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Exergoeconomic optimization of a district cooling network

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Abstract

A district cooling system (DCS) is superior to conventional air conditioning as it helps to reduce energy consumption and protect the environment by reducing carbon dioxide emissions. The main disadvantages of a DCS are the high initial investment costs and the long payback period. The distribution network (DN) represents a large share of initial investment costs; therefore, it has a great impact on the decision to construct a DCS. In order to ensure the competitiveness of DCS, the DN has to be optimized. In this paper the exergoeconomic concept is applied to evaluate a DN in a DCS. The objective function in the analysis is defined as the exergy based cost of the final product-cold. The exergy-based cost of cold depends on the total annual cost of a DN, the input exergy to the DN, the exergy losses and the exergy destruction. The aim of this study is to find the exergetic optimal pipe diameter and the insulation thickness, as well as the exergoeconomic optimal pipe diameter and the insulation thickness. The analysis was made for different cooling capacities and for two types of pipes: pre-insulated steel pipes, where the insulation material is polyurethane, and polyethylene pipes, without any insulation.

Keywords: Exergoeconomic Analysis, District Cooling Network, Optimal Diameter, Exergoeconomic Optimal Insulation Thickness

Nomenclature

c	Specific costs (€/kWh)	DCS	District cooling system
c_p	Specific heat capacity (J/(kg·K))	DN	District network

C	Internal pipe circumference (m)	el	Electric
$\dot{C}$	Cost rate associated with an exergy stream (€/s)	Ex	Exergy
d	Internal diameter (m)	F	Fuel
e	Specific exergy (J/kg)	g	Heat gains
E	Exergy (J, kWh)	i	Internal
$\dot{E}$	Exergy rate (W)	in	Inlet
f	Darcy's friction factor (/)	L	Losses
h	Specific enthalpy (J/kg)	m	Motor
L	Length (m)	OM	Operating and maintenance
$\dot{m}$	Mass flow rate (kg/s)	out	Outlet
P	Power (W)	P	Product
$\dot{q}$	Specific heat rate (W/m)	PE	Polyethylene
$\dot{Q}$	Heat rate (W)	$pipe$	Pipe
s	Specific entropy (J/(kg·K))	PU	Polyurethane
T	Temperature (K)	$pump$	Pump
U	Overall heat-transfer coefficient (W/m ² K)	Q	Heat
v	Velocity (m/s)	r	Return
$\dot{w}$	Specific pumping power (W/m)	s	Supply
$\dot{W}$	Pumping power (W)	$soil$	Soil
$\dot{Z}$	Non-exergy associated cost rate (€/s)	w	Water
z_f	Specific cost (€/m)	W	Work
$\dot{z}_f$	Specific cost rate (€/m·s)		
<i>Abbreviations</i>		<i>Greek symbols</i>	
0	Environment (reference state)	ε	Exergetic efficiency (/)
CI	Capital investment	η	Energetic efficiency (/)
<i>construction</i>	Construction costs	Θ	Temperature, (°C)
D	Destruction		

1. Introduction

An increase in the standard of working and living conditions is related to different cooling applications. The problems associated with cooling are frequently solved with partial solutions, such as small local cooling units installed on the façade of the building. These local cooling units are, in most cases, over-dimensioned for the rooms they are cooling, they often use environmentally harmful refrigerants, and the consumption of electrical energy is not well controlled or monitored. District cooling systems (DCSs) represent a solution to these problems. They reduce the emissions of hazardous substances, they can be more cost effective (due to economies of scale), and more energy efficient compared to small cooling devices. In the case of cold production by an absorption chiller the heat from cogeneration can be used in order to increase the energy efficiency of a cogeneration system. The disadvantages of a DCS are associated with the distribution network: the higher initial investment costs and the costs associated with heat gains and hydraulic losses. There are many studies

dealing with the thermodynamic optimization of different distribution network parameters, such as pipe insulation [1-5], pipe design [6-11] and pipe dimensioning [12, 13].

Exergoeconomics is a branch of engineering that combines the concepts of exergy and economic analyses. For an exergoeconomic analysis of any energy system, all the exergy flows and the use of exergy for each component have to be known. In the economic analysis the capital investment and operating costs of each component of the energy system are determined. Exergoeconomics enables the calculation of the exergoeconomic variables. In the case of an inadequate state, exergoeconomic optimization can be performed in order to find the minimum price of the product, in our case cold. An exergoeconomic analysis plays an important role in analysing, planning and optimizing the energy systems. Several studies were conducted in the field of the exergoeconomics of district energy systems [14-18]. Most of them were dealing with existing district heating systems.

In this paper the exergoeconomic concept is applied to the distribution network of a DCS. This was done in order to take into account all the exergy flows in the network, i.e., exergy associated with heat gains as well as the exergy associated with the viscous losses (pumping) due the distribution of the cold through the pipelines. Since the thermoeconomic concept cannot be efficiently applied in such cases (different exergy potential of heat and electricity), the exergoeconomic concept has been applied. It enables the evaluation of different types of energy which are related to the distribution network of the DCS. The distribution network has a crucial impact on the investment costs and, consequently, on the price of the final product, i.e., cold. The investment in the district cooling pipe network mainly depends on the pipe length and diameter [19]. An over-dimensioned district cooling network (DCN) increases the total installation and operating costs, while an under-dimensioned DCN can affect the supply of cold. The heat gains in a DCN are affected by the pipe diameters and the insulation material used, as well as the temperature of the medium in the supply and return pipes and the temperature of the soil. There are two main motivations for minimizing the heat gains in a distribution network [20]. Firstly, the temperature increase in the supply affects the designed cooling capacity for the consumers, and secondly, more cooling energy has to be supplied to the system to maintain the desired temperatures, which leads to higher primary energy consumption of the DCS. The pipe diameters also have an impact on the pressure losses and, consequently, on the electrical energy consumption of the pumps. The results of the exergoeconomic optimisation are in this study presented as a function of two crucial parameters: pipe diameter and insulation thickness.

