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ANALYSIS OF POTENTIAL OF HEAT RECOVERY AND STORAGE FROM THE COOLING SYSTEM OF A TUNNEL

Stanisław Rudy*, Paweł Kozak, Andrzej Krzykowski, Dariusz Dziki, Renata Polak, Jerzy Wójcik

Thermal Engineering Department, University of Life Sciences in Lublin

* *Contact details: Doświadczalna 44, 20-280 Lublin, e-mail: stanislaw.rudy@up.lublin.pl*

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ABSTRACT

The paper presents analysis of potential of recovering and storing thermal energy from a cooling installation of a fluidization tunnel freezer and then its storing. A plate heat exchanger placed behind a section of compressors was used for heat recovery and heated water that leaves the exchanger transferred heat through a pipe coil to a heat accumualtor. The amount of heat possible to be recovered from a cooling installation in relation to burdening of the cooling tunnel and the change of water temperature in a dispenser during the process was determined.

List of abbreviations:

$C = exp\left(\frac{k \cdot F}{m_a \cdot c_a}\right)$ constant value,

- Δt – temperature difference, (°C)
- c – specific heat of the liquid, ($J \cdot kg^{-1}K^{-1}$)
- C, a, b – constants depending on the conditions of heat exchange,
- c_a – specific heat of the heating fluid, ($J \cdot kg^{-1}K^{-1}$)
- c_b – specific heat of the heated fluid, ($J \cdot kg^{-1}K^{-1}$)
- d – double distance between the plates, - diameter of the coil (m),
- F – heat transfer surface area, (m^2)
- g – gravitational acceleration (g=9.81 $m \cdot s^{-2}$),
- Gr – Grashof number,
- h_o – conventional size – 1 mm,
- k – heat transfer coefficient, ($W \cdot m^{-2} \cdot K^{-1}$)
- l – replacement characteristic dimension, height of the tank, (m)
- m_a – mass flow of the heating fluid,
- m_b – heated liquid mass,
- Nu – Nusselt number,

Pr	– Prandtl number,
Re	– Reynolds number,
t	– temperature of the heating fluid, (90ºC)
t_k	– temperature of water in the tank heat when heated by a predetermined step, (t_p+ 5ºC)
t_p	– the initial temperature of the water in the water tank, (ºC)
u	– factor speed, ($m \cdot s^{-1}$)
α	– heat transfer coefficient, ($W \cdot m^{-2} \cdot K^{-1}$)
β	– the attenuation factor of turbulence at the surface of the wall,
β_1	– liquid volumetric expansion,
βt	– ratio of forced turbulence,
ΔT_{ln}	– logarithmic temperature difference,
ζ	–coefficient of flow resistance in the smooth channels,
η	– agent dynamic viscosity, (Pa·s)
λ	– thermal conductivity, ($W \cdot m^{-1} \cdot K^{-1}$)
v	– kinematic viscosity, ($m^2 \cdot s^{-1}$)
ρ	– density of medium, ($kg \cdot m^{-3}$)
τ	– tank heating time, (s)

Introduction

The issues related to energy savings, in addition to the alternative energy sources are in recent years, one of the cornerstones of the European energy policy strategy. Taking into account the increase in the price of conventional energy sources, environmental protection and financial benefits resulting from reduction of manufacturing costs, reducing energy consumption are of particular importance in case of production facilities (Staniszewski and Bonca, 2006; Wojdalski et al., 2008a; 2008b; Łapczyńska-Kordon et al., 2013). There is also work being carried out concerning analysis of energy effects of installations, including storage of excess heat in different objects (Kurpaska, 2003; 2007; Rutkowski 2008).

The plants using refrigeration systems may recover from condensers waste heat energy, which must be cooled, and originally the heat is transferred to the cooling medium (air, water) and eventually lost. Superheat of the refrigerant vapor and condensation heat can be used as needed, for example: for production of hot water for technological purposes - processing of raw materials, cleaning equipment and premises in the production process, and heating the ground under the cold chambers. If the plant is not connected to the district heating the hot water can be used for sanitary purposes and for heating of buildings (Oberg, 2005; Targański, 2011).

