Analysis of Technical Aspects of Choosing Air Conditioners and Temperature Parameter Control for Developing Air Conditioning Systems for Premises of Various Area at Fishery Enterprises

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Abstract. This article considers the theoretical and practical aspects of optimal construction of air conditioning systems for production enterprises of various area. The importance of this task is determined by the requirements on providing a comfortable work environment for maintenance personnel. The data on air conditioner operation, the theoretical surveys, and the factors that influence the temperature conditions and distribution of air streams are used to propose a number of recommendations for optimizing the construction of noise-proof conditioning systems. In addition, some guidelines are proposed for upgrading remote control systems for split conditioning systems.

1. Introduction

One of the key aspects of occupational safety is the maintenance of optimal air parameters in production premises. These parameters are temperature, humidity, and dust content in the air. Currently, the most favorable temperature in terms of comfort and on medical indications is provided by means of air conditioning equipment.

Air conditioning is the setting and automatic maintenance or adjustment of all or individual indoor parameters, for example, a certain level of temperature to provide optimal conditions for people’s general state and managing operating procedures [8].

Air conditioning is provided by a set of technical means referred to as air conditioning system (ACS). This system consists of air entrainment and treatment components: some of them (filters, heat exchangers, air humidifiers, air driers) serve to endow the air with certain parameters, move (fans) and distribute the air; others include cold and heat supply, automation, and remote control devices. ACS at large public and production buildings are usually serviced using integral automated control systems.

2. Materials and methods

The application of ACS at production premises has a number of peculiar features. Production premises, especially the ones used for fish processing, have unevenly distributed air streams. This is why, a situation occurs in which zones with uncomfortable conditions are formed, including too low or too
elevated temperatures, intensive cold air streams, etc. This issue can be partially tackled by choosing a suitable air conditioning system: a central ACS, a comfortable ACS, a reheat air conditioning plant, etc. In its respect, the choice is made on the basis of classifying conditioning systems and analyzing factors that influence the distribution of heat in a concrete premise.

Before getting down to the classification of conditioning systems, one should note that there is no conventional classification of ACS, which is due to the large number of wiring circuits, engineering and functional specifications that depend not only on the functionality of the systems themselves but also on their target facilities (conditioned premises) [11].

Modern conditioning systems can be classified by the following characteristics: [16]
- main purpose (target facility) (relief and process ACS);
- position of the conditioner relative to the conditioned premise (central and local ACS);
- availability of an in-system (embedded in the conditioner configuration) source of cold and heat (autonomous and non-autonomous ACS);
- action principle (direct-flow, recirculating, and combined ACS);
- number of serviced premises (local zones).

A classification according to other parameters is also possible. In addition to the above classifications, there are diverse conditioning systems for specialized operating procedures, including ACS with meteorological parameters varied in time according to a certain program.

**Relief ACS** are intended to generate and automatically maintain air temperatures, relative air humidity, air purity, and air flow rates that meet optimal sanitary and hygienic requirements on residential, public, and administration and amenity buildings or premises.

**Process ACS** are intended to provide air parameters compliant with production requirements to the full. The process conditioning of premises with people is provided, taking account of the sanitary and hygienic requirements on the state of the air environment.

**Central ACS** are supplied from outside with cold (with cold water or coolant), heat (with hot water, steam, or electricity), and electric power for actuating electric motors of fans, pumps, etc. Central ACS are mounted outside service premises and designed to condition one such premise, several of its zones, or a lot of separate premises. Central ACS are equipped with central non-autonomous conditioners made according to basic (standard) equipment layout drawings and their modifications.

**Local ACS** are developed on the basis of (non-) autonomous conditioners installed right in serviced premises. Local ACS are easy to install and mount.

**Autonomous ACS**, for example, split-system ACS, are supplied from the outside only with electric power. These conditioners have in-built compression chillers, usually run on Freon-22. Autonomous systems cool and dry the air, for which purpose the fan blows the recirculating air through chiller evaporators used as surface air-coolers. In the midseason and in winter these systems can heat the air with the help of electric heaters or by reversing the chiller’s operation in the so-called heat pump cycle.

Split-system conditioners can be considered the most basic option for decentralized maintenance of target indoor temperatures.

Air conditioning systems for production premises have several various configurations the most popular of which are single or multi-unit split-systems, central conditioning, chiller-fancoil water cooling systems, or multizonal systems.

