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PROSPECTS OF USING SECONDARY ENERGY RESOURCES

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Abstract:
The paper deals with the peculiarities of thermodynamic analysis of technological systems of food production with generation of secondary steam. System analysis is based on information about production, equipment operation, modern scientific achievements and modeling methods in chemical, physical, mathematical and other sciences. The information about applied energy and thermodynamic criteria, criteria of optimization of systems, list of sequence of actions on improvement of systems is given. Secondary energy resources of food industries are often represented by thermal energy of liquid media or secondary steam. Regeneration of the secondary steam allows, during its subsequent condensation, to return the heat potential of the steam to the media. This means that in the systems where the increase in the thermodynamic parameters of steam due to its compression, there is the use of Carno's return cycle. Such use occurs both in closed and open cycles. The description of the physical processes accompanying the work of the thermocompressors is subject to mathematical modeling based on the laws of conservation of energy, mass conservation and conservation of impulses. The assessment of the suitability for using secondary energy resources is based on exergy methods in order to determine the directions for their further improvement. For identical pressures in the mixing chamber, the efficiency of the actual compressor is higher than the ratio of the ejection coefficients of the real and ideal compressors, since internal irreversible losses along with the reduction of the ejection rate result in the increase of the specific exergy of the compressed flow.

Key words: *thermodynamics, recuperation, secondary energy resources, secondary steam, enthalpy, exergy, energy transformation.*

1. Introduction

Functioning of technological systems of food production is provided in the presence of chemical, energy, mechanical and microbiological processes at different stages of existence. Their cumulative flow, structural and parametric interconnection require the involvement of system analysis to assess the effectiveness of technologies objectively and identify areas and prospects for improvement and optimization. Information on the construction and operation of productions, modern scientific advances and modeling methods in chemical, physical, mathematical and biological sciences is the basis for system analysis.
The project of the existing system at the stage of its creation and implementation has its reflection in the form of the target function, and its optimization is the ultimate goal for system analysis [1, 2]. In the general definition, optimization tasks are divided into parametric and structural. Technological development of individual subsystems leads to the conclusion that the determining role belongs to the choice of optimal structure of the system, the economic effect of which is an order of magnitude higher than the effect of optimal process control [3].

Determination of the extremum of the target function without specifying conditions for other values leads to the case of unconditional optimization, and in cases when it is necessary to take into account certain conditions that are superimposed on other values, there is a conditional optimization. The type of criterion for an optimal or target function is determined by a specific optimization task. The list of criteria used in system evaluations includes economic, energy, technical, environmental, reliability, flexibility, etc. In connection with the further interests of the study, we will give information on energy and thermodynamic criteria.

To energy criteria include:

- criteria for the maintenance of heat

$$\eta_m = Q_g / Q_p = \eta_{lg} / \eta_{lt}, \quad (1)$$

where Q_g, Q_p, η_{lg} and η_{lp} are heat fluxes and coefficients of heat loss from the medium giving (g) and perceiving (p) heat;

- heat utilization factor

$$\eta_{uh} = Q / Q_{max} \rightarrow \max, \quad (2)$$

where Q and Q_{max} are real and maximum possible heat fluxes in the heat exchanger, J;

- Kirpichov's criterion

$$E_к = Q / AL \rightarrow \max, \quad (3)$$

where Q – real heat flux; AL – the work of the superchargers;

- Glaser's criterion

$$\eta_т = Q / N \rightarrow \max, \quad (4)$$

where N – power of the supercharger, W.

The thermodynamic criteria include:

- criterion of thermodynamic reversibility (Grassman)

$$\eta_{TR} = \Delta A_p / \Delta A_g \rightarrow \max, \quad (5)$$

where ΔA_g and ΔA_p – increase in the robustness of given and perceived warmth, J;

- thermodynamic coefficient of Gui-Stodol

$$\eta_T = 1 - \frac{\Delta A_g + \Delta A_p}{A_g + A_p} \rightarrow \max, \quad (6)$$

where A_g and A_p – robustness of given and perceived warmth, J.

