let. - vol. 46 (2000) št. - no. 8

STROJNIŠKI VESTNIK 8

JOURNAL OF MECHANICAL ENGINEERING

strani - pages 493 - 590

ISSN 0039-2480 . Stroj V . STJVAX

cena 800 SIT

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OGREVANJE IN KLIMATIZACIJA
ZA TRETJE TISOČLETJE
HEATING AND AIR-CONDITIONING
FOR THE THIRD MILLENNIUM

Maribor
6. - 8.5.2000

Organizatorji:
Organisers:

Mednarodni inštitut za hlajenje
International Institute of Refrigeration

SITHOK

Univerza v Mariboru
University of Maribor

Univerza v Ljubljani
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© Strojniški vestnik 46(2000)8,493
Mesečnik
ISSN 0039-2480
© Journal of Mechanical Engineering 46(2000)8,493
Published monthly
ISSN 0039-2480

Vsebina
Contents

Strojniški vestnik - Journal of Mechanical Engineering
letnik - volume 46, (2000), številka - number 8

© Strojniški vestnik 46(2000)8,494-502
ISSN 0039-2480
UDK 536.3:536.2:697.97
Pregledni znanstveni članek (1.02)

© Journal of Mechanical Engineering 46(2000)8,494-502
ISSN 0039-2480
UDC 536.3:536.2:697.97
Review scientific paper (1.02)

Analitična določitev srednje sevalne temperature zapletene geometrijske oblike prostora

An Analytical Determination of the Mean Radiant Temperature for a Complex Room Geometry

Matjaž Prek - Peter Novak

Sevalne toplotne izgube pomenijo sestavni del toplotnega ravnotežja človeka v prostoru. Pravilna določitev deleža sevalnega toplotnega toka je potrebna za oceno vpliva tega parametra na toplotno ugodje. Toplotno ravnotežje je odvisno predvsem od površinskih temperatur in kotnega faktorja med človekom in površinami prostora. Ker so notranje površine sestavljene iz različnih elementov, je izračun kotnega faktorja zahteven. V tem primeru pomeni računalniški program primerno orodje za izračun sevalnih toplotnih tokov, predvsem za zapleteno sestavljene površine, kakršne so v dejanskih razmerah.

Predstavljeni algoritem temelji na izračunu kotnih faktorjev, pri čemer je upoštevan zakon seštevnosti. Postopek izračuna omogoča določitev kotnih faktorjev za sestavljene površine in s tem upoštevanje zapletenega vpliva kotnih faktorjev na sevalni toplotni tok, kakršen je npr. pri sedeči osebi in znani usmeritvi. Matrično zasnovan postopek omogoča določitev vpliva različnih parametrov na srednjo sevalno temperaturo.

(Ključne besede: ugodje bivalno, prenos toplote, sevanje, temperature sevanja)

Radiative heat losses represent a substantial part of the total heat balance of the human body in a closed space. The correct determination of this contribution is necessary in order to gain an insight into the influence of this parameter on human thermal comfort. The thermal balance is strongly affected by surface temperatures and by the angle factor between a body and a wall surface. Since a room's internal surfaces are composed of various parts, the calculation of view factors becomes more complex. Therefore, a computer algorithm is a useful tool for determining the radiant heat exchange, particularly for the complex surface compositions encountered in practical situations.

The proposed algorithm is based on the computation of view factors, which are additive. This algorithm enables the computation of view factors for composite surfaces, thus allowing for the complex impact of view factors on thermal radiative heat exchange, as is the case for the seated posture and other orientations of the human body. The matrix-based approach makes it possible to determine the influence of various parameters on the mean radiant temperature.

(Keywords: thermal comfort, heat transfer, radiation, radiant temperature)

0 UVOD

Toplotno ugodje ljudi je odločilen dejavnik, ki določa kakovost bivalnega okolja v stavbah. Zagotovljeno je z vzdrževanjem vplivnih parametrov v predpisanih mejah ob hkratni izravnavi vseh motečih vplivov. Analiza sevalnega ogrevalnega sistema pokaže, da je pri zagotavljanju kakovosti bivalnega okolja odločilna razlika med temperaturo zraka v prostoru in dejansko temperaturo. Upoštevajoč toplotno ugodje imajo ljudje subjektivno nagnjenost k toplejšim obodnim

0 INTRODUCTION

The thermal comfort of people in confined environments is a crucial issue for the proper assessment of the indoor quality of buildings. It should be considered both as a requisite by itself and as a fundamental preliminary requirement for establishing other indoor needs. Thermal comfort is ensured by maintaining the declared values of the factors determining thermal comfort constant, while eliminating any disturbing influences causing local thermal discomfort. When analysing a radiant heating system, the most pronounced factor is stenam in hladnejšemu zraku; zato je primerna natančnejša določitev sevalnega dela prenosa toplote. Toplotno ravnotežje človeka je odvisno od sevalnega toplotnega toka med telesom in obodnimi stenami, ta pa je odvisen od površinskih temperatur in kota, katerega tvorita telo in opazovana površina. Za določitev toplotnega ugodja ali neugodja je zato treba najprej določiti kotne faktorje obodnih površin glede na položaj človeka.

Za preproste površine so algoritmi za izračun kotnih faktorjev že določeni. Vendar so v resničnem stanju površine sestavljene iz različnih neizotermnih elementov, ki onemogočajo neposredno uporabo algoritmov. Kot primer preprosto sestavljene neizotermne notranje površine so različni elementi, npr. stene, okna, vrata in ogrevala. Zaradi tega postane izračun kotnih faktorjev zelo zapleten in je potrebno dosledno upoštevanje temperaturnih karakteristik površin. Zato je vpeljan algoritem, ki omogoča izračun projekcijskega faktorja človeka glede na dano stanje.

Poprej je bila razvita metoda za izračun kotnega faktorja, ki temelji na neposrednem izračunu projekcijskega faktorja. Kotni faktorji so podani kot približki rešitev sistema enačb in so predstavljeni v obliki zbirke diagramov. V teh diagramih je kotni faktor podan neposredno kot funkcija brezdimenzijskega geometrijskega parametra. Enačbe so rešene za šest značilnih smeri (površin prostora), ki predstavljajo značilni prostor. Za te primere je predpostavljena razdelitev površin kvadra na podpovršine; zaradi simetrije človeškega telesa se število kotnih faktorjev zmanjša in so odvisni od usmeritve med človekom in delno površino. Za tipični prostor se število kotnih faktorjev zmanjša na šest za sedečega človeka, tri za stoječega in dva za primer neznane usmeritve.

Naslednji korak k poenostavitvi in s tem bolj uporabni obliki zapisa kotnih faktorjev je dobljen z metodo najmanjših kvadratov. Pri tem so kotni faktorji podani v obliki eksponentnih enačb v funkcijski odvisnosti od brezdimenzijskih geometrijskih parametrov. Ta poenostavljena metoda izračuna kotnih faktorjev zagotavlja dobro ujemanje izračunanih vrednosti z eksperimentalno izmerjenimi vrednostmi pa tudi z rešitvami sistema enačb. Način zapisa kotnih faktorjev skupaj s parametri omogoča preprosto uporabo računalniškega programa. Pri računanju resničnih stanj so obodne površine razdeljene na štiri površine kvadra (glede na položaj in usmeritev človeka) in z upoštevanjem zakona seštevnosti določen kotni faktor za celotno površino. Posplošen algoritem omogoča izračun kotnih faktorjev za poljubne neizotermne elemente obodnih površin prostora.

the difference between the indoor air temperature and the effective temperature. Regarding personal comfort, the human occupant has a subjective preference for a warmer building structure and cooler indoor air, indicating that radiative heat exchange should be favoured. The human body's thermal balance is strongly affected by radiative heat exchanges with surrounding surfaces, which are a function of the surface temperatures and the angle at which the human body senses them. This means that in order to establish local thermal comfort or discomfort, view factors of people with respect to the envelope surfaces must first be assessed.

Algorithms for computing these view factors for simple plane surfaces have already been determined. However, in practical cases, the room's internal surfaces are composed of various parts, each possessing a specific thermal situation. The complex internal surface of a room composed of a wall, a window, a door and a heating panel would be an example of this kind of mixture of composite plane surfaces. In this case, the calculation of view factors becomes much more complex and requires a careful management of the thermal and geometric properties of the surfaces. A comprehensive algorithm is introduced here which allows for the computation of angle factors of people with respect to the given complex situations.

In a previous study, a method of calculating the view factor was determined which avoids the direct calculation of the projected area factor. In this work, the view factors were given as solutions of equations and presented in the form of sets of graphs. In these graphs, the view factor was directly presented as a function of dimensionless geometrical parameters. The equations were solved for six relevant cases (room walls) that occur in a typical room. A division of parallel-piped surfaces into sub-surfaces has been proposed for these cases. Due to the symmetry of the human body, the number of view factors is reduced and the view factors are dependent on the orientation of the person and sub-surface. The number of view factors for the typical sub-surfaces of an enclosure is reduced to six for a seated person, three for a standing person and two when the orientation of a person is unknown.

Another step toward simplification, and thus to a more useful form of view factors, was suggested by the observation of graphs. There the view factors are given in the form of exponential equations dependent on dimensionless geometrical parameters. This simplified method for calculating the view factors enables good agreement between the calculated and actual values determined by solutions of equations and experimental data. The form of the view factors, along with the determined parameters, allows them to be used in computer algorithms. For practical applications, the surrounding surfaces should be divided into four rectangular sub-surfaces (with respect to the human body's position and orientation) and, by means of the additive property, computed to determine the whole view factor. This general algorithm could also be used for computing the view factor for non-isothermal elements of the wall.

1 KOTNI FAKTOR MED ČLOVEKOM IN POVRŠINO PROSTORA

Kotni faktor med človekom in pravokotno izotermno površino $F_{P\rightarrow A}$ lahko izračunamo po enačbi, podani v [1] in standardu [2]:

1 ANGLE FACTOR BETWEEN THE HUMAN BODY AND THE ROOM SURFACE

As presented in the work of Fanger [1] and determined with a standard [2], the angle factor between the human body and a rectangular surface $F_{P\rightarrow A}$ can be computed as:

$$F_{P\rightarrow A} = \frac{1}{\pi}\cdot \int_{\frac{x}{y}=0}^{\frac{x}{y}=\frac{a}{c}} \int_{\frac{z}{y}=0}^{\frac{z}{y}=\frac{b}{c}} \frac{f_P}{\sqrt{\left[1+\left(\frac{x}{y}\right)^2+\left(\frac{z}{y}\right)^2\right]^3}}\, d\left(\frac{x}{y}\right) d\left(\frac{z}{y}\right) \qquad (1)$$

kjer sta a in b širina in višina izotermne površine A ter f_P projekcijski faktor. Pri praktični uporabi enačbe (1) se pojavita dva problema: določitev projekcijskega faktorja f_P in vpliv neizotermne sestavljene površine.

Zato je bila razvita metoda, ki ne temelji na neposrednem računanju kotnih faktorjev. V [1] so podani kot rešitve enačb, prikazane v obliki zbirke diagramov. V njih je kotni faktor podan neposredno kot funkcija brezdimenzijskih parametrov *a/b* in *b/c*. Enačbe so rešene za šest značilnih primerov (sten prostora). Izračun temelji na delitvi površine kvadra na štiri delne površine. Zaradi simetrije človeka se število kotnih faktorjev zmanjša in so odvisni od usmeritve človeka glede na delno površino. Za površine kvadra se število kotnih faktorjev zmanjša na šest za sedeč položaj, tri za stoječ in dva za primer neznane usmeritve.

Naslednji korak k poenostavitvi in s tem uporabnejši metodi določitve kotnih faktorjev je predlagan v [3]. Na temelju grafično predstavljenih rešitev enačb je narejena analiza s postopkom najmanjših kvadratov, s katero je dobljena eksponentna enačba kot funkcija brez-dimenzijskih parametrov *a/c* in *b/c*:

where a and b are the width and height of the isothermal surface A, and f_P is the projected area factor. However, two problems arise in the practical application of equation (1): the determination of the projected area factor f_P and the influence of the non-isothermal composite surface.

In order to calculate the view factor, a method which avoids the direct calculation of the projected area factor has been established. In the previous work of Fanger [1], the view factors are determined as solutions of equations and presented in the form of sets of graphs. In these graphs, the view factor is directly presented as a function of the dimensionless geometrical parameters *a*/*c* and *b*/*c*. Equations are solved for six relevant cases (room surfaces) that occur in a typical room. For this, a division of the parallelepiped surfaces into sub-surfaces is proposed. Due to the symmetry of the human body, the number of view factors is reduced and is dependent on the orientation of the person to the sub-surface. The number of view factors for typical sub-surfaces of an enclosure is reduced to six for a seated person, three for a standing person and two for person with unknown orientation.

Another step toward simplification, and thus to a more useful form of view factors, was made by Rizzo et al. [3]. As suggested by the observation of graphs, the view factors are given in the form of exponential equations dependent on geometrical parameters, the dimensionless parameters *a*/*c* and *b*/*c* in the following equation:

$$F_{P\rightarrow A} = F_{sat,\max}\cdot\left[1-\exp\left(\frac{a}{c}\right)\Big/\tau\right]\cdot\left[1-\exp\left(\frac{b}{c}\right)\Big/\tau\right] \qquad (2)$$

kjer sta / where

$$\tau = A + B\cdot\frac{a}{c}$$

in / and

$$\gamma = C + D\cdot\frac{b}{c} + E\cdot\frac{a}{c}$$

$F_{sat,\mathrm{maks}}$ pomeni največjo vrednost projekcijskega faktorja za dano podpovršino v odvisnosti od usmeritve (znana ali neznana) in položaja človeka (sedeč ali stoječ). Koeficienti A, B, C, D in E so določeni z linearno regresijo za parameter τ in večkratno aproksimacijo za parameter γ. Primerjava rezultatov, dobljenih s to metodo, z rezultati rešitev enačb in izmerjenih vrednosti, je dokazala

$F_{sat,max}$ represents the maximum saturation value for a given sub-surface depending on human body orientation (known or unknown) and posture (seated or standing). The parameters A, B, C, D and E are determined by simple linear regression for the parameter τ and by multiple linear regression for the parameter γ. As shown by Nucara et al. [4], this simplified method for calculating the view factors enables good agreement between the upravičenost uporabe poenostavljene metode izračuna [4]. Oblika enačbe (2) omogoča preprosto uporabo koeficientov v računalniškem programu. Za reševanje dejanskih primerov je potrebna samo delitev obodnih površin na štiri delne površine (glede na usmeritev in položaj človeka), kotni faktor za celotno površino pa je določen z zakonom o seštevnosti.

calculated and actual values determined from solutions of equations, as well as experimental data. The form of equation (2) along with the determined parameters enables their use in computer algorithms. For practical applications, the surrounding surfaces should be divided into four rectangular sub-surfaces (with respect to the human body's position and orientation) and by means of the additive property computed for the whole view factor.

2 VPLIV SESTAVLJENIH POVRŠIN

2 THE INFLUENCE OF COMPOSITE ROOM SURFACES

Na sliki 1 je prikazan najpreprostejši primer enostavne izotermne površine. Če normala na površino A poteka skozi točko P (opazovana točka v prostoru, t. i. osrednja točka) in se ujema z ogliščem površine, potem lahko določimo kotni faktor z enačbo (2) v odvisnosti od usmeritve in položaja človeka. V primeru, da se normala ne ujema z ogliščem, kakor je prikazano na sliki (1), potem lahko določimo kotni faktor ob upoštevanju seštevnosti z enačbo (3):

The simplest example of an isothermal room surface is shown in Figure 1. If the normal from the subject P (the so-called generic point) coincides with the corner point of surface A, the view factor can be directly computed with equation (3) depending on the human body orientation and posture. In addition, if the normal does not coincide with the corner point (as shown in Figure 1), then the view factor can be determined by the application of the additive property in the following way:

$$F_{P\to A} = F_{P\to 1} + F_{P\to 2} + F_{P\to 3} + F_{P\to 4}$$

ali / or (3)

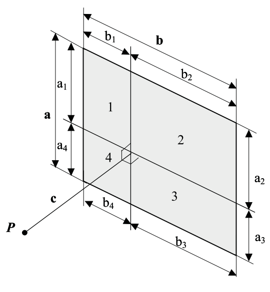
$$F_{P\to A} = \sum_{i=1}^{4} F_{P\to i}$$

kjer i-ta površina pomeni delno površino $a_i b_i$.

Enačbo (3) lahko uporabimo kot splošni algoritem za izračun kotnega faktorja za poljuben element površine. Z delitvijo površine, pri čemer element A že pomeni delni element stene, na naslednje štiri delne elemente (na sl. 2. delni elementi 1, 2, 3, 4), lahko izračunamo kotni faktor za vsako delno površino z enačbo (3). S ponovitvijo postopka za vse obodne površine prostora (ki so sestavljene iz različnih elementov) in za vse položaje človeka lahko določimo skupni kotni faktor.

where surface i represents sub-surface $a_i b_i$.

Equation (3) can be used as a general algorithm for calculating the view factor for an arbitrary element of the wall. By division of the surface, where element A of the wall represents the sub-surface, into four sub-surfaces (1, 2, 3, 4), the view factor can then be determined by solving equation (3) for every sub-surface. By repeating this procedure for all room walls (composed of different elements) and for all positions of the person in the room, the total view factor can be estimated.