The most expedient solution for small production premises of up to 200 sq. m in area are generic single-unit or multi-unit split-systems. The standard capacity of these conditioners from top specialized manufacturers allows maintaining optimal indoor air temperatures, usually, from 18 to 30°C, without forming low-temperature zones right near internal split conditioner units. Standard internal units can be placed so as to ensure air exchange across the entire air mass of the premise.

As an example, let us consider the specs of the LG LSK2460HL split conditioner: the cooling/heating rate is 24,000 WTE/h, the air circulation rate is 16/42 cubic meters per minute, the consumed capacity is ~ 2.7 kW. Four such conditioners mounted in the opposite corners of a premise of 250 sq. m in area
and four to five meters in height allow generating optimal temperature conditions (adjustable within a reasonable temperature range) and optimal air exchange conditions. It is also quite justified to mount simple split-systems in facilities of 300 to 500 sq. m in total area. Although this approach does have obvious drawbacks, it has some obvious advantages as well, notably, a relatively low cost, a high work delivery speed, and the option of stage-by-stage equipment installation. However, basic split-systems do not allow organizing centralized management. Central or multizone conditioning is a sensible solution for facilities of more than 500 sq. m in area. Conventional central conditioning involves supplying cool air from one ventilation plant to all the premises in the building along a network of air conduits. This approach has a number of advantages, including the synchronous provision of conditioning and ventilation and the possibility to save energy resources by using a regenerative heat exchanger in the plant. However, this method is not always optimal because the high air consumption necessary for compensating heat emissions requires using a system of heavy-gauge air conduits, which is not always possible. In addition, the air conduits must be insulated from heat to reduce heat losses and avoid the risk of condensate formation. The main issue with central conditioning is that it implies using low-heat capacity air as heat carrier. Another weakness of this method is that it requires using an automatic air consumption control system in every premise to precisely maintain the temperature in each single premise. This makes it way more complicated to manage the equipment, increases the system’s overall cost, and results in tough requirements on its maintenance. A modification of the described method is the VAV (variable air volume) system. Unlike conventional central conditioners, these plants allow regulating air supply to different premises. However, the enumerated flaws make it difficult to use these systems in premises of more than 500 to 1 000 sq. m. in area. The issue with conditioning these premises is solved using an intermediary coolant of high heat capacity, for example, water (in chiller-fancoil systems). Cold water is treated in the chiller and then supplied to the fancoils installed in the premises. Other coolants it is possible to use instead of water are freon and variable refrigerant flow (VRF) conditioning systems. Essentially, these are multi-systems with expanded capabilities. The outer unit (compressor-condenser unit) is either air-cooled or water-cooled. These systems have such advantages as a large number of internal units connected to the outer unit, the possibility to form an integral system with common control, and the capability to run internal units simultaneously for cooling and heating. Multizonal conditioners are used most often at such facilities as production premises divided in separate workshops and sections, and also production premises on different storeys of the building. In this case, the outer unit can be installed in a convenient location far away from workspaces. However, the broad capabilities mean certain difficulties with designing and choosing the assemblies, considering the need to maintain an optimal temperature and set the conditions for its adjustment. It is pointed above that VRF systems allow running the internal units in one such system for both, cooling and heating. In this case the air streams of various temperatures in one production premise are distributed among uninsulated areas placed close to each other. To choose conditioners and their capacity for ACS in design, it is proposed to follow the procedure described below. This express method is used mainly to develop ACS based on simple-design climate solutions such as split-system conditioners and also window-type and single-unit conditioners. To select a conditioner of required cold output, it is necessary to calculate the heat supplied to the premise with solar radiation, lighting, people, office automation equipment, etc. The components of the main heat leakages to the premise are enumerated below. 1)Heat leakages produced by the difference between the indoor and the outdoor air temperature as well as from the influence of solar radiation are found as: \[
Q_1 = V \cdot q_p, \tag{1}
\]
where \(V = S \cdot h; S\) is the premise area; \(h\) is the premise height; \(q_p\) is the specific heat load; it is taken that the heat leakage for the premise without sunlight is 30 to 35 W/m², the average heat leakage rate is 35 W/m², and the heat leakage to the premise with a large glazed area on the sunny side is 35-40 W/m². 2)Heat leakages \(Q_2\) produced by the office automation equipment used inside the premise.
The average value of \(Q_2\) is 300 W per full-packaged computer (or 30 % of equipment capacity).
3) Heat leakages $Q_3$ produced by the people in the premise.