The choice of the optimization criterion for the investigated energy technology system should correspond to the transparent scientifically substantiated physical content and reflect the most important aspects of the process. Sama (1995) notes that notes that classical thermodynamics with its second law is the fundamental basis for the synthesis of both new systems and for optimization of the parameters and structure of the existing ones, and since the processes proceed on the macroscopic level, this allows to establish direct interconnections between the parameters of heat, work and efficiency criteria.

The comprehensive goal of the thermodynamic analysis of the system is formulated by the following sequence:

- assessment of the degree of thermodynamic perfection of the system;
- definition of the cause and source of imperfection;
- finding ways to eliminate imperfections;
- formulation of measures to improve the efficiency of the system, which correspond to the chosen criterion of effectiveness.

The list of possible areas for optimization of thermodynamic systems is related to energy, entropy, exergy, thermoeconomic and Pinch analysis. These methods lead to thermodynamic relations, which relate to the possibilities of intensive energy-absorption, however, the in-depth improvement of technologies of energy transformations requires more extensive use of heat engineering, technological and biochemical variations.

An assessment of the suitability of using secondary energy resources requires recourse to the second law of thermodynamics, in which the correspondence to all types of heat flows is characterized by the quality of indicators, that is, the ability to perform work.

Work is possible only in a system that is not in equilibrium with the environment, when the pressure and temperature in the system are greater than those of the environment with which the system interacts. As the work is done, the isolated system approaches the equilibrium state with the environment where the work is stopped.

The quality indicator of various types of heat (exergy) is the maximal ability of matter to work in such a process, the final state of which is determined by equilibrium with the environment.

The exergy method is used for the purpose of energy optimization of technological systems and determination of directions of their further thermodynamic perfection [5].

Real energy processes are irreducible and this irreversibility is the cause of their imperfection. However, the physics of the essence of the energy balance does not reflect the loss of the system from non-failure [6].

The general representation of the components of exergy is associated with the concept of entropy. The types of energy that can be completely transformed into others include mechanical, electrical and nuclear (for them entropy $s' = 0$).

In the cases of the chemical energy of intermolecular bonds, the chemical potential and the heat $s \neq 0$, their transformations occur with certain losses.

In the general list of technological operations in the food industry, cases of heat treatment of media with intensive generation of secondary steam are presented, the recuperative return of the potential of which in the system greatly limits the costs of primary energy resources. Among the classical cases, it is possible to include processes of brewing beer wort, evaporation of beet juice, syrups, bards, milk, crystallization of sugar, etc. The hardware design of processes at the same time depends on the ultimate goal, which can take into account the technological features of their flow or is only a matter of concentration of solutions.

The classic theory of evaporation has been reflected in the processes of condensation of solutions during boiling, which occurs in volume, when the steam pressure of the solvent is equal to the total pressure in the ingenious volume. The feature of the process is the stability of the boiling temperature at this pressure and a certain composition of the solution. In this case, water evaporates and is removed in the form of secondary steam, and dissolved substances remain in solution in an un-changed amount. These processes are carried out in evaporators of technological purpose, and if their implementation takes place under vacuum, then such devices are called vacuum machines. For one-time evaporation in a separate evaporator, the cost of a heated steam is about 1 kg per 1 kg of evaporated water, which is economically unprofitable. The use of secondary steam is carried out in multi-body evaporation units or in evaporators with heat pumps.

In the latter case, the technological apparatus becomes a component of the system, which is the realization of the heat pump, since it simultaneously serves as an evaporator and a condenser [7-11].

2. Matherials and methods

The tasks of static optimization are solved for processes in tranquil modes with the decision of questions of creating optimal models, and in cases of dynamic optimization, the creation and implementation of systems with optimal control over non-sustained operating modes is carried out.