A review of the literature [1-13] shows that many thermoeconomic analyses were published in the field of optimization of district energy networks, especially district heating networks. However, the exergy value of all inlet and outlet energy flows is not taken into consideration in a thermodynamic analysis; therefore, electric energy and heat are not mutually comparable. The amount of energy as well as its quality is taken into consideration only by using the exergoeconomic approach. Our

research of the existing literature reveals that there are no studies that apply the exergoeconomic approach in optimization of a district cooling network.

2. Mathematical modelling of the problem

Cold is transported to every consumer in the DCS by a branched network. The network is composed of supply and return pipes, which are branched from the central cooling station, where cold is produced, to every consumer in the DCS. In order to reduce the heat gains, caused by the temperature difference between the water temperature inside the pipeline and the temperature of the soil, and the mutual interaction between the supply and return pipe, the pipe diameters can be reduced. The reduction of the pipe diameter has an impact on the pressure losses and consequently on the electrical energy consumption of the pumps.

2.1 Energy Equation

In a district cooling pipeline only heat and work can be exchanged across the boundary of the system. The water temperature along the length of the pipeline can be determined using an energy equation. **Fig. 1** shows the internal flow in a pipe, where the mass flow remains constant. The total pressure drop in the pipeline consists of the pressure loss due to viscous effects in the straight pipes (major losses) and the pressure drop in the various other components (minor loss) [21].

To cover the pressure losses, pumping power is needed; therefore, pumps have to be located along the network. It is assumed that a specific pumping power per length, see Eq. (1), is being supplied for the transport of the water through the pipeline. The friction factor for the turbulent flow in a straight pipe is calculated using the Colebrook correlation [23], and a friction factor of 20 % is considered to cover the minor losses [24]. The pump efficiency (includes internal mechanical, hydraulic and volumetric losses) is not considered in Eq. (1). The aim is to calculate the amount of heat gains generated because of the viscous losses which occur during transport of water through the pipeline and not the power supplied to the pumps. For more realistic results the pump efficiency, which is summarized in Ref. [25-28], could be considered in the calculations.

$$\dot{w} = \frac{(1.2 f) \dot{m} v^2}{2 d} . \quad (1)$$

The specific heat flux is defined as:

$$\dot{q}_g = U C_i (\Theta_{soil} - \Theta_w) . \quad (2)$$

where U is the overall heat-transfer coefficient, as in [29], and C_i is the internal pipe circumference. According to the control volume in **Fig. 1** the energy balance is defined as:

$$\dot{m} c_p [(\Theta_w + d\Theta_w) - \Theta_w] = d\dot{Q}_g + dP = \dot{q}_g dx + \dot{w} dx. \quad (3)$$

Eq. (3) is a first-order differential equation that can be solved by reorganization, substitution and integration. The outlet temperature of the water along the length of the pipe is a solution of Eq. (3) and is given by Eq. (4):

$$\Theta_{w,out} = \Theta_{soil} - \frac{\left(e^{\left(\frac{U C_i L}{\dot{m} c_p} \right)} \cdot \left(U C_i (\Theta_{soil} - \Theta_{w,in}) + \frac{(1.2 f) \dot{m} v^2}{2 d} \right) \right) - \frac{(1.2 f) \dot{m} v^2}{2 d}}{U C_i}. \quad (4)$$

The pumping power needed to supply the cold water to the consumers is calculated using Eq. (5). For the calculation of the pipe heat gains Eq. (6) is used:

$$\dot{W} = \dot{w} \int_0^L dx = \frac{(1.2 f) \dot{m} v^2}{2 d} L. \quad (5)$$

$$\dot{Q}_g = \dot{m} c_p (\Theta_{w,out} - \Theta_{w,in}) - \dot{W} \quad (6)$$

2.2 Exergetic Analysis

The system boundaries used in this paper are defined according to a widely used and accepted theory in exergoeconomics, presented in [30]. The reference state is the state where the DCS is in equilibrium with the environment. The central cooling station and the consumers are physically separated from the network. Thus, only thermal energy is being transferred in an interaction with the environment, while the pressure and chemical potential remain unchanged. The available exergy in a water flow is therefore defined by calculating the thermal part of the physical exergy:

$$e_w = \left[(h_w - h_{w,T_0}) - T_0 (s_w - s_{w,T_0}) \right]_{p=const.} \quad (7)$$

The exergy balance for a component as well as for the overall system can be written in the form: exergy of fuel (E_F)/exergy of product (E_P) [30]. The exergy of the product (E_P) is the desired result, expressed in exergy terms, achieved by the system (component). In our case it represents the exergy of the cold supplied to the consumers.

