Strojniški vestnik - Journal of Mechanical Engineering 53(2007)9, 548-555
UDK - UDC 697.44
Kratki znanstveni prispevek - Short scientific paper (1.03)

Toplotno-gospodarsko optimiranje toplovodnega sistema: Parametrična raziskava vpliva pogojev sistema

The Thermo-Economic Optimization of Hot-Water Piping Systems: A Parametric Study of the Effect of the System Conditions

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V prispevku smo predstavili toplotno-gospodarsko metodo optimiranja toplovodnega sistema. Metoda temelji na drugem zakonu termodinamike. Sočasno, ob upoštevanju eksergijske razgradnje zaradi trenja in eksergijske izgube zaradi toplotnih izgub in stroškov delovanja, smo določili optimalni premer cevi in debelino izolacije, medtem ko smo stroške cevovoda in izolacije obravnavali kot investicijo. S parametrično raziskavo smo predstavili vpliv masnega pretoka, letni čas obratovanja, amortizacijsko dobo ter temperaturo vode. Rezultati kažejo, da na optimalni premer cevi najbolj vpliva masni pretok. Letni čas obratovanja, amortizacijska doba in temperatura vode so odločilni vplivi pri izbiri primerne debeline izolacije.

(Ključne besede: toplovodni sistemi, optimiranje, eksergijska razgradnja, eksergijske izgube)

A thermo-economic optimization method for a hot-water distribution pipe is presented. The method is based on the second law of thermodynamics. Both the optimum pipe diameter and the insulation thickness are determined simultaneously, considering exergy destruction due to friction and exergy loss due to heat losses as the operation cost, while the piping and insulation costs are considered as an investment. The effect of mass flow rate, annual operation time, depreciation period and water temperature on the optimum pipe diameter and insulation thickness are presented with a parametric investigation. The results show that the mass flow rate dominates the optimum pipe diameter. The annual operation time, depreciation period and water temperature are the decision parameters for the optimum insulation thickness.

(Keywords: hot water networks, optimizations, exergy destruction, exergy loss)

0 INTRODUCTION

Hot-water network systems distribute thermal energy from a central source to residents; these are widely used in district and geothermal heating systems. They are also like the arteries of nearly all industrial applications. This is why hot-water networks should be designed carefully, in order to be able to operate the systems efficiently. The heat and pressure losses must be decreased to save energy, and also the investment and operation costs of the piping system must be as low as possibility to save money.

A literature review shows that there are a few methods for piping techniques. The classical piping techniques have been presented in various text books and handbooks ([1] to [3]). The classical piping techniques can be easily applied to a hot-water network layout, but they are not suitable for present-day requirements, as they consider only the fluid flow aspects.

A design procedure has been developed by Jones and Lior [4] that involves solar-heating-system piping and water-storage tanks. Their optimization method considers cost minimization with the first law of thermodynamics and fluid-flow aspects. Another method presented Wepher et al. [5], for selecting the optimum pipe size and insulation thickness, is based on the minimization of the total cost of piping and insulation capital and operation costs using the second law of thermodynamics. A bleeder steam line was presented as a case study.

Wechsatol et al. ([6] and [7]) proposed a design procedure involving the optimum geometric layout of schemes of hot-water distribution over an area. In these studies, investment and operation costs, exergy costs, lifetime and operation time of the system and other economic aspects were not considered. Lorente et al. [8] included the effect of the exergy in the design procedure used by Wechsatol et al.

In a recent study, four different thermo-economic techniques for the optimum design of hot-water piping systems were compared [9]. A simultaneous determination of the pipe diameter and the insulation thickness based on a thermo-economic optimization with the second law of thermodynamics method was recommended for use in design studies.

In this paper the aim is to study and present the parameters affecting the system conditions, such as the mass flow rate, the annual operation time, the depreciation period and the fluid temperature on the decision variables.