However, given the asynchronous availability and demand for waste heat the additional conventional heat source or heat storage accumulators must be installed.It is advisable to analyze the recoverable amount of thermal energy with reference to the existing energy needs (Gazda, 2002; Targański, 2009).

The objective of the study was to analyze the possibility of recovery and storage of the waste heat from the selected cooling system receiving heat from the tunnel, intended for pre-freezing of fruit and vegetables. The amount of recoverable heat from the refrigeration system in relation to loading of the cooling tunnel in a plate heat exchanger arranged behind the compressor system was determined. In the process of the heat storage tank temperature changes depending on the duration of the process were also determined.

Research facility

A fluidized freezing tunnel is used to freeze any whole or sliced fruit and vegetables. High-quality products are obtained through their individual freezing, and then the temperature is still reduced in two zones using cold air blast, due to the effect of fluidization. The research was conducted on a device located at the production plant Producers Group "Class" company based in Klementowice. Technical data concerning a fluidized cooling tunnel were presented in table 1.

Table 1.
Basic technical data of fluidized tunnel freezer

Specification	Technical parameters
Base performance, ($kg \cdot h^{-1}$)	1000
Cold demand for strawberries, (kW)	175
Refrigerant agent	R404
Refrigerant circuit (R404 , R407)	Expansion valves & distributor of Freon
The boiling point of the medium, (ºC)	- 40
The internal volume of coolers ammonia / Freon, (dm^3)	1x548 / 3x198
Defrosting coolers for Freon	Spraying water
Defrost time, (min)	25÷30
Re-cooling time, (min)	15÷20
Freezing time adjustment of the product, (min)	7÷35
The total installed electric power, (kW)	52
Normal power consumption during operation, (kW)	44
Electric power	3x400/50Hz
Compressed air for UDS – dried air	1 $m^3 \cdot min^{-1}$, 7.5 bar
Working width of the strip, (mm)	900
Width of the tunnel housing with Freon, (m)	4.4
Height of the housing tunnel, (m)	4.4
Length of the tunnel lining, (m)	5.12
Total length of the tunnel, (m)	6.67

A cooling tunnel is equipped with a system of three screw compressors with a capacity of 15 kW each, connected in parallel. Compressors, depending on the demand for cooling power, turn on automatically. A condenser of a freezing tunnel with a capacity of 100 kW is placed on the outside of the building and it is cooled by outside air, circulation of which is forced by a fan.

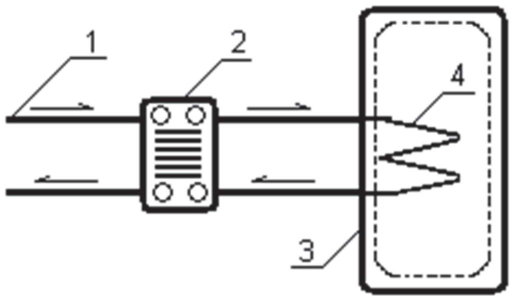

Figure 1. Schematic representation of the heat recovery installation, 1 – factor flowing from the compressor system, 2 – plate heat exchanger, 3 – water tank, 4 – pipe coil

Fans causing air fluidization in the tunnel have total performance of 30 $m^3 \cdot s^{-1}$. A schematic representation of heat recovery from a refrigerant was shown in Figure 1. A plate heat exchanger (soldered, made of stainless steel) which recovers heat from the tunnel freezing system was embodied in the cooling system behind the section of compressors. In the exchanger operating in the countercurrent, superheated vapor of refrigerant leaving the system of compressors is the cooled medium, while deionizer water is the heated medium. Table 2 summarizes the basic technical and structural parameters of the plate heat exchanger.