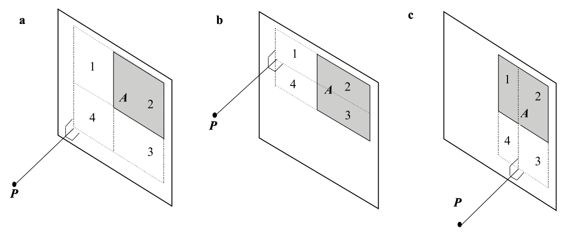
Sl. 1. *Kotni faktor med osrednjo točko P in površino A (z izmerami a x b)*
Fig. 1. *View factor between generic point P and surface A (dimensions a x b)*

Enačba (3) velja le v primeru, če se normala med točko in površino ujema z ogliščem površine kvadra (pogoj za uporabo eksperimentalno določenih kotnih faktorjev). Vendar lahko v vsakem primeru določimo normalo na ravnino, v kateri je opazovana površina tako, da se ujema s točko *P*. Kotni faktor površine določimo z delitvijo ravnine na štiri delne površine, od katerih je ena opazovana površina.

V običajnem primeru, ko je zid sestavljen iz elementov z različnimi površinskimi temperaturami (npr. okno, ogrevalo), lahko določimo kotni faktor za posamezni element po opisanem postopku. Štiri delne površine oblikujemo tako, da se normala skozi opazovano točko ujema z ogliščem površine na ravnini elementa; preostala oglišča delnih površin so določena z oglišči elementa. Na sliki 2 je prikazanih nekaj možnih razporeditev.

Equation (3) functions only if the normal from the subject coincides with the common corner point of the rectangles (a condition for using the experimentally determined view factors). Nevertheless, in the case of isothermal sub-surfaces with equal temperature, we are able to sweep the normal from the surface in such a way, that it coincides with the normal from the subject. The view factor for an arbitrary human position in the room can be determined by splitting the surface into four sub-surfaces and applying the additive property. This general algorithm can also be used for computing the view factor for non-isothermal elements of the wall.

In the usual case, where the wall is composed of elements with different temperatures (e.g. windows, heating panels), the view factor can be determined in a similar way to the sub-surface. The four sub-surfaces are constructed in such a way that the normal from the subject represents one corner of the surface; other corners and the middle point of the sub-surfaces are determined by corners of the element. Some possible arrangements are shown in Figure 2.



Sl. 2. *Primeri geometrijskih pogojev za izračun kotnega faktorja za osrednjo točko P*
Fig. 2. *Examples of geometrical conditions for calculating the view factor for generic point P*

Kotni faktor je določen z enačbo (3) z upoštevanjem seštevnosti:

The view factor is then determined using equation (3) by means of the additive property in the following way:

za primer na sl. 2a: / for the case in Figure 2a:

$$F_{P\to A} = F_{P\to(1+2+3+4)} - F_{P\to(1+4)} - F_{P\to(3+4)} + F_{P\to 4} \tag{4a}$$

za primer na sl. 2b: / for the case in Figure 2b:

$$F_{P\to A} = F_{P\to(1+2+3+4)} - F_{P\to 1} - F_{P\to 4} \tag{4b}$$

za primer na sl. 2c: / for the case in Figure 2c:

$$F_{P\to A} = F_{P\to(1+2+3+4)} - F_{P\to 3} - F_{P\to 4} \tag{4c}$$

Da lahko uporabimo računalniški program za izračun kotnih faktorjev za celotni tloris prostora, moramo ustrezno definirati koordinatni sistem. Delitev površine elementa ali ravnine na delne površine in upoštevanje zakona seštevnosti zahteva pravilno

In order to create a computer algorithm for calculating the view factor for an entire floor plan, attention must be paid to the proper determination of the coordinate system. The division of the surface into four sub-surfaces and an application of the addoločitev predznaka za posamezno delno površino. Glede na postopno razvrstitev delnih površin, kakor je prikazana na slikah 2a, b in c, ter določitev kotnega faktorja z enačbami (4a, b in c), lahko algoritem posplošimo:

ditive property requires the determination of the sign for certain sub-surfaces. According to the arrangements shown in Figures 2a, b, c and the equations (4) for computing the relevant view factor, one can deduce a general algorithm:

$$F_{P\to A}=\sum_{i=1}^{4}F_{P\to i}\cdot sign\left(a_i\cdot b_i\right) \qquad (5).$$

Ko določimo kotne faktorje za posamezne elemente, lahko izračunamo kotni faktor za sestavljeno površino z enačbo:

When the view factors for individual sub-surfaces are determined, the view factor for the composed wall can be expressed as:

$$F_{P\to \substack{stena|_{usmeritev}\\ wall|_{orientation}}}=F_{P\to \substack{celotna\ površina|_{usmeritev}\\ total\ surface|_{orientation}}}-\sum_{\substack{delna\ površina|_{usmeritev}\\ sub\text{-}surface|_{orientation}}}F_{P\to \substack{delna\ površina\\ sub\text{-}surface}} \qquad (6)$$

ter povprečno sevalno temperaturo T_{mrt} za znano usmeritev v prostoru:

and the mean radiant temperature T_{mrt} for a given orientation in the room is then:

$$T_{mrt}^4=\sum_{i=1}^{N}F_{P\to i}\cdot T_i^4 \qquad (7),$$

kjer je T_i absolutna površinska temperatura i-te notranje površine in $F_{P\to i}$ kotni faktor med človekom (osrednjo točko v prostoru) in i-to površino (ali podpovršino).

where T_i is the absolute temperature of the i-th internal surface and $F_{P\to i}$ is the view factor between the person (generic point in room) and i-th surface (or sub-surface).

3 PRIMER

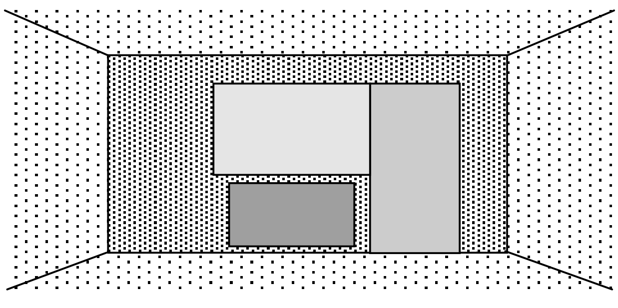
Uporabnost predstavljenega postopka je prikazana na primeru izračuna povprečne sevalne temperature za zid, sestavljen iz različnih elementov. Na sliki 3 je primer preproste zunanje stene, sestavljene iz vrat, okna in ogrevala (radiatorja).

V preglednici 1 so podane geometrijske in temperaturne predpostavke za posamezne površine. Zaradi preglednosti je pri računanju sevalne temperature upoštevana samo ena stena. Ker algoritem temelji na uporabi enačbe (2), so potrebni geometrijski parametri določeni avtomatično z definicijo elementov stene (ali prostora), razen višine osrednje točke, ki je definirana s predpostavljenim položajem človeka. Na ta način so izbrani koeficienti enačbe (2), ki so določeni glede na položaj človeka in s tem posredno odvisni od geometrijske oblike prostora in elementov.

3 CASE STUDY

The potential use of the introduced algorithm can easily be shown by means of an application aimed at the calculation of the mean radiant temperature for a wall which is composed of different elements. In Figure 3, an example of a wall composed of a door, window and heating panel is shown.

Table 1 contains the geometric and thermal assumptions for the wall surfaces; for the sake of clarity, the influence of only one wall on mean radiant temperature is analysed. Since the algorithm is based on the use of equation (2), the required geometrical parameters are automatically defined by given wall (or room) elements, except for the height of a generic point which is determined by the proposed body posture. The parameters used in equation (2) are defined by the body posture and, thus, are chosen depending on the geometry of the room and its elements.



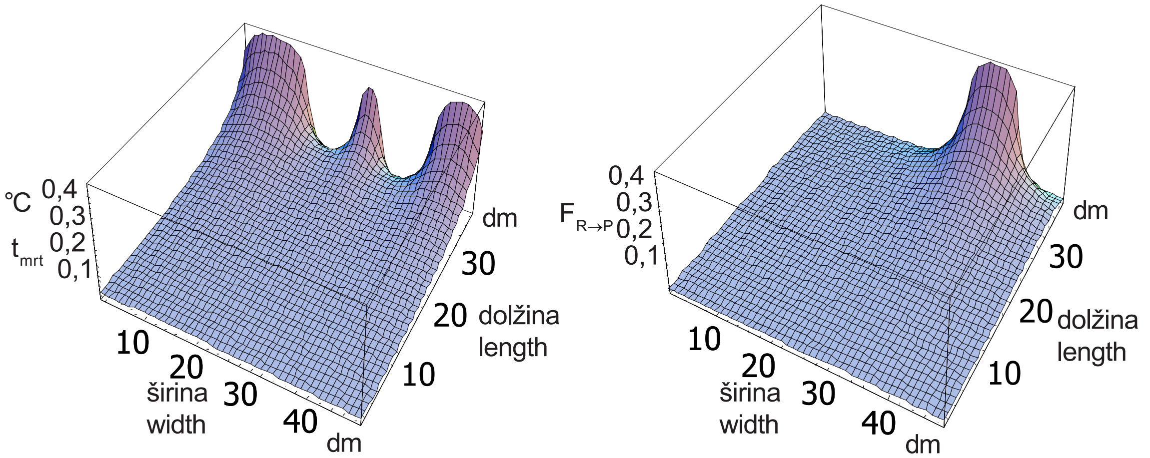
Sl. 3. *Primer predpostavljene sestave zunanje stene*
Fig. 3. *Assumed composition of the wall in this case*

Preglednica 1. *Parametri stenskih površin*
Table 1. *Parameters of the wall surfaces*

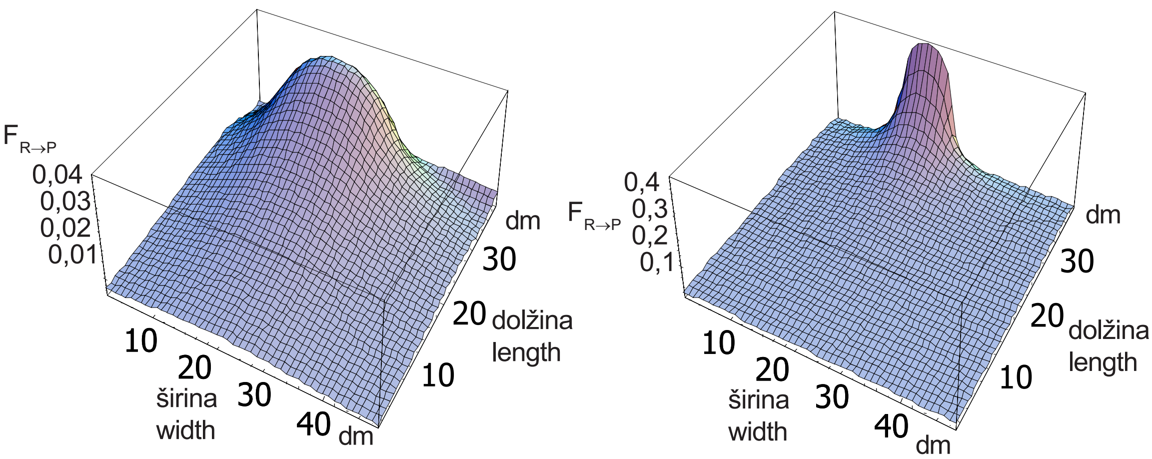
element	dolžina length m	višina height m	površinska temperatura surface temperature °C
stena wall	5	3	18
vrata door	1	2	16
okno window	2	1	14
ogrevalo heating panel	1	0,8	50

Na naslednjih slikah so prikazani izračunani kotni faktorji stene ter povprečna sevalna temperatura. Ker je algoritem splošen, ga lahko uporabimo za različne sestavljene površine, npr:
- poljubno lego elementa stene ali
- geometrijsko obliko elementa ali
- površinsko temperaturo.

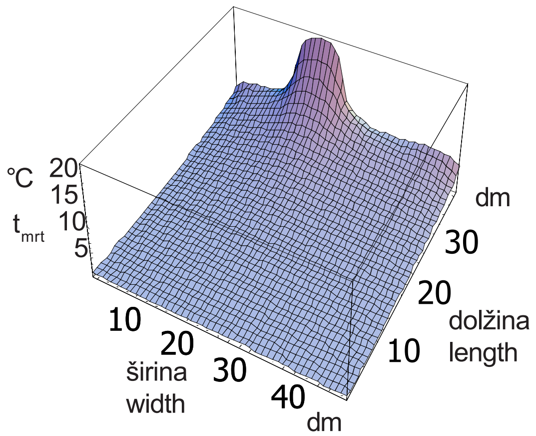
In the following figures, the calculated view factors and mean radiant temperatures are presented. In these examples, the height of the generic point is used as a variable. Since the algorithm possesses a general structure, it is easily applied to different surface compositions, such as:
- the position of the element within the wall,
- the arbitrary geometry of the element,
- the surface temperature of the element.

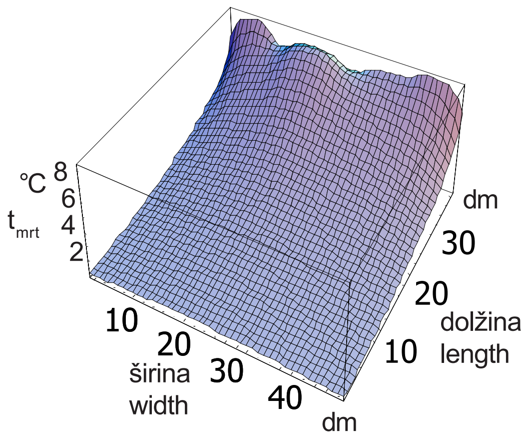


Sl. 4. *Izračunani kotni faktor za steno (levo) in vrata (desno)*
Fig. 4. *Calculated view factor for wall (left) and door (right)*



Sl. 5. *Kotni faktor za okno (levo) in ogrevalo (desno)*
Fig. 5. *Calculated view factor for window (left) and heating panel (right)*





Sl. 6. *Srednja sevalna temperatura t_{mrt} za različne višine: 0,6 m nad tlemi (levo) in 1,2 m (desno)*
Fig. 6. *Calculated mean radiant temperature t_{mrt} for different heights: 0.6 m above the floor (left) and 1.2 m (right)*

4 SKLEPI

Prikazana metoda predstavlja analitični postopek k določitvi optimalnega toplotnega okolja v stavbah. Predlagani algoritem temelji na izračunu kotnih faktorjev ob upoštevanju zakona seštevnosti. S tem je omogočen izračun povprečne sevalne temperature za sestavljene površine ob hkratnem upoštevanju zapletenega vpliva usmerjenosti človeka v prostoru. Zaradi matričnega načina izračuna je mogoča analiza vpliva različnih parametrov tako na povprečno sevalno temperaturo kakor tudi na pričakovano toplotno ugodje oz. neugodje. Hkrati je mogoče tudi upoštevanje drugih parametrov (temperaturni gradient, relativna hitrost zraka itn.), ki vplivajo na toplotno ugodje in so izraženi kot vrednost pričakovane povprečne presoje (PVM).

Za zagotovitev najboljših toplotnih razmer mora biti doseženo ravnotežje med toplotnimi viri in ponori. Metoda omogoča določitev vpliva karakteristik toplotnega vira - ogrevala (temperatura, geometrijska oblika) na povprečno sevalno temperaturo, ki predstavlja del toplotnega okolja. Za boljšo predstavljivost so lahko rezultati izračuna podani v obliki različnih izometričnih diagramov.

Ta metoda je učinkovito orodje za natančno določitev medsebojnega vpliva ogrevalnega sistema in gradbene konstrukcije, s čimer lahko dosežemo največje mogoče področje toplotnega ugodja. V idealnem primeru lahko to metodo uporabimo v fazi načrtovanja objekta in s tem določimo najboljše možno razmerje med toplotno-tehničnimi lastnostmi stavbe in ogrevalnega sistema.

4 CONCLUSIONS

The thermal comfort of people in confined environments is a crucial issue for the proper assessment of the indoor quality of buildings. The method presented in this paper represents an analytical tool for determining the optimum thermal environment for people in buildings. The proposed algorithm is based on the computation of view factors using the additive property. This algorithm enables the calculation of mean radiant temperature for composite room surfaces, even allowing for the complex impact of body posture. The matrix-based approach allows us to determine the effect of various parameters on mean radiant temperature, as well as on thermal comfort or discomfort. This approach also enables the consideration of other parameters (air temperature gradient, air velocity etc.) whose impact on thermal sensation is expressed as the predicted mean vote (PMV) value.

In order to achieve optimum thermal conditions, the thermal balance between heat sources and sinks must be established. This method enables the determination of the influence of heating source characteristics (temperature, geometrical parameters) on mean radiant temperature as a part of the overall environmental conditions. A graphical rendering of an isometric map is used for better visual interpretation of the results.

This method presents a useful tool for determining the correct interplay between heating system and building structure, thus achieving the maximum possible thermal comfort area. This method could, ideally, be incorporated into the architectural planning phase of buildings in order to determine the best relationship between the the building's structure and its heating system.

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Prejeto:
Received: 15.8.2000

Sprejeto:
Accepted: 10.11.2000

© Strojniški vestnik 46(2000)8,503-508
ISSN 0039-2480
UDK 697.97:519.682:628.83
Pregledni znanstveni članek (1.02)
© Journal of Mechanical Engineering 46(2000)8,503-508
ISSN 0039-2480
UDC 697.97:519.682:628.83
Review scientific paper (1.02)

Računalniška analiza ogrevalnih in hladilnih obremenitev za različne tipe stavb

A Computer Analysis of Heating and Cooling Loads for Different Types of Building

Uroš Stritih - Simon Muhič - Peter Novak

Prispevek prikazuje simulacije ogrevalnih in hladilnih obremenitev za različne tipe stavb s programskim paketom TRNSYS. Prikazujemo sedem različnih tipov stavb in štiri različne klimatske pogoje v Evropski zvezi. Skupaj je narejenih 28 simulacij, v tem prispevku pa prikazujemo rezultate za pisarniško zgradbo.