3. Results and discussion

Fig. 1 shows the scheme of the system with mechanical compression of the secondary steam and Fig. 2 is the corresponding diagram in coordinates *T-s.*

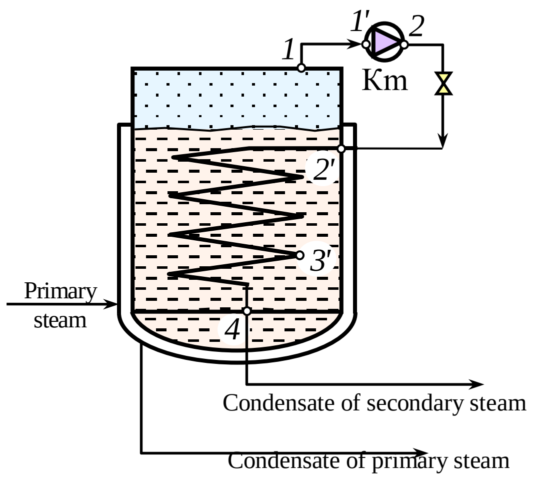

Fig. 1. Scheme of system with mechanical compression of secondary steam

The primary steam supplied in the heating shirt provides an increase in the energy potential of the solution to the boiling point, and the prescribed pressure level is stabilized, which results in the inclusion of a portion of the secondary steam with the compressor. Coupling of steam is accompanied by an increase in its temperature.

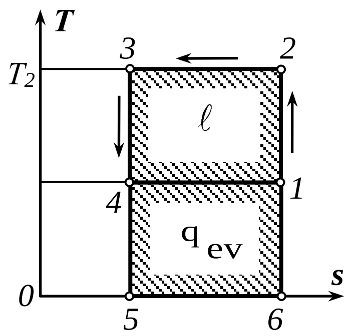

Fig. 2. Diagram of the system with mechanical compression of the secondary steam in coordinates T-s

This means that the energy potential of the secondary steam, determined at the temperature T_1, is supplemented by the compression energy. Taking into account the speed of compression in the first approximation, we will consider this process as adiabatic, and the theoretical work of the compressor ℓ is equal to the difference between enthalpies i''_2 and i''_1, that is,

$$\ell_{theor} = i''_2 / i''_1 \qquad (7)$$

From this assumption comes the dependence

$$T_2 = T_1 \left(\frac{P_2}{P_1} \right)^{\frac{k-1}{k}}$$ for an adiabatic process with gas it is possible to be transferred to the second steam. However, for the compilation of energy balances and, taking into account the possible need to increase the motive factor in the heat transfer process, it is possible to get into the zone of correlation between parameters i''_2 and the heat of vaporization r, in which the indicated balances are not achieved. An increase in the temperature of the secondary steam, which corresponds to a certain value i''_2, means simultaneous reduction of the heat of vaporization r and hence the increase of the difference should be deduced from it.

$$\Delta = i''_2 - r \qquad (8)$$

If the thermodynamic parameters of the wort melting remain constant during a flow of time, this should correspond to a stabilized input flow of secondary steam. In this case, the condition must be fulfilled (Fig. 3). It can be seen from the figure 3 that such condition can be achieved, so that the flow of the secondary steam will be stabilized.

$$\Delta = i_2'' - r = i_1'' \qquad (9)$$

At the same time, technological requirements, for example, of wort boiling may apply to situations in which the flow of secondary steam can increase or decrease. Obviously, the achievement of such changes will be related to the parameters Δ and pressure P_c, and obtaining the given stabilized parameters in the compressor with the use of modern methods of regulating does not cause the complexity.

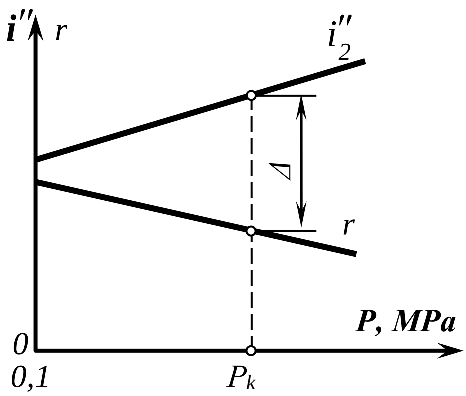

Fig. 3. Diagram to determine the parameter Δ

In Figure 1 the system is shown with two separate heating surfaces. Obviously, such an option is possible in its existence, but the prospect should have a variant, in which the transformed flow of the secondary steam returns to the heating shirt. This is due to the fact that the superficial heating shirt of the wortboiling apparatus, in its absolute value, is approaching the surfaces of the apparatus itself, which are in direct contact with the wort. Hence the conclusion is that the creation of an additional cooking surface with equivalent parameters looks problematic, which is amplified by the existing limitations of the temperature of the heating surface.

