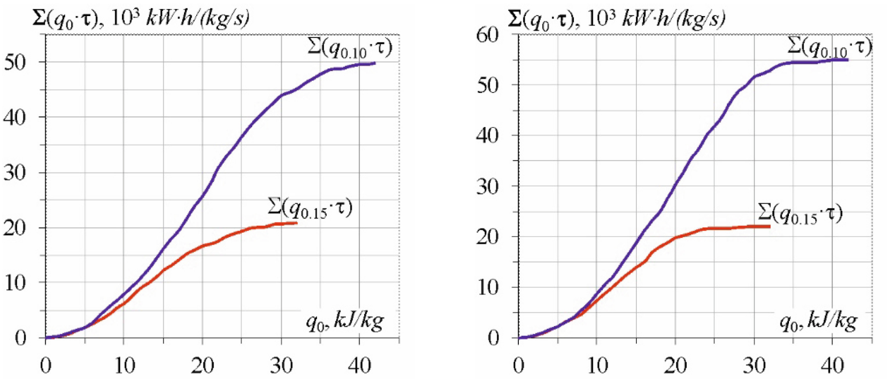
A rational specific refrigeration capacity $q_{0.rat}$ is determined to exclude unproductive expenses of refrigeration capacity q_0 caused by oversizing RM without obtaining a noticeable effect in increasing the annual refrigeration production $\sum(q_0{\cdot}\tau)$.

The further improved approach to enhance the efficiency of utilizing the ACS refrigeration capacity is based on shearing the overall refrigeration capacity, spent for ambient air processing, in two ranges with regard to current cooling loads. The first, booster, range of ambient air precooling to a certain intermediate threshold temperature, is characterized by considerable fluctuations in cooling load, whereas the second range of subsequent subcooling air to a target leaving temperature is characterized by the comparatively stable cooling load. The first booster range of cooling load requires regulation of the cooling capacity by the application of a variable speed compressor, whereas the second range of subsequent subcooling air can be covered by the operation of a conventional compressor in a mode closed to a nominal value.

Nomenclature		Δt_a	Ambient air temperature reduction
$\sum(q_0 \cdot \tau)$	Specific annual production of refrigeration energy, $\sum(q_0 \cdot \tau) = \sum(Q_0 \cdot \tau)/G_a$	φ_a	Ambient air relative humidity
AC	Air cooler	c_{ma}	Specific heat of moist air
ACS	Air conditioning system	*Subscripts*	
c	Specific heat of the air	10, 15	Parameters of air-cooled to temperatures 10 °C and 15 °C
G_a	Air mass flow rate	10–15	The difference in the parameters of air-cooled from 10 °C to 15 °C
OAP	Outdoor air processing	a	Ambient air at the inlet of cooler
q_0	Specific refrigeration capacity, $q_0 = Q_0/G_a$	$a2$	Cooled air at the outlet of the cooler
Q_0	Refrigeration capacity	b	Booster stage of air cooler
$q_{0.10}$, $q_{0.15}$	Specific refrigeration capacity for cooling ambient air to 10 °C and 15 °C	def	Deficit
$q_{0.10\text{–}15}$	Specific refrigeration capacity for cooling air from 10 °C to 15 °C	ma	Moist air
$q_{0.b}$	Booster stage specific refrigeration capacity	rat	Rational
$q_{0.bdef}$	Booster stage specific refrigeration capacity deficit	*Greek letters*	
$q_{0.rat}$	Rational specific refrigeration capacity	Δ	Difference
RM	Refrigeration machine	ξ	Specific heat ratio of total heat (sensible and latent) to sensible heat of moist air
t	Temperature	Σ	Sum
t_a	Ambient air temperature	τ	Time interval
t_{a2}	Air temperature at the air cooler outlet	φ	Relative humidity
VRF	Variable refrigerant flow		

4 Results

Specific annual refrigeration energy production $\sum(q_0 \cdot \tau)$ required for cooling the ambient air to t_{a2} = 10 and 15 °C against a design specific refrigeration capacity q_0, calculated for 2017 and 2019 in Mykolaiv region, south of Ukraine, is presented in Fig. 1.



a *b*

Fig. 1. Specific annual refrigeration energy production $\sum(q_0 \cdot \tau)$ required for cooling the ambient air to t_{a2} = 10 and 15 °C against a design specific refrigeration capacity q_0 for 2017 (*a*) and 2019 (*b*).