(Ključne besede: zgradbe pisarniške, energija toplotna, simuliranje obremenitev, analize računalniške)

This paper presents simulations of heating and cooling loads for different types of building with the TRNSYS programme package. We present seven different building types and four different weather conditions in the European Union. Altogether we made 28 simulations and the results for an office building are shown in this paper.

(Keywords: office buildings, thermal energy, load simulation, computer analysis)

0 UVOD

Toplotne in hladilne obremenitve so toplotna energija, ki jo moramo dovesti ali odvesti iz notranjosti prostora stavbe, da ohranimo karakteristike ugodja. Ko obremenitve določimo, se je treba lotiti določevanja opreme za ogrevanje in hlajenje.

Najpomembnejša skrb inženirjev je določiti največjo obremenitev, ker je od te odvisna moč opreme. Le-ta se ujema z ekstremnimi vrednostmi vročega ali hladnega vremena, ki jih imenujemo *projektni pogoji*. Naftna kriza je izostrila našo zavest do energije in računalniška revolucija je dala opremo za optimiranje projektiranja stavb in za izračun stroškov energije.

Svež zrak v stavbi je pomemben za ugodje in zdravje. Energija za pripravo tega zraka je pomemben dejavnik. Premalo zraka povzroča sindrom bolne stavbe, preveč zraka pa povzroča večjo rabo energije. Izmenjava zraka je mišljena kot tok zunanjega zraka, ki prečka mejo poslopja in ga je treba klimatizirati. Pogosto je primerno, da ga delimo s prostornino stavbe, kar izražamo v enotah izmenjav zraka na uro.

1 TRNSYS

Za simuliranje smo uporabili računalniški program TRNSYS [1] (‘tran-sys’), ki je komercialno

0 INTRODUCTION

Heating and cooling loads are thermal energy that must be supplied or removed from the interior of a building in order to maintain comfortable conditions. Once the loads have been established, one can proceed to the supply side and determine the performance of the required heating and cooling equipment.

Of primary concern to engineers are the peak loads, because they determine the capacity of the equipment. They correspond to the extremes of hot and cold weather, and are called *design conditions*. The oil crises have sharpened our awareness of energy, and the computer revolution has given us the tools to optimise the design of a building and to compute the cost of energy.

Fresh air in a building is essential for comfort and health, and the energy for conditioning this air is an important factor. Not enough air, and one risks sick-building syndrome; too much air, and one wastes energy. The supply of fresh air, or air exchange, is stated as the flow rate of the outdoor air that crosses the building boundary and needs to be conditioned. Often it is convenient to divide it by the building volume, expressing it in units of air changes per hour.

1 TRNSYS

For the simulation we have used the TRNSYS [1] (‘tran-sis’) computer programme, com-na voljo od leta 1975 in je namenjen simuliranju prehodnih pojavov toplotnih sistemov. TRNSYS je za reševanje problemov zasnovan modularno in uporablja računalniški jezik Fortran. Na primer podprogram Type 32 vsebuje model hladilne naprave. Podprogram Type 56 vsebuje model večconske zgradbe. Vsak modul ima vhod in izhod. Velikost tokov ter temperature vode in zraka so vstopni podatki za modul Type 32, medtem ko sta celotna in latentna toplotna moč izhodna podatka iz modula. Z izdelavo vhodnega modela uporabnik ukaže TRNSYSu, kako naj poveže module med seboj, ki tako tvorijo sistem. TRNSYS potem kliče posamezne podprograme glede na vhodno datoteko in iterira podatke v vsakem časovnem koraku, dokler sistem enačb ni rešen. Alternativa tej metodi je za raziskovalce, da napišejo enoten program, ki modelira samo en sistem. Toda vsakršne spremembe so pri takem programu bolj zapletene, kakor če uporabljamo TRNSYS.

2 REFERENČNE STAVBE

Pri naših simuliranjih smo uporabili naslednje objekte [4]:

- pisarno
- bolnišnico
- hotel
- šolo
- prodajalno
- stanovanjsko poslopje
- enodružinsko stavbo

Za **pisarniški objekt smo** uporabili World Trade Center v Ljubljani. Zgradba je usmerjena na sever-jug. Tloris ima izmere 28,35 x 25,6 m. Ima pritličje (4,5 m višine) + eno nadstropje (4,5m) + 15 nadstropij z višino 3,5m. Streha ima strmino 12°. Celotna višina objekta na jugu je 61,5m.

Za **bolnišnico** smo vzeli porodnišnico Univerzitetnega kliničnega centra v Ljubljani. Tloris poslopja je 44 x 38 m. Usmerjena je na sever-jug. Višina poslopja je 24 metrov (pritličje + 5 nadstropij, višina enega nadstropja je 4m). Prostornina poslopja je 44 x 38 x 24 = 40128 m^3.

Hotel ima tloris 18 x 40 m in prostornino 36000 m^3. 100 % je zaseden v poletni in 35% zimski sezoni. Povedali so nam, da je povprečna temperatura v hotelu pozimi 23 °C in poleti 21 °C.

Za **šolo** smo uporabili poslopje s tlorisom 30 x 80 m (okoli 8 razredov v vsakem od dveh nadstropij). Pouk teče (v razredih na levi) samo od 8:00 do 15:00 ure. V njej je 500 učencev in učiteljev. V popoldanskem času so samo dejavnosti v telovadnici (prostor na desni) in je v uporabi do 22:00 ure. Učence jemljemo kot standardne osebe.

mercially available since 1975, which is designed to simulate the transient performance of thermal energy systems. TRNSYS relies on a modular approach to solve large systems of equations described by Fortran subroutines. Each Fortran subroutine contains a model for a system component. For example, Subroutine Type 32 contains a model of a cooling coil. Subroutine Type 56 contains a model of a multizone building. Each component has inputs and outputs. The inlet flow rates and temperatures for the air and water are inputs to the Type-32 model, while the total and latent cooling rates are among the outputs of the model. By creating an input file, the user directs TRNSYS to connect the various subroutines to form a system. The TRNSYS engine calls the system components based on the input file and iterates at each timestep until the system of equations is solved. The alternative to this method is for the researcher to write a single, monolithic program that models only the system at hand. Subsequent changes to the system configuration are more difficult with monolithic programs than they are with modular programs such as TRNSYS.

2 REFERENCE BUILDING

In our simulations we have taken the following reference buildings [4]:

- office
- hospital
- hotel
- school
- store
- apartment house
- single house

For the **office building** we have used the World Trade Centre in Ljubljana. The building has a north-south orientation. The ground plan has dimensions of 28.35 x 25.6 m. It has a ground floor (height 4.5 m) + one floor (4.5m) + 15 levels with a height of 3.5m. The roof has an angle of 12°. The total height of the object to the south is 61.5 m.

For the **hospital** we chose the maternity hospital at University Clinical Centre. The ground plan of the building is 44 x 38 m. It is oriented north-south. The height of the building is 24 meters (ground floor + 5 floors, the height of 1 floor is 4m). The volume of the building is 44 x 38 x 24 = 40128m^3.

The **hotel** has a ground plan of 18 x 40 m and a volume of 36000 m^3. It has 100 % occupancy in the summer and 35 % in the winter. We were told that the temperature of the hotel in winter is 23°C and in summer 21°C.

For the **school** we took a building with a ground plan: 30 m x 80 m (cca. 8 classes on each of two floors). The lessons are only in progress (rooms on the left) from 8:00 till 15:00 hrs. There are 500 pupils and teachers. In the afternoon there are some activities only in the gymnasium (room on the right) which is in use till 22:00. The pupils are taken as standard persons.

Za primer **prodajalne** smo uporabili supermarket MERCATOR. Nova stavba, ki je bila zgrajena leta 1999, ima tloris 170 m x 110 m z višino 8 m. Daljša fasada je usmerjena na JV-SZ. Prostornina poslopja je V = 170 x 110 x 8 = 149600 m^3.

Za **stanovanjsko poslopje** smo uporabili večdružinsko stavbo. Stavba je usmerjena na sever-jug. Tloris meri 16 m x 64 m. Imamo pritličje in 12 nadstropij ter podstrešje. Višina nadstropja je 3 m in stavba je za 270 ljudi. Prostornina je 40960 m^3 (224 stanovanjskih enot). Streha je ravna in pohodna.

Za **enodružinsko hišo** smo analizirali hišo za eno družino v ljubljanskem predelu Murgle. Tloris stavbe je 15 x 8 metrov in je usmerjena na sever – jug.

As an example of the **store** we took a MERCATOR supermarket. A new building which was built in 1999 and has a ground plan of 170 m x 110 m with a heiht of 8 m. The long facade is oriented towards SE-NW. The volume of the building is 170 x 110 x 8 = 149600 m^3.

For the **apartment house** we took a multi-family building. The building is oriented north-south, the ground plan is 16 m x 64 m. There is a ground floor plus 12 flats, plus attics. The height of each flat is 3m and the building is for 720 people. The volume is 40960 m^3 (224 living units). The roof is flat and walking.

For the **single house** we have analysed a house for one family in the Murgle area of Ljubljana. The ground plan of the house is 15 x 8 meters and is oriented north-south.

3 KLIMATSKI PODATKI

Tipe stavb, predstavljene v 2. poglavju smo simulirali s štirimi klimatskimi pogoji v Evropi. Uporabili smo Testno referenčno leto (TRL - TRY) [2] za naslednje države:

- Velika Britanija – London
- Švedska – Stockholm
- Italija – Rim
- Slovenija – Ljubljana

Podatki za TRL so sestavljeni iz mesečnih vrednosti različnih let. Za Ljubljano je Testno referenčno leto sestavljeno iz podatkov v letih 1961 do 1980. Testno referenčno leto vsebuje naslednje podatke:

1) Številko dneva v letu (1 do 365),
2) Zunanjo temperaturo (°C)
3) Relativno vlažnost (%)
4) Hitrost vetra (m/s)
5) Globalno sevanje na vodoravno ploskev (kJ/hm^2) – če je vrednost nič, potem je podano neposredno sončno sevanje
6) Uro v dnevu
7) Neposredno sončno sevanje (kJ/hm^2) – če je vrednost nič, potem je dano globalno sevanje
8) Difuzno sončno sevanje (kJ/hm^2) - če je vrednost nič, potem je dano globalno sevanje

3 CLIMATE DATA

The building types described in section 2 have been simulated in four different climatic conditions for Europe. We have used a Test Reference Year (TRY) [2] for the following countries:

– United Kingdom – London,
– Sweden – Stockholm,
– Italy – Rome,
– Slovenia – Ljubljana.

Data for the TRY are made up from months of different years. For the Ljubljana Test Reference Year was taken data from years 1961 to 1980. Test Reference Year is a file of data which contains:

1) Number of days in the year (1 to 365),
2) External temperature (°C),
3) Relative humidity (%),
4) Wind velocity (m/s),
5) Total horizontal radiation (kJ/hm^2) – if the value is zero then data for direct radiation is given,
6) Hour in the day,
7) Direct solar radiation (kJ/hm^2) – if the value is zero then only the total solar radiation is given,
8) Difuse solar radiation (kJ/hm^2) - if the value is zero then only the total solar radiation is given.

4 SIMULIRANJA

Simuliranja smo izvajali s programom TRNSYS s podatki za objekte (poglavje 2) in s podatki za testno referenčno leto (poglavje 3).

Za vse primere smo vzeli nespremenljiv koeficient toplotne konvekcije znotraj in zunaj zidu. Po DIN 4701 – del 2 ([3] in [5]) smo uporabili:

- za zunanji prenos toplote 22,7 W/m^2K (81,7 kJ/hm^2K),
- za notranji prenos toplote 7,7 W/m^2K (27,7 kJ/hm^2K),
- za notranji prenos toplote – strop 5,88 W/m^2K (21,2 kJ/hm^2K).

4 SIMULATIONS

Simulations have been made using TRNSYS with building data (section 2) and with a Test Reference Year (section 3).

For all cases we took the constant convective heat-transfer coefficient inside and outside the wall. By DIN 4701 – Part 2 ([3] and [5]) we have used:

- for external heat transfer 22.7 W/m^2K (81.7 kJ/hm^2K),
- for internal heat transfer 7.7 W/m^2K (27.7 kJ/hm^2K),
- for internal heat transfer 5.88 W/m^2K (21.2 kJ/hm^2K).

Za vse primere je absorptivnost stene 0,6 (sončna absorptivnost stene). V primerih, ko imamo pred zidom steklo je celotna absorptivnost 0,6. Transmitivnost stekla reflectafloat je 57%, tako je absorptivnost stene 0,6 x 0,57 = 0,34. Reflektivnost okolice je 0,2.

Za vse primere smo uporabili bruto prostornino. V tem primeru je prostornina od 20 do 30% večja od prostornine zraka v stavbah. V vseh primerih smo vzeli infiltracijo 0,6 izmenjav zraka na uro. V vseh primerih je bilo vključeno ogrevanje/hlajenje z razvlaževanjem 50%.

Vsi primeri so bili narejeni tako, da je celoten zrak za ventilacijo vstopa v stavbo z zunanjo temperaturo (brez rekuperacije toplote). Zaradi tega so individualni toplotni tokovi prikazani posebej.

Za geografsko širino smo vzeli:
Ljubljana 46,22 S,
London 51,15 S,
Stockholm 59,35 S,
Rim 41,80 S.

Vsa simuliranja so bila narejena za vse leto. V datoteki z rezultati imamo naslednje podatke:

1. TIME	ura v letu (1 do 8760)
2. NO. OF DAY	številka dneva v letu (1 do 365)
3. DATE	datum dneva v letu (1.1. do 31.12)
4. Hour	ura v dnevu (0 do 24)
5. Toutside	zunanja temperatura (°C)
6. Tinside	notranja temperatura v stavbi. (°C) – temperatura v coni
7. Qsensible	senzibilni toplota (- ogrevanje, +hlajenje) (kJ/h)
8. Qsur	konvekcija zraka iz vseh sten v coni (kJ/h)
9. Qinf	infiltracijski energetski dobitki (kJ/h)
10. Qv	ventilacijski energetski dobitki (kJ/h)
11. Qg_c	notranji konvekcijski dobitki (kJ/h)
12. Qg_l	latentni energetski dobitki (kJ/h)
13. Qg_r	celotni notranji sevalni dobitki (kJ/h)
14. QUA_trans	stacionarne izgube sten in oken v coni z uporabo »k« vrednosti, podane v izračunu (kJ/h)

Prezračevalne izgube so izračunane pri svežem zraku z zunanjo temperaturo. Tako so te izgube/dobitki zelo pomembni in se izračunajo po enačbi:

For all cases the absorbtivity of the walls was 0.6 (Solar Absorbtance of wall). In the cases where we have glass in front of the wall the total absorbtivity is 0.6. The transmitivity of the glass reflectafloat is 57 % so the absorbtivity of the wall is 0.6 x 0.57 = 0.34. The reflectivity of the surroundings was 0.2.

For all cases we have used the gross volume. In this case the volume is from 20 to 30 % bigger than the volume of the air in the building. In all cases we have used an infiltration of 0.6 volume air-changes per hour. In all cases heating/cooling was on with dehumidisation at 50% humidity.

All cases were made so that all the air for ventilation comes into the building with the external temperature (without recuperation of heat). For this reason, individual heat fluxes are presented separately.

For the locations the following latitude was used:
Ljubljana 46.22 N,
London 51.15 N,
Stockholm 59.35 N,
Rome 41.80 N.

All simulations have been made for a whole year. In the results file we have the following data:

1. TIME	hour in the year (1 to 8760)
2. NO. OF DAY	number of day in the year (1 to 365)
3. DATE	date for the day in the year (January the 1st to December the 31st)
4. Hour	hour in the day (0 to 24)
5.Toutside	external temperature (°C)
6. Tinside	internal temperature in the building (°C) - air temperature of zone
7. Qsensibile	sensible energy demand (- heating, + cooling) (kJ/hr)
8. Qsur	total convection to air from all surfaces within zone (kJ/hr)
9. Qinf	infiltration energy gain (kJ/hr)
10. Qv	ventilation energy gain (kJ/hr)
11. Qg_c	internal convective gains (kJ/hr)
12. Qg_l	net latent energy gains (kJ/hr)
13. Qg_r	total internal radiative gain (kJ/hr)
14. QUA_trans	stationary U·A-transmission losses of walls and windows of zone using the u-values given in the transfer calculation section (kJ/hr)

Ventilation losses are calculated for necessary fresh air with external temperature. So these losses/gains are very important and they are calculated using the equation:

$$Q_v = \dot{V}.\rho.c_p(T_{vent} - T_{air})$$

kjer je: / where:

$$T_{\mathrm{vent}} = T_{\mathrm{outside}}$$

Ker so rezultati prikazani za vsak mesec posebej, lahko spremenimo prezračevalne rezultate zelo preprosto. Na enak način je izračunana tudi infiltracija, kjer smo uporabili 0,6 izmenjav zraka na uro.

