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RATIONAL DESIGNING OF TWO-STAGE ABSORPTION-EJECTOR CHILLING AIR SYSTEM

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ABSTRACT

The method to determine a rational design refrigeration capacity of ambient air chilling systems (ACS) to cover current thermal loads in response to actual intake air parameters and provide closed to maximum annual refrigeration avoiding oversizing is proposed. The method enables to determine a rational load distribution in ambient air processing to match actual climatic conditions by deviding the overall ACS thermal load into unstable range for ambient air precooling with load fluctuations and a comparatively stable load part for further air subcooling to target temperatures. The unstable and stable load ranges are covered by applying refrigeration machines of different types: the first – by high efficient chillers not sensitive to load changes (variable speed refrigeration compressor or waste heat recovery absorption chiller), the second – by operation of chillers at about rated thermal loads (traditional compressors or ejector chilles as the simplest). Such approach is reasonable for engine intake air cooling too.

Keywords: Refrigeration Machine, Current Thermal Load, Design Cooling Capacity, Climatic Condition

1. INTRODUCTION

The efficiency of ambient air chilling systems (ACS) and their refrigeration machines (chillers) performance depends on their loading and a duration of operation all the year round. The higher loading and longer duration of ACS operation during a year, the larger annual refrigeration energy generated according to current cooling duties. The rational design refrigeration capacity of ACS has to cover current thermal loads in response to actual intake ambient air parameters and to provide closed to maximum annual refrigeration effect avoiding refrigeration machines (RM) and the whole ACS oversizing simalteneously. In addition the method of ACS rational designing should allow to determine a rational thermal load distribution in ambient air processing to match actual changeable climatic conditions. With this the rational overall thermal load of ACS might be devided into unstable load range, corresponding to ambient air precooling with significant thermal load fluctuations in response to current climatic conditions, and a comparatively stable load part for further air subcooling to the target temperatures. The unstable and stable cooling load ranges could be covered by applying the RM of different types: the first load range – by application of the RM with high efficiency not strictly sensitive to load changes, the second range – by operation of the RM at about rated thermal loads.

A lot of publications are devoted to improving the performance of ACS by intensification of heat transfer processes (Bohdal et.al., 2015; Mikielewicz et al., 2013, 2016; Trushliakov et. al., 2018) and advanced scheme of RM (Butrymowicz et. al., 2013, 2018; Elbel et. al., 2016) and waste heat recovery technics (Forduy et. al., 2019; Konovalov et. al., 2018; Radchenko A. et. al., 2019), ambient air precooling by using an excessive refrigeration accumulated at decreased loads to cover peak loads (Radchenko R. et. al., 2018).

Numerous researchers studied the efficiency of ductless Variable Refrigerant Flow (VRF) systems. A Heating, Ventilation, and Air Conditioning (HVAC) system combined with roof top unit (RTU) was used as outdoor air processing (OAP) system in VRF system was proposed (Zhu et. al., 2014). It was obtained that the multi-split VRF system saved more than 20% energy compared to a variable air volume (VAV) system (Zhou et. al., 2007).

In a number of investigations the air conditioning is considered as one of the technologies for combined cooling, heating and power (CCHP) (Ortiga, et. al., 2013; Rodriguez-Aumente P.A. et. al., 2013). Some of principal technical innovations and methodological approaches in waste heat recovery refrigeration might be successfully applied for compression refrigeration technologies in air conditioning, in particular, two-stage air cooling (Radchenko A. et. al., 2019, 2020), methods to choose design refrigeration capacity to match current cooling demand (Canova et. al., 2009; Forsyth et. al., 2013; Kalhori et. al., 2012; Radchenko A. et. al., 2019, 2020).

All the typical methods, based on current or summarized annual refrigeration energy production, issue from the assumption of a design refrigeration capacity to cover maximum cooling needs over the full range of yearly operating conditions (Forsyth et. al., 2013; Kalhori et. al., 2012). Such approach inevitable leads to considerable oversizing the RM and the ACS in the whole, that requires to solve the problem of defying the correct design cooling load excluding oversizing, as it was shown in (Radchenko A. et. al., 2020; Radchenko M. et. al., 2020).

