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# Research of Innovative Energy-Drying Technology with the Use of a Spiral Heat Exchanger Filled with Thermal Factor in Simulated Conditions

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## Abstract

This publication presents the results of research on the innovative industrial technology of a cereal dryer in which a heat exchanger was used. Initial laboratory tests were used to determine the values characterizing the model installation work in the laboratory scale in order to verify the variables determining the selection of the most advantageous variant verified by cross-linking of dependent variables affecting the efficiency of the planned installation. Obtained results of research and analysis of variance of multifactorial experience in laboratory conditions, not affected by external factors introducing increased uncertainty of measurements, were supposed to provide information that would allow for selecting the type of exchanger liquid (the so-called thermal factor) circulating in the installation to receive and transfer heat and determine the optimal shape and the surface of the coil that fills the heat exchangers exchanged between air-liquid and fluid-air media.

**Keywords:** Heat Exchanger; Dryer; Maize Dryer; Thermal Factor

## Introduction

Maize (*Zea mays*) belongs to one of the most popular crops, whose annual production in the world is about 1 billion tons per year [1]. According to literature reports, drying of corn grain is an expensive process, absorbing about 30% of cultivation costs, while with improper conduct of the material preservation process or low yield, up to 50% of inputs can be made [2,3]. To store corn grain safely, it should be dried to moisture content below 15% and cooled to below 10°C. Drying can be carried out by a high-temperature method in special dryers or divided into two stages and in the second stage using an economical low-temperature technique. The division of the process into two stages is dictated by economic reasons, because after achieving grain moisture below 20%, high temperature drying becomes unprofitable and creates the possibility of grain being dried at a lower temperature in a silo or flat store [4]. Grain dried up to 13-14% humidity is suitable for storage, without exposing it to food loss or germination. The maize drying system is characterized by high energy consumption resulting from 2-4 times higher moisture content in grain during field harvest [5]. The drying process is very complex-grain and air are guided simultaneously through the dryer shaft in parallel flow, counter-flow and cross-flow [6]. This is possibly one of the reasons why there have so far been only few scientific papers studying sub-processes of the air flow, the particle motion and the heat and mass transfer [7]. During drying, a moisture content of approx. 12% should be achieved in a short period of time. Acceleration of this process entails large fuel losses and decreases the daily drying capacity. The most effective is the drying process carried out in two phases. The first phase is high-temperature drying, where the temperature of the drying agent reaches up to 100°C, and the temperature of the grain slightly increases and does not change until 24%-28% of humidity [8].

In light of the above, an important feature of the solution being developed is the introduction of a thermal energy recuperation system in the dryer air circulation system. For this purpose, it is planned to design a recirculating system [9]. The main element is the heat exchanger between the air-liquid medium at the outlet of the ventilation duct that discharges warm and moist air from the dryer, the function of which is to receive heat from the air leaving the dried bed. The second element of the installation is the heat exchanger between the air-liquid medium installed in the air intake introduced into the burner chamber. Its function is the transfer of heat accumulated by the circulating medium in the exchanger system in order to heat the air entering the combustion chamber. In the combustion chamber, the air is heated by means of a burner and then it is drawn through the dried bed [10]. Initial laboratory tests were used to determine the values characterizing the model installation work in the laboratory scale in order to verify the variables determining the selection of the most advantageous variant verified by cross-linking of dependent variables affecting the efficiency of the planned installation.

Obtained results of research and analysis of variance of multifactorial experience in laboratory conditions, not affected by external factors introducing increased uncertainty of measurements, were to provide information that will allow selecting the type of exchanger liquid (so-called thermal factor) circulating in the installation to receive and transfer heat and determine the optimal shape and surface of the coil that fills the heat exchangers exchanged between the air-to-liquid and liquid-air media [11].