It is usually taken for calculations that the heat leakage rate is 100 W per capita in the office and 100 to 300 W per capita in restaurants and premises, where people take to physical labour;

$$Q = Q_1 + Q_2 + Q_3.$$  \hspace{1cm} (2)

The calculated heat leakages are added to 20% of unaccounted heat leakages so that

$$Q_{\text{ttl}} = (Q_1 + Q_2 + Q_3) \cdot 1.2 \text{ W}.$$  \hspace{1cm} (3)

If the premise has additional heat-emitting equipment (electric cookers, production equipment, etc.), the given calculation must take account of the respective load as well.

The main user-relevant indicator of comfort and indoor air quality is indoor temperature. The so-called optimal indoor air temperature used in the design of air conditioning systems ranges from 20 to 25°C in residential, public, and administration and amenity buildings, depending on the time of the year and the speed of indoor air streams. According to GOST 30494-26, the optimal microclimate parameters are those values of microclimate indicators that ensure the sensation of comfort and normal body thermal status at the minimal stress of thermal control mechanisms for at least 80% of the people indoors at regular and consistent exposure.

The design engineer uses this range to choose the design indoor air temperature against which he determines the indoor thermal balance in the warm and the cold part of the year as well as in the midseason. Moreover, the maximal and the minimal indoor air temperature are often chosen from the range of optimal temperatures in the warm and the cold period, respectively, for the purpose of power supply.

This choice of design temperatures is absolutely adequate from the standpoint of the existing reference documents and broadly used in the design of conditioning systems. In case of multizonal systems (VRF, VAV, chiller-fancoil system, sometimes, split-systems), however, this choice makes it impossible to sustain the optimal air parameters in serviced premises.

This is caused by the fact that these systems allow the user to individually choose indoor temperature values necessary for a given premise. The range of necessary indoor temperatures is fairly broad; in most systems, it extends from 18 to 30°C. The temperature is chosen at random, which depends on individual thermal control peculiarities. The chosen temperature often falls out of the range of standardized optimal indoor air parameters, which is conditioned by the very definition of optimal microclimate parameters that are optimal only for 80% of the people. The statutory indicators calculated for the average person can be used only in designing large premises for lots of people, for example, cinemas, conference rooms, and sports centres. Multizonal conditioning systems providing comfort to particular individuals must be designed, taking account of their subjective characteristics.

The procedure of attaining comfortable specifications is described in work [1].

3. Results

It is usually accepted in practical calculations of air stream distribution that jet speeds and temperatures drop according to the regularities set for free jets (Fig. 1). In multizonal conditioning, however, the interaction of jets that will redistribute air streams and temperatures is inevitable [13]. The jet’s behavior will depend, first of all, on the correlation of forces the most critical of which for the task in question will be inertia, viscosity, gravity, and pressure. Surface tension, elastic strains, and unsteady motion forces can be ignored because our case covers real stationary gas streams. The relative change in air volume and, therefore, density, will be considered negligibly small because conditioners in on mode have a narrow range of temperature variations.

Then the main criteria of similarity that characterize relations of applied forces will include

1) Reynolds criterion that reflects the inertia-viscosity ratio as

$$\text{Re} = \frac{ul}{\nu},$$  \hspace{1cm} (4)

where $u$ and $l$ are the characteristic speed and size, $\nu$ is the kinematic viscosity factor;
2) Froude density coefficient (reverse to Archimedes criterion) that characterizes the inertia-buoyancy force ratio as

$$\frac{Fr_p}{\rho} = \frac{u^2\rho}{gl(\rho_0 - \rho)},$$  \hspace{1cm} (5)

where $\rho$ and $\rho_0$ are the jet and indoor air density, $g$ is the acceleration of gravity;

3) Eulerian number found as

$$Eu = \frac{p}{\rho u^2},$$  \hspace{1cm} (6)

where $p$ is the pressure in the fluid.

According to the experimental investigation results presented in the paper by V. A. Bakharev and V. I. Troyanovskii, the air chute sizes that exclude the influence of side walls on jet spread must meet the following ratio:

$$\frac{\sqrt{\omega}}{d_0} \geq 71.5,$$  \hspace{1cm} (7)

where $\omega$ is the passage area of the air stream indoors.

Usually, the criterial equation in the theory of similarity is recorded as

$$Eu = f(Re, Fr_p),$$  \hspace{1cm} (8)

which allow formulating the research task as to determine the influence of friction, gravity, and inertia on pressure variations in the air stream.

Let us distinguish the influence domains of each of the indicated criteria and respective forces.

