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SPECIFICS OF AIR CONDITIONING IN THE GERMINATION PROCESS OF MALT

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Abstract: *Information is provided on the specifics of stabilizing the temperature of germinated malt in the spring-summer and winter seasons. The situation is complicated by the rated values of the grain mass temperatures ranging between 12 and 18º C, which are rather far from the ambient temperatures in the specified seasons with maintained relative humidity of air used for aeration. The possibility and advisability of using exhaust air masses for recuperation are estimated; calculation formulas are provided concerning the organization of recuperation modes in the aeration processes. Information regarding the thermodynamic air preparation in the winter by using open steam and the results of calculations in support of advisability of energy conservation through recuperation are provided. The thermodynamic peculiarities of preparing the air for aeration with the need to achieve the rated temperature and the relative humidity φ=100 % are demonstrated.*

Keywords: *germination, grain, malt, heat, air, aeration, temperature, humidity, recuperation.*

1. Introduction

The synthesis of advanced technology systems requires a comprehensive analysis in the courses and transformations of material and energy flows with the organization of their interaction by using appropriate information systems.

Such analysis may be useful even for quite simple systems, or rather for the traditional and conventional technologies. The latter entirely concerns the processes of grain mass germination in the production of malts. The presence of biochemical processes creates conditions and physical consequences, to which the germinated grain aeration system should respond. Biochemical changes trigger the release of heat and carbon dioxide. The secondary effects are the reduced relative humidity of the air that passes through the grain mass and the drying of the latter. However, it is a process disturbance, as grain humidity and temperature must be stabilized [1-3].

In view of the above, the purpose of this study is to overcome the thermodynamic incompatibility regarding the requirements to the interaction of grain masses and air streams for aeration.

2. Matherials and methods

The study is a theoretical one and is based on information about biochemical transformations in grain mass and the processes of energy and mass interchange with the aeration air [4-9].

3. Results and discussion

Even a cursory analysis of air preparation processes leads to the conclusion about the diversity of problems and solutions regarding the winter and summer seasons.

For the summer season, the mixing of fresh and recuperative air flows can ensure a nominal result in a considerable range of temperatures. The temperatures of recirculation air $t_{r.a.}$being entirely predictable, it is possible to determine the required ratios of their volumes with variable values of the fresh air temperatures $t_{f.a}$.

Suppose we have

$$\varepsilon = V_{f.a.} / V_{r.a.}, \tag{1}$$

where $V_{f.a.}$ i $V_{r.a.}$ are volumes of fresh and recuperative air.

Considering the latter dependence, the temperature of the selected volumes mixture will be found according to the formula

$$t_m = \frac{(1+\varepsilon)\, t_{f.a.} t_{r.a.}}{t_{r.a.} + \varepsilon\, t_{f.a.}}. \tag{2}$$

By using the above formula it is possible find the ratio of air volumes, which will satisfy the condition of obtaining the rated mixture temperature:

$$\varepsilon = \frac{t_{f.a.} t_{r.a.} - t_m t_{r.a.}}{t_m t_{f.a.} - t_{f.a.} t_{r.a.}}. \tag{3}$$

where $t_{f.a.}$ – the temperature of the fresh air; $t_{r.a.}$ – the temperature of regenerative air.

To assess the importance of using recuperative modes let us refer to the conditions of aeration air preparation in the winter season, the fresh air temperature being $t_{f.a.}$ = –20 °C. Calculations will be done for the volume $V_{f.a.}$ = 1000 m^3, with its final temperature being 16 °C. In this case the amount of heat energy to be transmitted to the air in the heater will be:

$$Q = V_{f.a.} c_{f.a} (16 - (-20)) = 46{,}740.4 \text{ kJ}.$$

The further part of calculations will be done for the conditions of grain germination in 8 drums, each containing 8,000 kg of green malt. The level of aeration for each drum is about 10,000 m^3 of air per hour. Then, in the absence of recirculation, the energy consumption under the conditions formulated above will amount to:

$$Q_d = Q \frac{10{,}000}{1{,}000} = 467{,}600 \text{ kJ/h},$$

and

$$Q_{8d} = 8Q_d = 8 \cdot 467{,}600 = 3{,}740{,}800 \text{ kJ/h}$$

for eight drums, respectively.