The exergy of the fuel E_F is the exergetic resource expended to generate the E_P . According to **Fig. 2**, the E_P and E_F of a DC network are calculated as:

$$\dot{E}_P = \dot{E}_{s,v2} - \dot{E}_{r,v2} \quad (8)$$

$$\dot{E}_F = \dot{E}_{s,v1} - \dot{E}_{r,v1} + \dot{W}_s + \dot{W}_r. \quad (9)$$

The exergetic efficiency for the selected control volume is calculated using Eq. (10):

$$\varepsilon = \frac{E_p}{E_F}. \quad (10)$$

A very important step in an exergy analysis is the system definition. The choice of boundaries can determine whether the effect of heat transfer should be charged as exergy destruction ($\dot{E}_D$) or exergy losses ($\dot{E}_L$). **Fig. 2** shows the boundaries of the control volume for the internal flow pipe. The heat gains of the pipe are indicated as $\dot{Q}_{g,s}$ for the supply and $\dot{Q}_{g,r}$ for the return pipe. Two different boundaries, denoted I and II, were defined [30]. Boundary I includes only the supply and the return pipe. Boundary II is located outside the system, where the temperature corresponds to the ambient temperature T_0 . The exergy balance for one pipe can be written as:

$$\dot{E}_F = \dot{E}_p + \dot{E}_D + \dot{E}_L. \quad (11)$$

With boundary I the rate of exergy loss equals the rate of exergy transfer associated with the heat transfer $\dot{E}_Q$, and thus given by Eq. (12) for supply pipeline and by Eq. (13) for return pipeline:

$$\dot{E}_{L,s}^{\text{Boundary I}} = \dot{E}_Q = \int_0^L \left(\frac{T_0 - T_w}{T_w} \right) \cdot \dot{q}_{g,s} \cdot dx. \quad (12)$$

$$\dot{E}_{L,r}^{\text{Boundary I}} = \dot{E}_Q = \int_0^L \left(\frac{T_0 - T_w}{T_w} \right) \cdot \dot{q}_{g,r} \cdot dx \quad (13)$$

With boundary II, heat transfer occurs at temperature T_0 . There is no associated exergy transfer, which means that $\dot{E}_Q = 0$, as well as $\dot{E}_L = 0$. The exergy destruction term accounts for the total exergy destruction owing to the friction and the irreversibility of the heat transfer. For boundary II, this can be written as:

$$\begin{aligned} \dot{E}_D^{\text{Boundary II}} &= \dot{E}_F - \dot{E}_p = \\ &= \dot{E}_{s,v1} - \dot{E}_{r,v1} + \dot{W}_s + \dot{W}_r - \dot{E}_{s,v2} + \dot{E}_{r,v2}. \end{aligned} \quad (14)$$

The sum of the exergy destruction and exergy losses, $\dot{E}_D + \dot{E}_L$, remains constant for given inlet and outlet states, regardless of which boundary, I or II, is chosen. The relation between the exergy destruction and the exergy loss for the different boundaries is described by:

$$\dot{E}_D^{\text{Boundary II}} = \dot{E}_D^{\text{Boundary I}} + \dot{E}_{L,s}^{\text{Boundary I}} + \dot{E}_{L,r}^{\text{Boundary I}}. \quad (15)$$

According to Eq. (12), (13), (14) and (15) the exergy destruction for boundary I can be written as:

$$\begin{aligned}\dot{E}_D^{\text{Boundary I}} &= \dot{E}_D^{\text{Boundary II}} - \dot{E}_{L,s}^{\text{Boundary I}} - \dot{E}_{L,r}^{\text{Boundary I}} \\ &= \dot{E}_{s,v1} - \dot{E}_{s,v2} + \dot{W}_s - \dot{E}_{L,s} + \dot{E}_{r,v2} - \dot{E}_{r,v1} + \dot{W}_r - \dot{E}_{L,r}\end{aligned}\quad (16)$$

2.3 Economic Analysis

The total costs for the network are made up of the investment costs for the pipes, the investment costs for the pumps, the pumping costs and the construction costs. In reference [32] the pipe costs and construction costs were determined with a polynomial equation of the 2nd degree. In reference [33] the polynomial equation is used to calculate the costs of insulated pipes, depending on the pipe diameter. In reference [34] the market costs of the insulated pipes, the installation and the operating costs are determined for Sweden and Canada. There are seven different types of pipe included in the cost analysis. The average values of the market costs for those pipes as a function of the different diameters are presented in reference [17].

The manufacturers of DCS equipment usually offer a warranty for their products. For the equipment that represents the main part of the district cooling network, i.e., the pipes, the warranty is 5 years for a lifetime of 30 years [35, 36]. In some research reports a lifetime of 40 [32] or even 50 years [37] is used in the calculations. The amortization period in the literature for district energy networks is 20 [7, 19] or 25 years [34].

Two types of pipes for the distribution of cold are considered in this study: pre-insulated steel pipes (insulation is polyurethane) and polyethylene pipes (no insulation).

The following calculations were used for the investment costs of the pipes:

- The outer diameter of the pre-insulated steel pipes is determined as a linear function of the inner diameter, using actual data from the manufacturers [35, 37]:

$$d_{out,PU} = 1.315 \cdot d_{in} + 59.36 . \quad (17)$$

- The specific cost of the pre-insulated steel pipes is determined as a polynomial function of the 2nd degree, using average values of the actual data from the manufacturers in Slovenia [36, 38]. The coefficients are $a=0.0012 \text{ €/}(\text{m} \cdot \text{mm}^2)$, $b=0.2229 \text{ €/}(\text{m} \cdot \text{mm})$ and $c=9.117 \text{ €/m}$.

$$z_f = a \cdot d_{in}^2 + b \cdot d_{in} + c . \quad (18)$$

- The price of the polyethylene pipes (PE) is determined using Eq. (18). Data are obtained from the manufacturers of PE in Slovenia [39]. The coefficients are: $a=0.001 \text{ €/}(\text{m} \cdot \text{mm}^2)$, $b=-0.0023 \text{ €/}(\text{m} \cdot \text{mm})$ and $c=0.2227 \text{ €/m}$.