Table 2.
Technical parameters of the plate heat exchanger

Design data	Medium 1	Medium 2	Unit
Medium	Superheated vapour R- 407 C	Water	
Flow rate	0.85	0.5	$kg \cdot s^{-1}$
Speed of the flow	16.8	0.78	$m \cdot s^{-1}$
Temperature input	97	10	ºC
Temperature output	50	90	ºC
Density	55.6	999.6	$kg \cdot m^{-3}$
Specific heat	829	4215	$J \cdot kg^{-1}K^{-1}$
Thermal conductivity	0.0112	0.573	$W \cdot m^{-1}K^{-1}$
Dynamic viscosity	$0.0128 \cdot 10^{-3}$	$1.304 \cdot 10^{-3}$	$Pa \cdot s$
Kinematic viscosity	$0.23 \cdot 10^{-6}$	$1.3 \cdot 10^{-6}$	$m^2 \cdot s^{-1}$
Operating pressure	1.8	0.7	MPa
Distance of plates	0.01	0.01	m
Prandtl number	0.947	9.56	
Surface of the plate	0.05	0.05	m^2
Number of plates all active	10/8	10/8	
Drop of pressure (max)	12	12	m H_2O

Table 3.
Technical parameters of the heat accumulator

Parameter	Unit	Value
Tank volume	(dm^3)	1500
Volume of heat exchanger	(dm^3)	22.8
Surface op pipe coil	(m^2)	3.6
Length of pipe coil	(m)	45
Diameter of pipe coil	(m)	0.0254
Diameter of cylinder without insulation	(mm)	1200
Diameter of tank with insulation	(mm)	1400
Width	(mm)	1210
Heat circulation	($kWh \cdot 24\ h^{-1}$)	5.3
Maximum operating pressure	(bar)	8
Maximum operating overpressure	(bar)	3
Maximum temperature	(ºC)	95

The constant temperature of water leaving the heat exchanger plate of 90°C was adopted and the water flow rate of 0.78 $m \cdot s^{-1}$ was assumed, which corresponds to the flow rate of 0.5 $kg \cdot s^{-1}$. The heated water leaving the plate heat exchanger is fed by thermally insulated wires to the heat storage, the operating parameters and design parameters of which are presented in table 3.

Due to the fact that the temperature of hot water in the heat storage increases with the duration of the heat exchange process, the heat transfer coefficients and heat penetration coefficient were determined using the temperature jump at 5°C within the range of 10°C to 90°C.

Results

Heat transfer coefficient of the plate heat exchanger between the heat exchanging agents was calculated using the equation-criteria given by (Zander and Zander, 2003). This equation takes the following form:

$$Nu = 0{,}022 \cdot \sqrt{\zeta_0} \cdot \beta \cdot \beta_t \cdot \mathrm{Re}^{0{,}825} \cdot \mathrm{Pr}^{0{,}54} \tag{1}$$

Determination of the Nusselt number, which allows determination of heat transfer coefficients of water and superheated vapour to the surface of the heat exchanger is possible after determination of factors included in the equation (1). The Reynolds number and parameters necessary to determine the Nusselt number, as well as the transfer coefficients and heat transfer coefficients and the heat penetration coefficients were determined on the basis of the data included in table 2. Calculated parameters for superheated refrigerant vapor and the water heated in the heat exchanger are summarized in table 4.

Table 4
Numbers indispensable for determination of Nusselt number from equation (1), coefficients of heat transmission and penetration

Parameter	Method of determination	Superheated steam	Water
Reynolds number	$\mathrm{Re} = \frac{u \cdot l}{\nu}$	$1.46 \cdot 10^6$	12000
Flow resistance coefficient	$\zeta_0 = \frac{0.3164}{\mathrm{Re}^{0.25}}$	0.0091	0.03
Turbulence damping factor	$\beta = 4 - \frac{1.65 \cdot h_0}{d}$	3.9175	
Coefficient of forced turbulence	$\beta_t = 1 + \left(0.33 - \frac{0.66 \cdot h_0}{d}\right) \cdot \ln \frac{4.23 \cdot \left(0.65 + 1.07 \cdot \lg \frac{h}{h_0}\right)}{0.3164}$	1.9312	
Nusselt number	$Nu = 0.022 \cdot \sqrt{\zeta_0} \cdot \beta \cdot \beta_t \cdot \mathrm{Re}^{0.825} \cdot \mathrm{Pr}^{0.54}$	1877.6	226.25
Heat transfer coefficient	$\alpha = \frac{Nu \cdot \lambda}{h} \quad \left[\frac{W}{m^2 \cdot K}\right]$	2102.9	12964.1
Heat penetration coefficient	$k = \frac{1}{\frac{1}{\alpha_s} + \frac{\delta}{\lambda} + \frac{1}{\alpha_w}} \quad \left[\frac{W}{m^2 \cdot K}\right]$	4018.88	