1 THERMODYNAMIC MODEL FOR PIPE FLOW

The thermodynamic model based on the second law will be summarized as follows. The details can be found in [9]. A schematic representation of the insulated pipe segment is shown in Fig. 1. It is a long straight conduit segment, installed above the ground in an environment at a temperature and pressure (T_{out}, P_{out}) that are identical to those of the dead state. The assumptions are a constant environmental temperature for T_{out} and constant thermodynamic properties at an appropriate mean temperature. The pipe segment consists of a stainless-steel pipe, insulation material and a galvanized steel cover sheet. Hot water is pumped through the conduit; the thermodynamic variables at the entrance and the exit of the conduit segment are as shown in Fig. 1.

Assuming the pumping energy is very small in comparison to the energy transported through the pipe segment, the hot-water exit temperature, T_{exit}, from the pipe is [10]:

$$T_{exit} - T_{out} = (T_{\sup} - T_{out})\ e^{-\Gamma_1 \Gamma_2} \tag{1}$$

with:

$$\Gamma_1 = \left(\frac{2\pi}{\begin{array}{l}(1/r_{in}h_{in}) + (1/k_{pipe})\ln(r_{pipe}/r_{in}) + \\ (1/k_{ins})\ln(r_{out}/r_{pipe}) + (1/r_{out}(h_{out}+h_{rad}))\end{array}} \right) \tag{2}$$

$$\Gamma_2 = \left(\frac{L}{\dot{m} C_p} \right) \tag{3}$$

where the convection heat-transfer coefficient inside pipe, h_{in}, and the convection and radiation heat transfer coefficients outside pipe, h_{out} and h_{rad}, are calculated as [10]:

$$\frac{h_{in}\, D_{in}}{k} = 0.023\,\mathrm{Re}^{0.8}\,\mathrm{Pr}^{0.4} \tag{4}$$

$$h_{rad} = \frac{\sigma\varepsilon(T_{ms}^4 - T_{ms}^4)}{T_{ms} - T_{out}} \tag{5}$$

$$\begin{array}{l} h_{out} = 11.58\left(1/D_{out}\right)^{0.2}\left[2/\left(T_{ms}+T_{out}\right)\right]^{0.181} \\ \left(T_{ms}-T_{out}\right)^{0.266}\left(1+0.7935\,V_{wind}\right)^{0.5} \end{array} \tag{6}$$

Eq. (6) is valid for turbulent air flow and L/D_{out} >10. It is a general equation of the ASTM Standard C680 for computer calculations [11]. The mean outside surface temperature, T_{ms}, is calculated iteratively in Eqs. (5) and (6).

Assuming a constant heat capacity at the arithmetic mean temperature, the heat loss is calculated as:

$$\dot{Q}_{loss} = \dot{m} C_p (T_{\sup} - T_{exit}) \tag{7}$$

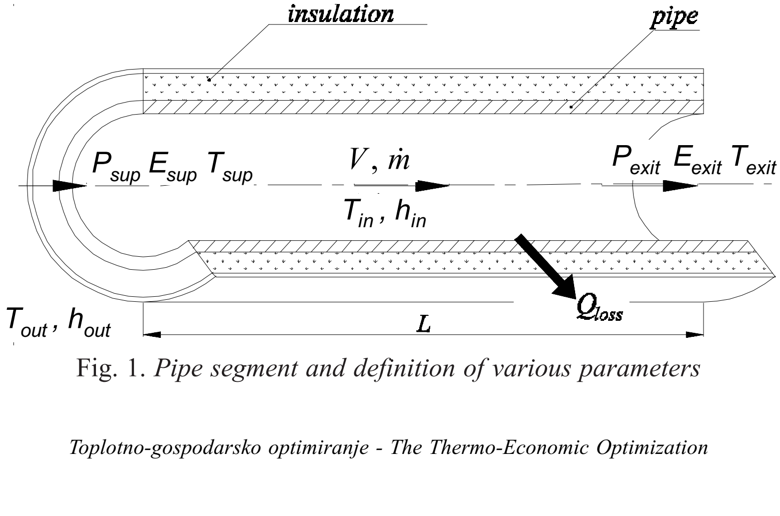

Fig. 1. *Pipe segment and definition of various parameters*

The pumping power is:

$$\dot{W}_p = \frac{\dot{m}\,\Delta P}{\rho \eta_p} \quad (8),$$

where ΔP is the pressure drop through the pipe, which is calculated using the Darcy-Weisbach equation:

$$\Delta P = \left(f \frac{L}{D} + \sum \xi \right) \rho \frac{V_m^{\ 2}}{2} \quad (9).$$

Here f is the Darcy–Weisbach friction factor, which is evaluated using the Colebrook correlation [12]. ξ is the pressure-loss coefficient of the fittings. The numerical values of ξ are evaluated as suggested in [2].