- The separate prices of steel pipes and polyurethane insulation are calculated using the actual data from the manufacturer [38, 40]. The actual prices of the insulation do not depend on the thickness, but on customer demand (although the average price of insulation is determined as 372.84 €/m³ (see Fig. 3)).

- The price of steel pipes is determined according to reference [38, 40] as the difference between the price of pre-insulated pipes and the price of insulation for these pipes. Eq. (18) is used to calculate the price of the steel pipes. The coefficients are: $a=0.0009 \text{ €/}(\text{m} \cdot \text{mm}^2)$, $b=0.153 \text{ €/}(\text{m} \cdot \text{mm})$ and $c=7.8578 \text{ €/m}$.

- The specific construction costs were calculated by using a polynomial equation of the 2nd degree in reference [32]. The coefficients are: $a=0.0009 \text{ €/}(\text{m} \cdot \text{mm}^2)$, $b=0.4751 \text{ €/}(\text{m} \cdot \text{mm})$ and $c=284.06 \text{ €/m}$.

- The investment costs of the pumps are determined as in reference [32]. The pumping costs are calculated using Eq. (19). The cost of electricity and the efficiency of the motor are given in **Table 1**. The efficiency of the motor is not a fixed number but it changes with the load on the motor. The fixed efficiency of the motor 0.9, as an example, is used in our calculation [25].

$$\dot{Z}_{\text{pump},el} = c_{el} \cdot \frac{P_{\text{pump}}}{\eta_m}. \quad (19)$$

2.4. Exergoeconomic Model

The total cost of an element is made up of fixed and variable costs. In the case of a district cooling network the fixed costs include the material costs, costs of service, labour costs, maintenance costs, depreciation, financial expenses and other operating expenses. The variable costs include the costs of energy, the costs of electricity to power the pumps and the costs of chemically prepared water.

The exergoeconomic analysis of the k-th component [30] consists of a cost balance written as:

$$\sum_{out} \dot{C}_{out,k} = \dot{C}_{W,k} + \sum_{in} \dot{C}_{in,k} + \dot{Z}_k. \quad (20)$$

A cost balance shows that the sum of cost rates associated with all outlet exergy streams equals the sum of cost rates of all inlet exergy streams plus the appropriate charges due to capital investment ($\dot{Z}_k^{CI}$) and operating and maintenance expenses ($\dot{Z}_k^{OM}$). The sum of the last two terms is denoted by $\dot{Z}_k$. When a component receive the power (i.e. pump) the term $\dot{C}_{W,k}$ is on the right side of the expression. The Eq. (20) can be rearranged as:

$$\sum_{out} (c_{out,k} \cdot \dot{E}_{out,k}) = c_{W,k} \cdot \dot{W}_k + \sum_{in} (c_{in,k} \cdot \dot{E}_{in,k}) + \dot{Z}_k, \quad (21)$$

where the inlet and outlet exergy rates ($\dot{E}_{out,k}$, $\dot{E}_{in,k}$ and $\dot{W}_k$) are calculated in the exergy analysis in the previous stage, the variables $c_{out,k}$, $c_{in,k}$ and $c_{W,k}$ are the levelized costs per unit of exergy for the exergy streams associated with the k-th component.

In **Fig. 4** the cost balance in an elementary component of a DCS is shown. The most basic and elementary component of a DCS is an insulated pipe. As is clear from **Fig. 4**, an exergy stream is

entering the pipe with costs related to its generation (c_{in}). The specific exergy flow in the form of pumping power is supplied to the pipe ($\dot{w}$) and the specific cost rate of the pipe are specified as ($\dot{z}_f$). The cost balance for the elementary component of a DCS presented in Fig. 4 can be rewritten by using Eq. (20) and (21) as:

$$c_{out} \cdot \dot{E}_{out} = c_{in} \cdot \dot{E}_{in} + c_{el} \cdot \dot{w} \cdot L + \dot{z}_f \cdot L. \quad (22)$$

Exergy destruction and exergy losses affect the costs per unit of exergy of the exiting stream (c_{out}), which is higher compared to the corresponding costs of the entering exergy stream (c_{in}).

3. Results and discussion

The exergoeconomic model of a district cooling network was used to calculate the costs of cold at the primary side of the consumer. Two different types of pipes were considered in this analysis: pre-insulated steel pipes, where polyurethane was considered as an insulation material, and polyethylene pipes, without insulation. Firstly, the exergetic and exergoeconomic optimal diameter was determined for both types of pipes. In this case it was assumed that the thickness of the insulation material is linearly dependent on the inner diameter of the pipe (Eq. (17)). Real data from the manufacturers were used in this study. Secondly, the exergetic and exergoeconomic optimal thickness of the insulation material of the pre-insulated pipe was determined. Lastly, the exergetic and exergoeconomic optimal diameters with the corresponding optimal insulation thicknesses were determined.

The optimal values were determined by an iterative optimization. In this procedure the individual parameters are defined, for which a given amount of cold is transported with the highest exergetic efficiency or the lowest cost of exergy for cold to the consumer. To transport a sufficient amount of cold the mass flow-rates are also changed. With the optimal pipe diameters and the optimal insulation thickness, the object is a complete reconstruction of the network in an existing temperature regime.

In **Table 1** the parameters used in this study are listed. For the purpose of presenting the mathematical model, a 1000-m-long pipeline was selected.