## Methodology

The research stand was built in cooperation with Polnet and the Institute of Technology and Life Sciences. It consists of two cassette heat exchangers facing each other. The heat exchanger cassettes were

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connected by ventilation ducts between which an air heater with an axial fan was placed, which produced a stream of heating air passing between both exchangers [12]. The principle of operation of the test stand consisted of simulating the process of air heating in ventilation ducts imitating the interior of the dryer and the heat flow between the tested heat exchangers. The hot air coming out of the heater simulates heated air coming out of the dryer [13]. Passing through the heat exchanger, it releases part of the heat to the thermal medium circulating in the installation, which is forced to the heat exchanger in the place of sucking in cold ambient air [14]. In this heat exchanger, the heat is transferred to the air sucked in from the environment and in this way; air gets into the heater at a temperature higher than the ambient temperature. The view of the test bench during the tests is shown in Figure 1 [15,16].

The coils filling the inside of the heat exchangers are connected by means of thermally insulated copper pipelines [17]. A closed circulation of the thermal medium was applied to the heat exchanger system. Circulation of the circulating liquid-the thermal factor-was forced by the circulating pump installed in the system with the possibility of regulating the flow velocity. In addition, the exchanger installation includes a closed type leveling vessel, used to stabilize the pressure of the liquid in the system, service valves, and the vent valve for the installation [18].

The test stand has been equipped with control and measurement equipment. A heat meter installed in the thermal medium circuit:Apator Powogaz (type:ELF 90-0.6) with a single-stream flow transducer, designed to measure the consumption of thermal energy (GJ or kWh) was charged and given by the circulating thermal factor [19]. The flow meter is equipped with a multifunctional microprocessor heat converter that allows for data archiving and configuration of calculation parameters according to the requirements of the test series [20].

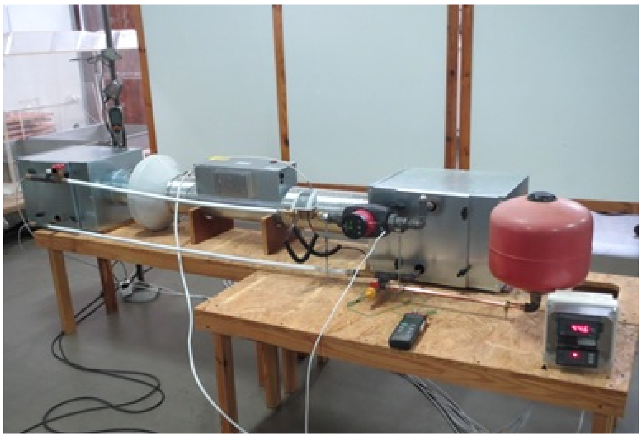

**Figure 1:** General view of the test stand and control and measurement apparatus in laboratory conditions.

## Results and Discussion

### Evaluation of technical solutions of the heat exchanger system for air flow suppression

In order to determine the influence of the type of heat exchangers used to reduce the efficiency of the air-forced fan, the average air velocity [m/s] in the air stream on the test bench was determined. For the heat exchanger installation, after removing the frames with tubular heat exchangers from the cassettes, the air flow rate was calculated from the formula:

$$S = c_{av} F \left(m^3 s^{-1}\right)$$

Where, F-cross-sectional area of the exchange conductor at the test stand ($m^2$),

$c_{av}$: Average air velocity ($m.s^{-1}$)

The air velocity of the cav was calculated after the dynamic pressure measurement of the Prandtl tube using the formula:

$$c_{av} = \sqrt{\frac{2\Delta p_d}{\rho_t}} \left(m\,.\,s^{-1}\right)$$

Where, ρt: Air density at temperature t(°C) at the place of measurement ($kg.m^{-3}$].

For air at +20°C, assuming that γ=11,825 N/$m^3$, g=9,81 m/s and that że $\Delta_d$ in kG/$m^3$, the air velocity is:

$$C_{sr} = 4,04\sqrt{\Delta\rho_d} \left(m\,.\,s^{-1}\right)$$

In order to accurately determine the average air velocity of the $c_{av}$ through the conduit, the field of the circular conduit on the test stand is divided into rings of the same cross-section and measures $\Delta_d$ at four points on two mutually perpendicular straight rings. Because the diameter of the air transporting pipe is 160 mm (R=80 mm), the measuring points have been determined in three rings. The distance $r_n$ [mm] of the measuring points from the center of the cross-section is defined by the formula:

$$r_n = R\sqrt{\frac{2n-1}{i}} \text{ (mm)}$$

Where, n: Order number of the ring counting from the center of the cross-section,

i: Number of rings,

R: Inner radius of the wire (mm)