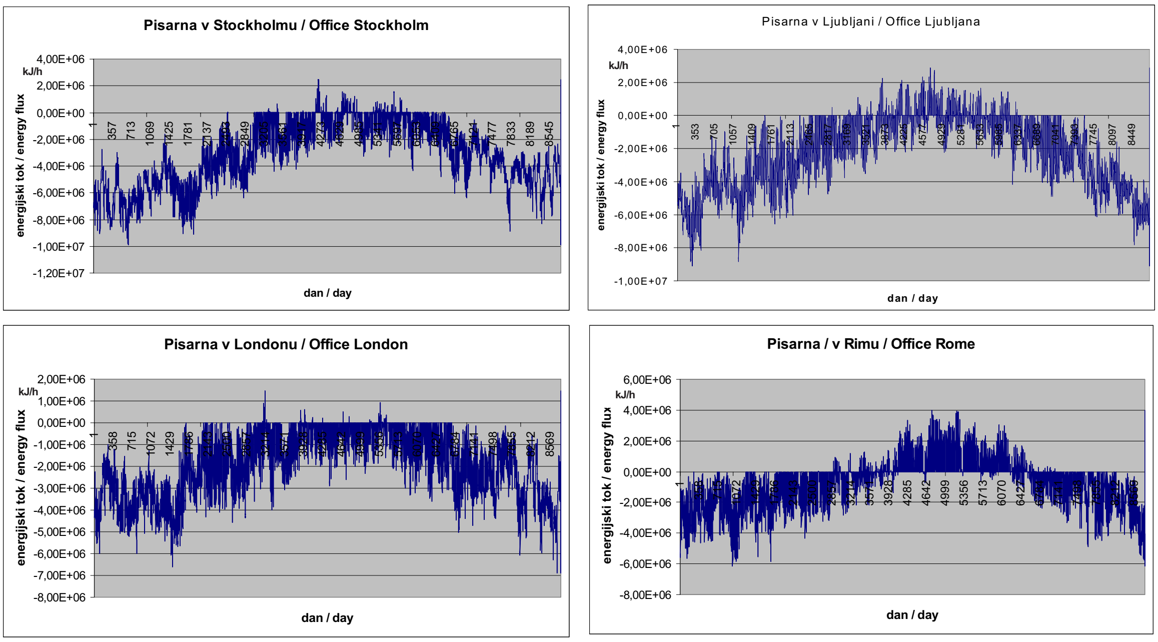
Since the results are shown separately for each hour we can change the ventilation results very easily. In the same way, infiltration is also calculated where we have used 0.6 of volume air-changes per hour.

5 REZULTATI

Skupaj smo naredili 4 x 7 = 28 simuliranj za 8760 ur. Na naslednji sliki predstavljamo rezultate za pisarniško poslopje. Pozitivne vrednosti pomenijo hlajenje, negativne pa ogrevanje.

5 RESULTS

Altogether we have made 4 x 7 = 28 simulations for 8760 hours. In the next figure we present the results for the office building . Positive values mean cooling and negative values mean heating.



Sl. 1. *Primer senzibilnih obremenitev v času enega leta*
Fig. 1. *An example of sensible heat loads for a period of one year*

Za inženirsko uporabo so najpomembnejše najmanjše in največje vrednosti. Na sliki 2 predstavljamo te rezultate za pisarniško poslopje.

For engineering use the most important data are the minimum and maximum values. In Fig. 2 we present those values for the office building.

6 SKLEP

Ugotovili smo, da imata Stockholm in Ljubljana zelo podobne toplotne obremenitve, podobno kakor London in Rim. Razmerje obremenitev med Ljubljano in Rimom pa je 3:2. Na drugi strani so hladilne obremenitve podobne za Ljubljano in Stockholm, medtem ko za London in Rim ugotavljamo razlike, kar je posledica različnih klimatskih razmer.

6 CONCLUSION

We found that Stockholm and Ljubljana have very similar heating loads as do London and Rome. But the ratio between loads in Ljubljana and Rome is 3:2. On the other hand, cooling loads are similar for Ljubljana and Stockholm whereas for London and Rome we found a difference which is the consequence of different climatic conditions.

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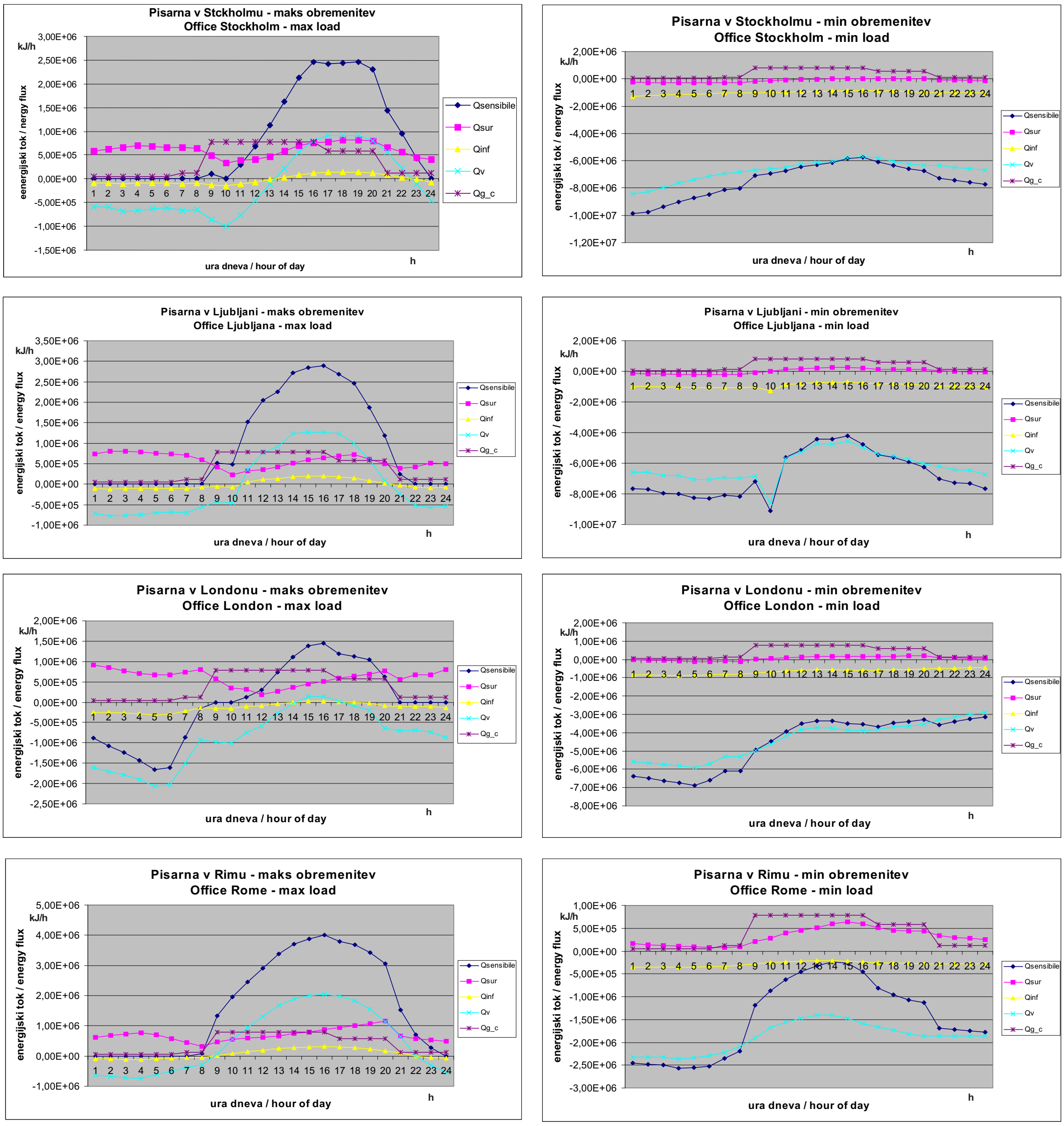

Sl. 2. *Največje in najmanjše obremenitve za pisarniško zgradbo*
Fig. 2. *Maximum and minimum loads for office building*

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Prejeto:
Received: 15.8.2000

Sprejeto:
Accepted: 10.11.2000

© Strojniški vestnik 46(2000)8,509-516
ISSN 0039-2480
UDK 536.24:546.212:66.06:628.16.04/.09:537.84
Pregledni znanstveni članek (1.02)
© Journal of Mechanical Engineering 46(2000)8,509-516
ISSN 0039-2480
UDK 536.24:546.212:66.06:628.16.04/.09:537.84
Review scientific paper (1.02)

Preprečevanje izločanja vodnega kamna na površinah prenosnikov toplote z uporabo naprave za magnetno obdelavo vode

The Prevenation of Surface Precipitation on Heat Exchangers Using a Magnetic Water-Treatment Device

Andrej Pristovnik - Lucija Črepinšek Lipuš - Jurij Krope

V nalogi predstavljamo metodo za nadzor vodnega kamna na temelju magnetne obdelave vode (MOV) v prenosnikih toplote.

Podali smo teoretičen pregled tvorbe kotlovca pri industrijskih prenosnikih toplote s poudarkom na obarjanju kalcijevega karbonata ($CaCO_3$) in kalcijevega sulfata ($CaSO_4$) ter osnovne izračune za uspešno uporabo naprav MOV pri preprečevanju nastajanja vodnega kamna.

(Ključne besede: prenosniki toplote, zaščita proti kotlovcu, magnetna obdelava vode, magnetohidrodinamika)

Magnetic water treatment (MWT), a water-conditioning method for scale control in heat exchangers (HEs), is discussed.

The theoretical possibilities of scale formation in industrial processes with the emphasis on the precipitation of $CaCO_3$ and $CaSO_4$ as the main scale components, are reviewed. Some preliminary calculations for a theoretical understanding of the scale problem in HEs and its prevention using MWTs are contributed.

(Keywords: heat exchangers, scale control, magnetic water treatment, magnetohydrodynamic)

0 UVOD

Problem izločanja vodnega kamna se pojavlja pri vseh tehnoloških procesih, ki uporabljajo naravno vodo. To pa še posebej velja v primeru uporabe prenosnika toplote, pri katerem pride do povišanja temperature in posledično do prenasičenja soli, ki tvorijo vodni kamen (predvsem $CaCO_3$ in $CaSO_4$). Obstaja več dobro znanih in uporabnih metod za preprečevanje nastajanja vodnega kamna. Uporaba nekaterih pomeni velik finančni strošek, druge pa onesnažujejo okolje. V zadnjih letih se kot alternativa kemičnim metodam vedno bolj uveljavlja t.i. magnetna obdelava vode (MOV).

Čeprav je metoda znana že petdeset let in z ekonomskega in okoljevarstvenega vidika zelo sprejemljiva, prav procesna industrija še naprej dvomi o njeni učinkovitosti in uporabi ([1] do [4]).

1 NASTANEK VODNEGA KAMNA

Naravna voda je dejansko bogata raztopina/disperzija mnogih ionov: Ca^{2+}, Mg^{2+}, Na^+, K^+, HCO_3^-, SO_4^{2-} in Cl^-. Ioni Na^+, K^+ in Cl^- so inertni, preostali pa so vključeni v t.i. medfazno ravnotežje. Zaradi

0 INTRODUCTION

The build-up of scale deposits is a common and costly problem in many industrial processes which use natural water supplies, especially in heat-exchange processes, where a high oversaturation of scale-forming components (i.e. $CaCO_3$ and $CaSO_4$) is established. There are many well-known scale-prevention methods, but they are costly and environmentally unfriendly. MWT is being used more and more as an alternative method for scale control.

The process industry remains skeptical about this non-chemical method despite its long history and examples of favorable economic benefits ([1] to [4]).

1 SCALE FORMATION

Natural waters are rich solution/dispersion systems which contain the ions: Ca^{2+}, Mg^{2+}, Na^+, K^+, HCO_3^-, SO_4^{2-} and Cl^-. The Na^+, K^+ and Cl^- ions are inert, while the others are incorporated into an inter-sprememb obratovalnih razmer (sprememba tlaka, temperature, vrednosti pH) pride do prenasičenja in soli se v obliki vodnega kamna izločajo na stene cevi, prenosnikov toplote in drugih naprav, ki so v stiku z vodo.

Najpomembnejši parameter za nadzor vodnega kamna je delež kalcijevih ionov Ca^{2+}. Določimo ga s pomočjo t.i. karbonatnega ravnotežja ((1) do (4)). Parametra (c) in (K) pomenita koncentracijo in konstanto ravnotežja.

phase equilibrium. Due to the natural supersaturation of the supplied water or supersaturating due to changed operating conditions (such as a pressure drop, temperature or pH increase) hard scale precipitates in pipelines and on the walls of equipment.

The most important parameter in scale control is the concentration of Ca^{2+} ions, determined by carbonate equilibrium ((1) to (4)), where the parameter c is the concentration and parameters K is the equilibrium constant.

$$CO_{2(g)} + H_2O \leftrightarrow H_2CO_3, \qquad K_g = \frac{c_{H_2CO_3}}{p_{CO_2}} \tag{1}$$

$$H_2CO_3 \leftrightarrow H^+ + HCO_3^-, \qquad K_1 = \frac{c_{HCO_3^-} \cdot c_{H^+}}{c_{H_2CO_3}} \tag{2}$$

$$HCO_3^- \leftrightarrow H^+ + CO_3^{2-}, \qquad K_2 = \frac{c_{H^+} \cdot c_{CO_3^{2-}}}{c_{HCO_3^-}} \tag{3}$$

$$CaCO_{3(s)} \leftrightarrow Ca^{2+} + CO_3^{2-}, \qquad K_s = c_{Ca^{2+}} \cdot c_{CO_3^{2-}} \tag{4}$$

Iz pogoja o električni nevtralnosti (5) in z upoštevanjem ionskega produkta vode (6) lahko izpeljemo odvisnost koncentracije kalcijevih (Ca^{2+}) ionov kot funkcije vrednosti pH in temperature.

From the condition of the solution's electric neutrality (5) and the water dissociation equilibrium (6), the concentration of Ca^{2+} ions can be derived as a function of pH and temperature.

$$2c_{Ca^{2+}} + c_{H^+} = 2c_{CO_3^{2-}} + c_{HCO_3^-} + c_{OH^-} \tag{5}$$

$$H_2O \leftrightarrow H^+ + OH^- \qquad K_w = c_{H^+} \cdot c_{OH^-} \tag{6}$$

$$c_{Ca^{2+}} = \frac{K_W - c_{H^+}^2 + \left[\left(c_{H^+}^2 - K_W\right)^2 + 8c_{H^+}^2 \cdot K_s\left(2 + \frac{c_{H^+}}{K_2}\right)\right]^{1/2}}{4c_{H^+}} \tag{7}$$

Konstante ravnotežja so odvisne od temperature (7). V naravnih vodah (pH < 7) vodi zvišanje temperature in vrednosti pH do znižanja ravnotežne koncentracije Ca^{2+} ionov (7). Pri znižanju tlaka pride do znižanja koncentracije H_2CO_3 (1) in posledično s povečevanjem vrednosti pH pospešeno obarjanje $CaCO_3$ ((2) do (4) in (7)).

S temperaturo (do 40 °C) se zvečuje topnost $CaSO_4$, pri višjih temperaturah (okoli 100 °C) pa naglo zmanjšuje. Iz opisanega je razvidno, da se bo v nizkotemperaturnih sistemih v glavnem izločal kalcijev karbonat ($CaCO_3$) in v visokotemperaturnih sistemih (toplovodi, uparjalniki, prenosniki toplote) pa kalcijev sulfat ($CaSO_4$).

The equilibrium constants in equation (7) are temperature dependent. In natural waters (with a pH less than 7), a rise in temperature and pH leads to a reduction of the Ca^{2+} equilibrium concentration according to equation (7). The pressure drop leads to a lower concentration of H_2CO_3 according to equation (1) and causes $CaCO_3$ precipitation with a pH increase according to eqs. ((2) to (4) and (7)).

The solubility of $CaSO_4$ increases as the temperature increases to approximately 40°C and then rapidly decreases at higher temperatures around 100°C. As a result, $CaCO_3$ is the main scale component in low-temperature water systems, while in high-temperature water systems (especially in high-pressure heat exchangers and boilers) $CaSO_4$ prevails.

2 ZMANJŠANJE UČINKOVITOSTI PRENOSA TOPLOTE

Obloge vodnega kamna, ki nastanejo na površinah prenosov toplote, zmanjšujejo pretočne zmogljivosti in predvsem učinkovitost prenosnikov toplote ter s tem zvišujejo investicijske, obratovalne in vzdrževalne stroške. Brez primerne obdelave napajalne

2 HEAT EXCHANGE REDUCTION

The scale formed on heated surfaces reduces the flow capacity and heat exchange efficiency which leads to higher investment, operation and maintenance costs. Hard scale can be a severe industrial problem without properly supplied water condition-vode so tako nastale trdovratne obloge težak industrijski problem; terjajo periodično čiščenje z mehanskimi postopki in jedkanjem s solno kislino.

Naslednja ocena bo pokazala, kako vodni kamen izrazito znižuje prenos toplote.

Moč toplotnega toka P_1 skozi kovinsko steno površine S_{stene} pri temperaturni razliki ΔT je za nov prenosnik (sl. 1.a) določena z enačbo (8). Prestopnostni koeficient α_1 je tu praktično enak konvekcijskemu koeficientu plasti vode na obeh straneh stene. Konvekcijski koeficient kovine je namreč bistveno višji kakor za vodo

ing. It demands periodic cleaning using mechanical methods and HCl etching.

The following preliminary calculations show how scale drastically reduces the exchange of heat.