3.1. Optimization of the pipe diameter – pipes available on the market

The objective of this study is to find the pipe diameter where the exergetic efficiency has its maximum value and the cost of exergy for cold achieves its minimum. A given amount of cold in a required temperature regime can be transported to the consumers through pipelines with different diameters. By increasing the diameter of the pipes, the velocity and the corresponding pumping power, decrease. Simultaneously, the overall heat-transfer coefficient decreases and the surface area for heat transfer increases. The internal diameters considered in this study were selected in the range from 40 mm to 600 mm. The outer diameter, in which the insulation thickness is taken into account, is calculated using Eq. (17). The exergetic efficiency is calculated using Eq. (10). The cost of exergy for the cold is calculated using Eq. (22). Real data from the manufacturers were used in this study.

In **Table 2** and **Fig. 5** the exergetic efficiency and the cost of exergy for the cold in the case of pre-insulated pipes are shown. The study was made for several cooling capacities: 50 kW, 100 kW, 200 kW, 500 kW and 1500 kW.

The exergetic efficiency increases by increasing the cooling capacity. The exergetic optimal diameter of the pre-insulated pipe is 144 mm ($\delta_i=52.4$ mm) when the cooling capacity is 50 kW. When the cooling capacity is 500 kW, the exergetic optimal diameter is 533 mm ($\delta_i=113.6$ mm).

The cost of exergy for cold can be decreased by increasing the cooling capacity. The exergoeconomic optimal diameter of the pre-insulated pipe is 66 mm ($\delta_i=40.1$ mm) when the cooling capacity is 50 kW, 185 mm ($\delta_i=58.8$ mm) when the cooling capacity is 500 kW and 299 mm ($\delta_i=76.8$ mm) when the cooling capacity is 1500 kW.

Fig. 6 shows the exergetic efficiencies and cost of exergy for cold in the case of polyethylene pipes without insulation. The same cooling capacities as in the previous case were taken into account.

The optimal diameters in the case of polyethylene pipes are smaller compared to the pre-insulated steel pipes because the pipe roughness of the polyethylene pipes is substantially lower (see **Table 1**). Therefore, the pressure drop is lower and a smaller mass flow is needed to transport the same amount of cold. The exergetic efficiency in the case of polyethylene pipes is lower compared to the pre-insulated pipes. The main reason is the higher heat gains when non-insulated polyethylene pipes are used to supply the cold.

In the exergoeconomic analysis the optimal diameters are slightly larger compared to the pre-insulated steel pipes. The minimum cost of exergy for cold at a cooling capacity 50 kW is 4.64 €/kWh_{Ex} (for d=69 mm) when polyethylene pipes are used, and 4.03 €/kWh_{Ex} when pre-insulated pipes are used (for d=66 mm). By increasing the cooling capacity to 1500 kW the cost of cold decreases in both cases: it is 2.83 €/kWh_{Ex} (for d=300 mm) for polyethylene pipes and 2.81 €/kWh_{Ex} (for d=299 mm) for pre-insulated pipes.

With the sensitivity analysis the impact of different parameters on the final cost of exergy for cold is studied. Four different parameters are discussed: specific cost of pre-insulated steel pipes ($z_{f,pipe}$), specific construction costs ($z_{f,construction}$), price of electricity (c_{el}) to drive the circulation pumps and the price of inlet exergy flow of cold (c_{in}) which covers all costs of the production of the cold. The values of those parameters vary for $\pm 30\%$ from the reference values given in Section 2.3 or in **Table 1**. The results of sensitivity analysis for cooling capacities 500 kW and 1500 kW are shown in **Fig. 7**.

The impact of specific cost of pre-insulated steel pipes, specific construction costs and price of electricity to drive the circulation pumps on the cost of exergy for cold is very small for the selected operating conditions given in **Table 1**. The price of inlet exergy flow which depends on the total production costs has the greatest impact on the cost of exergy for cold. By increasing the c_{in} for 30 % the c_{Ex} increases for approx. 30 % in both cases on **Fig. 7**.

3.2. Optimization of the insulation thickness

In this section the exergetic and exergoeconomic optimal thicknesses are calculated. The insulation thickness was selected in the range from 0 mm to 300 mm. The insulation material is the polyurethane (see **Table 1**). In this particular case cooling capacities of 50 kW, 200 kW and 500 kW were considered in the analysis. The calculations were performed for different diameters.

In **Fig. 8** the exergetic efficiency and the cost of exergy for cold as a function of insulation thickness are shown. The cooling capacity is 50 kW.

The exergetic efficiency increases when increasing the insulation thickness because the heat gains decrease. The lowest exergetic efficiency is reached when a pipe with $d=50$ mm is considered. The main reason is attributed to higher pressure losses and, consequently, to a greater pumping power. The pressure drop in the pipeline as well as the heat gains affect the exergetic efficiency. A larger diameter reduces the pressure losses, but due to the larger surface area the heat gains increase.

The lowest value of the cost of exergy for cold (cooling capacity 50 kW) is achieved in the case of a pipe with $d=66$ mm. The exergoeconomic optimal insulation thickness for this case is 51 mm.

Fig. 9 shows the exergetic efficiency and the cost of the exergy for cold for a cooling capacity of 200 kW.

When $d=80$ mm, the exergetic efficiencies are the lowest. By increasing the insulation thickness, the exergetic efficiency increases. The lowest cost of exergy for cold is achieved at an

optimal pipe diameter $d=123$ mm when the insulation thickness is 55 mm. The highest values are achieved for $d=300$ mm due to the high investment costs for the insulation material and for $d=80$ mm due to the high pressure losses.

Fig. 10 shows the exergetic efficiency and the cost of the exergy for cold for a cooling capacity of 500 kW. In the case of the pipe with $d=80$ mm the required cooling capacity cannot be ensured, because the pressure drop is too high. Therefore, this example is not shown in **Fig. 10**.