After measuring the dynamic pressures at individual measuring points, the average dynamic pressure $\Delta_{d\ av}$, corresponding to the average air velocity from the formula is determined:

$$\sqrt{\Delta\rho_{dsr}} = \frac{\sqrt{\Delta\rho_{d1}} + \sqrt{\Delta\rho_{d2}} + \ + \sqrt{\Delta\rho_{dn}}}{n} \text{(m/s)}$$

On the basis of the obtained value $\sqrt{\rho_{dsr}}$ in different conditions (different types of heat exchangers or lack thereof different efficiency of the fan), on the basis of the above presented formulas, the cav and the air flow S are calculated.

The results of calculations of air flow efficiency on the test bench of various systems are presented in Table 1.

| lp. | $U_n$ | $S_1$ | $S_2$ | $S_3$ |
|---|---|---|---|---|
| | (V) | ($m^3.h^{-1}$) | ($m^3.h^{-1}$) | ($m^3.h^{-1}$) |
| 1 | 230 | 471.5 | 438.6 | 431.1 |
| 2 | 207 | 451.6 | 417.2 | 413.3 |
| 3 | 184 | 418.8 | 387.6 | 384.0 |
| 4 | 161 | 390.5 | 363.2 | 359.5 |
| 5 | 138 | 347.9 | 325.8 | 325.0 |
| 6 | 115 | 292.6 | 270.8 | 269.8 |
| 7 | 92 | 259.5 | 247.5 | 241.3 |

**Table 1:** Airflow efficiency for different types of heat exchangers depending on the fan supply voltage. $S_1$-no heat exchangers, $S_2$-tube heat exchangers without sipes, $S_3$-tube heat exchangers with sipes.

The use of two tubular heat exchangers with lamellae on the test bench at the same time caused the greatest reduction in the efficiency of the conveyed air. At the rated supply voltage of the fan ($U_n$=230V), the installation of these exchangers caused a drop of approx. 7% of the capacity of the conveyed air. This corresponded to a reduction in the average air velocity in the conduit from 6.76 to 7.40 $m.s^{-1}$. At a power supply of 0.6 $U_n$ (138 V), the loss of conveyed air capacity was around 15%. The use of tubular exchangers without sipes caused a drop in the efficiency of the conveyed air slightly smaller than with the same exchangers, but with lamellas. The above tests were carried out on a test bench in the laboratory at ambient temperature in the range of 18-20°C, and the values of density and specific gravity for the temperature of 20°C were adopted for the calculation.

## Test in laboratory conditions for heat removal by a circulating liquid

During the test of heat transfer by the thermal factor circulating in the heat exchanger system, measured in various operating conditions of the system (thermal medium flow efficiency, air flow rate through exchangers, air temperature), the amount of heat that was taken from the heated air (simulating air coming out of the column drying) and passed to cool air was sucked in from the environment. During the tests, the thermal medium was pumped in the exchanger system with a circulating pump with 3 settings, flow rate, in the range of 150-320 $dm^3/h$. In the case of water transfer, the pumping efficiency at individual levels did not differ by more than 3%. The same was true for the transfer of 36% ethylene glycol with water, but with slightly lower yields at individual levels.

The flow rate of the conveyed air, depending on the supply voltage of the fan, was in the range of 320-470 $m^3.h^{-1}$. At the cross-sectional area of the exchanger frame 0.05 $m^2$ at the test stand, under real conditions of the dryer, with a fan capacity of approx. 20,000 $m^3$, similar type exchangers should have a cross-sectional area between 2 and 3 $m^2$, with the smaller area corresponding to the highest efficiency used in the research. Obtaining such exchanger air flow surface on sucking in air from the environment presents no problem. In the case of an exchanger that receives heat from the air dryer removed from the block, in order to obtain such a surface, it is suggested to mount the exchanger in front of the fan.

Also, the temperature of hot air, simulating the air coming out of the drying block after passing through the grain bed, during the tests took 3 values:50, 60, 70°C. Such temperature values were adopted on the basis of the average actual temperature of the air coming out of the dryer under operating conditions, which was approximately 60°C. During the tests, this temperature was obtained by periodically switching on the electric air heater and it was the average temperature between the temperature of the outgoing air at which the control block switched the heater on and off. Therefore, the temperature obtained for individual measurements was in the range of ± 3°C.