The exergy destruction due to friction is [13]:

$$\dot{E}_{des} = \frac{T_{out}}{T_m} \dot{W}_p \quad (10).$$

Here, $T_m=(T_{sup}+T_{exit})/2$ is the mean water temperature.

The exergy flow in the pipe is [13]:

$$\dot{E} = \dot{m} C_p \left[(T - T_{out}) - T_{out} \int_{Tout}^{T} \frac{dT}{T} \right] \quad (11).$$

The difference between the exergy supplied, $\dot{E}_{\sup}$ plus $\dot{W}_p$, and the exergy at the exit, $\dot{E}_{exit}$, will be the exergy loss from the pipe segment:

$$\dot{E}_{loss} = \dot{E}_{\sup} + \dot{W}_p - \dot{E}_{exit} \quad (12).$$

1.1 The objective function

Exergy destruction due to friction and exergy loss due to heat losses are considered as operation costs, while the piping and insulation costs are considered as investment. Using a similar method in [5], the objective function can be written as:

$$\dot{C}_{tot} = \dot{Z}_{pipe} + \dot{Z}_{ins} + c_{el} \dot{E}_{des,p} + c_e \dot{E}_{loss} \quad (13),$$

where c_{el}, c_e are the specific exergy costs for the electrical energy and the hot water.

The unit exergy cost of the hot water in a boiler, c_e, is calculated in a similar way to [3]:

$$c_e = \frac{c_f \dot{E}_f + \dot{Z}_b}{\dot{E}_{\sup} - \dot{E}_{ret}} \quad (14).$$

Here, c_f is the energy cost of the fuel. $\dot{E}_f$ is the annual fuel exergy flow rate. $\dot{Z}_b$ is the cost of capital annualized over the boiler lifetime and the cost of operation and maintenance. $\dot{Z}_b = (CRF + \varsigma) C_b$, $CRF = i/(1-(1+1)^{-n})$ is the capital recovery factor, ς (= 0.01) is a coefficient that accounts for part of the fixed operation and maintenance cost and C_b is the total capital cost of the boiler. $(\dot{E}_{\sup} - \dot{E}_{ret})$ is the hot-water supply and return exergy flow rates added in the boiler.

The capital costs of the pipe and the insulation segment annualized over its lifetime and the annual cost of the operation and maintenance are given in [3]:

$$\dot{Z}_{pipe} = (CRF + \varsigma) C_{pipe} \quad (15)$$

$$\dot{Z}_{ins} = (CRF + \varsigma) C_{ins} \quad (16),$$

where C_{pipe} and C_{ins} are the total cost of the pipe and the insulation, which are evaluated according to the data given in [14], based on US currency as follows:

$$C_{pipe} = (1.308032 + 0.54011\, m_{pipe} + 1.4933x10^{-5}\ m_{pipe}^2) L_{pipe} \quad (17)$$

$$C_{ins} = (11.156 + 299.308 e_{ins} - 471.830\ e_{ins}^2) A_{ins,surf} \quad (18)$$

here, m_{pipe} is the pipe mass per length for any diameter, e_{ins} is the insulation thickness, $A_{ins,surf}$ is the insulation surface area. Eq. (17) is valid for 1.22 to 248 kg/m and Eq. (18) is valid for 10 to 240 mm of insulation thickness.

Finally, Eq. (13) is referred to as *the objective function*. It can be minimized through either analytical or numerical methods. In any case it is necessary to define the decision variables for each pipe segment. The decision variables are taken as the pipe diameter and the insulation thickness.

In practice, above-ground installation is not usual, most of the pipes in district-heating networks are installed under the ground. If an underground application is considered, the digging and burying costs should be included in the objective function as new investment parameters. Since these new cost terms may vary significantly from one application region to another, this parametric study is conducted only for above-ground installation.