Although many researchers consider current or annual refrigeration energy generated in response to actual air conditioning duties along with time elapsed, only a few studies focus on analyzing the behavior of yearly cumulative cooling profiles in dependence on loading to determine a design refrigeration capacity (Forsyth et. al., 2013; Kalhori et. al., 2012; Radchenko A. et. al., 2019, 2020).

Thus, the methods should allow to define a rational design refrigeration capacity issuing from the actual ambient thermal loading and at the same time makes it possible to provide efficient operation of ACS at rational thermal loading and to develop corresponding scheme decisions of ACS and RM. They would be quite reasonable for rational designing of central HVAC systems and their combined versions with indoor air conditioning system, that typically operate in the range of 40% to 80% of design capacity (Goetzler, 2007). It might be also adopted for designing ductless Variable Refrigerant Flow (VRF) systems with Outdoor Air Processing (OAP) system and their advanced VRF-OAP version with common OAP unit, for which the actual OAP refrigeration capacity should be less than 30% of the design outdoor unit capacity to prevent a lack of cooling capacity in indoor units (Im, 2016). Proceeding from the results of researches mentioned above the priority of ambient air processing rational loading in ACS for space and energetic conditioning is evident.

The aim of research is to develop a novel method of rational designing of ACS that provides defining the rational values of the overall refrigeration capacity of the ACS and its distribution between unstable and stable cooling load ranges in response to current climatic conditions covered by refrigeration machines (chillers) of different types to provide efficient operation with achieving closed to maximum annual refrigeration energy generation according to cooling needs without refrigeration machines overestimation and the ACS oversizing and enables to develop corresponding scheme decisions of advanced two-stage air cooling by refrigeration machines of combined type. The methodology is simple and quite reasonable for rational designing of ACS for space conditioning and combustion engine intake air cooling. The application of annual refrigeration energy generation as primary criterion enables to simplify futher detailed economical calculations for concret object.

2. THE RESULTS

The efficiency of ACS and their RM performance depends on their loading and a duration of their yearly operation. Therefore the annual refrigeration energy generated in response to current duties is considered as a primary criterion for the choice of a rational design thermal load of ACS. For this the current refrigeration energy generated by the RM at any time period needed for ambient air cooling to the target temperature, have been summarized over the year to determine the rational design thermal load. To conduct this procedure the were two methods developed: the first – by using the annual summarized refrigeration energy dependence on the design refrigeration capacity of the RM to choose its value, that provides closed to maximum annual refrigeration generation, and the second – according to the maximum rate of annual refrigeration energy increment to choose optimal design refrigeration capacity, that provides minimum sizes of RM and ACS.

The further development of the method of ACS designing consists in rational distribution of the overall design refrigeration capacity between two ranges. Proceeding from a different character of behaviour of thermal loads, the ambient air processing in the ACS is considered as a two-stage processing and includes a fluctuation range as the first (high-temperature) stage and comparatively stable range as the second (low-temperature) stage. The threshold air temperature is determined for rational distribution of design overall cooling capacity of ACS between two stages with differrent character of heat load behaviour.

The proposed method to determine the rational design refrigeration capacity of ACS and their RM is based on the yearly loading cumulative characteristic of ACS as annual summarized refrigeration energy production dependence on a design refrigeration capacity of the RM: $\Sigma(Q_0 \cdot \tau) = f(Q_0)$. The current refrigeration energy, generated by the RM during time periods for ambient air cooling to the target temperature t_{a2} = 10, 15 or 20°C, has been summarized over the year.

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 RM not in absolute Q_0, but in relative (specific) values per unit air mass flow rate through the air cooler (G_a = 1 kg/s) – as specific refrigeration capacity q_0, kW/(kg/s) or kJ/kg:

$$q_0 = Q_0 / G_a \quad \text{or} \quad q_0 = \xi \cdot c_{ma} \cdot (t_a - t_{a2}), \tag{1}$$

where Q_0 is the total refrigeration capacity when cooling the air with mass flow rate G_a:

$$Q_0 = (c_a\, \xi \cdot \Delta t_a)G_a\,, \tag{2}$$

where $\Delta t_a = t_a - t_{a2}$ – decrease in air temperature; t_a – ambient air temperature, K or °C; t_{a2} – air temperature at the air cooler outlet; ξ – relative heat ratio; c_{ma} –specific heat of moist air, kJ/(kg·K).