The logarithmic temperature difference needed to determine the heat exchanger capacity was determined on the basis of the temperature factors contained in Table 2.

$$\Delta T_{\ln} = \frac{\Delta T_1 - \Delta T_2}{\ln \frac{\Delta T_1}{\Delta T_2}} = \frac{(97-90)K - (50-10)K}{\ln \frac{7K}{40K}} = 18.93K \cdot \tag{2}$$

Thermal power of the plate heat exchanger is:

$$\dot{Q} = F \cdot k \cdot \Delta T_{\ln} = 0.4\, m^2 \cdot 4018.88 \frac{W}{m^2 \cdot K} \cdot 18.93K = 30431W \tag{3}$$

The maximum heat output of the heat exchanger is slightly higher than the value of 30 kW. Achieving this thermal power is possible during operation of all compressors, assuming that the temperature of condensing agent is 40°C.

It was assumed that the process of heat exchange between the coil pipe and the water in the tank can occur by natural unlimited convection, while the heat transfer coefficient between the water and the heating coil pipe in accordance with forced convection with the laminar flow. Nusselt number, which is used to determine the coefficient of heat transfer from the heated water to the surface of the coil, can be determined from the following relation:

$$Nu = C \cdot (Gr \cdot \mathrm{Pr})^n \tag{4}$$

Based on the obtained values of the Grashoff and Prandtl criterion numbers the constants C and n are 0.54 and 0.25, respectively. Since the fluid flow is an intermediate flow, it appears from the values of the criterion numbers, the equation becomes:

$$Nu = 0.54 \cdot (Gr \cdot \mathrm{Pr})^{0,25} \tag{5}$$

Table 5.
Coefficients indispensable for determination of characteristic numbers of equation (4) for heat exchange between a pipe coil and water in an accumulator and the value of coefficient of heat penetration

Coefficients										
T (°C)	ρ ($kg \cdot m^{-3}$)	c ($J \cdot kg^{-1}K^{-1}$)	λ ($W \cdot m^{-1}K^{-1}$)	$\beta_1 \times 10^4$ (K^{-1})	$\eta \cdot 10^3$ ($Pa \cdot s$)	$\nu \cdot 10^6$ ($m^2 \cdot s^{-1}$)	Pr -	Gr -	Nu -	α_1 ($W \cdot m^{-2}K^{-1}$)
10-15	999.6	4215	0.57	88.00	1.30	1.30	9.56	9411404	52.59	24.9
15-20	998.9	4211	0.59	156.00	1.13	1.10	8.15	18484951	59.83	28.93
20-25	998.2	4207	0.60	207.00	1.00	1.00	7.06	25182190	62.35	30.76
25-30	996.9	4207	0.61	248.00	0.90	0.91	6.20	30785681	63.47	31.84
30-35	995.6	4203	0.62	304.00	0.80	0.81	5.50	39378038	65.51	33.35
35-40	993.9	4203	0.62	342.00	0.72	0.72	4.85	45402673	65.78	33.92
40-45	992.2	4203	0.63	390.00	0.65	0.66	4.30	51425002	65.85	34.39
45-50	990.1	4203	0.64	105.00	0.60	0.62	3.90	13352169	45.87	24.22
50-55	988	4203	0.65	46.00	0.55	0.56	3.56	5751328	36.32	19.39
55-60	985.6	4203	0.65	50.00	0.51	0.52	3.25	5905490	35.74	19.26
60-65	983.2	4207	0.66	53.00	0.47	0.48	3.00	5768817	34.83	18.94
65-70	980.5	4211	0.66	56.00	0.44	0.45	2.75	5467555	33.63	18.4
70-75	977.7	4215	0.67	58.00	0.41	0.42	2.56	4857748	32.07	17.65
75-80	974.8	4215	0.67	60.00	0.38	0.39	2.35	4062627	30.02	16.6
80-85	971.8	4219	0.67	63.00	0.36	0.37	2.23	2991470	27.44	15.26
85-90	968.5	4224	0.68	66.00	0.34	0.35	2.10	1652759	23.31	13.02

The values of the heat transfer coefficient from the outer surface of a coil pipe to heated water in the water tank increases slightly in the temperature range from 10ºC to 40ºC. Above this temperature range the values of the heat transfer coefficient are reduced. The coefficient of heat penetration into the water heated in the tank has a small value in the whole considered temperature range.