With a new wall of a HE (Fig. 1/a), the heat-flow intensity (P_1) through the metallic wall of area S at a temperature difference ΔT is determined by equation (8), where the heat transition coefficient (α_1) is practically equal to the convection coefficient of the water layer on both sides of the wall due to the much higher value of the heat-conduction coefficient of the metal

$$P_1 = \alpha_1 \cdot S \cdot \Delta T \qquad (8).$$

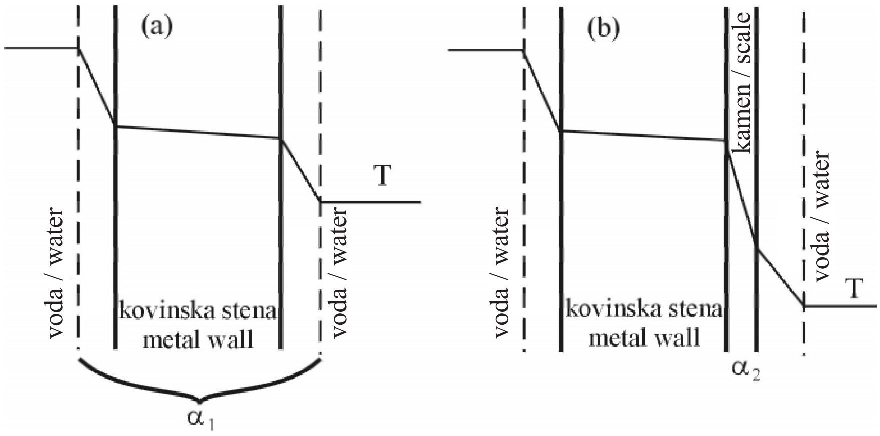
Obloge vodnega kamna (sl. 1.b) znižujejo moč toplotnega toka P' in je ta določen z enačbo (9). Tukaj se lahko celokupni prestopnostni koeficient α' izračuna iz α_1 nove stene in α_2 nastalih oblog po enačbi (10). Velja za postavko iste temperaturne razlike med ogrevano in hladilno vodo $\Delta T = \Delta T_1 + \Delta T_2$. Koeficient α_2 je odvisen od celotne debeline oblog Δy_2 po zvezi (11), kjer je λ_2 toplotna prevodnost vodnega kamna.

The formation of scale (Fig. 1/b) reduces the heat flow intensity (P') according to equation (9), where the total heat-transition coefficient (α') can be calculated using α_1 of the new wall and α_2 of the formed scale according to equation (10) at the same temperature difference $\Delta T = \Delta T_1 + \Delta T_2$. The coefficient α_2 depends on the total scale lining thickness (Δy_2) according to equation (11), where λ_2 is the heat conductivity of the scale.

$$P' = \alpha' \cdot S \cdot \Delta T = \alpha_1 \cdot S \cdot \Delta T_1 = \alpha_2 \cdot S \cdot \Delta T_2 \qquad (9)$$

$$\alpha' = \frac{1}{1/\alpha_1 + 1/\alpha_2} \qquad (10)$$

$$\alpha_2 = \frac{\lambda_2}{\Delta y_2} \qquad (11).$$



Sl. 1. *Temperaturni krivulji skozi: (a) novo kovinsko steno in (b) skozi kovinsko steno z oblogo vodnega kamna*

Fig. 1. *The temperature curve through a new metallic wall (a) and through a metallic wall covered with scale (b)*

Preglednica 1 prikazuje nekaj vrednotenj relativnega zmanjšanja učinkovitosti prenosa toplote ξ, ki je definirana z enačbo:

Table 1 represents some estimations for the relative drop of the heat-exchange efficiency (ζ) defined by equation:

$$\zeta = \frac{P_1 - P'}{P_1} = 1 - \frac{\alpha'}{\alpha_1} = 1 - \frac{1}{1 + \alpha_1 \Delta y_2 / \lambda_2} \qquad (12)$$

Preglednica 1. *Relativna zmanjšanja učinkovitosti prenosa toplote (pri izbrani praktični vrednosti α_1 = 500 W/m²K za kovinsko steno) zaradi oblog $CaCO_3$ (λ_2 = 1,75 W/mK) oziroma $CaSO_4$ (λ_2 = 0,50 W/mK)*
Table 1. *Relative drops of heat-exchange efficiency at chosen practical values α_1 = 500 W/m²K due to $CaCO_3$ lining (λ_2 = 1.75 W/mK) and $CaSO_4$ lining (λ_2 = 0.50 W/mK), respectively*

Δy_2	1,5 mm	5,5 mm	20 mm
$\zeta(CaCO_3)$	30%	60%	85%
$\zeta(CaSO_4)$	60%	85%	95%

Rezultati potrjujejo praktične izkušnje, da zaradi nizke toplotne prevodnosti $CaCO_3$ in $CaSO_4$, celo tanke obloge vodnega kamna izrazito zmanjšujejo učinkovitost prenosa toplote. V visokotlačnih grelnih napravah je ta problem še posebej močno izražen, saj se v večinskem deležu izloča kalcijev sulfat, ki ima manjšo toplotno prevodnost od kalcijevega karbonata.

V mnogih primerih se je izkazalo, da omogočajo naprave za magnetno obdelavo vode razmeroma učinkovit sistem za nadzor vodnega kamna. Eden od uspešnih preskusov naprav MOV domačega proizvajalca Panorama Ptuj [6] pomeni vgradnja le-teh v prenosnik toplote Toplotne oskrbe Maribor (TOM) [5].

Naprave so bile instalirane na ceveh s hladno napajalno vodo in so učinkovito preprečile nastanek vodnega kamna. V preglednici 2 sta predstavljena rezultata vgradnje naprav za magnetno obdelavo v prenosnika toplote.

Results prove that even thin scale linings drastically reduce the heat-exchange efficiency because of the low heat of conductivity of the scale components $CaCO_3$ and $CaSO_4$. In high-pressure boilers the problem will be even greater due to the main scale component, $CaSO_4$, which has a lower heat of conductivity than $CaCO_3$.

These theoretical predictions are in accordance with many practical results, where scale formation on HE surfaces demanded a preliminary treatment of the supplied water. In many cases MWT turned out to be a very efficient method for scale control. The installation of MWT devices to prevent hard scale in the HEs in the TOM town heating station [5] was one of the successful domestic tests of the Panorama Ptuj magnetic device [6].

These devices were installed on the cold water pipeline entrance of the HE and efficiently solved any problems with hard scale. Table 2 represents some observations on the scale in the two HEs which were supplied with magnetic ally treated water.

Preglednica 2. *Rezultati naprave za magnetno obdelavo proizvajalca Panorama Ptuj v toplotni postaji TOM-a*
Table 2. *Results of Panorama Ptuj devices in the TOM station*

Prenosnik toplote HE	star cevni register U old U-pipe register	nov spiralni register new spiral register
obloge ob vgradnji scale at MWT installation	da yes	ne none
prvi pregled time of the first control	8 mesecev po vgradnji 8 months after installation	11 mesecev po vgradnji 11 months after installation
stanje po prvem pregledu state after the first control	obloge, odstranitev z vodnim visokotlačnim curkom present scale was removable with high-pressure water jet	tanke plastne obloge, odstranitev z vodnim visokotlačnim curkom thin powder scale was removed with jet
drugi pregled time of the second control	16 mesecev po vgradnji 16 months after installation	17 mesecev po vgradnji 17 months after installation
stanje ob drugem pregledu state after the second control	oblog ni bilo,površina je bila veliko bolj čista kakor pred samo vgradnjo without a new scale, surfaces were cleaner than before the installation time	enako kakor pri prvem pregledu the same as at the first control

3 NADZOR VODNEGA KAMNA V PRENOSNIKIH TOPLOTE

V naravni vodi, bogati z raztopljenimi/dispergiranimi snovmi, delujejo naprave za magnetno obdelavo vode neposredno na samo stabilnost in

3 THEORETICAL PRINCIPLES OF MWT SCALE PREVENTION ON HE SURFACES

The nature of MWT devices acting on supplied water as a rich solution/dispersion system is to alter its crystallization habits and dispersion stability to form kristalizacijo dispergiranih delcev. Kristali, ki se izločajo po obdelavi, so večji in modificirani. Prav na teh kristalih se neposredno iz vode izloči večji del soli, tako da se na stenah naprav nabere neprimerno manj vodnega kamna.

Ob pretakanju vode skozi napravo za magnetno obdelavo prihaja do sprememb, ki pa se izražajo (najverjetneje) v spremenjeni ionski hidrataciji prek magnetohidrodinamičnega premika ionov in koncentracijskega vpliva na dispergirane delce v sami napravi MOV [7].

Izračuni kažejo, da se med magnetno obdelavo vode agregatne tvorbe, sestavljene iz $CaCO_3$ in $CaSO_4$ trdno sprimejo. Iz samega načela staranja kristalov namreč kosmiči, v katerih so delci med seboj šibko povezani, niso tako zaželeni kakor goste agregatne tvorbe [8].

Po teoriji DLVO (Deryagin, Landau, Verwey, Overbeck) ([9] in [10]) smo opravili numerično analizo koagulacije in kosmičenja nemagnetnih delcev vodnega kamna in prišli do sklepa, da v naravnih vodah prevladuje koagulacija, ki je odvisna od same naprave MWT, medtem ko je zaradi nizke vrednosti Hamakerjeve konstante in nizke magnetne susceptibilnosti pri večjih delcih ($a > 0{,}1$ μm) mogoča le kosmičenje.

Po drugi strani pa se bodo magnetohidrodinamično nastali kosmiči pod vplivom turbulentne pulzacije razbile. Do pulzacije prihaja v večini naprav MOV, kjer je priporočena pretočna hitrost od 0,5 do 2 m/s.

Ob preseženi vrednost Reynoldsovega števila (10^4) imamo opraviti s turbulentnim tokom

bigger modified crystals, which in suspended form offer surfaces for scale precipitation and in that way hard scale formation indirectly prevails on equipment walls.

The change in the water's behaviour when the water flows through the magnetic field is most probably a result of altered ion hydration, by magnetohydrodynamic shifts of ions and concentration effects on the dispersed particles in the working channel of the MWT device [7].

Some calculations have been made showing that all aggregates, formed from scale components ($CaCO_3$ and $CaSO_4$) during MWT, are compact-strongly adhered. In other words, the flocks in which constituent particles are weakly bonded are not as favorable for scale prevention as the compact aggregates according to the principles of crystal aging [8].

A numerical analysis of the coagulation and flocculation of the nonmagnetic scale components, based on the Deryagin, Landau, Verwey, Overbeck theory ([9] and [10]), has been made. It offered an estimation that in natural waters only flocculation from big particles (with radius $a > 0{,}1$ μm) is possible due to the low Hamaker constant and low magnetic susceptibility of these components, while a coagulation prevails and depends on the MWT working conditions.

On the other hand, the magnetohydrodynamically formed big flocks will be shattered by turbulent pulsations which appear in the majority of practical MWT devices, where the recommended values of water flow velocity are in range from 0.5 to 2 m/s for efficient anti-scale treatment.

The Reynolds number Re, defined by equation (13), characterizes turbulent flow, if it is greater than 10^4

$$\mathrm{Re} = \frac{\rho \cdot v \cdot d}{\eta} \qquad (13).$$

Parametra η in ρ pomenita viskoznost in gostoto vode. Pri pretočni hitrosti 0,5 m/s je kritična velikost delovnega preseka znotraj naprave MOV 2 cm in pri 2 m/s pa 0,5 cm.

Iz pulzacijske teorije [11] smo za izračun pulzacijske dolžine (b) in pulzacije delcev(v_b) s polmerom (a) izpeljali sistem enačb:

The parameter η is the viscosity and ρ is the mass density of water. For a water flow of velocity 0.5 m/s, the critical thickness of the working channel (d) is 2 cm, and for 2 m/s, the critical thickness is 0.5 cm.

From the turbulent pulsations theory [11], the equation system was obtained for the evaluation of the pulsation length (b) and the pulsation for a particle with radius a (v_b).

$$b = 207 d \frac{\log \mathrm{Re}/7}{\mathrm{Re}^{7/4}} \qquad (14)$$

$$v_b = 0{,}17 v \left(\frac{b}{a}\right)^{1/3} \mathrm{Re}^{1/4} \qquad (15)$$

Stabilni kosmič z $10k_BT$ vezno energijo med delci (k_B je Boltzmannova konstanta) lahko razbijemo s turbulentno pulzacijo samo, če je gostota kinetične energije $\rho v_b^2/2$ večja od gostote vezne energije $10k_BT/(4\pi a^3/3)$. Določimo lahko t.i. kritični polmer kristalnega delca (a^*):

A stable flock with a $10k_BT$ bonding energy between constituent particles (k_B is the Boltzmann constant) would be shattered by turbulent pulsation, if the kinetic energy density $\rho v_b^2/2$ were greater than the bonding energy density $10k_BT/(4\pi a^3/3)$. A crystal particle radius is therefore:

$$a^* = \sqrt[3]{15 \mathrm{k_B} T / \pi \rho v_b^2} \tag{16}$$

Tako je pri $v = 0{,}5$ m/s kritični polmer 0,25 μm in 0,13 μm pri $v = 2$ m/s. Povzamemo lahko, da bo za priporočene pretočne hitrosti proizvajalcev naprav MOV turbulenca razbila $CaCO_3$ in $CaSO_4$ kosmiče. V suspendirani obliki bodo ostali le najbolj močno vezani agregati.

V primerjavi s kemijskimi metodami priprave vode za nadzor vodnega kamna je magnetna obdelava še najbolj podobna suspendiranju kristalnega prahu.

Naslednji izračuni določajo potrebno količino prahu za preprečevanje izločanja $CaCO_3$ na stenah prenosnikov toplote z relativno površino $S_{HE} = S_{stene}/V_{vode}$, kjer sta S_{stene} površina sten in V_{vode} prostornina vode.

Da bi se zagotovil hiter prenos toplote, so v skladu z enačbo (17) [12] priporočane visoke vrednosti S_{HE}, in sicer med 100 in 1000 /m.

The critical radius a* is 0.25 μm for a water flow of velocity $v = 0.5$ m/s and a* is 0.13 μm for $v = 2$ m/s. A theoretical conclusion could be made for all recommended ranges of water flow velocity that turbulence will deaggregate $CaCO_3$ and $CaSO_4$ flocks. Only highly adhered aggregates will remain in a suspended form.

In a comparison with chemical scale-prevention methods, the suspending of crystal powder is the most similar to the MWT method.

The following calculation estimates that the necessary amount of powder for the prevention of $CaCO_3$ precipitation on the walls of a HE with a relative surface: $S_{HE} = S_{wall}/V_{water}$, where S_{wall} is the area and V_{water} is the water volume.

To ensure a quick heat exchange, high values of S_{SE} from 100 to 1000 /m are recommended according to equation (17) [12].

$$\frac{dT}{dt} = \frac{\alpha}{c_p \rho} \Delta T S_{HE} \tag{17}$$

V enačbi (18) je iz kvocienta ξ oborjene mase v jedru vode (dm_v) in mase na stenah (dm_s) razvidno, da se bo vodni kamen nalagal v tanjših oblogah pri nižjih vrednostih S_{HE}.

In this relationship for the heating rate dT/dt, the parameter c_p is the heat capacity of water. A thinner scale lining will be formed at lower values S_{HE}, as can be predicted from the ξ-quotient (of precipitated mass in the bulk of water-dm_v and precipitated mass on the walls - dm_s) in equation (18). So, the optimal value S_{HE} in HE designing should be found.

$$\xi = \frac{dm_v}{dm_s} = \frac{r_v V_{water}}{r_s S_{wall}} = \frac{r_v}{r_s S_{HE}} \tag{18}$$

V modificirani obliki sta enačbe za hitrost kristalne rasti (r) določila Nancollas in Reddy [13], in sicer na podlagi obarjanja iz jedra raztopine s temperaturo T_1 na površino naprav s temperaturo T_2 ((19) in (20)). Pri tem velja, da je parameter k določen empirično, M_{CaCO_3} je relativna molska masa kalcijevega karbonata in R splošna plinska konstanta.

The relationship of crystal growth rate r has been determined by Nancollas and Reddy [13] and is represented by equations (19) and (20) in a modified form for precipitation in the bulk of a solution with temperature T_1 and on the equipment walls with temperature T_2, where k is an empirical parameter, M_{CaCO_3} is the relative molecular mass of $CaCO_3$ and R is the universal gas constant.

$$r_v = k M_{CaCO_3} \exp\left(\frac{-\Delta G_1}{R T_1}\right) S_{powder} \beta_1 \tag{19}$$

$$r_s S_{HE} = k M_{CaCO_3} \exp\left(\frac{-\Delta G_2}{R T_2}\right) S_{wall} \beta_2 \tag{20}$$

$$\beta = c_{Ca^{2+}} \cdot c_{CO_3^{2-}} - K_s \tag{21}$$

Kristalna rast je odvisna od sestave raztopine in trdnine:

– stopnje prenasičenja β, ki je ob stenah prenosnikov toplote (β_2) višje kakor v jedru raztopine (β_1), in od
– aktivacijske energije ΔG, ki je odvisna od kristalne faze.

V primeru obarjanja $CaCO_3$ sta kristalni fazi kalcit in aragonit. V suspendiranem prahu, nastalem z magnetno obdelavo, je opažen povečan delež

The crystal growth rate depends on solution and solid phase composition by:

– β supersaturation degree (defined by 21), which is higher at the HE walls (β_2) than in the bulk of solution (β_1);
– ΔG activation energy depending on crystal phase.

In the case of $CaCO_3$ precipitation, crystal phases aragonite and calcite are formed. A powder, formed from magnetically treated water, has an increased aragonita. Za hipotetični primer vzamemo vrednost ΔG_1 za aragonit in vrednost ΔG_2 za kalcit.

Z zamenjavo r_v in $r_s S_{HE}$ v enačbi (18) z izrazoma (19) in (20) dobimo zvezo (22) za količino prahu, ki je potrebna za učinkovit nadzor $CaCO_3$ oblog:

fraction of aragonite. In an ideal case a ΔG_1 value could be taken for aragonite and a ΔG_2 for calcite.

With the substitution r_v and $r_s S_{HE}$ in (18) by (19) and (20), equation (22) is obtained and a necessary powder surface is estimated:

$$\frac{S_{\text{powder}}}{S_{\text{wall}}} = \xi \left(\frac{\beta_1}{\beta_2} \right) \exp \left(\frac{\Delta G_1}{\mathrm{R} T_1} - \frac{\Delta G_2}{\mathrm{R} T_2} \right) \tag{22}$$

Da bi se učinkovito preprečile obloge trdega vodnega kamna za temperaturno območje 40 do 100°C in za zahtevano učinkovitost, je potrebna količina prahu S_{prah} istega reda, kakor je površina sten prenosnikov toplote S_{stene}

For efficiency request $\xi \approx \beta_1/\beta_2$ and operational temperatures between 40°C and 100°C, the necessary powder surface S_{powder} should be of the same order as the surfaces of the heat exchanger walls S_{wall} to effectively prevent hard scale.