However, the system with an additional heating surface can be used in devices with significant dimensions, for which designers are required to provide internal additional heating surfaces or external circulating contours with such surfaces. Although the root cause of such problems is known and has the form of a correlation of the heating surface to the volume of the apparatus, however, in such cases the use of heat pumps based on the mechanical compression of the secondary steam is perspective.

Of course, the presence of the internal surface of the heating should affect the hydrodynamics of the steam-liquid medium in the apparatus. By the traditional implementation of cylindrical apparatus with semi-spherical bottoms, a stable circulation circle with a rising peripheral flow and a downstream center is achieved. The introduction of an additional heating surface does not displace the ascending circular peripheral flow, but it is accompanied by a central upstream flow, and the downstream flow will be formed between the two ascending.

However, the features of the heating surfaces and their arrangement do not ne-glecte the main conclusion concerning the possibility and necessity of full use of the potential of the secondary steam. Moreover, such use must be carried out within the closed circuit circuits.

The practical implementation of this idea continues to interfere with the possible presence of air in the secondary steam. Such air in the system of heat transfer forms air "coat" and requires special intervention.

The limitations of this disadvantage are achieved by blowing out or discharging part of the steam-water mixture into the environment or providing the whole system of airtight nature so that the air does not enter the circulation circuit along with the secondary steam.

Recent features of the arrangement of systems with closed circulating power supply circuits refer to cases in which the output of the formed secondary steam is compulsory. This leads to the use of a heat-exchanging surface. An example with a wortboiling device is not typical, but it was considered in connection with the need to achieve three results of heat treatment, namely, the achievement of an aseptic state, a deep extraction of substances from the hops, and output from the process of 10% liquid phase in the form of secondary steam.

The choice of the thermodynamic parameters of the system with the mechanical compression of the secondary steam is carried out taking into account the desired temperature difference on the surface of the heat transfer. Received characteristics of saturated water steam in the range of possible use are given below (Table 1).

Table 1

Parameters of saturated water steam (by pressure)

P, MPa	t, °C	i', kJ/kg	i'', kJ/kg	r, kJ/kg	S', kJ/(kg·K)	S'', kJ/(kg·K)
0.100	99.640	417.40	2675	2258	1.3026	7.360
0.120	104.81	439.40	2683	2244	1.3606	7.298
0.140	109.33	458.50	2690	2232	1.4109	7.246
0.160	113.32	475.40	2696	2221	1.4550	7.202
0.180	116.94	490.70	2702	2211	1.4943	7.163
0.200	120.23	504.80	2707	2202	1.5302	7.127
0.220	123.27	517.80	2711	2193	1.5630	7.096
0.240	126.09	529.80	2715	2185	1.5929	7.067

Numerical energy relations in this case are determined by the equations (7) - (9), and the coefficient of recuperative return of the energy of the secondary steam is reduced to the form:

$$\beta = \frac{r_{(c)}}{\ell_{theor}} = \frac{r_{(c)}}{i''_2 - i''_1} \qquad (10)$$

where the parameter of the compressed steam corresponds to the heat of condensation $r_{(c)}$. During the transition to the use of a thermocompressor in the form of an ejector (Fig. 4), the necessary pressure in the system is formed by a combination of the primary and secondary steam to form a mixture that corresponds to the equation of thermal and material balance:

$$\begin{cases} m_{mix}\, i''_{mix} = m_{prim}\, i''_{prim} + m_{sec}\, i''_{sec}; \\ m_{mix} = m_{prim} + m_{sec}, \end{cases} \qquad (11)$$

where m_{mix}, m_{prim} and m_{sec} – respectively the mass of the mixture, primary and secondary steam.