For the considered climatic conditions when the ambient air is cooled to the temperature t_{a2} = 10 °C a design specific (at G_a = 1 kg/s) refrigeration capacity of RM $q_{0.10rat}$ = 35 kJ/kg provides close to the maximum annual refrigeration production $\sum(q_0 \cdot \tau) \approx 48 \cdot 10^3$ kW·h/(kg/s*)* while maintaining its increment with a noticeable high rate. A design specific refrigeration capacity $q_{0.10rat}$ = 35 kJ/kg for cooling the ambient air to t_{a2} = 10 °C is assumed as rational one to provide closed to a maximum annual refrigeration production $\sum(q_0 \cdot \tau)$. However, with decreased specific refrigeration capacity $q_{0.10rat}$ = 35 kJ/kg versus its maximum value $q_{0.10}$ = 42 kJ/kg, id est. by more than 15%. Similarly, for cooling ambient air to t_{a2} = 15 °C the rational value of specific refrigeration capacity is $q_{0.15rat}$ = 25 kJ/kg is lower than its maximum value $q_{0.15}$ = 32 kJ/kg by about 20% with the corresponding reduction in sizes of RM and other equipment.

In order to prove the approach to the analysis of the efficiency of using design refrigeration capacities of ACS chillers, taking into account the change in cooling loads by actual 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 and 15 °C, respectively $q_{0.10}$ and $q_{0.15}$ for July of 2017 and 2019 year, Mykolaiv region, south of Ukraine, have been considered (Fig. 2).

As calculation results in Fig. 1 shows, when the ambient air is cooled from its current temperatures t_a to t_{a2}, the cooling load fluctuations of $q_{0.10}$ and $q_{0.15}$ are very significant. The almost equidistant trend lines of the specific cooling load $q_{0.10}$ and $q_{0.15}$ indicate that these fluctuations are caused primarily by changes in the specific cooling load $q_{0.15}$ for precooling the ambient air to the temperature t_{a2} = 15 °C, within which there is practically damping of the fluctuations of the current cooling load. The intermediate temperature t_{a2} = 15 °C is assumed as threshold temperature for the rational distribution of design overall cooling capacity of ACS between two ranges with different characters of cooling load behavior.