4 SKLEP

Za nadzor vodnega kamna na stenah prenosnikov toplote je potrebna optimizacija velikosti površine za prenos toplote glede na obratovalne razmere. Zraven kemičnih postopkov za zmanjšanje koncentracije Ca^{2+} ionov se priporoča uporaba naprav MOV.

Dobro načrtovana naprava MOV, ki zagotavlja zadostno količino suspendiranih delcev v obliki praška, lahko učinkovito prepreči nastanek kotlovca. Problem učinkovitega načrtovanja naprav MOV je nezadostno poznavanje samega mehanizma delovanja teh naprav. Mehanizem je zapleten in je neposredno odvisen tudi od obratovalnih razmer in sestave napajalne vode. Na srečo so na temelju empiričnih izkušenj izdelali lepo število učinkovitih naprav MOV. S tem zadovoljujejo veliko povpraševanje po tej preprosti in cenovno ugodni rešitvi za preprečevanje nastanka vodnega kamna.

4 CONCLUSION

For scale control in HEs an optimization of the heat-exchange surface area is recommended for simultaneous high heat transition and scale prevention. In addition, besides chemical methods for the reduction of the Ca^{2+} concentration, the alternative method of MWT is recommended.

A well-designed MWT device which assures the formation of a suspended scale powder with a surface area comparable to the exchange surface area can effectively prevent hard-scale formation. The problem with designing MWT devices is an insufficient theoretical understanding of the MWT mechanism. The mechanism is complex and depends directly on operational conditions and the composition of the supplied water as a solution/dispersion system. Fortunately, numerous MWT devices of different constructions have been designed on an empirical basis resulting from several decades of testing and are available to satisfy a large demand for such easy and cheap solutions to industrial scale problems.

5 OZNAKE
5 SYMBOLS

premer delca	a	m	particle radius
dolžina pulza	b	m	pulsation length
koncentracija	c	mol/L	concentration
specifična toplota	c_p	J/kgK	heat capacity
premer cevi	d	m	thickness of working channel
aktivacijska energija	ΔG	J/mol	activation energy
prva konstanta ravnotežja pri disociaciji H_2CO_3	K_1	ml/L	equilibrium constant of the first step of H_2CO_3 dissociation
druga konstanta ravnotežja pri disociaciji H_2CO_3	K_2	ml/L	equilibrium constant of the second step of H_2CO_3 dissociation
plinska konstanta ravnotežja	K_g	mol m²/NL	gas equilibrium constant
topnostni produkt	K_s	mol²/L²	soluble product
ionski produkt vode	K_w	mol²/L²	dissociation product of water
empirična konstanta hitrosti kristalne rasti	k	1/mol m³s	empirical constant of crystal growth rate
Boltzmannova konstanta	k_B	J/K	Boltzman constant
molska masa	M	kg/mol	molar mass
masa	m	kg	mass
moč toplotnega toka	P	J/s	heat flow intensity
tlak	p	N/m²	gas pressure

splošna plinska konstanta	R	J/molK	universal gas constant
Reynoldsovo število	Re	-	Reynolds number
hitrost kristalne rasti s raztopini	r_v	kg/m³s	crystal growth rate in bulk of water
hitrost kristalne rasti na kovinskih stenah	r_s	kg/m²s	crystal growth rate on walls
površina	S	m²	surface area
absolutna temperatura	T	K	temperature
čas	t	s	time
hitrost pretoka	v	m/s	flow velocity
hitrost turbulentne pulzacije	v_b	m/s	turbulent pulsation velocity
debelina sloja vodnega kamna	Δy	m	scale thickness
koeficient toplotne prehodnosti	α	J/m²sK	heat transition coefficient
stopnja prenasičenja	β	mol²/L²	supersaturation degree
viskoznost	η	Ns/m²	water viscosity
koeficient toplotne prevodnosti	λ	J/msK	heat conductivity
gostota snovi	ρ	kg/m³	mass density
učinkovitost nadzora vodnega kamna	ξ	-	scale control efficiency
učinkovitost prenosa toplote	ζ	-	heat exchange efficiency

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6 REFERENCES

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Prejeto:
Received: 15.8.2000

Sprejeto:
Accepted: 10.11.2000

© Strojniški vestnik 46(2000)8,517-524
ISSN 0039-2480
UDK 621.574.013:620.92
Pregledni znanstveni članek (1.02)
© Journal of Mechanical Engineering 46(2000)8,517-524
ISSN 0039-2480
UDC 621.574.013:620.92
Review scientific paper (1.02)

Analiza eksergijskih tokov absorpcijske hladilne naprave

An Analysis of Exergy Flows in an Absorption Chiller

Vasilije Vasić - Jurij Krope - Darko Goričanec

V prispevku je podan postopek analize eksergijskih in anergijskih tokov v enostopenjski absorpcijski hladilni napravi z delovnim medijem LiBr/H_2O ter postopek izračuna eksergijskega izkoristka v odvisnosti od stopnje uporabe naprave.

Nakazana je prednost absorpcijske hladilne naprave pred kompresijsko hladilno napravo, ki se izkazuje v možnosti uporabe eksergijsko revne odpadne toplote in v rabi alternativnih energetskih virov.

(Ključne besede: naprave hladilne, naprave absorpcijske, tok energijski, izkoristek eksergijski)

This paper presents our analysis of the exergy and anergy flows and exergy efficiency of an absorption chiller which has been calculated for all areas of the device's exploitation. This analysis has been implemented on a single-stage absorption chiller with LiBr/H_2O as a working media.

We wish to show the advantages of sorption chillers, in our case absorption chillers, in comparison to compressor chillers in utilizing low exergy, rejected heat and in the use of alternative energy sources.

(Keywords: thermodynamics, absorption chiller, exergy flow, efficiency)

0 UVOD

Potreba po hlajenju postaja v sodobnem svetu vedno večja zaradi večjih zahtev po bivalnem ugodju, kar pa je povezano z zahtevnejšimi tehnološkimi procesi. Hlad tako postaja enakovreden produkt električni energiji in toploti.

Hlad se pridobiva najpogosteje s kompresorskimi hladilnimi napravami, katerim postajajo vse bolj konkurenčne absorpcijske hladilne naprave in namesto mehanske (električne) energije uporablja toploto.

Absorpcijska hladilna naprava (sl. 1) je sestavljena iz dveh obtokov – gretja in hlajenja, ki sta med seboj povezana. Posebnost naprave je toplotni kompresor, medtem ko so elementi hladilnega obtoka enaki kakor pri kompresorski hladilni napravi.

Delovne snovi absorpcijske hladilne naprave so okolju precej bolj prijazne od tistih pri kompresijski hladilni napravi. Najpogosteje se uporabljata delovni snovi – binarna zmes LiBr/H_2O ali H_2O/NH_3.

0 INTRODUCTION

The demand for cooling is growing as peoples' expectations of a more comfortable life, surrounded by technology continues to increase. Cooling is becoming a product equivalent to electricity and heat.

The cooling process is most frequently carried out compressor chillers, of which absorption chillers are becoming increasingly significant. In contrast to the compressor chillers, absorption chillers are driven by heat rather than electricity.

An absorption chiller (Figure 1) works using two interconnected cycles - heating and cooling. The device's speciality is its thermal compressor, while the cooling cycle is undertaken using the same components found in compressor chillers.

The working media of absorption chillers are environmentaly friendly compared to those used by compressor chillers because the most frequently used working media for absorption chillers are binary mixtures of LiBr/H_2O and H_2O/NH_3.

1 DELOVANJE NAPRAVE

Delovanje absorpcijske hladilne naprave poteka, prikazano posplošeno, na dveh tlačnih nivojih, treh temperaturnih nivojih in treh nivojih koncentracije hladiva [1].

Generatorju in uparjalniku se toplota dovaja na najvišjem oz. najnižjem temperaturnem nivoju, medtem ko se okolici toplota predaja na srednjem temperaturnem nivoju iz kondenzatorja in absorberja (sl. 1).

Vezava zunanjega obtoka kondenzatorja in absorberja je lahko ločena (vzporedna vezava) ali pa povezana (serijska vezava). Učinkovitejši način je serijska vezava [1], ki je uporabljena v primeru analize eksergijskih tokov.

Izhodišče vsake termodinamične analize procesa ali postroja je energijska bilanca (prvi glavni zakon termodinamike). V apsorbcijski hladilni napravi

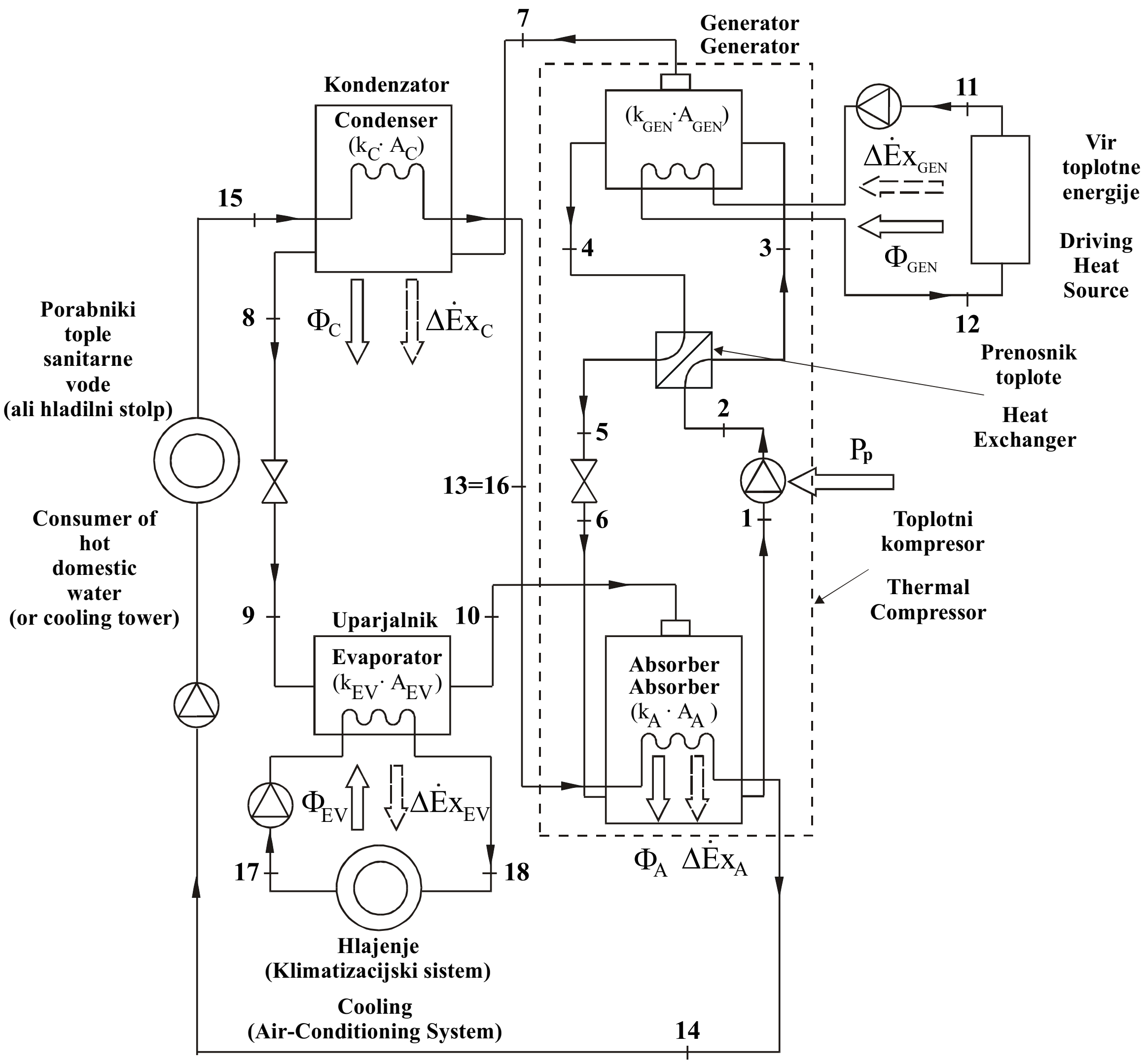
1 THE PERFORMANCE OF THE ABSORPTION CHILLER

The functioning of an absorption chiller can be described simply as a process between two pressure levels, three temperature levels and three levels of coolant concentration [1].

Heat is provided to the generator and evaporator at either the highest or lowest temperature levels, respectively. In the mean time, heat is delivered from the absorber and the condenser to the surroundings at the medium temperature level (Figure 1).

The external-flow connection to the condenser and the absorber can be either independent (parallel flow) or connected (serial flow). Serial connection has proved to be the more efficient method [1] and has been adopted when presenting our analysis of exergy flows.

At the beginning of each thermodynamic analysis of thermal processes or a thermal plant we use the energy balance (First Law of Thermodynam-



Sl. 1. *Shema absorpcijske hladilne naprave*
Fig. 1. *Model of the absorption chiller*

imamo različne oblike energij (npr. toplotna, električna), katerih vrednost se kaže v stopnji zmožnosti za pretvarjanje v druge oblike energij. Popolnoma spremenljiv del energije v druge oblike energij se imenuje eksergija, nespremenljiv del pa anergija.

Vse energije preračunamo na enako osnovo – eksergijo. Analiza eksergijskih tokov, ki temelji na drugem glavnem zakonu termodinamike, ima nalogo določiti: mesta nepovračljivosti, velikost nepovračljivosti, smer odvijanja procesov in celotno učinkovitost naprave.

Pri analizi eksergijskih tokov se ne omejimo samo na analizo popolnosti procesov v napravi, temveč tudi na procese s toplotno menjavo delovne snovi s toplotnimi prejemniki in medsebojnimi vplivi na okolje [2].

Velikost nepovračljivosti – anergijskih tokov je praviloma odvisna od vrste opreme, delovne snovi in pogojev obratovanja.

Osnovni vzroki nepovračljivosti energijskih procesov v absorpcijski hladilni napravi so [3]:

- ohlajanje pare s temperature v generatorju na temperaturo kondenzacije,
- dušenje hladilnega sredstva z dušilnim ventilom s tlaka kondenzacije na tlak uparjanja,
- segrevanje pare z uparjalne temperature na temperaturo absorpcije,
- prenos toplote v prenosniku toplote termičnega kompresorja,
- segrevanje s hladivom bogate raztopine,
- dušenje s hladivom revne raztopine za prenosnikom toplote in
- prenos toplote na zunanje nosilce toplote v zunanjih obtokih.

Kljub želji po zmanjšanju ali celo preprečitvi nastanka nepovračljivosti procesov imamo v dejanski absorpcijski hladilni napravi na nekatere nepovračljivosti zelo malo ali povsem nobenega vpliva.

ics). In the absorption chiller different forms of energy occur (e.g. heat, electricity), their values are reflected through their ability to convert to other forms of energy. The fully convertible part of energy as another form of energy is called exergy (availability) and the unconvertible is called anergy.

We therefore calculate all energy forms on the same basis – exergy. The analysis of exergy flows, according to the Second Law of Thermodynamics, is the a task of determining: places of irreversibility, the direction of occurred processes and the effectiveness of the whole device.

When performing the analysis of exergy flows we did not only analyze the perfection of the processes in the device, but we also focussed on the processes which consider heat interaction with the working media and the heat recipients as well as the device's mutual interaction with the environment [2].

How big are the irreversibilities – anergy flows usually depend on the type of equipment, the working fluid and the operating conditions.

The basic causes of irreversibilities of the energy processes in absorption chillers are [3]:

- steam cooling from the temperature in the generator at the condensing temperature;
- throttling the cooling media with a throttle valve from the condensing pressure to the evaporation pressure;
- warming up the vapor from the evaporation temperature up to the absorption temperature;
- warming up with coolant-rich solution;
- throttling with coolant-poor solution behind the solution heat exchanger;
- heat exchange on the working media in external circulation loops.

Despite the desire for a reduction of or even prevention of irreversibilities occurring in the process, in the real absorption chiller there is avery small or no influence on the irreversibilities.

2 PRERAČUN EKSERGIJSKIH TOKOV

Preračun eksergijskih tokov je izveden na podlagi podatkov s preglednice 1 in literature ([1] in [4]) z določitvijo specifične eksergije snovnega toka, v karakterističnih točkah naprave (sl. 1 in 2).

Specifična eksergija snovnega toka pove, koliko dela pridobi enota masnega toka pri povračljivem medsebojnem delovanju z okoljem [2].

Specifična eksergija snovnega toka v karakterističnih točkah naprave v hladilnem in zunanjem tokokrogu določimo z enačbo:

2 CALCULATION OF EXERGY FLOWS

The calculation of exergy flows is performed using the data from Table 1 and any available literature ([1] and [4]) with a determination of specific flow exergy in labelled positions of the chiller (Fig. 1 and 2).

The specific flow exergy tells us how much work has been produced with the unit of mass flow by reversible and mutual interaction with the environment [2].