According to the jet theory, turbulent jets are self-similar, beginning from $Re \geq 200$. In this work’s tasks self-similarity means the independence of the process from Re numbers and, therefore, from viscosity forces.

The significance of the $Fr_p$ criterion has no unequivocal definition. It is known from (Abracham, 1972) that at $Fr_p \geq 20$ buoyancy forces stop affecting the jet spread and the streams can be treated as turbulent nonbuoyant. At $20 > Fr > 4$ the jet spread is influenced by both, gravity and inertia forces; at $Fr < 4$ the dominant forces are buoyancy forces and the jets go down or up, depending on the sign of the difference between the jet and the ambient temperature.

Abracham’s investigations have allowed finding critical Froude’s air stream numbers; once these are exceeded, the jets become self-similar relative to buoyancy forces, whereas inertia forces begin to exert the dominant influence that determines the regularities in the jet’s speed drops and geometrical parameters.

In multizonal conditioning air streams can spread as jets in an unbounded space (Fig. 1) or as jets with circulating zones (Fig. 2). These zones begin to form when the values of the Bakharev-Troyanovskii criterion are exceeded.

The pressure for free jets spreading in a space with infinitely remote boundaries is taken constant, and the equation of motion for the jet spreading along the x axis (Fig. 1) is recorded as Eq. 8

$$u \frac{\partial u}{\partial x} + v \frac{\partial u}{\partial y} = \pm 2c^2x^2 \frac{\partial^2 u}{\partial y^2} \frac{\partial^2 u}{\partial y^2} \hspace{1cm} (9)$$

where $V$ is the kinematic viscosity factor, $c$ is the coefficient of turbulent mixing in Prandtl’s free turbulence theory.
Prandtl’s, Taylor’s, and Kármán’s half-empirical turbulence theories have been used to develop self-similarity solutions that allow attaining the results close to the test data for pairings of half-infinite flat parallel streams, turbulent free jets, and tracks at a large distance from streamlined bodies. The drop rate of the maximal speed along the axis, jet width, initial section length, and speed and temperature profiles for such jets are determined according to regularities for free boundary layers. The successes in describing free jets and tracks as part of half-empirical solutions apply, to a lesser extent, to more complex solutions such as the system of interacting jets with circulating zones (Fig. 2). Jets with circulating zones form when the stream expands in a bounded space. The indicated limitations can be created by the walls on the sides and butt ends of jets and by the neighbouring jets that create low pressure gradients the value of which found on the basis of D. Bernoulli’s equation corresponds to stream speed values.

Thus proper correlations for calculating speed and temperature profiles and impurity concentrations must be proposed for each of the highlighted jet spread regions (nonbuoyant turbulent jet Fr>20, buoyant turbulent jet Fr<4, and intermediate jet 20<Fr>4). It is assumed at a first approximation that these profiles coincide.

As shown above, turbulent nonbuoyant air jets are calculated using the results of solving free turbulence equations. The results of the theoretical and experimental investigations conducted by C. J. Chen and W. Rodi in 1980 [14] can be used for buoyant and mixed jets without zones of circulation. Work [Bolshakov, Konstantinov, Popov et al., 1984] presents a list of relations for calculating self-similar jet flows, turbulent buoyant jets in a stationary, an unstratified, and a linearly stratified medium, and also turbulent buoyant round jets in a stationary unstratified medium, taking account of the influence of the envelope. In this paper the initial jet pulse is usually ignored at $Fr_p \to 0$ and the flow is referred to as blast (plume in foreign literary sources). At $Fr_p \to \infty$ buoyancy effects are ignored and the flow is referred to as jet. In addition, the jet flow ($x' < 0.50$), the intermediate ($0.5 < x' < 5$), and the plume zone ($x' > 5$) are distinguished in the positive-buoyancy round jet, where

$$x' = \frac{x+x_o}{d_o} \left(\frac{\rho_o}{\rho_e}\right) \frac{u_o}{\sqrt{\frac{\rho_o}{\rho_e}g d_o}}$$

(10)
The successes in describing the spread of various jets, jets with zones of circulation included, were achieved by using model tasks based on the joint solution of equations of motion, equation of turbulent energy balance $k$, and balance equation of turbulent energy dissipation $\varepsilon$.