2 SOLUTION PROCEDURE

In the present study the minimization of the objective function was obtained with a numerical method using a computer code written in FORTRAN. The computer code was written in a general form that can optimize the whole network. In this study the calculations are presented only for a pipe segment. The code obtains the minimum value of Eq. (13) for a pipe segment utilizing all the available defined pipe diameters and insulation thickness. For the parametric study, the inner diameter of the pipe in the computations varies from 100 mm to 1000 mm. The thickness of the pipe was assumed to be 1 % of the pipe diameter. The insulation thickness varies from 0 to 150 mm.

3 PHYSICAL AND FINANCIAL PARAMETERS

The validation of the objective function Eq. (13) for a single pipe was carried out using the same piping system as shown in Fig. 1. The fixed parameters are as follows: pipe segment length, L = 100 m; water supply temperature, T_{sup} = 100 °C; water return temperature to boiler, T_{ret} = 80 °C; ambient temperature, T_{out} = 10°C; and the mass flow rate, $\dot{m}$ =25 kg/s. The economic and financial parameters are as follows: annual interest rate, i = 6%; depreciation period, n = 20 years; and annual operation time, t = 3500 h/year. The energy costs: natural gas was used as a base fuel with c_f = 3.90 \$/GJ, with a lower heating value of the fuel H_u = 0.0495 GJ/kg and assuming a boiler efficiency of η_b = 0.9. The exergy cost is calculated from Eq. (14) as c_e = 18.60 \$/GJ. The electricity price: electricity is assumed to be c_{el} = 26.11 \$/GJ. The thermo-physical parameters: the conductivities of the pipe and the insulation materials were assumed to be k_{pipe} = 54 and k_{ins} = 0.045 W/mK, respectively. The emissivity of the insulation jacket was ε = 0.26. The range of the fluid velocity was 0.3 m/s to 6.3 m/s.

4 DISCUSSION OF RESULTS

Fig. 2 shows the cost as a function of the pipe's inside diameter with the cost components. The mass flow rate in the pipe is assumed to be 25 kg/s. The total cost of the pipe at the optimum point is 0.19 \$/hour. The optimum insulation thickness is 73 mm. The optimum pipe diameter is 158 mm. As can be seen in the figure the determining parameter of the total costs is the cost of the frictional loss. The other parameters being almost linear functions, increasing with the pipe's diameter. For $D > D_{opt}$ the cost of the heat loss and the insulation are significantly higher than the costs of the pipe and the pressure loss. At the optimum point the heat loss cost is about 40 %, the insulation cost is 35 %, the frictional loss cost is 15 % and the pipe cost is 10 % of the total cost.

Figures 3 to 6 show the effect of the system conditions on the decision variables. For simplicity and a better comparison the results in these figures are presented in a non-dimensional form. The parameters in these plots are non-dimensionalized as $(\phi - \phi_{fix})/\phi_{fix}$. Here, ϕ is the parameter being investigated and ϕ_{fix} represents the fixed or reference value of ϕ. The ϕ_{fix} values are the optimum values mentioned in Fig.2.