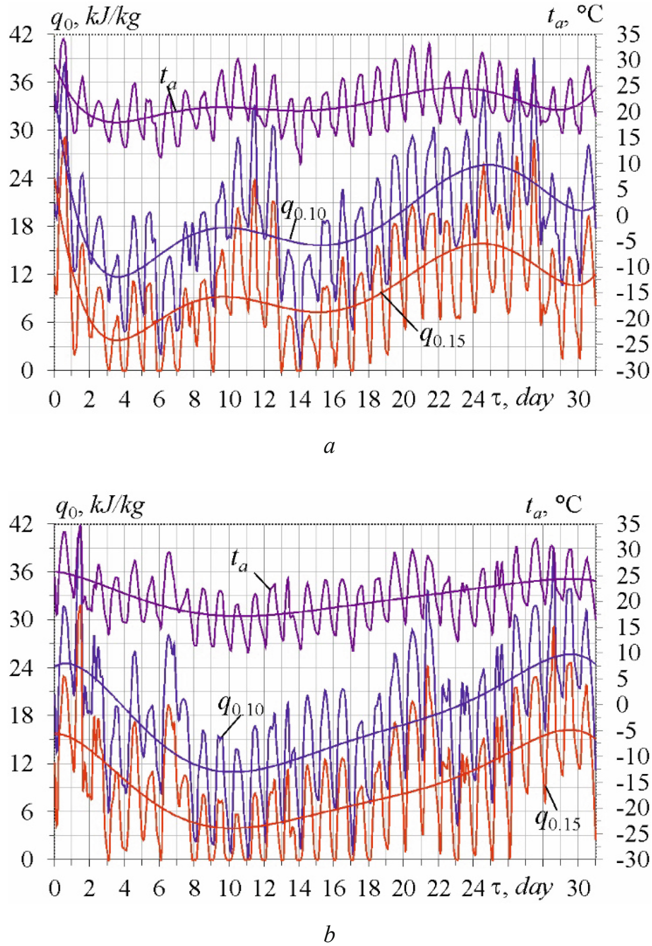


Fig. 2. Current values of ambient air temperature t_a, specific refrigeration capacity $q_{0.10}$, needed for cooling ambient air from t_a to t_{a2} = 10 °C, specific refrigeration capacity $q_{0.15}$, needed for cooling ambient air from t_a to intermediate temperature t_{a2} = 15 °C for July 2017 (*a*) and 2019 (*b*).

At the same time, within further cooling of the air from the intermediate temperature t_{a2} = 15 °C to t_{a2} = 10 °C the fluctuations of the specific cooling loads on the ACS $q_{0.10–15} = q_{0.10} - q_{0.15}$ are comparatively small (within a range of 6–10 kJ/kg in Fig. 3a versus 24–2 kJ/kg for $q_{0.15}$ in Fig. 2a and within a range of 4–10 kJ/kg in Fig. 3b versus 24–0 kJ/kg for $q_{0.15}$ in Fig. 2b for a bit lower monthly ambient air temperature) 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. 3).

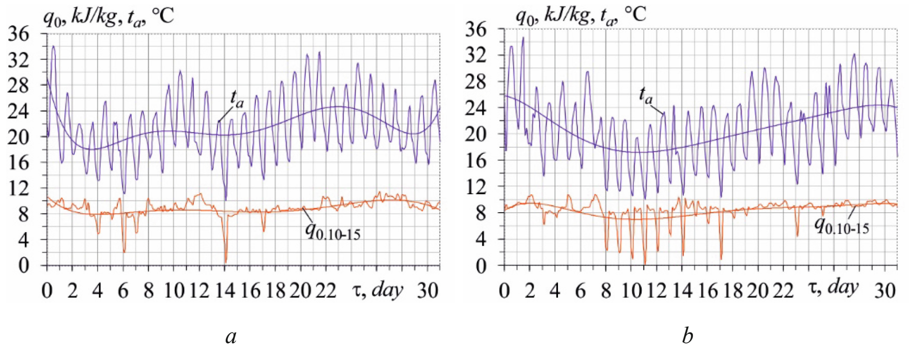
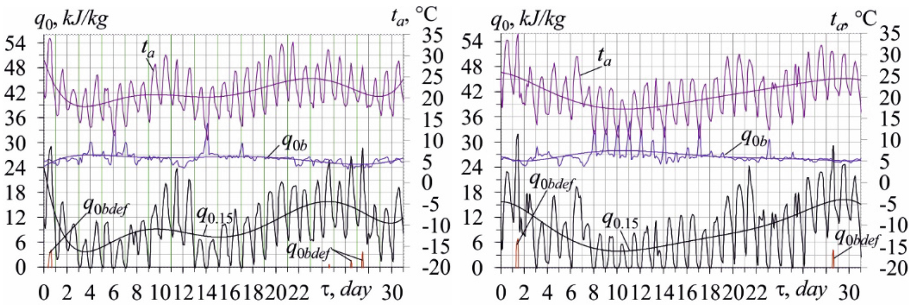


Fig. 3. Current values of ambient air temperature t_a, specific refrigeration capacity $q_{0.10–15} = q_{0.10} - q_{0.15}$, needed for subcooling air from t_{a2} = 15 °C to t_{a2} = 10 °C for July 2017 (*a*) and 2019 (*b*).

Proceeding from a different behavior of current cooling loads, the ambient air treatment in the ACS is considered as two-stage processing and includes a range of cooling load fluctuation as the first booster stage ambient air precooling and a range of comparatively stable cooling load as the second air subcooling stage.