With the calorific value of natural gas amounting to 41,000 kJ/m^3, the gas consumption will be:

$$V_g = \frac{Q_{8б}}{41000\eta} = \frac{3{,}740{,}800}{41{,}000 \cdot 0.8} = 114 \text{ m}^3\text{/h},$$

where η is the efficiency factor of the air heating system.

Under the value ε = 1 gas consumption will be reduced by half, while a further increase in recirculation will improve the results.

The above part of calculations to find out the energy consumption for heating up the air has been performed in accordance with the provisions of thermodynamics. However, we do not take into account the process requirement of bringing the relative fresh air humidity to 100 percent.

We will consider the specifics of the course of operations by using Diagram I–d (Fig. 1).

The initial temperature t_1 and the relative humidity φ_1 of air on the diagram correspond to point 1, which also corresponds to the absolute moisture content d_1. The phase of air heating through heat-exchange surface is carried out provided that $d = const = d_1$. If we assume that point 2 in terms of the previously accepted value corresponds to $t_2 = 16°C$, then the further isenthalpic saturation process leads to point 3 on the diagram, which corresponds to an isotherm with the temperature t_3. It means that the task will fail through the above combination of heating and cooling. At the same time, a full-fledged option would be a semi-stage or even a single-stage process, which, however, will require preliminary determination of the positions of points 2 and 3.

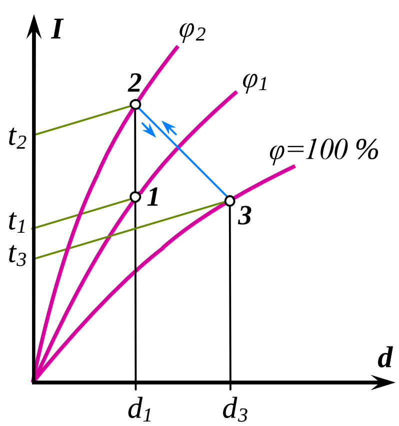

Fig. 1. Diagram I–d representing the single-stage air heating and humidification mode with the values $t_3 = t_{rated}$ and $\varphi = 100$ %

The calculations should reasonably begin with finding the position of point 3 on the curve $\varphi = 100$ %, while the second coordinate is the rated temperature isotherm t_3 (Fig. 2).

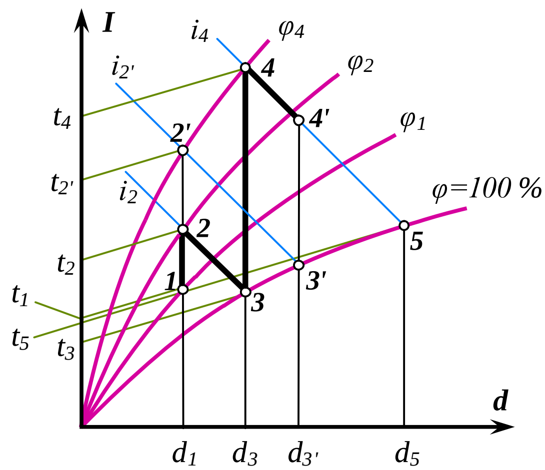

Fig. 2. Diagram I–d representing the two-stage air preparation process

Also, the absolute moisture content of the air d_1 should be known.

We can find the position of point 2 and the corresponding temperature t_2, to which air should be heated in the single-stage mode, by moving along the isoenthalpy from point 3 to the intersection with the ordinate, which corresponds to d_1.

The two-stage process is reflected in the diagram provided in Fig. 2.