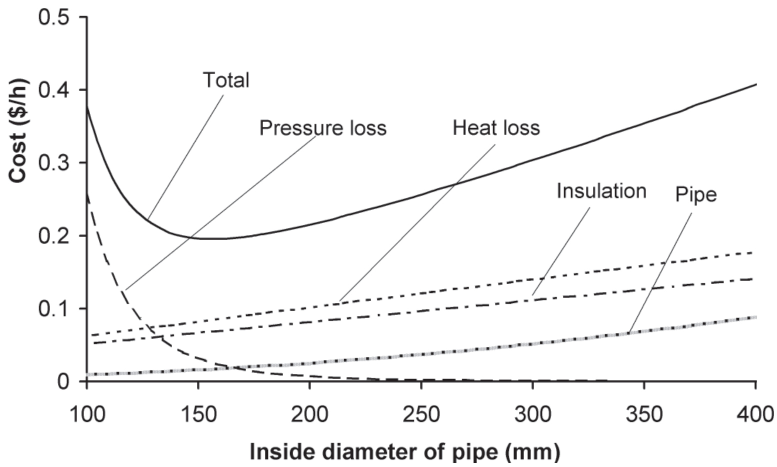

Fig. 2. *Variation of costs with pipe inside diameter for* e_{opt}*=73 mm*

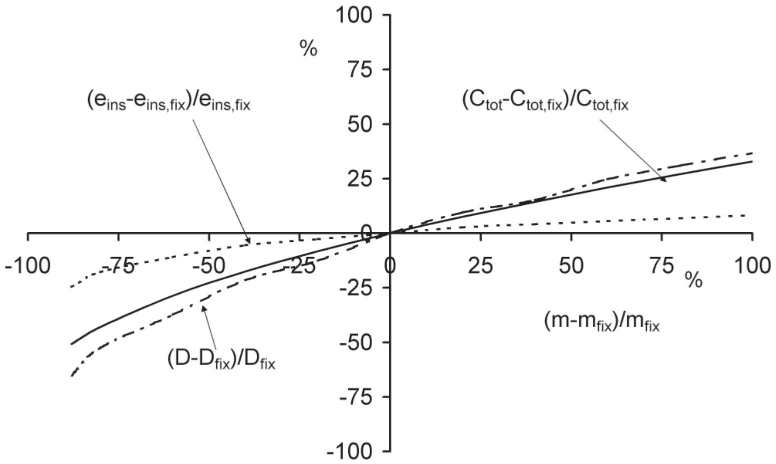

Fig. 3. *Effect of the mass flow rate on the decision variable:* m_{fix}*=25 kg/s,* D_{fix}*=158 mm,* $e_{ins,fix}$*=73mm,* $C_{tot,fix}$*=0.19 $/hour*

Fig. 3 shows the non-dimensional variations of the total cost, the optimum pipe diameter and the insulation thickness with the non-dimensional mass flow rate. The mass flow rate $\dot{m}$ altered from 5 to 50 kg/s, and the pipe diameter, insulation thickness and the total cost for the optimum conditions were obtained using Eq.(13) for each mass flow rate. The parameters shown in the figure are non-dimensionalized using the related ϕ_{fix} values, which are the optimum values of Fig.2. The results in Fig.3 show that the mass flow rate is the determining parameter for the optimum pipe diameter and the total cost. The effect of mass flow rate on the optimum insulation thickness is less significant, compared to the other parameters. This is an important outcome for the designer. Consider a tree-shaped hot-water distribution network. If the flow in the pipe in this network is divided equally into two branches, the flow rate in the new branches will be 50% of the main pipe and the non-dimensional optimum pipe diameter of the new branches reduces about 29%. The non-dimensional total cost for new pipes is reduced by about 24%. However, the non-dimensional insulation thickness is reduced by only 9%. As a result, this figure suggests that the pipe diameters in the considered hot-water distribution-network system should be reduced satisfactorily, from the energy plant to the end user. But the insulation thickness should not be changed too much. In addition, the ratio of the pipe diameter of the branches to the main pipe diameter is about 0.707, which agrees well with the value reported in [6].

Fig. 4 shows the effect of the annual operation time on the total cost, the optimum pipe

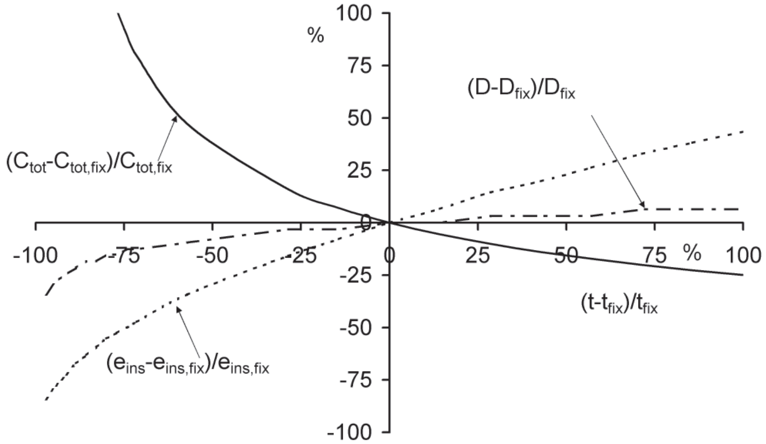

Fig. 4. *Effect of the annual operation time on the decision variables:* t_{fix}*=3500 hours/year,* D_{fix}*=158 mm,* $e_{ins,fix}$*=73mm,* $C_{tot,fix}$*=0.19 $/year*