Calculating the amount of heat that was removed from the hot air coming out of the dryer and then transferred to the air pressed into the heater, the following formula was used:

$$Q = m_{cg} . C_{cg} \Delta T_{cg} \text{ (J)}$$

Where, $m_{cg}$: Mass of the thermal medium (circulating liquid) (kg)

$c_{cg}$: Specific heat of the thermal medium ($J.(kg.K)^{-1}$)

$\Delta T_{cg}$: Temperature by which the thermal factor (°C) has been heated/cooled.

The mass of the thermal medium that was involved in heat transfer was calculated based on the readings from the flow meter mounted on the basis of the dependence:

$$m_{cg} = v_{cg} \rho_{cg} t \text{ (kg)}$$

Where, $v_{cg}$: Flow rate of thermal medium ($dm^3.hr^{-1}$),

$_{cg}$: Specific mass of the thermal medium (circulating liquid) ($kg.dcm^{-3}$)

: Duration of the measuring contact (h)

When used as a thermal agent (circulating liquid) of water, the specific heat of water was taken in the calculation cw 4190 ($J/(kg.K)^{-1}$), and the specific gravity $\rho_w$=1.0 ($kg.dm^{-3}$).

In order to protect the unused installation against freezing, liquids with a reduced level of freezing are used. During the laboratory tests, the circulating installation was filled with a non-freezing liquid, i.e. a mixture of 36% water and ethylene glycol, which protects the installation during standstill before freezing, at ambient temperature of-20°C. The specific heat of the mixture with a concentration of 36% water and ethylene glycol in the temperature range of 20 to 70°C is increased from approximately 3390 ($J/(kg.K)^{-1}$) to approximately 3710 ($J/(kg.K)^{-1}$). In the given temperature range, the density of the mixture being a thermal medium decreases from 1056 ($kg/m^3$) to 1038 ($kg.m^{-3}$). For the hard freeze fluid prepared in this way, the specific heat of the fluid cg=3630 ($kg.(J.K)^{-1}$) was assumed in the calculations, and the specific gravity of the fluid ρg=1.047 ($kg.(dcm)^{-3}$). These values were adopted for the above-described mixture at a temperature of about 40°C, which is the average temperature of the thermal medium in the circulation of the test bench during the experiment.

## Accuracy of measurements

The measuring probes used to measure the temperature of the thermal medium type PT100 with an accuracy of ± 0.3°C/0°C in accordance with EN:60751B in combination with a 0.1°C meter allowed measurements in the temperature range from about 20° to 70°C with uncertainty of ± 0.8°C to ± 2.5°C. A thermometer with a K-type thermocouple input was used to measure the air temperature with a reading resolution of 0.1˚C and accuracy of measurements equal to ± 0.1% of the reading and ± 0.7˚C, which corresponded to measurement uncertainties in the range of measured air temperatures from 20˚ to 70˚C, respectively (± 0.72˚C to ± 0.77˚C).

The flow of the circulating medium in the pipeline between the exchangers was measured with an integrated flow meter with an accuracy of 0.5%, which corresponded to the measurement uncertainty with the smallest flows equal to ± 8 $dcm^3 \cdot h^{-1}$ and respectively ± 15 $dcm^3.h^{-1}$ at the largest flows. Measurements of electric power consumed by the heating element were made with an ammeter and a laboratory voltmeter with an accuracy class of 0.2. The measurement of the amount of electricity consumed by the heating device the heater was performed with uncertainty of ± 0.3A, and the voltage measurement of the heating device was carried out with an uncertainty of ± 4.8 V. Measurement of electrical power consumed at nominal conditions was calculated with uncertainty of ± 0.19 kW. The thermal energy values received/transferred from the thermal factor to the air and vice versa were calculated with uncertainty in the range from ± 0.35 MJ to ± 1.5 MJ, with the highest values referring to the calculations made at maximum flows and large temperature increases. The expanded uncertainties given above result from the standard uncertainty multiplied by the expansion coefficient k=2, which for the normal distribution provides a confidence level of approximately 95%.