The coefficient of heat transfer from the heating fluid to the inner surface of the coil pipe can be determined from the criterial equation:

$$Nu = C \cdot \mathrm{Re}^a \cdot \mathrm{Pr}^b \cdot \frac{d}{L} \tag{6}$$

Coefficients C, a, b are determined after previous determination of the Reynolds number, in order to determine the type of flow in the coil pipe. The Reynolds number included in Table 6 is greater than 3,000, on this basis, we conclude that the flow in the coil pipe is turbulent. The coefficient d/L is skipped because its value is less than 1/50. After being simplied the criterial equation takes the following form:

$$Nu = C \cdot \mathrm{Re}^{a} \cdot \mathrm{Pr}^{b} \tag{7}$$

Constants included in the equation for turbulent flow are respectively: C=0.023, a=0.8, b=0.4. Criterial numbers of Reynolds, Prandtl and, determined on the basis of these numbers, the Nusselt number and the heat transfer coefficient for the heat transfer process between the heating water and the inner surface of the coil pipe are presented in Table 6.

The coefficients of heat transfer from the heating fluid to the inner surface of the coil pipe have a much higher values than in the case of the plate heat exchanger, which is a consequence of forced convection and turbulent flow of the heating fluid.

The heat penetration coefficient for the considered ranges of temperature, was determined according to the formula presented in table 2.

The value of heat transfer coefficient, thermal conductivity coefficient of the coil pipe wall, the heat penetration coefficient and the coil pipe wall are listed in table 7.

Table 6.
Value of characteristic numbers and coefficient of heat transmission concerning heat permission between heating water and internal surface of pipe coil

Temperature (ºC)	Kinematic viscosity ($m^2 \cdot s^{-1}$)	Reynolds number	Prandtl number	Nusselt number	α_2 ($W \cdot m^{-2}K^{-1}$)
10-15	1.30	15240.00	9.56	125.98	6447.8
15-20	1.10	18010.91	8.15	135.09	5850.35
20-25	1.00	19812.00	7.06	137.66	5419.69
25-30	0.91	21771.43	6.20	140.93	5049.07
30-35	0.81	24611.18	5.50	148.18	4696.26
35-40	0.72	27516.67	4.85	154.06	4367.06
40-45	0.66	30063.73	4.30	157.6	4088.91
45-50	0.62	32214.63	3.90	160.18	3878.37
50-55	0.56	35633.09	3.56	167.41	3664.57
55-60	0.52	38469.90	3.25	171.62	3479.7
60-65	0.48	41361.17	3.00	176.14	3321.7
65-70	0.45	44521.35	2.75	180.43	3161.08
70-75	0.42	47739.76	2.56	185.41	3029.34
75-80	0.39	51459.74	2.35	190.25	2883.71
80-85	0.37	54131.15	2.23	194	2795.43
85-90	0.35	57095.10	2.10	197.65	2700.18
90	0.33	60773.01	1.95	201.7	2588.75

The determined values of heat transfer coefficients in the considered range of measurement are low, this is due to the specificity of the relation which defines this coefficient and mainly due to the lowest sum of components.

The heating time of liquid in the heat accumulator to the desired temperature can be determined by balancing the heat transfer equation and the equations of heat collected by the heated water and the equation of the heat given by heating water:

$$\tau = \frac{c}{c-1} \cdot \frac{m_b \cdot c_b}{m_a \cdot c_a} \cdot \ln \frac{t - t_p}{t - t_k} \tag{8}$$

The constant C used for determination of the duration of the process of heating water in the heat accumulator for each temperature ranges were presented in table 7.