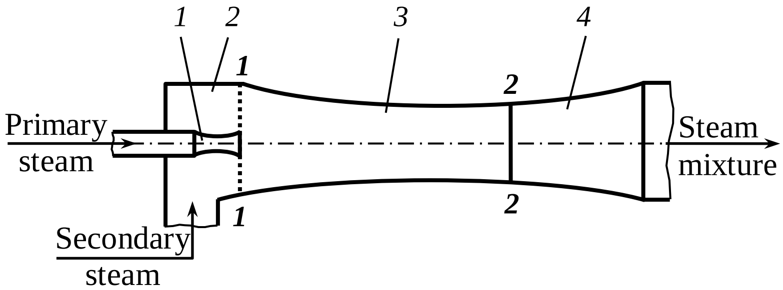

Fig. 4. Scheme of the ejection thermocompressor: 1 - working nozzle; 2 - receiving camera; 3 - mixing chamber; 4 – diffuser

Hence the relation between masses or mass flows:

$$m_{prim} = \frac{m_{sec}\left(i''_{mix} - i''_{sec}\right)}{i''_{prim} - i''_{mix}} \quad (12)$$

In the absence of a heat pump, the primary steam consumption to generate the secondary one will be:

$$m_{prim} = \frac{m_{sec} i''_{sec}}{i''_{prim}} \quad (13)$$

The perfection of jet apparatus-transformers is determined by the magnitude of the efficiency, which is the ratio of the exergy received by the secondary steam flow to the exergy spent by the flow of the primary steam:

$$\eta = \frac{u\left(e_{mix} - e_{sec}\right)}{e_{prim} - e_{mix}} \quad (14)$$

where e_{mix}, e_{prim} and e_{sec} – specific exergy of mixture, primary and secondary streams respectively; $u = m'_{sec}/m'_{prim}$ – ejection coefficient; m'_{sec} and i m'_{prim} – mass flows of the secondary and primary steam respectively.

Specific exergy is determined by the dependence:

$$e = i_0 - i_{env} - T_{env}\left(s_0 - s_{env}\right), \text{ kJ/kg} \quad (15)$$

where i_0 and s_0 – respectively specific enthalpy and specific entropy of the primary steam in the isoentropically inhibited state; i_{env} and s_{env} –specific enthalpy and the specific entropy respectively of the primary steam in a state of equilibrium with the environment.

Let's compare the efficiency of the ideal and real jet transformers at identical initial parameters of the primary and secondary steam:

$P_{prim} = idem$; $e_{prim} = idem$; $P_{sec} = idem$; $i_{sec} = idem$; $P_{mix} = idem$. In fig. 5 we have a diagram in the coordinates i-s with the specified parameters of flow interactions.

The state of the primary flow of steam in front of the apparatus is determined by the point p: entropy s_{prim}; enthalpy i_{prim}; pressure p_{prim}. The state of the secondary steam flow in front of the transformer is determined by the point n: entropy s_{sec}; entalpiy i_{sec}; pressure p_{sec}.

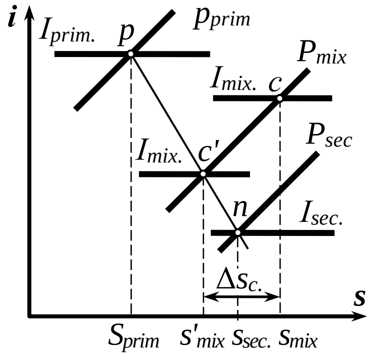

Fig. 5. Parameters of the interaction of the flow of ejection thermal transformer

In terms of the coefficient of ejection u' in an ideal apparatus, enthalpy i'_{mix} is determined on the basis of the law of conservation of energy:

$$i'_{mix} = \frac{i_{prim} + u' s_{sec}}{1 + u'} \quad (16)$$

In an ideal jet device without loss, the entropy of the system does not change. Therefore, the entropy of the compressed stream is:

$$s'_{mix} = \frac{s_{prim} + u' s_{sec}}{1 + u'} \quad (17)$$

The point c' in the section p-n, which

connects in the diagram points p and n corresponding to the initial thermodynamic parameters of the system satisfies equations (16) and (17).
Obviously, in a nonideal ejection machine, the process takes place with losses, so the entropy of the system increases. Then some point c may correspond to a state of compressed stream, which entropy is $s_{mix} > s'_{mix}$.
Under the same pressure ($p_0 = idem$), the enthalpy of the compressed stream in the real apparatus is $i_{mix} > i'_{mix}$. Herewith:

$$i_{mix} = \frac{i_{prim} + us_{sec}}{1+u} \quad (18)$$

As $i_{prim} > i_{sec}$, then $u < u'$ corresponds to the condition $i_{mix} > i'_{mix}$. This means that for given pressures p_{prim}, p_{sec} and p_{mix} the ejection coefficient of actual jet engine is less than ideal.
Taking into account the condition (14) for the ideal compressor, we will write:

$$\eta = \frac{u'(e'_{mix} - e_{sec})}{e_{prim} - e'_{mix}} \quad (19)$$

where u' – the ideal compressor ejection coefficient; e'_{mix} – specific exergy of the compressed stream in the ideal compressor.
Dividing the condition (14) into (19), we obtain:

$$\eta = \frac{u}{u'} = \frac{e_{mix} - e_{sec}}{e'_{mix} - e_{sec}} \cdot \frac{e_{prim} - e'_{mix}}{e_{prim} - e_{mix}} \quad (20)$$

It is obvious that $e_{mix} > e'_{mix}$.

Condition (15) follows:

$$\begin{aligned} e_{prim} - e'_{mix} &= i_{mix} - i'_{mix} - \\ &- T_{env}(s_{mix} - s'_{mix}) = \\ &= \Delta i_{mix} - T_{env}\Delta s \end{aligned} \quad (21)$$

where i_{mix} and s_{mix} – enthalpy and entropy of the medium in a state that corresponds to a point c; i'_{mix} і s'_{mix} – the same in the state of the point c'.
But $\Delta i_{mix} = T_{med}\Delta s_{mix}$; approximately $T_{med} = \frac{T_c + T_{c'}}{2}$, where T_c and $T_{c'}$ – appropriate temperatures to the points c and c'.
Then

$$e_{mix} - e'_{mix} = (T_{med} - T_{env})\Delta s \quad (22)$$

at $T_{med} > T_{env}$ $e_{mix} > e'_{mix}$. As $e'_c < e_c$, then $\frac{e_{mix} - e_{sec}}{e'_{mix} - e_{sec}} > 1$ and

$$\frac{e_{prim} - e'_{mix}}{e_{prim} - e_{mix}} > 1$$

Hence,

$$\eta = \frac{u}{u'}. \quad (23)$$

Condition (23) indicates that for the same pressures p_{mix}, the efficiency of the real compressor is greater than the ratio of the ejection coefficients of the real and ideal compressors, since internal irreversible losses along with the reduction of the ejection coefficient result in the increase of the specific exergy of the compressed flow.

4. Conclusion

Secondary energy resources of food industries are often represented by thermal energy of fluid media or secondary steam. The use of energy potentials of liquids or their approximate flow structures is more often achieved by heat exchange with countercurrent flows. The result of this interaction is the energy recovery.

It is shown that for the use of secondary energy resources of steam or gas phases, the regenerative heat exchange by physical strength can be supplemented by regeneration and the proving of thermodynamic parameters to values that allow to remove from systems or to limit the action of energy primary sources. Regeneration of the secondary steam allows the process of its subsequent condensation to return to the environment the potential of heat of vaporization. This means that in the systems in which the increase of thermodynamic parameters of steam due to its compression is restored, there is a use of Carnot's reverse cycle. Such application takes place both in closed and open loops.

The work of compressing the secondary steam is much less than the heat of its condensation. That is what determines the high efficiency of regenerative regimes. However, the direct use of secondary steam in condensation modes eliminates the expediency of the presence of thermal pumps in such systems.

The basis of the calculations of secondary steam recovery systems is material and thermal balances. The description of the physical processes that accompany the work of the thermocompressors is subject to mathematical modeling based on the laws of conservation of energy, mass conservation and conservation of impulses. The assessment of the suitability for using secondary energy resources is based on exergy methods in order to determine the directions for their further improvement.

For identical pressures in the mixing chamber, the efficiency of the real compressor is greater than the ratio of the ejection coefficients of the real and ideal compressors, since internal irreversible losses along with the reduction of the ejection coefficient result in the increase of the specific exergy of the compressed flow.

Mathematical models of transformation of energy-material flows of secondary steam are developed. It is shown that the secondary steam with respect to the medium in which it is generated has an exergy that is zero, since without the transformation of its thermodynamic parameters, it is impossible to return the heat of condensation.

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