The exergetic efficiency is higher in the case of a pipe with $d=300$ mm compared to $d=150$ mm. The reason is the lower pressure drop due to a larger pipe diameter. The cost of exergy for cold is the lowest when pipes with $d=185$ mm are considered. The optimal exergoeconomic insulation thickness for this case is 57 mm.

Comparing the values from **Fig. 8, 9 and 10**, it can be seen that the exergetic efficiency increases and the cost of cold decreases by increasing the cooling capacity. The same results were also established in **Section 3.1**.

3.3. Optimization of the pipe diameter and the insulation thickness

In this section the maximum exergetic efficiency and the minimum cost of exergy for cold are determined for each cooling capacity discussed in the previous sections. The exergetic and exergoeconomic optimal diameters with the corresponding optimal insulation thicknesses were determined using the iterative optimization.

In **Table 3** and **Fig. 11** the results of exergetic and exergoeconomic analyses for different cooling capacities are presented.

The optimal diameters in the exergetic analysis are larger compared to the optimal diameters in the exergoeconomic analysis because the pumping power has a large impact on the exergetic efficiency (see Eq. (9) and (10)).

Comparing the results from **Sections 3.1 and 3.3** it is clear that the pre-insulated steel pipes available on the market achieve slightly lower exergy efficiencies and slightly higher costs of exergy for cold compared to the ideal case where optimal diameters and corresponding optimal insulation thicknesses are calculated. In the exergy analysis the exergetic efficiency can be improved by decreasing the pipe diameter and increasing the insulation thickness. In the exergoeconomic analysis the thickness of the insulation material has the greatest impact on the exergy costs for cold. Therefore, the insulation thickness must be increased.

4. Conclusions

The exergetic efficiency of supplying cold in a district cooling network can be used as a performance criterion for designing more exergy-efficient systems. With an exergoeconomic approach, cost optimization can also be included in the consideration.

For the purpose of the analysis, the exergetic and exergoeconomic optimal diameters and the optimal insulation thicknesses were studied for several cooling capacities, using particular prices of heat and electric energy in Slovenia. The study was conducted for two different types of pipes: the pre-insulated steel pipes, where polyurethane is used as an insulation material, and the polyethylene pipes, without any insulation. The length of the pipeline was 1000 m and the properties of pure water without anti-corrosion and anti-icing inhibitors were applied in the calculations. In a real-case scenario the addition of inhibitors is required, which affects the heat transfer and the pressure drop. When the exergoeconomic approach is considered, the optimal pipe diameters are smaller compared to the case where the maximum exergetic efficiency of the cold supply is the objective function. The exergetic efficiency increases by increasing the cooling capacity. On the other hand, the cost of the exergy for cold decreases under such conditions. By comparing the optimal diameters of the pre-insulated steel pipes and the polyethylene pipes it can be concluded that the exergetic and exergoeconomic optimal diameters are smaller in the case of pre-insulated pipes. By increasing the insulation thickness, the exergetic efficiency increases.

The exergoeconomic analysis presented in this paper deals with the exemplary cases that are based on the particular prices of heat and electric energy in Slovenia. The results presented in Section 3 reveal the parameters at which the minimum cost of exergy of cold and maximum exergetic efficiency for the considered case can be obtained. The results of the analysis provided in this paper can be used as an illustration of the application of the exergoeconomic approach. The new exergoeconomic model can be used as a tool for the optimisation of future design and for the optimisation of existing district cooling networks. In the future, further work is required in order to determine the minimum exergoeconomic cost of cold for the entire district cooling system. This cost of cold will be the indicator of the actual cost that is paid by the consumer. In our future work not only will the district network be included, but also central cooling stations, where cold is produced using different cooling technologies, and the consumer substation. The results will be used to determine the exergoeconomically most efficient technology for a district cooling system.

References

- [1] Başoğlu, Y.; Keçebaş, A., Economic and environmental impacts of insulation in district heating pipelines. *Energy* 2011, 36, (10), 6156-6164.