Accordingly, a design refrigeration capacity for precooling the ambient air from the current temperature t_a to t_{a2} = 15 °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 according to Fig. 1, and its stable component $q_{0.10–15}$ for further subcooling air: $q_{0b} = q_{0.10rat} - q_{0.10–15}$ (Fig. 4).



a *b*

Fig. 4. Current values of ambient air temperature t_a, specific refrigeration capacity $q_{0.15}$, needed for cooling ambient air from t_a to intermediate temperature t_{a2} = 15 °C, booster specific refrigeration capacity $q_{0b} = q_{0.10rat} - q_{0.10\text{–}15}$ and booster specific refrigeration capacity deficit $q_{0bdef} = q_{0.15} - q_{0b}$ within subcooling air from t_{a2} = 15 °C to t_{a2} = 10 °C for July 2017 (*a*) and 2019 (*b*).

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 $q_{0.15}$, which corresponds to precooling the ambient air from t_a to t_{a2} = 15 °C, at elevated current cooling loads $q_{0.15}$ there is some deficit of the booster component q_{0bdef} of refrigeration capacity calculated by the residual principle, $q_{0bdef} = q_{0b} - q_{0.15}$, compared to the current specific thermal loads $q_{0.15}$. However, the booster refrigeration capacity deficit q_{0bdef} occurs very seldom 1, 27, 28 July of 2017 (Fig. 4a) and 2 and 29 July of 2019 (Fig. 4b). It proves the appropriate results of the application of the proposed method for analyzing the efficiency and rational distribution of a design refrigeration capacity used for ambient air processing in ACS according to current cooling loading.

5 Conclusions

The approach to analyzing the efficiency of utilizing a refrigeration capacity used for ambient air processing in ACS and method of determining its rational values and their distribution during the operation in actual changeable current climatic conditions is presented. The annual refrigeration production is assumed as a primary criterion for estimation of utilizing a refrigeration capacity, and its dependence on a design refrigeration capacity is used as a basic yearly loading characteristic curve for determining a rational design refrigeration capacity.

The method to determine a rational design cooling capacity to provide a closed to maximum annual refrigeration production and, at the same time, to avoid oversizing a refrigeration machine and its enlarged cost has been developed.

The ambient air treatment in ACS is proposed to consider as two-stage processing, which includes a range of cooling load fluctuation as the first booster stage of air cooler for ambient air precooling and a range of comparatively stable cooling load as the second stage for further air deep cooling. The first booster range of cooling load requires regulation of the cooling capacity, for instance, by the application of a variable speed compressor, whereas the second range of subsequent subcooling air can be covered by the operation of a conventional compressor in a mode closed to a nominal value.