For the needs of practice, however, half-empirical correlations are used most often; according to them, circulating zone length $L$ and distance to whirlpool centre $L_u$ (Fig. 2) are found as

$$L = 3B, \quad L_u = 2B.$$  \hspace{1cm} (11)

The distribution of all of the parameters along two thirds of the whirlpool zone length (to the whirlpool centre) corresponds to the distribution in a free jet; then the air stream sharply expands to the limiting surfaces or to the line of contact with a neighbouring jet. The speed profile in this part of the jet also remains universal but the profile becomes more slanting.

In view of the foregoing it can be concluded that the use of multizonal systems to condition large production premises makes it necessary to rightly choose the design air temperature. The approach to solving this problem is exposed in work [2].

Considering that the air space of any premise can be presented as a closed heterogenous environment, the existence of air streams with various temperatures will lead to the heat exchange that can be tentatively described as

$$P = \xi S (Q_2 - Q_1) + mc \frac{d}{dt} (Q_2 - Q_1),$$  \hspace{1cm} (12)

where $P$ is the absorbed (released) capacity;
$\xi$ is the heat transfer coefficient;
$S$ is the heat exchange surface;
$Q_2 : Q_1 : m$ are the temperature and mass parameters of the environment.

The heat exchange will affect the steady-state temperature, especially at a significant difference in the steady-state temperatures of inner units of a multizonal system. The aspects of controlling these parameters are exposed in work [3].

The temperature control issue matters in engineering solutions applied in the design of conditioning systems. They can be controlled in an optimal manner when the true temperature in a certain point of the internal ecosystem is known. In addition, the actuating devices of the conditioning system are tied with the data from thermal sensors installed in the system’s devices.

The device within can be represented as a closed heterogeneous environment.

The temperature control in heterogeneous closed environments can be the parameter to which one can confine the diverse processes of thermal change in both, various engineering procedures and macrosystems, for example, the control over the air temperature in air-conditioned amenity premises.

To make the right choice of a circuitry and design solution of a temperature control system, optimal for a concrete engineering procedure or macrosystem, it is necessary to distinguish the main criteria of controlling devices. It is expedient to include in these requirements the range of measured temperatures, accuracy of control, delayed action, and thermal pollution level of a measured object.

The range of measured temperatures can be found as

$$\Delta \Theta = \Theta_2 - \Theta_1,$$  \hspace{1cm} (13)

where $\Theta_2$ is the highest control temperature;
$\Theta_1$ is the lowest possible control temperature.

The range of measured temperatures must be taken into account when choosing thermal elements of sensors for control instruments. The accuracy of control is found as

$$\sigma = \frac{\Theta_2 - \Theta_1}{\Theta_1} \cdot 100\%,$$  \hspace{1cm} (14)

where $\Theta_2$ is the measured temperature;
Ө₁ is the true temperature.

The accuracy of control is affected most of all by circuitry and design solutions chosen for control devices [4, 5].

Heat retention has the strongest influence on the accuracy of control, especially for express control and intermittent control with short time spans. The need for solving the heat retention problem arises in two cases. In the first one the sensor with mass \( m \) and specific heat capacity \( C \) is located in a medium with constant temperature \( \Theta₁ \) and sets into action at instant \( t₁ \). The sensor in on mode releases capacity \( p \) (active sensor), and its temperature \( \Theta₂(t) \) begins to rise. When the balance with the medium is reached, the temperature stops changing. The process can be tentatively described as

\[
P = \xi S (\Theta₂ - \Theta₁) + mc \frac{d(\Theta₂ - \Theta₁)}{dt},
\]

where \( S \) is the heat exchange surface (sensor surface area plus lead-in parts area); \( \xi \) is the heat transfer coefficient.

This equation corresponds to the aperiodic link equation with the time constant found as

\[
T = \frac{mc}{\xi S}.
\]

In the second case the sensor with \( m \), heat capacity \( c \), and temperature \( \Theta₂ \) at instant \( t \) is placed in a controlled medium with temperature \( \Theta₁ \). As a result of the heat exchange with the medium, the sensor temperature begins to tend to \( \Theta₂ \). This process is described as

\[
\Theta₁ = \Theta₂ + \frac{mc}{\xi S} \cdot \frac{d\Theta₂}{dt}.
\]

The experimental curve of the transition process in the heat converter is similar to the plot of the transition process in the aperiodic link. The difference between the curve and the aperiodic link plot is that in initial section \((t₁ - t₂)\) of the curve subregular mode related to the distribution of heat inside the sensor body is observed plus the thermal gradients correspondent to the unidirectional flow of heat takes place are established. The subregular mode section curvature depends on the uniformity of the sensor’s structure and can be changed on intention by using a sensor with a high non-uniformity of masses. The regular temperature setting mode is observed in section \((t₂ - t₃)\), whereas a steady-state thermal equilibrium occurs after \( t₃ \).