In this case, the fixed points on the diagram are points 1 and 5, while the positions of points 2 or 2' and 4 or 4' are variable. However, the choice of particular laws in the air heating and humidifying processes does not influence the overall energy consumption results, as they are theoretically determined by the dependence

$$Q = V_{f.a.} c_{f.a.} (t_n - t_{f.a.}) + \\ + (d_{(f)} - d_{(i)}) r V_{f.a.} \rho_{f.a.}, \quad (4)$$

where $d_{(i)}$ and $d_{(f)}$ are the initial and final absolute air humidity.

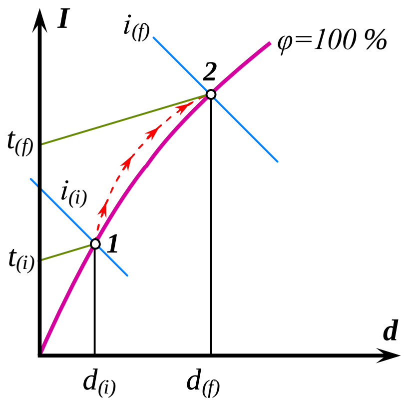

Fig. 3. Diagram I–d representing the air and steam mixing process

Based on the performed analysis we may conclude that it is possible and reasonable to combine fresh air heating and humidification operations by introducing steam into the latter. The course of such a process may be represented by the same Diagram I–d in Fig. 3.

The transition from point 1 to point 2 may coincide with or exceed the position of the curve φ = 100%. However, the final temperature $t_{(f)} = t_i$ must be reached along with the full steam saturation of air. It is obvious that the course of the process and its final results depend *inter alia* on steam parameters. In this case the following correlation must be met:

$$Q = V_{f.a.} c_{f.a.} \left(t_{(f)} - t_{(i)}\right) + i_{steam} M_{steam}, \quad (5)$$

where M_{steam} is the steam mass supplied to the process, and i_{steam} is the steam enthalpy.

Hence, from the perspective of achieving the rated temperature, the steam flow must be as follows:

$$M_{steam} = \frac{V_{f.a.} c_{f.a.} \left(t_{(f)} - t_{(i)}\right)}{i_{steam}}. \quad (6)$$

However, three conditions may arise, in particular:

$$M_{steam} > \left(d_{(f)} - d_{(i)}\right) V_{f.a.} \rho_{f.a.}; \quad (7)$$

$$M_{steam} = \left(d_{(f)} - d_{(i)}\right) V_{f.a.} \rho_{f.a.}; \quad (8)$$

$$M_{steam} < \left(d_{(f)} - d_{(i)}\right) V_{f.a.} \rho_{f.a.}; \quad (9)$$

Condition (7) corresponds to the flows ratio, under which one portion of steam will be condensed, while the other portion will become part of the air saturated to the extent of φ = 100 %. Whereas expression (8) involves the case of steam fully becoming part of the air, the rated air temperature mode is achieved in the situation corresponding to expression (9). However, the relative humidity in this case is φ < 100 %.

In the latter case complete saturation of the air with moisture should be achieved by spraying water of rated temperature.

It is obvious that in most cases a necessity arises to heat up the air and to increase its enthalpy in the winter time. For humid air, the latter shall be found based on the superposition principle, under which

$$i_{h.a.} = i'_{d.a.} + i_s, \quad (10)$$

where $i'_{d.a.}$ is dry air enthalpy, and i_s is steam enthalpy.