Table 7.
Numbers indispensable for determination of the value of heat transmission coefficient and determined value of heat permission (k) and time of heating a tank(τ)

Temperature (ºC)	α_l ($W \cdot m^{-2}K^{-1}$)	α_l ($W \cdot m^{-2}K^{-1}$)	δ (m)	λ ($W \cdot m^{-1} \cdot K^{-1}$),	k ($W \cdot m^{-2} \cdot K^{-1}$),	C	τ (s)
10-15	24.9	6447.8	0.002	25	24.755	1.043193	1833
15-20	28.93	5850.35	0.002	25	28.721	1.12447	1868
20-25	30.76	5419.69	0.002	25	30.512	1.132861	1896
25-30	31.84	5049.07	0.002	25	31.561	1.13773	1982
30-35	33.35	4696.26	0.002	25	33.027	1.144716	2065
35-40	33.92	4367.06	0.002	25	33.568	1.147254	2228
40-45	34.39	4088.91	0.002	25	34.01	1.149331	2433
45-50	24.22	3878.37	0.002	25	24.023	1.103304	3774
50-55	19.39	3664.57	0.002	25	19.258	1.081999	5286
55-60	19.26	3479.7	0.002	25	19.125	1.08141	6149
60-65	18.94	3321.7	0.002	25	18.804	1.079911	7399
65-70	18.4	3161.08	0.002	25	18.267	1.077466	9320
70-75	17.65	3029.34	0.002	25	17.523	1.074124	12506
75-80	16.6	2883.71	0.002	25	16.483	1.069575	18717
80-85	15.26	2795.43	0.002	25	15.159	1.06375	34739
85-90	13.02	2700.18	0.002	25	12.944	1.054121	64254

Heating time of the heat accumualtor increases with the increase of temperature. Extending the heating time at higher levels of temperature is caused by lower temperature difference between the heat exchanging factors. Water heating to the temperature above 80ºC, due to duration of the process, is pointless. Typically, sufficient water temperature stored in the tank is at the level of 65-70ºC. This range of temperature allows its use both for central heating, hot water, as well as to thawing of cooling tunnel and other processes.

Conclusions

1. The process of heat exchange between the superheated vapour of refrigerant and water, taking place in the heat exchanger, has considerable values of the heat transfer coefficients, because of its turbulent nature. Much greater values (about 6 times) are obtained for the coefficient of heat transfer penetration from the liquid to the inner surface of the heat exchanger.

2. The maximum heat output of the heat exchanger is slightly higher than the value of 30 kW. Obtaining such power is possible when all compressors work and the refrigerant condensing temperature of 40ºC. Actual thermal power of the exchanger is usually smaller and depends primarily on the condensation temperature depending on ambient temperature conditioned and instantaneous loads of freezing tunnel.
3. The heat penetration coefficients determined in case of natural convection have low values in the range within aaproximately 13 to 35 $W \cdot m^{-2} \cdot K^{-1}$. The values of these coefficients in the analyzed temperature range initially increase with increasing temperature, and then decrease.
4. In the investigated temperature range the values of heat transfer coefficients from the heating liquid to the inner surface of the coil pipe decrease with increasing temperature.
5. Determined heat penetration coefficients in the analyzed exchanger, take small values. Improving the conditions of heat transfer in this case would be possible by applying a mixer in the heat accumualtor
6. The heating time of the heat accumulator increases with the temperature increase. In this case heating water to the temperature above 80ºC, is not justified from the energetic point of view.

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ANALIZA MOŻLIWOŚCI ODZYSKIWANIA I MAGAZYNOWANIA CIEPŁA Z INSTALACJI CHŁODNICZEJ TUNELU

Streszczenie. W pracy przeprowadzono analizę możliwości odzyskiwania i maga-zynowania energii cieplnej z instalacji chłodniczej fluidyzacyjnego tunelu zamrażalniczego a następnie jej magazynowania. Do odzysku ciepła wykorzystano płytowy wymiennik ciepła umieszczony za sekcją sprężarek a podgrzana woda opuszczająca wymiennik przekazywała ciepło poprzez wężownicę do zasobnika ciepła. Wyznaczono ilość ciepła możliwego do odzyskania z instalacji chłodniczej w zależności od obciążenia tunelu chłodniczego i zmiany temperatury wody w zasobniku w trakcie trwania procesu.

Key words: odzysk ciepła z instalacji chłodniczych, tunel zamrażalniczy, ciepło odpadowe