The specific flow exergy in the device's labelled positions, in the cooling and heating cycle is determined by the equation:

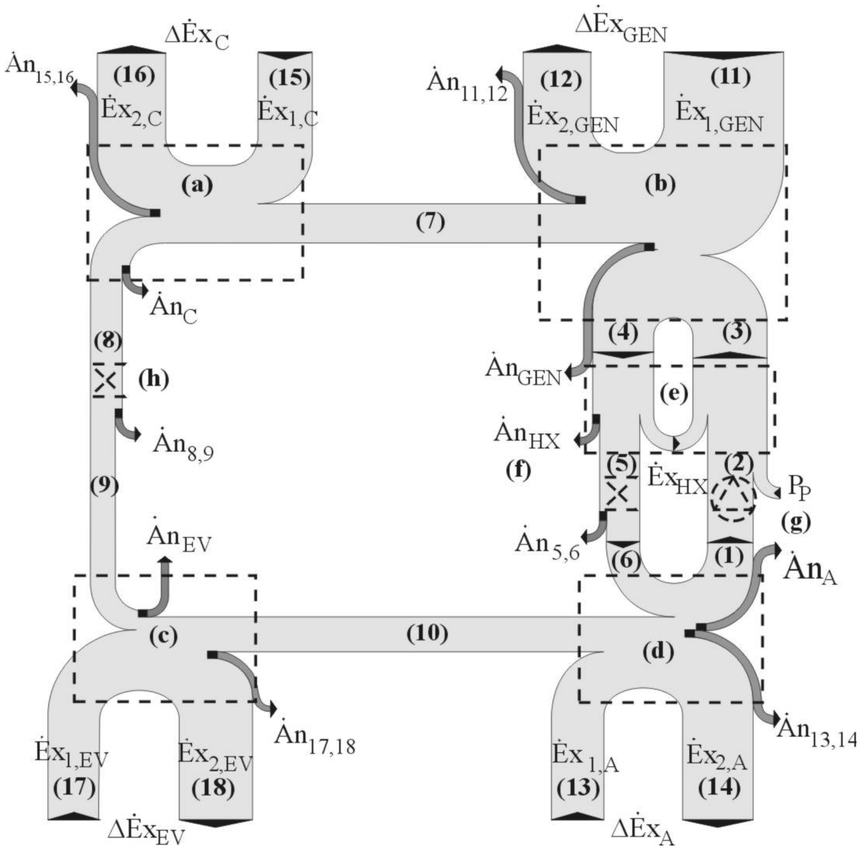
$$e_x = h - h_o - T_o \cdot (s - s_o) \quad (1).$$

Za določitev vrednosti specifične eksergije v toplotnem kompresorju običajno uporabimo eksergijski diagram na sliki 3.

Usually we use the exergy diagram (Fig. 3) to determine values of specific flow exergy in the thermal compressor.

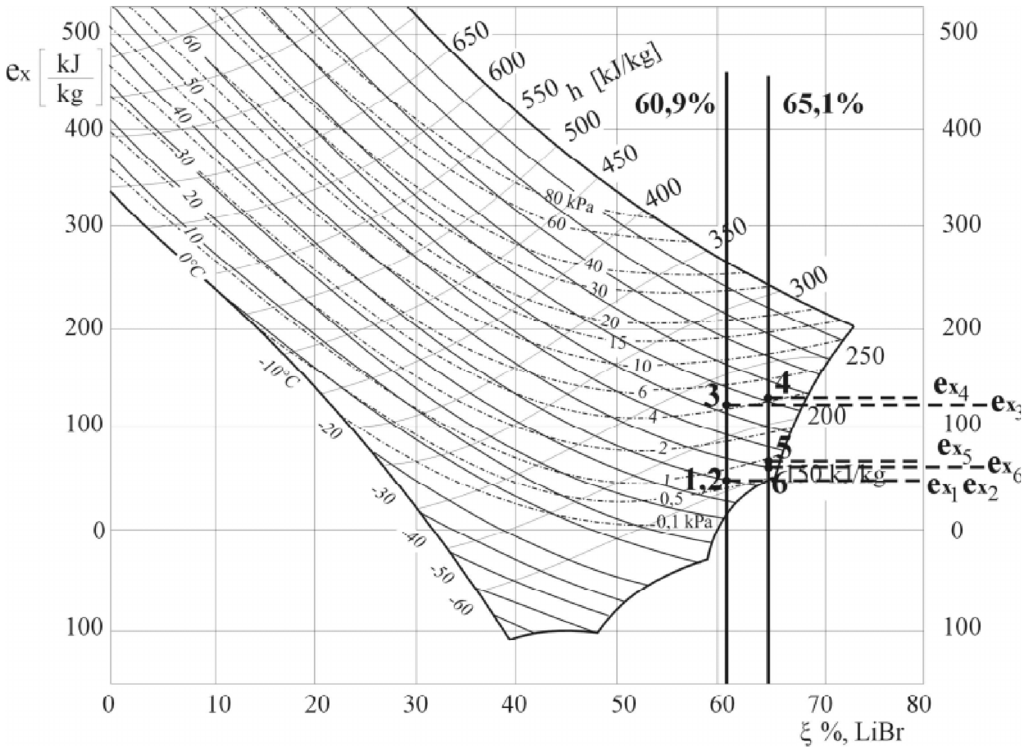
Preglednica 1. *Osnovni podatki absorpcijske hladilne naprave*
Table 1. *Basic data for the absorption chiller*

Osnovni podatki prenosnikov toplote **Basic data for the heat exchangers**			
$k{\cdot}A_{EV}$ =	11,9 kW/K	$k{\cdot}A_{HX}$ =	2,0 kW/K
$k{\cdot}A_{C}$ =	17,9 kW/K	$k{\cdot}A_{A}$ =	6,1 kW/K
$k{\cdot}A_{Gen}$ =	8,5 kW/K		
masni pretok zunanjega obtoka **mass flow in the external circuits:**			
kondenzator in absorber **condenser and absorber**	$q_{m,k} = q_{m,a}$ = 4,2 kg/s		
uparjalnik **evaporator**	$q_{m,up}$ = 2,3 kg/s		
generator **generator**	$q_{m,gen}$ = 3,2 kg/s		
koncentrirana raztopina **with coolant-rich solution**	$q_{m,rr} = q_{m,1}$ = 0,45 kg/s		



(a)	Kondenzator	(a)	Condenser
(b)	Generator	(b)	Generator
(c)	Uparjalnik	(c)	Evaporator
(d)	Absorber	(d)	Absorber
(e)	Izmenjevalnik toplote	(e)	Solution heat exchanger
(f)	Dušilni ventil 2	(f)	Throttle valve 2
(g)	Črpalka raztopine	(g)	Solution pump
(h)	Dušilni ventil 1	(h)	Throttle valve 1

Sl. 2. *Diagram pretoka eksergij v absorpcijski hladilni napravi*
Fig. 2. *Exergy flow diagram in the absorption chiller*



Sl. 3. *Diagram e_x,ξ delovne snovi LiBr/H_2O [5]*
Fig. 3. *e_x,ξ diagram for working media LiBr/H_2O [5]*

Negativni predznak specifične eksergije dobimo v primeru, ko se povečuje entropija toka snovi pri hkratnem padcu tlaka pod tlak okolice [2]. To se pojavi, v analiziranem primeru, za stanja v točki 9 in 10 (slika 1). Rezultati preračuna eksergijskih tokov absorpcijske hladilne naprave so podani v preglednicah 2 in 3, kjer oznake posameznih veličin ustrezajo oznakam na sliki 1 in diagramu eksergijskih tokov na sliki 2.

A negative sign for specific flow exergy was obtained for the case where the entropy of flow increased with a simultaneous pressure drop below the value of the environment's pressure [2]. This occured, in our case, for positions 9 and 10 (Fig. 1). The calculated results of exergy flows in the absorption chiller are given in Table 1 and Table 2, where the thermodynamic properties correspond to those named in Figure 1 and exergy flow diagram in Figure 2.

Preglednica 2. *Vrednosti specifične eksergije, masnega pretoka in temperature v posameznih točkah naprave*
Table 2. *Values of mass flow, temperature and specific flow exergy in the labelled position of the absorption chiller*

	Masni pretok Mass flow	Temperatura Temperature	Specifična eksergija Specific exergy		Masni pretok Mass flow	Temperatura Temperature	Specifična eksergija Specific exergy
	$q_{m,i}$ kg/s	t °C	$e_{x,i}$ kJ/kg		$q_{m,i}$ kg/s	t °C	$e_{x,i}$ kJ/kg
1	0,4500	41,9	48,3000	**10**	0,0287	2,1	-155,1000
2	0,4500	41,9	48,3000	**11**	3,2000	95,0	34,6500
3	0,4500	71,1	120,7000	**12**	3,2000	88,3	29,1200
4	0,4216	85,7	131,0500	**13**	4,2000	29,3	0,5079
5	0,4216	52,5	69,0000	**14**	4,2000	34,5	1,4550
6	0,4216	50,3	62,1000	**15**	4,2000	25,0	0,0737
7	0,0287	76,2	102,6000	**16**	4,2000	29,3	0,5079
8	0,0287	31,5	0,8369	**17**	2,3000	12,0	0,4333
9	0,0287	2,1	- 5,1740	**18**	2,3000	5,0	1,7700

Preglednica 3. *Vrednosti eksergijskih in anergijskih tokov enostopenjske absorpcijske hladilne naprave*
Table 3. *Values of exergy and anergy flows in the single-stage absorption chiller*

Sestavina naprave Component of device	Sestavina Component		Zunanji obtok External loop	
	Eksergijski tok Exergy flow	Anergijski tok Anergy flow	Eksergijski tok Exergy flow	Anergijski tok Anergy flow
	$\dot{E}x_i$ kW	$\dot{A}n$ kW	$\Delta\dot{E}x_i$ kW	$\Delta\dot{A}n$ kW
kondenzator condenser	3,1200	-68,47	1,54	69,73
absorber absorber	67,7200	-85,87	3,59	82,98
prenosnik toplote heat exchanger	6,6100	6,77	-	-
generator generator	13,5700	75,48	-18,4	75,48
uparjalnik evaporator	3,7900	71,41	3,05	71,41
dušilni ventil – 1 throttle valve – 1	-	0,11	-	-
dušilni ventil – 2 throttle valve – 2	-	0,85	-	-
črpalka raztopine solution pump	0,001033			

2.1 Razlaga rezultatov

Splošno je znano, da se pri termodinamični analizi toplotnih procesov odločamo za eksergijsko bolj varčne naprave.

Med naštetimi vrstami nepovračljivosti, ki se pojavljajo v napravi (sl. 2), se izkaže, da se največje nepovračljivosti pojavljajo v generatorju in absorberju (pregl. 3), kjer se namreč poleg nepovračljivosti pri prenosu toplote pojavljajo še nepovračljivosti zaradi mešanja delovne snovi.

Na sliki 1 opažamo nasprotno smer toplotnega in eksergijskega toka v uparjalniku. Ta pojav je specifičen v tem, da se telesom, katerih temperatura je pod temperaturo okolice in se jim toplota odvaja, vrednost eksergije veča in nasprotno [6]. Prav zaradi tega se lahko utemeljeno sklepa, da lahko termodinamično pravilno zapišemo izkoristek vložene energije v napravo, v kateri hkrati potekata gretje in hlajenje, samo z eksergijskim izkoristkom.

Za absorpcijsko hladilno napravo, pri kateri izkoriščamo samo hlad, lahko določimo eksergijski izkoristek z enačbo:

2.1 Comments on the results

It is a well-known fact, that with a thermodynamic analysis of thermal processes we are concentrating on the more exergy-saving type of plants.

The named forms of irreversibilities, which occurred in the device – Figure 2, become obvious according to the presented analysis, because the greatest irreversibilities occured in the generator and absorber (Table 3). Besides the irreversibilities in the heat transfer, irreversibilities due to the mixture processes were also present.

In Figure 1 we can see the opposite directions of energy and exergy flows by evaporator. This phenomenon is specific for the systems (bodies) whose temperature are below the surrounding's temperature and whilst they are rejecting heat (being cooled), their exergy value is growing and vice versa [6]. Therefore, this brings us to the using conclusion that thermodynamics is the only correct way to express the efficiency of consumed energy in the device where heating and cooling are simultaneously performed only with exergy efficiency.

For the absorption chiller, where we utilize only cooling, we can determine the exergy efficiency using the following equation:

$$\psi_{AHN} = \frac{\Delta \dot{E}x_{UP}}{\Delta \dot{E}x_{GEN} + P_p} = \frac{3,05}{18,4} = 0,166 \qquad (2).$$

V primeru, da poleg hladu izkoriščamo še oddano toploto kondenzatorja in absorberja (npr. segrevanje sanitarne vode), lahko eksergijski izkoristek absorpcijske hladilne naprave določimo z enačbo:

In the case where we also utilize rejected heat from the absorber and the condenser, we can determine the exergy efficiency with the following equation:

$$\psi_{AHN} = \frac{\Delta \dot{E}x_{UP} + \Delta \dot{E}x_K + \Delta \dot{E}x_A}{\Delta \dot{E}x_{GEN} + P_p} = \frac{3,59 + 1,54 + 3,05}{18,4} = 0,445 \qquad (3).$$

Moč potrebne črpalke se zaradi doslednosti definicije v enačbah (2) in (3) zapiše, vendar se zaradi majhne vrednosti, (pregl. 3) v računu ne upošteva.

3 SKLEP

Prednost sorpcijskih hladilnih naprav, v obravnavanem primeru absorpcijskih hladilnih naprav, pred kompresorskimi hladilnimi napravami je v možnosti uporabe eksergijsko revne odpadne toplote in rabe alternativnih energijskih virov.

Kompresorske hladilne naprave so sicer energetsko učinkovitejše in manjše, vendar eksergijsko manj učinkovite in porabljajo čisto eksergijo [7].

Za pogon absorpcijskih hladilnih naprav se lahko uporablja tudi odpadna toplota postroja soproizvodnje. Postroj sočasne proizvodnje električne energije in toplote ter hladu imenujemo trigeneracijski postroj [8].

Pri trigeneracijskih postrojih je zaželeno, da ima absorpcijska hladilna naprava kar se da velik eksergijski izkoristek in s tem dosežemo večji eksergijski izkoristek celotnega trigeneracijskega postroja [8].

We have to consider the power of the pump to obtain the correct definitions, equation (2) and (3), which due to its small value, is unimportant.

3 CONCLUSION

The advantage of sorption chillers, in our case absorption chillers, in comparison to compressor chillers is in utilizing low exergy, rejected heat and in the use of alternative energy sources.

Compressor chillers have a higher energy efficiency and are more compact, but they are less exergy efficient and utilize pure exergy [7].

To drive the absorption chiller we can also utilize the rejected heat from the cogeneration plant. This kind of plant, with simultaneous production of heat, electricity and cooling is also called the trigeneration system [8].

In the trigeneration plant all the requirements are present, the absorption chiller has the highest possible exergy efficiency, which enables it to achieve higher exergy efficiency in the whole trigeneration plant [8].

4 OZNAKE
4 SYMBOLS

anergijski tok	$\dot{An}$	W	anergy flow
specifična eksergija snovnega toka	ex	J/kg	mass flow specific exergy
energijski tok	$\dot{Ex}$	W	exergy flow
specifična entalpija	H	J/kg	specific enthalpy
moč	P	W	power
masni pretok	q_m	kg/s	mass flow
temperatura	T	K	temperature
masni delež	ξ	%	mass ratio
eksergijski izkoristek	ψ	%	exergy efficiency
Indeksi:	**Subscripts:**		
absorber	A		absorber
absorbcijska hladilna naprava	AHN		absorption chiller
generator	Gen		generator
kondenzator	K		condenser
referenčno stanje okolice	o		reference state
črpalka	p		pump
prenosnik toplote	PT		heat exchanger
uparjalnik	UP		evaporator

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Prejeto:
Received: 15.8.2000

Sprejeto:
Accepted: 10.11.2000

© Strojniški vestnik 46(2000)8,525-531 © Journal of Mechanical Engineering 46(2000)8,525-531
ISSN 0039-2480 ISSN 0039-2480
UDK 532.5:621.644:697.4 UDC 532.5:621.644:697.4
Pregledni znanstveni članek (1.02) Review scientific paper (1.02)

Zmanjšanje tlačnih izgub v vročevodnih cevnih mrežah

The Reduction of Friction Losses in District-Heating Pipelines

Andrej Krope - Jurij Krope - Igor Tičar

V prispevku je predstavljen vpliv kationskih površinsko aktivnih dodatkov na zmanjšanje intenzivnosti turbulence v ceveh primarnih vročevodnih mrež sistemov za daljinsko ogrevanje. Že zelo majhne količine dodatkov vroči vodi, povzročijo znatno zmanjšanje odpora pri pretoku po ceveh in zato manjše izgube tlaka, kar vodi do manjše potrebne moči črpalk, znižanja črpalnih stroškov, povečanja zmogljivosti, zmanjšanja stroškov plina za ogrevanje vode in zmanjšanja toplotnih izgub. Pri načrtovanju in izgradnji novih vročevodnih mrež lahko uporabljamo cevi z manjšimi premeri in tako znatno znižamo investicijske stroške.

(Ključne besede: pretok fluida, izgube tlačne, dodatki, sistemi toplovodni)

In the paper the impact of cationic surfactant additives on reducing the turbulence intensity in the hot-water pipelines of district heating systems is presented. With small amounts of cationic surfactants in district-heating water the friction losses in pipelines can be reduced significantly. Because of this effect the pressure drops are decreased what leads to reductions in pump energy, pumping costs, costs of gas for heating the supply water, heat losses and to an increase in the heat capacity. New district-heating networks can be designed with smaller pipe diameters and so investment costs can be reduced significantly by applying friction-reducing additives.

(Keywords: fluid flow, pressure drops, additives, heating pipelines)

0 UVOD

Zmanjšanje zalog primarnih goriv in s tem v zvezi varčevanje z energijo terja na področju toplotne tehnike iskanje novih tehnično-znanstvenih spoznanj, kar je v zadnjem času pomembna tema številnih državnih in mednarodnih raziskovalnih projektov. Težišče raziskav temelji na boljšem izkoriščanju primarne energije.