Then a specific annual refrigeration energy production $\Sigma(q_0 \cdot \tau)$, kWh/(kg/s) or kJ/(kg/h):

$$\Sigma(q_0 \cdot \tau) = \Sigma(\xi \cdot c_{ma} \cdot (t_a - t_{a2}) \cdot \tau \cdot 10^{-3}), \tag{3}$$

where τ – time interval, h.

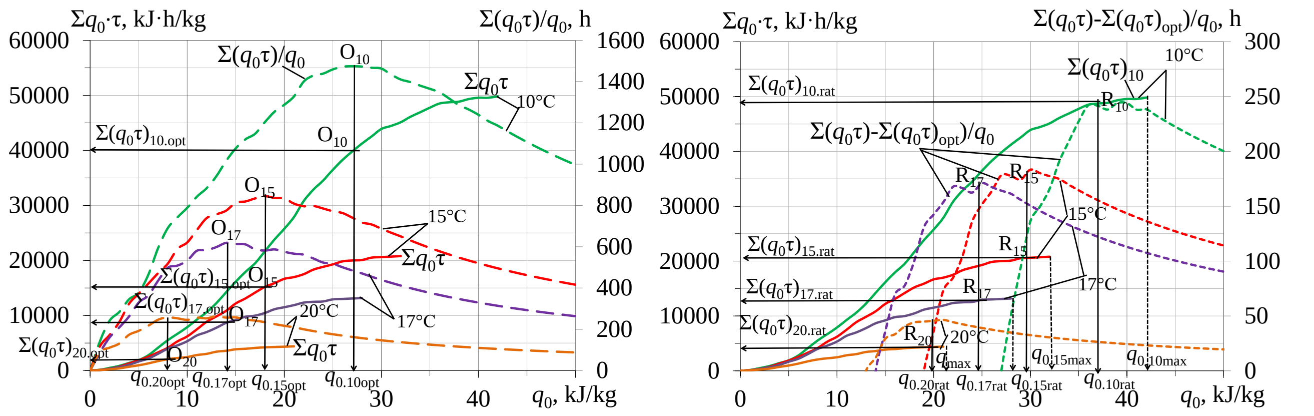
The real input data on site actual climatic conditions (ambient air temperature t_{amb} and relative humidity φ_{amb}) were selected by using the well known program "meteomanz" (http://www.meteomanz.com).

According to a developed method the fluctuations of the current specific refrigeration energy production $q_0 \cdot \tau$ were considered by different rate of their annual summation $\sum(q_0 \cdot \tau)$ increment in response to specific refrigeration capacity q_0 needed. With this a rate of annual refrigeration energy increment as its relative value $\sum(q_0 \cdot \tau)/q_0$ according to increasing a refrigeration capacity q_0 required is used as an indicator to efficient utilizing a design refrigeration capacity. This is a principally novel approach to ACS designing versus a traditional approach to cover the maximum current thermal duties and maximum annual refrigeration energy production accordingly that leads to RM oversizing.

In order to conduct this approach the advanced methodology of ACS designing includes addition stages focused to determine an optimal value of specific refrigeration capacity $q_{0.opt}$, providing the maximum rate of annual specific refrigeration energy $\sum(q_0 \cdot \tau)$ increment as its relative value $\sum(q_0 \cdot \tau)/q_0$ and minimum sizes of ACS accordingly. With this a relative parameter $\sum(q_0 \cdot \tau)/q_0$ is used as indicator for selecting the specific refrigeration capacity (optimum) $q_{0.opt}$, providing minimum ACS sizes with corresponding annual value $\sum(q_0 \cdot \tau)_{opt}$ (Fig.1a).

The rational value of design specific refrigeration capacity $q_{0.rat}$, providing a closed to maximum annual refrigeration energy production $\sum(q_0 \cdot \tau)$ without ACS oversizing (chiller overestimating) is associated with the second maximum rate of annual specific refrigeration energy production $\sum(q_0 \cdot \tau)$ increment within its range beyond the first maximum rate: $q_0 > q_{0opt}$ and $\sum(q_0 \cdot \tau) > \sum(q_0 \cdot \tau)_{opt}$ accordingly. With this a familiar relative parameter $[\sum(q_0 \cdot \tau) - \sum(q_0 \cdot \tau)_{opt}]/q_0$ is used as indicator to choose a rational value $q_{0.rat}$, providing a closed to maximum annual refrigeration energy production $\sum(q_0 \cdot \tau)$ (Fig.1b).