## The test verifying the assumed operating parameters of the exchanger installation on the test bench

Preliminary research was carried out on the above-described standpoint, which was to eliminate the observed errors and verify the assumed assumptions of the correctness of the research station installation. Firstly, on the bench with an exchanger installation without a circulating liquid-a heating element was connected-an electric heater and air forced through it by an axial fan to temperatures used in subsequent experiments and power consumption was observed to heat this air. The tests were carried out at three air flow rates corresponding to the supply voltage of the fan $U_w$=1.0 Un, 0.8 Un and 0.6 Un to obtain the average temperature corresponding to the temperature of the outgoing air from the dryer at 60˚C, and additionally at this temperature of 70˚C efficiency of the fan corresponding to the supply voltage Uw=0.6 Un and the temperature 50˚C at the capacity of the fan corresponding to the voltage supply Uw=1.0 Un. During the test, tubular exchangers with lamellae were fixed in the cassettes. Energy consumption was determined based on the power consumption of the heater and working time during an hour test cycle. The measurement results are presented in Table 2.

| S.No. | Temperature assumed | Temperature obtained | Air transfer rate | Energy consumption |
|---|---|---|---|---|
| | (°C) | (°C) | ($m^3.h^{-1}$) | (kWh) |
| 1 | 70 | 68,5 | 325,1 | 5,43 |
| 2 | 60 | 58,3 | 431,1 | 5,98 |
| 3 | 60 | 60,9 | 384,0 | 5,26 |
| 4 | 60 | 58,6 | 325,1 | 4,40 |
| 5 | 50 | 48,8 | 431,1 | 4,35 |

**Table 2:** Electricity consumption for heating air between heat exchangers without a circulating liquid on a laboratory bench.

The results obtained experimentally were verified by control calculations of the amount of heat consumed to heat the same amount of air as in the conducted test on the test bench. The calculations were carried out for air flow at the level of 460.8 m3.h-1, with the assumed temperature of the air leaving the dryer equal to 60˚C at an incoming air temperature of 20˚C, i.e. heating of dry air from 20˚ to 60˚C. The amount of heat collected by air in these conditions was calculated according to the formula:

$$Q_p = m_p \cdot c_{pp} \Delta T_p \text{ [J]}$$

Where, $m_p$: Mass of heated air (kg)

$c_{pp}$: Specific heat of conveyed air (J.(kg.K)-1)

$\Delta T_p$ The temperature at which the forced air was heated (˚C)

The mass of conveyed air that received heat was calculated on the basis of the air flow capacity depending on the fan supply voltage based on the following:

$$m_p = v_p \rho_p t \text{ (kg)}$$

Where, $v_p$: Air flow efficiency corresponding to the given fan supply voltage ($m^3.h^{-1}$)

$_p$: Specific mass of air ($kg \cdot m^{-3}$)

Time of measurement of heat transfer (h)

In the calculations, the specific heat of transported air was assumed cpp=1020 (($J.(kg.K)^{-1}$) and the specific mass of the conveyed air ρp=1,2 [$kg.m^{-3}$]. Under these conditions, the calculated amount of heat needed to heat the air is:

$$Q_p = 21.11\ MJ$$

After recalculation, the amount of energy needed to heat the air is 5.87 kWh, which is similar to the amount of electricity used to heat the air in these conditions. The obtained similar real and theoretical results allow for the applied research method to be used to assess the energy efficiency of the experimental installation of a heat exchanger on a test bench in laboratory conditions.