The irregular mode takes fairly little time at a small nonuniformity of masses in the sensor, and the thermal lag design for the sensor can be conducted as for the aperiodic link. The thermal time constant is determined by the full heat capacity of the converter and the conditions of the heat exchange between the converter and the ambient medium; this is why, one and the same converter has different time constants, depending on heat-exchange conditions. To calculate the time constant, it is necessary to find the overall heat capacity of the elements included in the sensor. Approximate calculations can be conducted proceeding from the average heat capacity of metals within 400 to 600 J/kg·K, the heat capacity of inorganic insulation materials (mica, china) of 800 to 1 000 J/kg·K, and the heat capacity of organic materials (resin-dipped fabric laminate, SRB paper laminate) of 1 200 to 1 400 J/kg·K.

The heat transfer factor depends on the medium, sensor surface treatment, and convection characteristics of the control medium. Persistent variables differ in very broad limits. Commercial heat meters have a time constant of three to six minutes. To shorten this period, it is expedient to use p-h thermal sensors because these are much lighter and, therefore, have a smaller heat retention.

The general analysis of the choice of temperature-sensitive sensors for designing circuits for heat meters meant to control the temperature in closed heterogeneous environments must rely on choosing the main criteria of assessing a particular kind of sensors for applicability. Some of these criteria are the range of temperatures, required accuracy of control, delayed action, allowed thermal pollution level. It is also expedient to assess sensors of various kinds against the chosen criteria by means of computer programs.
The above exposed considerations applied to macrosystems, in particular, to split air conditioning systems used in closed premises, allows optimizing the conditioner management system and cut energy expenditures.

4. Discussion
As a rule, modern conditioning systems have remote control consoles (RCC) that allow the operator to change the system’s running modes by directly manipulating the operating controls on this console [12, 17].

The RCC of a split conditioning system has an on button, an off button, buttons for different running modes (cooling, heating, drying), buttons for changing the fan’s blow angle and intensity, and also temperature-setting buttons. The RCC electronic circuit converts preset parameters to a coded signal transferred with an infrared beam to the inner unit receiver. After a certain temperature is set using the RCC, this mode sets up within a certain period of time, when the air blown by the internal unit fan reaches the preset temperature, because the temperature sensor (thermal resistor) is installed right in the internal unit radiator. In this case, the temperature in the zone, where the operator with the RCC is found, will differ from the preset temperature. For the diagram of the control based on temperature parameters see Fig.3.

![Diagram](image)

**Figure 3.** Control based on temperature parameters: 1 is the receiver of control signals; 2 is the control circuit; 3 is the temperature sensor; 4 is the RCC with temperature-setting buttons.

The difference between the temperature in the zone, where the operator with RCC is found, and the preset temperature depends on the conditioner’s internal unit, the air stream direction angle, the existence of eddy zones (walls or items on the way of the air stream), etc. That said, the actual temperature may differ from the preset one by several degrees, which does not allow creating comfortable conditions for the personnel in the operator’s zone. The difference between the actual and the preset temperature is illustrated by the plot in Fig. 4.
Figure 4. Changes in the difference between the actual and the preset temperature, depending on
the distance and the stream angle.

This plot is valid only at a certain difference between the actual and the preset temperature. The greater
is this difference, the more pronounced is the nonlinearity in the temperature rise along the plot’s curves.
Sometimes, it is expedient to use an RCC with a corrective thermal resistor connected by the control
signal to the main thermal resistor in the internal unit. This configuration is fairly easy to implement by
introducing minor modifications to the RCC because the basic RCC package has a temperature sensor
for informing the operator about the ambient temperature.
This upgrade may allow improving labour conditions for people who are found in the conditioned zone
and use the RCC.

5. Conclusion
The above exposed considerations are an attempt to determine an approach to solving the problem of
optimizing the choice of conditioning systems for production premises. The aspects of calculating air
streams inside large premises have been considered. The influence of thermal error with modifications
in remote control system has been analyzed. The recommendations for upgrading remote control
systems are provided.
When designing conditioning systems for large production premises, it is expedient
to make a
preliminary mathematical modeling by probabilistic techniques for identifying internal temperature
fields and their distribution indoors. This will allow making a more optimal choice of type and capacity
of conditioning systems being designed.

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