The results of calculations of optimum $q_{0.opt}$ and rational $q_{0.rat}$ specific refrigeration capacity for set cooled air temperatures t_{a2} = 10, 15 and 20 °C and temperate climatic conditions of Nikolaev region, southern Ukraine, 2017 year, are presented in Fig.1.



a **b**

Figure 1: Specific annual refrigeration energy generation $\sum(q_0 \cdot \tau)$ and its relative values $\sum(q_0 \cdot \tau)/q_0$ referred to design specific refrigeration capacity q_0 (a) and relative annual specific refrigeration energy $(\sum(q_0 \cdot \tau) - \sum(q_0 \cdot \tau)_{opt})/q_0$ within the range beyond the optimal value $\sum(q_0 \cdot \tau)_{opt}$ (b) for cooling ambient air to t_{a2} = 10, 15 and 20 °C versus a specific refrigeration capacity q_0 : $q_{0.opt}$ – optimum and $q_{0.ratt}$ –rational values

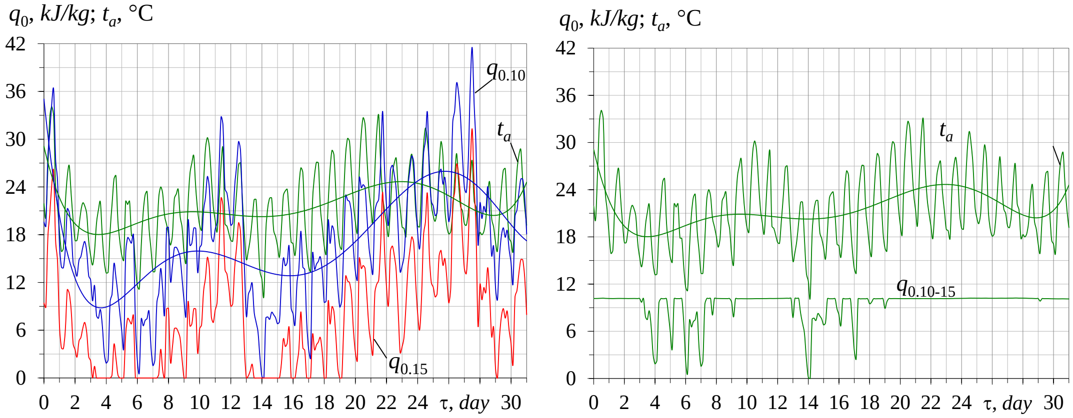
As Fig. 1a shows, a maximum rate of annual specific refrigeration energy increment $\sum(q_0 \cdot \tau)/q_0$ for cooling ambient air to t_{a2} = 10 °C takes place at the optimal design refrigeration capacity $q_{0.opt}$ is about 27 kJ/kg and provides annual refrigeration production $\sum(q_0 \cdot \tau)_{opt}$ significantly less than its maximum value 50 MJ·h/kg.

Fig. 1b shows, a maximum rate of annual specific refrigeration energy increment $(\sum(q_0 \cdot \tau) - \sum(q_0 \cdot \tau)_{opt})/q_0$ within the range beyond its value $\sum(q_0 \cdot \tau)_{opt}$ = 40 MWh/(kg/s) corresponding to optimal design specific refrigeration capacity $q_{0.opt}$=27 kJ/kg for t_{a2} = 10 °C takes place at the rational design specific refrigeration capacity $q_{0.rat}$= 36 kJ/kg and provides precise value of annual specific refrigeration energy production $\sum(q_0 \cdot \tau)_{rat}$ = 49 MJ·h/kg that is very closed to its maximum value 50 MJ·h/kg but at a design refrigeration capacity $q_{0.rat}$=36 kJ/kg much less than oversized its value $q_{0.max}$ = 42 kW/(kg/s), i.e. 15% less.

As Fig. 1 shows, the maximum rate of specific annual refrigeration capacity increment as ratio value $\sum(q_0 \cdot \tau)/q_0$ for cooling ambient air to t_{a2} = 10 ºC is achieved at a design optimum specific refrigeration capacity value $q_{0.opt}$ = 27 kJ/kg with less annual refrigeration energy generation $\sum(q_0 \cdot \tau)_{10}$ = 40 MJ·h/kg. In this case a maximum value of specific annual refrigeration energy generation $\sum(q_0 \cdot \tau)_{10}$ = 50 MJ·h/kg might be achieved by applying the energy conserving technologies through using excessive refrigeration capacity, accumulated at decreased current cooling loads, to cover the peak refrigeration demand as an example.