- 348 [2] Kaynakli, O., Economic thermal insulation thickness for pipes and ducts: A review study. *Renewable*
349 *and Sustainable Energy Reviews* 2014, 30, (0), 184-194.
- 350 [3] Keçebaş, A.; Ali Alkan, M.; Bayhan, M., Thermo-economic analysis of pipe insulation for district
351 heating piping systems. *Applied Thermal Engineering* 2011, 31, (17–18), 3929-3937.
- 352 [4] Keçebaş, A., Determination of insulation thickness by means of exergy analysis in pipe insulation.
353 *Energy Conversion and Management* 2012, 58, (0), 76-83.
- 354 [5] Ljubenko, A.; Poredoš, A.; Zager, M., Effects of Hot-Water-Pipeline Renovation in a District Heating
355 System. *Strojniški vestnik-Journal of Mechanical Engineering* 2011; Vol. 57.
- 356 [6] Dalla Rosa, A.; Li, H.; Svendsen, S., Method for optimal design of pipes for low-energy district heating,
357 with focus on heat losses. *Energy* 2011, 36, (5), 2407-2418.
- 358 [7] Yildirim, N.; Toksoy, M.; Gokcen, G., Piping network design of geothermal district heating systems:
359 Case study for a university campus. *Energy* 2010, 35, (8), 3256-3262.
- 360 [8] Chan, A. L. S.; Hanby, V. I.; Chow, T. T., Optimization of distribution piping network in district
361 cooling system using genetic algorithm with local search. *Energy Conversion and Management* 2007,
362 48, (10), 2622-2629.
- 363 [9] Li, X.-l.; Duanmu, L.; Shu, H.-w., Optimal design of district heating and cooling pipe network of
364 seawater-source heat pump. *Energy and Buildings* 2010, 42, (1), 100-104.
- 365 [10] Chow, T. T.; Au, W. H.; Yau, R.; Cheng, V.; Chan, A.; Fong, K. F., Applying district-cooling
366 technology in Hong Kong. *Applied Energy* 2004, 79, (3), 275-289.
- 367 [11] Öztürk, İ. T.; Karabay, H.; Bilgen, E., Thermo-economic optimization of hot water piping systems: A
368 comparison study. *Energy* 2006, 31, (12), 2094-2107.
- 369 [12] Tol, H. İ.; Svendsen, S., Improving the dimensioning of piping networks and network layouts in low-
370 energy district heating systems connected to low-energy buildings: A case study in Roskilde, Denmark.
371 *Energy* 2012, 38, (1), 276-290.
- 372 [13] Tol, H. İ.; Svendsen, S. In Optimal dimensioning of low-energy district heating networks with
373 operational planning—case study for existing buildings, *Proceedings of the 11th International*
374 *Conference on Sustainable Energy Technologies*, 2012; 2012; pp 113-122.
- 375 [14] Hepbasli, A., A review on energetic, exergetic and exergoeconomic aspects of geothermal district
376 heating systems (GDHSs). *Energy Conversion and Management* 2010, 51, (10), 2041-2061.
- 377 [15] Keçebaş, P.; Gökgedik, H.; Alkan, M. A.; Keçebaş, A., An economic comparison and evaluation of two
378 geothermal district heating systems for advanced exergoeconomic analysis. *Energy Conversion and*
379 *Management* 2014, 84, (0), 471-480.

- 380 [16] Ozgener, L.; Hepbasli, A.; Dincer, I.; Rosen, M. A., Exergoeconomic analysis of geothermal district
381 heating systems: A case study. *Applied Thermal Engineering* 2007, 27, (8–9), 1303-1310.
- 382 [17] Ljubenko, A.; Poredoš, A.; Morosuk, T.; Tsatsaronis, G., Exergoeconomic analysis of heat supply by a
383 district heating system. *Proceedings of ECOS 2013-The 26th international conference on efficiency,*
384 *cost, optimization, simulation and environment impact of energy systems*, Guilin, China, 2013; Guilin,
385 China, 2013.
- 386 [18] Oktay, Z.; Dincer, I., Exergoeconomic analysis of the Gonen geothermal district heating system for
387 buildings. *Energy and Buildings* 2009, 41, (2), 154-163.
- 388 [19] Zinko, H.; Bøhm, B.; Kristjansson, H.; Ottosson, U.; Rama, M.; Sipila, K., District heating distribution
389 in areas with low heat demand density. *International Energy Agency* 2008.
- 390 [20] Calance, M. A., *Energy Losses Study on District Cooling Pipes: Steady-state Modeling and Simulation.*
391 2014.
- 392 [21] Munson, B. R.; Young, D. F.; Okiishi, T. H., *Fundamentals of fluid mechanics.* New York: 1990.
- 393 [22] Ljubenko, A.; Poredoš, A.; Morosuk, T.; Tsatsaronis, G., Performance Analysis of a District Heating
394 System. *Energies* 2013, 6, (3), 1298-1313.
- 395 [23] Bergman, T. L.; Incropera, F. P.; Lavine, A. S., *Fundamentals of heat and mass transfer.* John Wiley &
396 Sons: 2011.
- 397 [24] Li, H.; Svendsen, S., Energy and exergy analysis of low temperature district heating network. *Energy*
398 2012, 45, (1), 237-246.
- 399 [25] Nourbakhsh, A.; Jaumotte, A.; Hirsch, C.; Parizi, H.B., *Turbopumps and pumping systems.* Springer
400 Science & Business Media: 2007.
- 401 [26] Girdhar, P.; Moniz, O., *Practical centrifugal pumps.* Elsevier: 2011.
- 402 [27] Lobanoff, V.S.; Ross, R.R., *Centrifugal pumps: design and application.* Elsevier: 2013.
- 403 [28] Pumping System Assessment Tool (PSAT). U.S. Department of Energy, Oak ridge National
404 Laboratory, <https://www.energy.gov/eere/amo/articles/pumping-system-assessment-tool>; 2008
405 [accessed 26.5.2017].
- 406 [29] Bøhm, B., On transient heat losses from buried district heating pipes. *international Journal of energy*
407 *research* 2000, 24, (15), 1311-1334.
- 408 [30] Bejan, A.; Moran, M. J., *Thermal design and optimization.* Wiley. com: 1996.
- 409 [31] Morosuk, T.; Tsatsaronis, G., Graphical models for splitting physical exergy. *Shaping our future energy*
410 *systems* 2005, 1, 377-384.