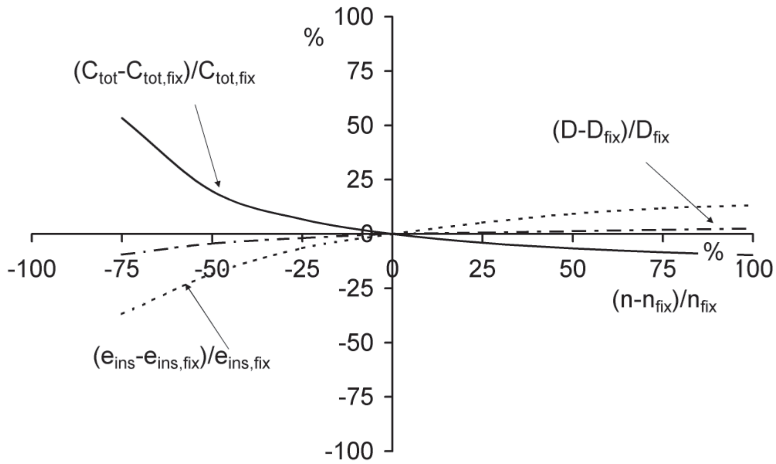

Fig. 5. *Effect of the depreciation period on the decision variables:* n_{fix} *= 20 years,* D_{fix} *= 158 mm,* $e_{ins,fix}$ *= 73mm,* $C_{tot,fix}$ *= 0.19 \$/year*

diameter and the insulation thickness. A longer annual operation time reduces the total cost of the conduit, which is as expected. The insulation thickness is affected significantly by the annual operation time. However, the effect of the annual operation time on the pipe diameters is less significant. For a prearranged hot-water distribution network, if the annual operation time of the system is extended by 100%, the optimum pipe diameter increases by about 5%. However, the optimum insulation thickness for the new system rises by about 40%. This shows that the annual operation time influences the heat-loss cost and the insulation cost significantly. The designer should be careful when specifying the annual operation time, which dominates the insulation thickness drastically.

Fig. 5 shows the effect of the depreciation period on the decision variables. The effect of the depreciation period is less significant with respect to the optimum pipe diameter. The heat-loss cost is more crucial than the frictional loss. The insulation thickness increases by about 15% when the depreciation period is extended by 100%.

Fig. 6 shows the effect of the water-supply temperature on the decision variables. Here, the supply temperature was altered from 50°C to 200°C. The water-supply and return temperature difference is assumed to be 20°C. The effect of the pressure variation with hot-water temperature on the pipe's thickness was ignored. Referring to Fig. 6, a higher fluid temperature increases the heat loss and the insulation thickness, which raises the total cost of the conduit. The pipe diameter is reduced slightly by a temperature rise, which causes a reduction in the heat-transfer area.

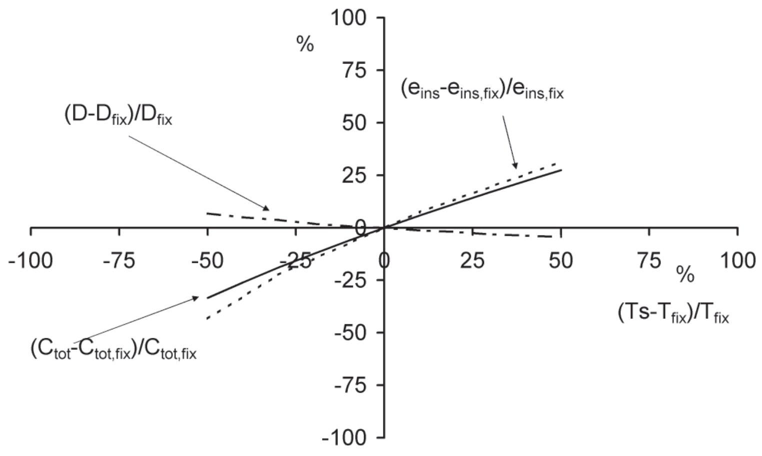

Fig. 6. *Effect of the water supply temperature on the decision variables:* T_{fix} *= 100 °C,* D_{fix} *= 158 mm,* $e_{ins,fix}$ *= 73mm,* $C_{tot,fix}$ *= 0.19 \$/year*