The further development of the method of ACS designing consists in rational distribution of the overall design refrigeration capacity between two ranges. Proceeding from a different character of behaviour of thermal loads, the ambient air processing in the ACS is considered as a two-stage processing and includes a fluctuation range as the first (high-temperature) stage and comparatively stable range as the second (low-temperature) stage (Fig. 2). The threshold air temperature is determined for rational distribution of design overall cooling capacity of ACS between two stages with different character of heat load behaviour.

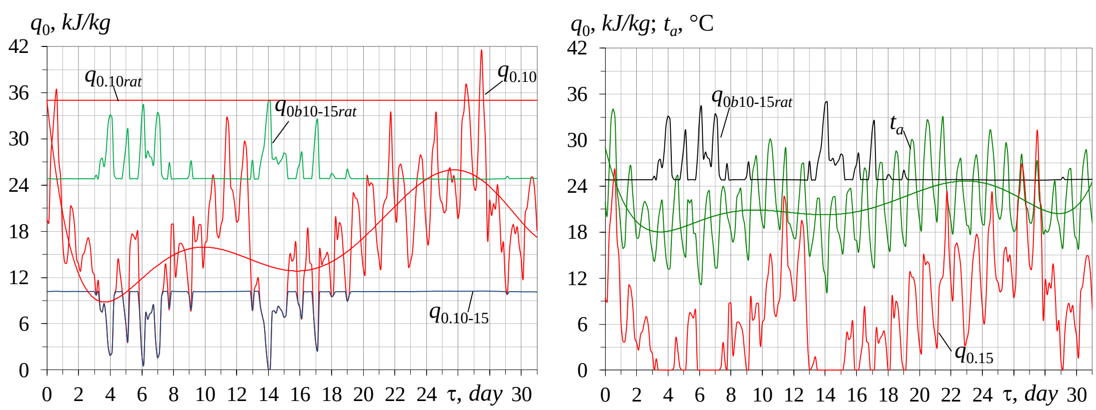
To prove a methodological approach to determine the rational design heat load, matching current changeable climatic conditions, the values of specific refrigeration capacity $q_{0.15}$, needed for cooling ambient air from its current temperature t_a to t_{a2} = 15 °C, and $q_{0.10}$, needed for cooling ambient air to t_{a2} = 10 °C, and the specific refrigeration capacity $q_{0.10\text{-}15}$ as their difference $q_{0.10\text{-}15} = q_{0.10} - q_{0.15}$, needed for cooling air from t_{a2} = 15 °C to t_{a2} = 10 °C, is calculated for climatic conditions in Nikolaev region in July 2017 (Fig.2).



a b

Figure 2: Current values of ambient air temperature t_a , specific refrigeration capacity $q_{0.10}$ and $q_{0.15}$ needed for cooling ambient air to t_{a2} = 10 and 15 °C (a) and specific refrigeration capacity $q_{0.10\text{-}15} = q_{0.10} - q_{0.15}$ (b) needed for cooling air from t_{a2} = 15 °C to t_{a2} = 10 °C for July 2017

As can be seen, with cooling of the ambient air from t_{amb} to the temperature t_{a2} = 15 °C the fluctuations of the current thermal load $q_{0.15}$, i. e. of the current refrigeration capacity of the compressor, spent for the ambient air precooling, are gradual. Such considerable changes in the current thermal loads $q_{0.15}$ on the ACS point out that if the design maximum current thermal load is chosen, this will result in a significant amount of an excessive refrigeration capacity in the temperate daily hours. At the same time, when air is being cooled from t_{a2} = 15 °C to t_{a2} = 10 °C, the fluctuations in the thermal load on the ACS $q_{0.10\text{-}15} = q_{0.10} - q_{0.15}$ are comparatively small: around 10 kJ/kg. The temperature of cooled air t_{a2} = 15 °C is quite reasonably to use as the threshold temperature for shearing the overall design thermal load on the ACS into a comparatively stable thermal load range $q_{0.10\text{-}15}$ and the unstable range of ambient air precooling accompanied by gradual fluctuations of current heat loads. So, the stable thermal load value $q_{0.10\text{-}15}$ is chosen as design basic stable part $q_{0.10\text{-}15} = q_{0.10} - q_{0.15}$ of the rational overall thermal load $q_{0.10rat}$ = 36 kJ/kg on the ACS, determined according to closed to maximum annual energy generation (Fig. 1). Accordingly, the remaining part of the overall thermal load $q_{0.10rat}$ might be used for precooling the ambient air from the current changeable ambient temperature t_{amb} to the threshold temperature t_{a2} = 15 °C and determined as boost specific refrigeration capacity $q_{0.b10\text{-}15rat} = 36 - q_{0.10\text{-}15}$ (Fig. 3).