- 411 [32] Dobersek, D.; Goricanec, D., Optimisation of tree path pipe network with nonlinear optimisation
412 method. *Applied Thermal Engineering* 2009, 29, (8–9), 1584-1591.
- 413 [33] Verda, V.; Ciano, C., Procedures for the search of the optimal configuration of district heating
414 networks. *International Journal of Thermodynamics* 2005, 8, (3), 143-153.
- 415 [34] Dalla Rosa, A.; Boulter, R.; Church, K.; Svendsen, S., District heating (DH) network design and
416 operation toward a system-wide methodology for optimizing renewable energy solutions (SMORES) in
417 Canada: A case study. *Energy* 2012, 45, (1), 960-974.
- 418 [35] LOGSTOR, Pipe pricelist. In 2016.
- 419 [36] Olsen, P.; Lambertsen, H.; Hummelshøj, R.; Bøhm, B.; Christiansen, C.; Svendsen, S.; Larsen, C.;
420 Worm, J. In A new low-temperature district heating system for low-energy buildings. *Proceedings of*
421 *the 11th International Symposium on District Heating and Cooling, Iceland, 2008; 2008.*
- 422 [37] Sartor, K.; Quoilin, S.; Dewallef, P., Simulation and optimization of a CHP biomass plant and district
423 heating network. *Applied Energy* 2014, 130, 474-483.
- 424 [38] PURLEN, Price list of pre-insulated steel pipes: Series 1. In 2016.
- 425 [39] TATRA, Pipes PE 80 and PE 100 for water supply network. In 2016.
- 426 [40] PURLEN, Pricelist of insulation for ductil pipes. In 2016.
- 427 [41] LOGSTOR, Product Catalogue -The bonded pipe system. In LOGSTOR, Ed. Denmark, 2012.
- 428 [42] Kraut, B.; Puhar, J.; Stropnik, J., Kraut's Engineering Handbook. In *Littera picta*, Ljubljana: 2001.
- 429 [43] Duh, T., Technical and economic analysis of district cooling system [BSc Thesis]. Ljubljana:
430 University of Ljubljana; 2013.
- 431 [44] Duh Čož, T.; Morosuk, T.; Kitonovski, A.; Poredoš, A., Exergoeconomic evaluation of different
432 types of absorption and compression chillers. *Proceedings of ECOS 2015-The 28th international*
433 *conference on efficiency, cost, optimization, simulation and environment impact of energy systems,*
434 *Pau, France, 2015; France: 2015.*

Figures

Fig. 1. Control volume for the internal flow in a pipe.

Fig. 2. Control volume for the internal flow in a pipe.

Fig.3. Real data and average value of the price of insulation.

Fig. 4. Cost balance in an elementary component of a district cooling system.

Fig. 5. Exergetic efficiency and cost of exergy for supplying cold through a 1000-m-long supply pipeline as a function of the cooling power and the pipe diameter (temperature of supply/return 7/12 °C: pre-insulated pipe).

Fig. 6. Exergetic efficiency and cost of exergy for supplying cold through a 1000-m-long supply pipeline as a function of the cooling power and the pipe diameter (temperature of supply/return 7/12 °C; polyethylene pipe without insulation).

Fig. 7. The impact of specific cost of pre-insulated steel pipes, specific construction costs, price of electricity to drive the circulation pumps and the price of inlet exergy flow of cold which covers all costs of the production of cold, on the final cost of exergy for cold, for the cooling capacities of 500 kW (Fig. 7a) and 1500 kW (Fig. 7b).

Fig. 8. Exergetic efficiency and the cost of exergy for supplying cold through a 1000-m-long supply line as a function of pipe diameter and insulation thickness, cooling capacity is 50 kW.

Fig. 9. Exergetic efficiency and cost of exergy for supplying cold through a 1000-m-long supply line as a function of pipe diameter and insulation thickness, cooling capacity is 200kW.

Fig. 10. Exergetic efficiency and cost of exergy for supplying cold through a 1000-m-long supply line as a function of pipe diameter and insulation thickness, cooling capacity is 500 kW.

Fig. 11. Exergetic efficiency and cost of exergy for supplying cold through a 1000-m-long supply line as a function of pipe diameter and insulation thickness.

Tables

Table 1

Parameters used in this study.

Fuel	PEF
Input temperature to the network	7 °C
Temperature of the soil	15 °C
Ambient (reference) temperature	25 °C
Operating period of the DCS	4000 h/y
Pressure in the pipeline	3 bar
Pipe-network lifetime	30 years [35]
Pipe roughness (steel pipe)	0.05 mm [41]
Pipe roughness (polyethylene pipe)	0.002 mm [42]
Depth of buried pipes	1.5 m
Length of pipeline	1000 m
Pump lifetime	10 years [32]
Efficiency of the motor	0.9 [25]
Thermal conductivity (polyurethane)	0.027 W/(m K) [35]
Thermal conductivity (polyethylene)	0.38 W/(m K) [39]
Thermal conductivity of soil	1.4 W/(m K)
Pump price	150 €/kW [32]
Price of electricity in Slovenia	0.132 €/kWh [43]
Price of inlet exergy to the network	2.7 €/kWh _{Ex} [44]

Table 2

Results from exergetic and exergoeconomic analyses.

$\dot{Q}_C$, kW	Exergetic analysis			Exergoeconomic analysis		
	d, mm	δ_i , mm	ε	d, mm	δ_i , mm	c_{Ex} , €/kWh
50	144	52.4	0.938	66	40.1	4.03
100	211	62.9	0.963	90	43.9	3.44
200	313	79.0	0.979	123	49.1	3.13
500	533	113.6	0.990	185	58.8	2.92
1500	>600	N/A	N/A	299	76.8	2.81

Table 3

Results from exergetic and exergoeconomic analyses.

$\dot{Q}_C$, kW	Exergetic analysis			Exergoeconomic analysis		
	d, mm	δ_i , mm	ε	d, mm	δ_i , mm	c_{Ex} , €/kWh
50	134	56.2	0.941	66	51.6	4.02
100	187	57.6	0.962	90	53.6	3.43
200	260	58.4	0.976	123	55.6	3.12
500	399	58.6	0.986	186	57.4	2.92
1500	>600	N/A	N/A	303	58.5	2.81