5 CONCLUSIONS

In this study the effect of the hot-water piping-system conditions on the decision variables is investigated with a thermo-economic optimization method based on the second law of thermodynamics. Important parameters, such as the ambient conditions, the interest rate, the fuel and electricity prices, and the thermo-physical parameters, were assumed to be the same and constant for the calculations. The main results obtained in this study can be summarized as follows:

- The results for the optimized pipe show that heat-loss cost, including the insulation cost, is about 75% of the total cost. The effect of the frictional cost and the pipe cost on the total costs is about 25%. The priority in the piping-design techniques for a hot-water delivery system should be on the heat-transfer calculation rather than the pressure-drop calculations.
- The mass flow rate is a determining parameter for the optimum pipe diameter. There is a secondary effect of the mass flow rate on the insulation thickness.
- The annual operation time is a determining parameter for the insulation thickness. The insulation thickness increases significantly when the annual operation time increases. The effect of the annual operation time on the pipe diameter is less significant.
- The effects of the depreciation period on the decision variables are similar to the annual operation time, but they are less significant. Heat-loss and insulation costs are more important parameters affecting the optimum values of the total cost of the hot-water piping network over the lifetime of the system.
- For an optimization study the fluid temperature is also an important parameter. A higher fluid temperature increases the heat loss, which requires a higher insulation thickness. There is a less significant effect of the fluid temperature on the pipe diameter.

Finally, the system conditions have a strong effect on the optimum values of the design parameters. Therefore, one should start to design the hot-water distribution network by specifying the precise amount of capacity, annual operation time, depreciation period and fluid temperature. For an existing hot-water network, any increase in the capacity by adding new users or changing the system parameters and keeping the pipes unchanged might affect the system's efficiency in a negative way.

6 NOMENCLATURE

C	\$	cost
c	\$/kJ	specific cost
$\dot{C}$	\$/s	cost rate
C_p	kJ/kgK	heat capacity
CRF		capital recovery factor
D	m	diameter
$\dot{E}$	kJ/s	exergy flow rate
e	m	thickness
f		friction coefficient
h	W/m^{2}K	convective heat-transfer coefficient
h_{rad}	W/m^{2}K	radiative heat-transfer coefficient
i		annual interest rate
k	W/mK	conductivity
L	m	pipe length
$\dot{m}$	kg/s	mass flow rate
n	year	depreciation period
P	Pascal	pressure
Pr		Prandtl number
$\dot{Q}$	kJ/s	heat flow rate
Re		Reynolds number
r	m	radius
T	K	temperature
T_{ms}	K	mean outside-surface temperature
t	h/year	annual operation time
V	m/s	velocity
$\dot{W}_p$	kJ/s	pumping power
Z	\$	capital cost
$\dot{Z}$	\$/s	capital cost rate

Greek letters

ε		surface emissivity
η		efficiency
ϕ		exergy efficiency
ρ	kg/m^3	density
σ		Stefan-Boltzmann constant
ξ		pressure loss coefficient

Subscripts

b	boiler
des	destruction
e	exergy
el	electricity
$exit$	exit value
f	fuel
in	inside, inner value of conduit

ins	properties of insulation	*pipe*	properties of pipe
m	mean value	*rad*	radiative
opt	optimum value	*ret*	return value
out	outside, outer value of conduit	*sup*	supply value
p	pumping	*tot*	total value

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Prejeto:
Received: 13.7.2006

Sprejeto:
Accepted: 27.6.2007

Odprto za diskusijo: 1 leto
Open for discussion: 1 year