a *b*

Figure 3: Current values of thermal load $q_{0.10}$ for cooling ambient air to t_{a2} = 10 °C and covered by remaining boost specific refrigeration capacity $q_{0.b10\text{-}15rat} = 35 - q_{0.10\text{-}15}$ and by basic specific capacity $q_{0.10\text{-}15} = q_{0.10} - q_{0.15}$, needed for cooling air from t_{a2} = 15 °C to t_{a2} = 10 °C (a) and thermal load $q_{0.15}$ for cooling ambient air to t_{a2} = 15 °C and covered by boost specific capacity $q_{0.b10\text{-}15}$

As Fig. 3b shows, a design boost specific refrigeration capacity $q_{0.b10\text{-}15rat}$ generally covers current thermal loads $q_{0.15}$ for precooling the ambient air from to t_{a2} = 15 °C , except the warmest quite short periods of daylight hours.

The unstable and stable cooling load ranges could be covered by applying the refrigeration machines of different types: the first load range – by application of the chillers with high efficiency not strictly sensitive to load changes (with variable speed refrigeration compressors or waste heat absorption chiller), the second range – by operation of the chillers at about rated thermal loads (traditional refrigeration compressors without speed variation or ejector chilles as the simplest in design).

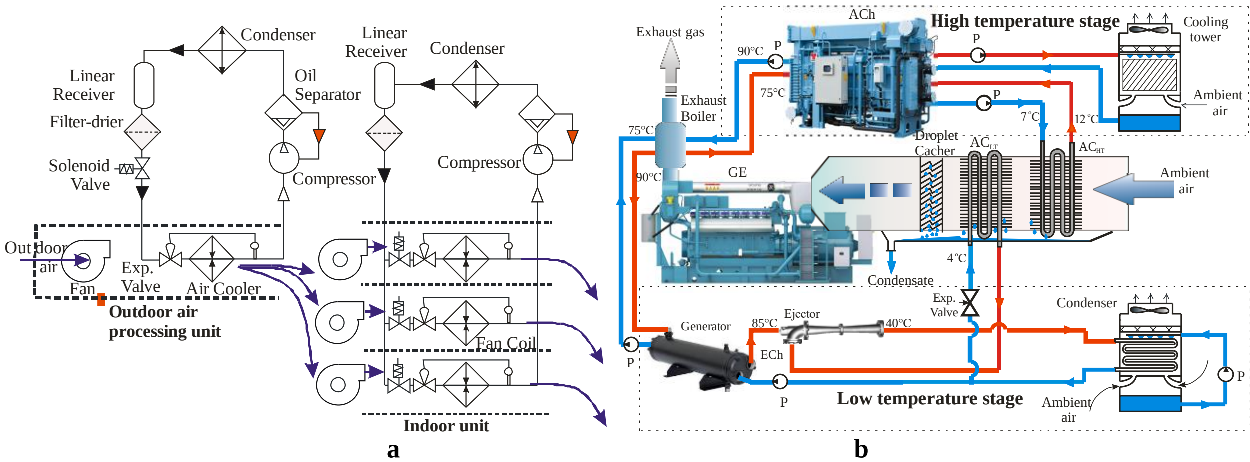


Figure 4: The scheme of combined outdoor and indoor ACS with VRF (b) and two-stage gas engine intake air cooling system in absorption-ejector chiller (AECh): AC_{HT} and AC_{LT} – high-and low temperature stages of air cooler