Then a preliminary comparative study was carried out, where the thermal factor in the circulating heat exchanger system was water and a frame with a tube exchanger without sipes and with lamellae was used. The study was conducted for a simulated temperature of the air coming out of the dryer at 50˚C and the flow rate of air conveyed between exchangers, depending on the type of exchanger, in the range of 384.0-387.6 $m^3.h^{-1}$. Next, a test was carried out for the simulated temperature of the air coming out of the dryer at the level of 70°C and the flow rate of the conveyed air in the range of 431.1-438.6 $m^3.h^{-1}$. During the tests, the energy consumption necessary to heat the air and the amount of energy transferred by the thermal factor of the heated air were measured. On this basis, the amount of heat recovered by means of a heat exchanger extracted from the dryer and the amount of heat transferred to the exchanger used to heat the air sucked in from the installation's environment was calculated. The obtained test results are presented in Table 3, supplemented additionally by the temperatures to which the air simulating the air coming out of the dryer was heated.

| **Tube exchanger with sipes** | | | | | **Tube exchanger without sipes** | | | |
|---|---|---|---|---|---|---|---|---|
| **T** | **Eel** | **Ecob** | **ΣE** | **temperature obtained** | **Eel** | **Ecob** | **ΣE** | **temperature obtained** |
| **(°C)** | **(kWh)** | **(kWh)** | **(kWh)** | **(°C)** | **(kWh)** | **(kWh)** | **(kWh)** | **(°C)** |
| 50 | 3,02 | 1,20 | 4,22 | 48,0 | 3,67 | 0,66 | 4,33 | 47,2 |
| 70 | 5,10 | 1,94 | 7,04 | 66,9 | 6,0 | 0,87 | 6,87 | 62,6* |
| *temperature obtained during continuous operation of the heater | | | | | | | | |

**Table 3:** Energy demand for obtaining a specific temperature on the test bench with various heat exchangers.

The obtained results clearly indicate a lower efficiency of recovery and heat transfer by various tube heat exchangers from the thermal factor. In the case of an experiment with a tube heat exchanger at a temperature of 70°C, without installing an additional heater, the assumed air temperature after the heater could not be obtained. The preliminary test results presented above made it possible to decide on eliminating the factor series with a tube exchanger without sipes and carrying out a complete factor test in the conditions described in point. 2.3 when used in a circulation system as a thermal water factor and then 32% of an ethylene glycol/water mixture and tubular exchangers with sipes, which have been verified as more effective in the function of heat transfer between air-liquid-air centers.

## Simulation tests of the parameters of the exchanger installation on the test bench

The results of calculations of the amount of thermal energy transferred between air-liquid and air on a test stand equipped with tubular heat exchangers with lamellae with two types of thermal circulation medium are shown in Table 4. These values were calculated on the basis of values of air temperatures and temperatures recorded during the experiment circulating liquid at the indicated measuring points.

| **Temperature assumed** | **Air flow rate** | **Flow rate of thermal medium** | **Energy transferred from heated air** | |
|---|---|---|---|---|
| | | | **Water** | **Mixture** |
| **(°C)** | **($m^3.h^{-1}$)** | **($m^3.h^{-1}$)** | **(MJ)** | **(MJ)** |
| 70 | 431.1 | 0.15 | 6.39 | 5.36 |
| 70 | 431.1 | 0.25 | 6.48 | 6.05 |
| 70 | 431.1 | 0.32 | 7.01 | 6.45 |
| 70 | 384.0 | 0.15 | 5.34 | 4.93 |
| 70 | 384.0 | 0.25 | 5.83 | 5.51 |
| 70 | 384.0 | 0.32 | 6.25 | 5.90 |
| 70 | 325.0 | 0.15 | 4.94 | 4.23 |
| 70 | 325.0 | 0.25 | 5.34 | 4.94 |
| 70 | 325.0 | 0.32 | 5.67 | 5.26 |
| 60 | 431.1 | 0.15 | 5.00 | 4.26 |
| 60 | 431.1 | 0.25 | 5.31 | 4.89 |
| 60 | 431.1 | 0.32 | 5.55 | 5.22 |
| 60 | 384.0 | 0.15 | 4.64 | 4.07 |

| | | | | |
|---|---|---|---|---|
| 60 | 384.0 | 0.25 | 5.25 | 4.67 |
| 60 | 384.0 | 0.32 | 5.37 | 5.24 |
| 60 | 325.0 | 0.15 | 4.47 | 3.66 |
| 60 | 325.0 | 0.25 | 4.77 | 4.10 |
| 60 | 325.0 | 0.32 | 5.10 | 4.22 |
| 50 | 431.1 | 0.15 | 3.46 | 3.26 |
| 50 | 431.1 | 0.25 | 3.87 | 3.57 |
| 50 | 431.1 | 0.32 | 4.08 | 3.87 |
| 50 | 384.0 | 0.15 | 3.80 | 3.13 |
| 50 | 384.0 | 0.25 | 4.20 | 3.78 |
| 50 | 384.0 | 0.32 | 4.33 | 4.06 |
| 50 | 325.0 | 0.15 | 3.19 | 2.81 |
| 50 | 325.0 | 0.25 | 3.55 | 3.32 |
| 50 | 325.0 | 0.32 | 3.87 | 3.59 |