References

1. Li, H., Liu, E., Zhou, G., Yang, F., Su, Z., Xu, Y., Jian, C., Shi, S.: Influence of fin thickness on heat transfer and flow performance of a parallel flow evaporator. Therm. Sci. **23**(4), 2413–2419 (2019)
2. Saab, R., Ali, M.I.H.: Variable-refrigerant-flow cooling-systems performance at different operation-pressures and types-of-refrigerants. Energy Procedia **119**, 426–432 (2017)
3. Dąbrowski, P., Klugmann, M., Mikielewicz, D.: Channel blockage and flow maldistribution during unsteady flow in a model microchannel plate heat exchanger. J. Appl. Fluid Mech. **12**, 1023–1035 (2019)
4. Dąbrowski, P., Klugmann, M., Mikielewicz, D.: Selected studies of flow maldistribution in a minichannel plate heat exchanger. Arch. Thermodyn. **38**, 135–148 (2017)
5. Saeed, M.U., Qureshi, S.R., Hashmi, K.J., Khan, M.A., Danish, S.N.: Performance assessment of alternate refrigerants for retrofitting R22 based air conditioning system. Therm. Sci. **22**(2), 931–941 (2018)
6. Konovalov, D., Trushliakov, E., Radchenko, M., Kobalava, H., Maksymov, V.: Research of the aerothermopressor cooling system of charge air of a marine internal combustion engine under variable climatic conditions of operation. In: Tonkonogyi V., et al. (eds.) Advanced Manufacturing Processes. InterPartner-2019, LNME, pp. 520–529. Springer, Cham (2020)
7. Trushliakov, E., Radchenko, M., Bohdal, T., Radchenko, R., Kantor, S.: An innovative air conditioning system for changeable heat loads. In: Tonkonogyi, V., et al. (eds.) Advanced Manufacturing Processes. InterPartner-2019, LNME, pp. 616–625. Springer, Cham (2020)
8. 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., Pavlikov, V., Kritskiy, D. (eds.) Integrated Computer Technologies in Mechanical Engineering. Advances in Intelligent Systems and Computing, vol. 1113, pp. 461–467. Springer, Cham (2020)
9. Forduy, S., Radchenko, A., Kuczynski, W., Zubarev, A., Konovalov, D.: Enhancing the gas engines fuel efficiency in integrated energy system by chilling cyclic air. In: Tonkonogyi V., et al. (eds.) Advanced Manufacturing Processes. InterPartner-2019, LNME, pp. 500–509. Springer, Cham (2020)
10. Radchenko, A., Mikielewicz, D., Forduy, S., Radchenko, M., Zubarev, A.: Monitoring the fuel efficiency of gas engine in integrated energy system. In: Nechyporuk, M., Pavlikov, V., Kritskiy, D. (eds.) Integrated Computer Technologies in Mechanical Engineering. Advances in Intelligent Systems and Computing, vol. 1113, pp. 361–370. Springer, Cham (2020)
11. Volintiru, O.N., Scurtu, I.C., Stefănescu, T.M.: Modeling and optimization of HVAC system for special ships. J. Phys: Conf. Ser. **1122**(1), 012004 (2018)
12. Mikielewicz, D., Mikielewicz, J.: A thermodynamic criterion for selection of working fluid for subcritical and supercritical domestic micro CHP. Appl. Therm. Eng. **30**, 2357–2362 (2010)
13. Bohdal, L., Kukielka, L., Świłło, S., Radchenko, A.M., Kułakowska, A.: Modelling and experimental analysis of shear-slitting process of light metal alloys using FEM, SPH and vi-sion-based methods. In: AIP Conference Proceedings, vol. 2078, p. 020060 (2019)

14. Bohdal, L., Kukielka, L., Radchenko, A.M., Patyk, R., Kułakowski, M., Chodór, J.: Modelling of guillotining process of grain oriented silicon steel using FEM. In: AIP Conference Proceedings, vol. 2078, p. 020080 (2019)
15. 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. **00**, 154 (2020)
16. Ortiga, J., Bruno, J.C., Coronas, A.: Operational optimisation of a complex trigeneration system connected to a district heating and cooling network. Appl. Therm. Eng. **50**, 1536–1542 (2013)
17. Preda, A., Scurtu, I.C.: Thermal image building inspection for heat loss diagnosis. In: 5th International Scientific Conference SEA-CONF 2019, Journal of Physics: Conference Series, vol. 1297, no. 012004, 6 p. (2019)
18. Balasubramanian, A., Jayaraman, V., Sivan, S., Vairavan, M.: Experimental analysis of triple fluid vapour absorption refrigeration system driven by electrical energy and engine waste heat. Therm. Sci. **23**(5B), 2995–3001 (2019)
19. Radchenko, A., Bohdal, L., Zongming, Y., Portnoi, B., Tkachenko, V.: Rational designing of gas turbine inlet air cooling system. In: Tonkonogyi V., et al. (eds.) Advanced Manufacturing Processes. InterPartner-2019, LNME, pp. 591–599. Springer, Cham (2020)
20. Eidan, A.A., Alwan, K.J.: Enhancement of the performance characteristics for air conditioning system by using direct evaporative cooling in hot climates. Energy Procedia **142**, 3998–4003 (2017)
21. Im, P., Malhotra, 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: 16th International Refrigeration and Air Conditioning Conference, Purdue, West Lafayette, USA (2016)
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. 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)
24. 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), 15 (2018)
25. 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)
26. 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)
27. 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)
28. 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)
29. 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)
30. Enteria, N., Yamaguchi, H., Miyata, M., Sawachi, T., Kuwasawa, Y.: Performance evaluation of the variable refrigerant flow (VRF) air-conditioning system subjected to partial and unbalanced thermal loadings. J. Therm. Sci. Technol. **11**(1), 1–11 (2016)