Such an approach of two-stage ambient air conditioning might be a quite reasonable for rational designing of ACS for space conditioning as independent central ACS and as the outdoor air processing subsystems to run the combined outdoor (OACS) and indoor (IACS) air conditioning systems more efficiently due to air feeding to indoor environments at the target temperature level to avoid overloading the IACS, as well as for cooling air at the inlet of combustion engines by waste heat recovery absorption and ejector chillers (ACh and ECh) to enhance combustion engine fuel efficiency. A two-stage ACS based on combined absorption-ejector chillers (AECh) could be considered as a perspective trend in ACS for energetic application (combustion engine intake air cooling) and for space conditioning as a subsystem of trigeneration installation for combined cooling, heat and power generation (CCHP).

The application of such hybrid intake air coolers is especially expedient for operation of ACS in temperate climatic conditions. They are able to provide deep intake air cooling with corresponding increase of ACS operation duration and much more annual effect as result: fuel reduction or power output increase, compared to traditional cooling in ACh.

3. CONCLUSIONS

A novel methodological approach is proposed to determine a rational design thermal load on the ACS that provides closed to maximum annual refrigeration energy generation in response to current conditioning duties without refrigeration capacity overestimating. It was shown that rational designing of ACS provides decrease of installed refrigeration capacity, refrigeration machine sizes and capital expenses accordingly by around 15%.

According to developed method the fluctuations of the current refrigeration energy production were considered by different rate of their annual summation increment in response to refrigeration capacity needed. With this a rate of annual refrigeration energy increment as its relative value refered to refrigeration capacity required is used as an indicator to efficient utilizing a design refrigeration capacity. This is a principally novel approach to ACS designing versus a traditional approach to cover the maximum current thermal duties and maximum annual refrigeration energy production accordingly that leads to inevitable oversizing.

In order to conduct this approach the advanced methodology of ACS designing includes addition stages focused to determine an optimal refrigeration capacity providing the maximum rate of annual refrigeration energy increment and minimum sizes of ACS accordingly. With this a relative annual refrigeration energy increment as its annual refrigeration energy, refered to refrigeration capacity required, is used as an indicator.

The rational value of design specific refrigeration capacity providing a closed to maximum annual refrigeration energy production without ACS oversizing is associated with the second maximum rate of annual refrigeration energy production increment within its range beyond the first maximum rate.

Proceeding from a differrent character of behaviour of thermal loads, the ambient air processing in the ACS is considered as a two-stage processing and includes a fluctuation range as the first (high-temperature) stage and comparatively stable range as the second (low-temperature) stage. The method to determined a threshold air temperature for rational distribution of design overall cooling capacity of ACS between two stages is developed. The stable thermal load value is chosen as design basic stable load part for cooling ambient air from threshold to target temperatures, for instance, from 15 to 10 °C. Accordingly, the remaining part of the overall rational thermal load is used for precooling the ambient air to the threshold temperature. The remaining booster load part needs the application of various methods of refrigeration capacity modulation as by applying variable speed compressors or other energy conserving technologies through using excessive refrigeration capacity, accumulated at decreased current heat loads, to cover the peak load, as an example.

The unstable and stable cooling load ranges could be covered by applying the refrigeration machines of different types: the first load range – by application of the chillers with high efficiency not strictly sensitive to load changes (with variable speed refrigeration compressors or waste heat absorption chiller), the second range – by operation of the chillers at about rated thermal loads (traditional refrigeration compressors without speed variation or ejector chilles as the simplest in design). Thus, the method allows at the same time to provide efficient operation of ACS at rational thermal loading and to chose corresponding scheme decisions of ACS and its refrigeration machines (chillers).

The approach of two-stage ambient air conditioning are quite reasonable for rational designing of ACS for space conditioning as independent central ACS and as the outdoor air processing subsystems to run the combined ACS more efficiently due to air feeding to indoor environments at the target temperature to avoid its overloading.

A two-stage ACS based on combined absorption-ejector chillers (AECh) could be considered as a perspective trend in ACS for energetic application (combustion engine intake air cooling) and for space conditioning as a subsystem of trigeneration installation. The application of such combined two-stage intake air cooling technologies are especially expedient for operation of ACS in temperate climatic conditions. They inable to provide deep intake air cooling with corresponding increased annual refrigeration energy generation in response to its duties or fuel saving or power output in the case of energetic application for engine intake air cooling.

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