**Table 4:** Factor test results in a tubular tube heat exchanger with sipes on a test bench in laboratory conditions.

As the presented results of the study show in the determined laboratory conditions in the course of the experiment, better results were obtained by using water as a circulating thermal medium transferring heat between air-liquid centers. This is due to the greater specific heat of water compared to the ethylene glycol mixture. In turn, the advantage of using a mixture of water and ethylene glycol is its lower freezing point. It is advisable to prepare a mixture with a concentration adjusted to the expected lowest temperatures in the season of drying grains falling for the period of September-November, when it is possible that low air temperatures may occur. It is assumed that the lower the concentration of a mixture, the higher its specific heat, and so at 0°C for a mixture of 46% (freezing point about-35°C), specific heat is 3.31 kJ.(kg.K)$^{-1}$, for a mixture of 36% (freezing temperature about-20°C), specific heat is 3.49 kJ.(kg.K)$^{-1}$, and for a mixture with a concentration of 26% (freezing temperature about-10°C), specific heat is about 3.68 kJ.(kg.K)$^{-1}$ (Table 5).

| Energy balance between heat exchangers | | | | |
|---|---|---|---|---|
| **Liquid flow ($m^3.h^{-1}$)** | **Temperature obtained (°C)** | **Energy from heat recovery (kWh)** | **Electric energy (kWh)** | **Total energy for heating (kWh)** |
| 0.149 | 57.6 | 1.30 | 3.83 | 5.16 |
| 0.245 | 58.1 | 1.46 | 3.86 | 5.29 |
| 0.319 | 58.0 | 1.49 | 3.74 | 5.23 |

**Table 5:** It represents the balance of energy recovered from cooling the air exiting the exchanger and electricity consumed to heat the air entering the exchanger to the desired temperature of 60°C at an air flow rate of 384 $m^3.h^{-1}$.

The obtained results of the total energy consumption required to heat the air from 20°C to 60°C with the energy consumed under the same conditions on the stand without the energy recovered from warm air are approximately the same. In the system without heat recovery, approximately 5.26 kWh of electric energy was consumed and in the system with heat removal the average total energy consumption was 5.23 kWh, the flow of liquid had no significant effect, as confirmed by statistical analysis performed by ANOVA variance analysis. On this basis, it can be concluded that the share of energy recovered in total energy consumed for heating air was about 27%.

## Conclusion

This publication presents the results of research on the innovative industrial technology of a cereal dryer in which a heat exchanger was used. The amount of thermal energy transferred between air-liquid and liquid-air on the test stand equipped with tubular heat exchangers with lamellae with two types of thermal circulation medium was calculated on the basis of air temperature values and circulating liquid temperatures recorded during the experiment. As the presented results of the study show in the determined laboratory conditions in the course of the experiment, better results were obtained by using water as a circulating thermal medium transferring heat between air-liquid centers. This is due to the greater specific heat of water compared to the ethylene glycol mixture. In turn, the advantage of using a mixture of water and ethylene glycol is its lower freezing point. It is advisable to prepare a mixture with a concentration adjusted to the expected lowest temperatures in the season of drying grains falling for the period of September-November, when it is possible that low air temperatures may occur. It is assumed that the lower the concentration of a mixture, the higher its specific heat, and so at 0°C for a mixture of 46% (freezing point about-35°C), specific heat is 3.31 kJ.(kg.K)$^{-1}$, for a mixture of 36% (freezing temperature about-20°C) specific heat is 3.49 kJ.(kg.K)$^{-1}$, and for a mixture with a concentration of 26% (freezing temperature about-10°C), specific heat is about 3.68 kJ.(kg.K)$^{-1}$.

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