Substitution of values $i'_{d.a.}$ and i_s leads to the form

$$i_{h.a.} = c_p t + (2{,}500 + 1.93)t, \text{ kJ/kg}. \quad (11)$$

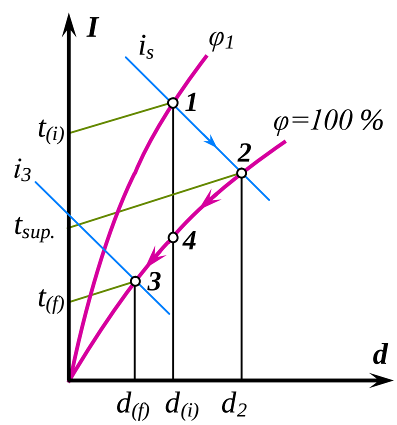

Fig. 4 Scheme for sequencing the operations in Diagram I–d

Since winter time is characterized by low values of absolute moisture content, air saturation up to φ = 100% and an increase in temperature are simultaneously performed operations. Thus the total energy consumption is determined by the difference of air enthalpies under final and initial thermodynamic parameters.

Air conditioning in the summer period has some distinctions, due to which the specifics and sequence of operations for changing the thermodynamic parameters should be considered. The coordinates of the point of initial thermodynamic parameters are determined by the values of temperature and relative humidity (Fig. 4). Thus the final values are also known and constitute the values $t_{(f)}$ = 10 °C and φ = 100 %.

Suppose that the specified coordinates and the relevant parameters in the diagram correspond to points 1 and 3, while point 2 corresponds to the parameters of fully saturated humid air with φ = 100 %. If saturation is taken as the first operation, its completion provides only for the rated relative humidity, while the rated temperature is not reached. To provide for the latter, it is necessary to continue cooling the saturated humid air.

The diagram (Fig. 4) shows that the subsequent transformation of the thermodynamic parameters of saturated humid air involves a decrease in the absolute moisture content from d_2 to $d_{(f)}$. It is obvious that, from a physical point of view, this means the presence of a steam condensing mode, the energy cost of which will be the condensation heat extraction and

$$\Delta i_{1-2} = 0; \qquad \Delta i_{2-3} = i_2 - i_3. \quad (12)$$

The result of this process will be a state of thermodynamic equilibrium of the humid air with parameters of point 3, while the load on the refrigeration unit will correspond to the value Δi_{2-3}.

In the diagram, air cooling without prior moisture saturation corresponds to the segment 1–4, and point 4 corresponds to temperature $t_4 > t_3$ and enthalpy $i_4 > i_3$.

Here we have

$$\Delta i_{1-4} = i_1 - i_4; \quad (13)$$

$$\Delta i_{4-3} = i_4 - i_3, \quad (14)$$

while the load on the air cooling system is

$$q_{k.y.} = \Delta i_{1-4} + \Delta i_{4-3} = i_1 - i_4 + i_4 - i_3 = i_1 - i_3. \quad (18)$$

As $i_1 = i_2$ in the isenthalpic process, the choice in favour of the first or second option does not affect the result regarding the cooling system load.

However, if there is no difference regarding the material balance, there are some distinctions concerning the heat balance. The cooling process following the arc 2-3 on the curve φ = 100 % must be accompanied by moisture condensation in the amount of $d_2 - d_3$. At the same time, the process of condensation requires the relevant conditions, condensation centers, energy security, creating an interphase surface, etc. Given the brevity of the said process and the above conditions, it may be concluded that the air stays in a metastable state, i.e. in a state of oversaturation with moisture.

The interaction of such air with the grain mass having the temperature 2-3° C higher should not be accompanied by condensation. However, as the air temperature increases, the level of thermodynamic disequilibrium is reduced and thus the drainage capacity is lowered.

4. Conclusion

Based on the analysis of air preparation for germinated grain aeration in the production of malt the following conclusions can be made:

1. Preparation of air flows for germinated grain aeration in the summer and winter seasons has significant differences not only in terms of the input

thermodynamic parameters but also regarding the techniques of achieving the specified output parameters;
2. In the winter period, it is reasonable to heat up the incoming flows of fresh air by using live steam;
3. The thermodynamic inconsistency of air in terms of temperature and relative humidity may be limited by increasing the intensity of aeration.

The use of recuperation modes used in grain aeration processes reduces energy consumption by 50 %. The results and recommendations may be applied to various types of malt houses.

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