Energetski sistemi za daljinsko ogrevanje zagotavljajo prihranke pri porabi primarne energije in ekološko sprejemljivo oskrbo s toplotno energijo [1]. Z ekonomskega vidika gre v primeru daljinskega ogrevanja za nasprotje med nizkimi stroški proizvodnje toplotne energije in relativno visokimi prenosnimi in razdeljevalnimi stroški. Večino stroškov, povezanih s sistemi daljinskega ogrevanja pomenijo naložbe v cevno mrežo ter stroški črpanja. Zaradi tega so sistemi daljinskega ogrevanja cenovno razmeroma ugodni le pri majhnih pretočnih razdaljah. Ena od možnosti za izboljšanje učinkovitosti in gospodarnosti takšnih sistemov je dodajanje snovi za

0 INTRODUCTION

Decrease of primary energy supplies and in this connection saving with energy demands investigations for new technical-scientific cognition in the field of heat engineering, what is an important topic of numerous national and international research projects particularly in recent years. The centre of this researches is more rational exploitation of primary energy.

District-heating systems ensure savings by consumption of primary energy and ecological heat-energy supply [1]. From the economic point of view there is a contradiction between low heat-generation costs and relatively high transport and distribution costs of district heat. The investments in pipelines and pumps, together with the pumping costs, form a major cost item of district-heating systems. For this reason the costs for such systems are relatively favourable only at short transport distances. One of the possibilities to improve effectiveness and economic viability of district-heating systems is applicazmanjšanje odpora pri pretoku tekočin. Na ta način lahko znatno zmanjšamo tlačne izgube, povečamo pretok in s tem prenos toplotne energije, kar vodi do občutnega znižanja naložb v cevi in črpalke ter znižanja stroškov električne energije, ki je potrebna za pogon črpalk. Zmanjšanje tlačnih izgub se kaže tudi v primernejši rabi primarne energije in nižji obremenitvi okolja.

Vpliv majhnih količin dodatkov vodi na odpor pri pretoku tekočin in padec tlaka v ravni cevi je že leta 1948 odkril Toms [2]. Od tedaj je bilo izvedenih že na stotine preskusov, ki so potrdili takratno odkritje. Značilnosti tega pojava so naslednje:

– pri pretoku vode v ceveh je mogoče z dodatkom raztopine dodatka koncentracije 5 ppm zmanjšati odpor za 70 %,
– večje znižanje odpora se pojavlja le pri turbulentnem toku,
– z dodatkom raztopine dodatka je mogoče povečati pretok za 30 %,
– raztopine dodatkov so učinkovitejše pri ceveh manjšega premera.

V preteklosti so za znižanje odpora uporabljali različne polimerne dodatke z veliko molekulsko maso, vendar so se ti izkazali za manj uporabne zaradi nepovračljive razgradnje, ki se pojavi pri velikih strižnih silah. Danes se za znižanje odpora v vročevodnih cevnih sistemih uporabljajo kationski površinsko aktivni dodatki majhnih molekulskih mas, ki povzročajo znižanje tlačnih izgub že v zelo majhnih koncentracijah in imajo povračljivo strukturo.

1 DELOVANJE KATIONSKIH POVRŠINSKO AKTIVNIH DODATKOV

Učinek zmanjšanja viskoznosti in s tem odpora pri pretoku tekočin, ki ga povzročajo vodne raztopine dodatkov, temelji na zmanjšanju intenzivnosti turbulence in ga lahko pojasnimo s tvorbo in oblikovanjem micelijev.

Površinsko aktivni dodatki so nizkomolekularne snovi z majhno kemijsko aktivnostjo in nizko topnostjo, ki so sestavljene iz hidrofilnega in hidrofobnega dela [3]. Kadar so takšne molekule v vodi ali v topilu, ki ima podobne lastnosti kakor voda, se pod določenimi pogoji združejejo v združbe, ki jih imenujemo miceliji. Miceliji so aglomerati nekaj sto molekul in lahko imajo različne oblike; lahko so okrogli, palični ali pa ploščati. Potrebni pogoj za zmanjšanje odpora so palični miceliji. Kritično micelarno koncentracijo, nad katero se molekule dodatka združujejo v micelije, prikazuje slika 1. Če je v vodni raztopini dodatka presežena koncentracija CMC_1, pride do tvorbe krogelnih micelijev s premerom približno dvakratne dolžine posamezne molekule. Ta koncentracija je le malo odvisna od temperature. Če

tion of friction-reducing additives in hot-water supply pipelines. In this way pressure drops can be significantly reduced and the flow rate can be increased enabling a reduction in the investment in pipelines and pumps and savings in the costs of electrical energy used for drive of pumps. A reduction of friction loss is also shown to be more rational consumption of primary energy and results in a impact on the environment.

As early as 1948 Toms [2] reported on friction loss and pressure drop when minute amounts of soluble polymer additive where added to water flowing through a straight pipeline. Since then, hundreds of experiments have confirmed his initial findings. The essential features of this phenomenon are as follows:

– the friction loss of water flow in pipelines can be reduced as much as 70% with additives in concentrations as low as 5 ppm,
– significant reductions in friction loss occur only for turbulent flow,
– additives can increase the flow rate by 30%,
– additive solutions are more effective on small pipelines than on large ones.

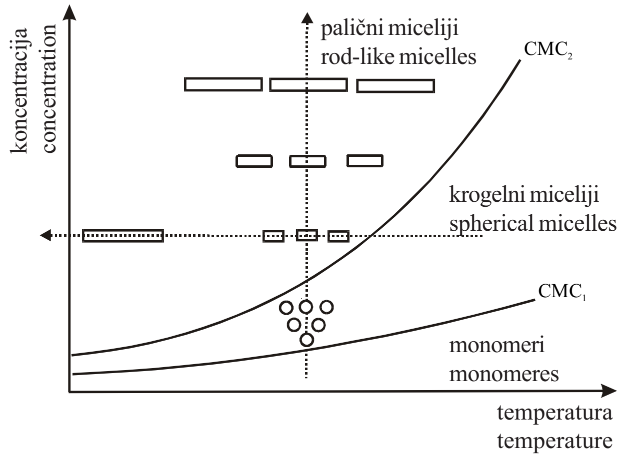
In the past a variety of polymer-based additives with high molecular weight have been used for reducing the friction losses, however, they have proved as less applicable because of their irreversible degradation which occurs at high values of shear stress. Nowadays, low molecular cationic surfactant additives, which effect on reduction of pressure drops already in small quantities and have reversible structure, are used for drag reduction in hot-water-pipe systems.

1 OPERATING PRINCIPLE OF SURFACTANTS

The phenomenon of friction reduction with surfactants in aqueous solutions is based on the decrease of the turbulence intensity and can be explained with the formation and the shape of micelles.

Surfactants are low-molecular-weight substances with low chemical activity and low solubility, but great interfacial activity. The molecules consist of a hydrophilic group and a hydrophobic part [3]. Under certain conditions the surfactant monomers form micelles in aqueous solutions. Micelles are clusters of approximately a hundred surfactant molecules and can take any of a variety of shapes, such as spheres, bars or disks. The presence of rod-like micelles is considered to be a necessary condition for the friction-reducing effect. Critical micelle concentrations, above which surfactant molecules form micelles, are shown in Figure 1. When the critical micelle concentration (CMC_1) in an aqueous solution is exceeded the surfactants form spherical micelles. This concentration is almost temperature independent. If the concentration is increased still further, the se koncentracija še naprej povečuje, se število molekul dodatka na micelij povečuje, dokler ni celotna prostornina micelija popolnoma izpolnjena z ogljikovimi verigami. Ko je presežena koncentracija CMC_2, tvorijo dodatki palične micelije, ker je takšna prostorska oblika energijsko ugodnejša. Dolžina paličnih micelijev se povečuje z naraščajočo koncentracijo. Kritična koncentacija CMC_2 je močno odvisna od temperature.

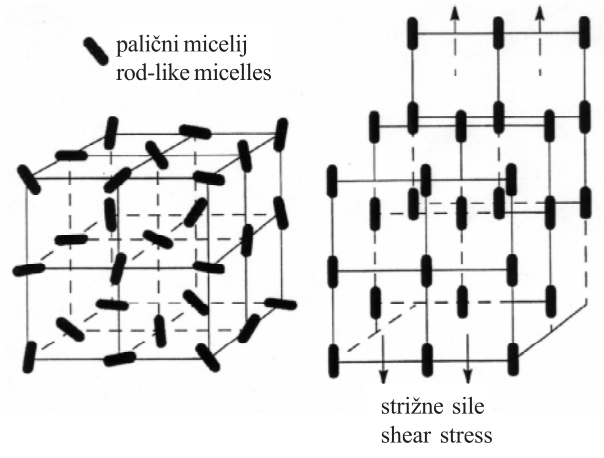
number of surfactant molecules per micelle will increase until the micelle volume is completely filled with carbon chains. When the second critical micelle concentration (CMC_2) is exceeded the surfactants form rod-like micelles, because these boundary faces are more energetically favourable. The length of bar-shaped micelles increases with increasing concentration. The CMC_2 concentration is strongly temperature dependent.



Sl. 1. *Kritična micelarna koncentracija [3]*
Fig. 1. *Critical micelle concentrations [3]*

Vodne raztopine dodatkov, ki tvorijo krogelne micelije se obnašajo podobno kakor voda, viskoznost takih raztopin je včasih celo večja od viskoznosti čiste vode in zato ne povzročajo učinka znižaja odpora pri pretoku tekočine. Pri koncentracijah, večjih od CMC_2, pa se v raztopini dodatka oblikujejo palični miceliji, ki kažejo viskoelastično obnašanje. Takše celice micelijev se zaradi turbulentnega toka in strižnih sil usmerjajo v smeri toka in tvorijo viskoelastično prostorsko mrežo, ki razširi prehodni sloj in zmanjša turbulentno jedro glavnega toka (sl. 2).

Aqueous surfactant solutions that form spherical micelles behave in the same way as the water, at high concentrations the viscosity becomes somewhat higher than that of the water, so this clusters do not perform friction-reducing effect. At concentrations that are higher than CMC_2, the surfactant solutions in which rod-like micelles have formed exhibit a favourable viscoelastic behaviour. Such cells become oriented by the pulse loads of turbulent flow and form a permanently oriented viscoelastic network which expands the buffer layer and reduces the layer of turbulent main-stream flow (Fig. 2).



Sl. 2. *Viskoelastična mreža in usmeritev paličnih micelijev zaradi delovanja strižnih sil [3]*
Fig. 2. *Viscoelastic network, orientation of micelles, shear induced structure [3]*

Pri koncentracijah okoli CMC_2 se oblikuje le nekaj relativno velikih micelijev, ki pa so le omejeno sposobni oblikovati usmerjene mreže, zato je njihov vpliv na znižanje odpora majhen. Za zadovoljivo znižanje je zato potrebna večja koncentracija, ki povzroči trajno usmerjene viskoelastične mreže, ki dušijo razvijanje turbulentnih vrtincev in tako povzročajo laminarni tok.

Funkcionalno odvisnost vodnih raztopin kationskih površinsko aktivnih dodatkov od Reynoldsovega števila prikazuje slika 3, na kateri opazimo 4 različna področja [4]:

⇒ Področje I: v laminarnem področju toka z majhnimi strižnimi silami ali brez njih oblikujejo palični miceliji prostorsko mrežo z elektrostatskim odbojem, ki je posledica njihovega površinskega naboja, in v njej zasedejo energetsko ugodna mesta. V tem stanju se raztopine površinsko aktivnih dodatkov obnašajo kot newtonske tekočine.

⇒ Področje II: povečanje strižnih sil in turbulentni tok vplivata na usmerjanje micelijev in oblikovanje viskoelastične mreže, kar povzroča laminaren tok. V tem stanju se raztopine površinsko aktivnih dodatkov obnašajo kot pseudoplastične tekočine.

⇒ Področje III: nadaljnje povečevanje strižnih sil vpliva na povečanje učinka zniževanja odpora. Miceliji so zmožni sprejeti več energije, ker deformiranje in raztezanje mreže povzroča sile, ki delujejo proti turbulentnemu vrtinčastemu gibanju in zato manjšajo oddajo energije. V tem področju je, ob uporabi dodatka, katerega delovno območje se ujema z obratovalnimi razmerami v sistemu daljinskega ogrevanja, učinek znižanja odpora neodvisen od koncentracije vodne raztopine dodatka. Raztopine dodatkov se tudi v tem področju obnašajo pseudoplastično, vrsto toka, ki se pojavi v takšnih razmerah pa imenujemo pseudolaminarni tok.

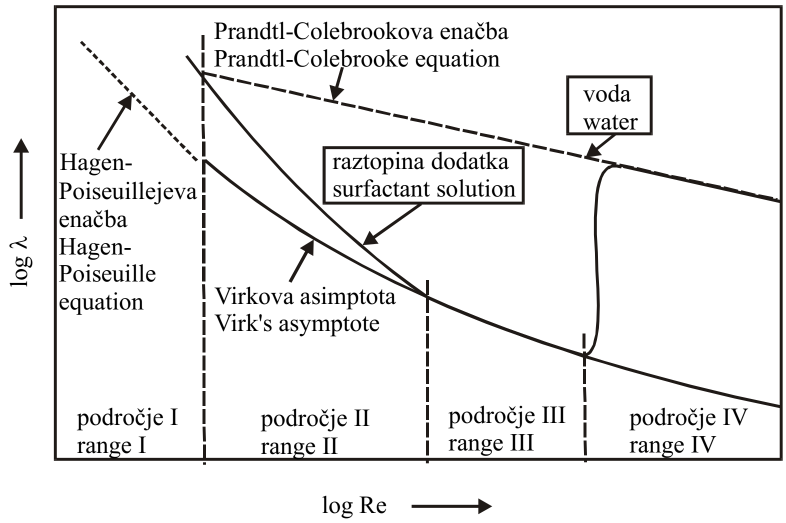
If the concentration is only just above CMC_2, then only few relatively large micelles will be formed. These micelles are not well capable of forming an oriented network, which is why their friction effect is only small. Therefore, for a significant reduction of friction losses a higher concentration is required. These concentrations generate permanently oriented viscoelastic networks which suppress the formation of turbulent whirls and produce a laminar flow in this way.

The functional relationsip between surfactant-solution behaviour and Reynolds number is shown in Figure 3, where we can see four different ranges [4]:

⇒ range I: In the laminar region of flow with little or no shear stress the rod-like micelles form a spatial network with the electrostatic repulsion caused by their surface charge, in which they occupy energetically favourable positions. In this state the surfactant solution shows Newtonian behaviour.

⇒ range II: A rise of shear stress and turbulent flow lead to orientation of the rod-like micelles and formation of the viscoelastic network what causes laminar flow. In this range the surfactant solution shows pseudoplastic flow behaviour.

⇒ range III: A further rise in shear stress leads to an increase in friction reduction. In this range the maximum reduction of friction losses appears. Micelles are able to incorporate more energy because deforming and stretching causes reset forces which act against the turbulent fluctuation movement and therefore reduce the energy dissipation. In this range pseudoplastic behaviour exist as well and this flow condition is known as pseudolaminar flow.



Sl. 3. *Darcyjev koeficient linijskih izgub v odvisnosti od Reynoldsovega števila za vodne raztopine dodatkov [4]*
Fig. 3. *Darcy's friction coefficient with its dependence on the Reynolds number for aqueous surfactant solution [4]*

⇒ Področje IV: zelo visoke strižne sile, ki jih povzroča povečana hitrost toka povzročijo razpad viskoelastične micelarne mreže in s tem konec vpliva dodatkov na znižanje odpora. Značilna krivulja raztopine dodatka doseže krivuljo vode. V tem področju nastane celovit turbulentni tok, raztopine površinsko aktivnih dodatkov se ponovno obnašajo kot newtonske tekočine.

Hkrati s pozitivnim učinkom znižanja odpora se pri uporabi dodatkov, zaradi spremenjenih pretočnih razmer (newtonsko obnašanje tekočine zamenja pseudoplastično) pojavijo tudi negativni učinki: to so zmanjšanje prenosa toplote, korozija in onesnaževanje okolja [1]. Problem zmanjšanega prenosa toplote rešujemo z modeliranjem vrste in lokacije prenosnikov toplote ter tako, da v menjalnike vstavljamo pregrade, ki povečujejo turbulenco in s tem prenos toplote. Novejše raziskave so pokazale, da kombinirani površinsko aktivni dodatki s tržnimi imeni Habon G / NaSal, Obon G / NaSal in Dobon G / NaSal ne kažejo nobenega vpliva na korozijo materialov, ki se uporabljajo v sistemih daljinskega ogrevanja. Problem strupenosti dodatkov in s tem povezanega onesnaževanja okolja rešujemo tako, da jih uporabljamo le v zaprtih sistemih s posredno povezavo obrata za proizvodnjo toplote in porabnikov prek toplotnih postaj in sekundarne mreže.

⇒ range IV: Very high shear rates finally affect the destruction of the viscoelastic micelle network so that the friction-reducing effect disappears and the characteristic surfactant solution curve approaches that of water. In this case a fully developed turbulent flow appears, which again shows Newtonian behaviour.

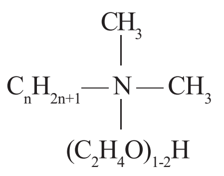
As well as the positive effects of drag reduction, negative effects due to the change in flow behaviour (pseudoplastic behaviour instead of Newtonian) [1] like heat-transfer reduction, corrosion and contamination of environment also occur. The phenomenon of radial turbulence and the associated reduction of heat transfer in heat exchangers can be solved by installing turbulence-increasing obstacles inside the heat exchangers to improve the heat-transmission properties. Some new investigations have shown that combined cationic surfactants with the trade names Habon G / NaSal, Obon G / NaSal and Dobon G / NaSal do not show any impact on the corrosion rates of materials which are built in district-heating systems. The problem of contamination and pollution of environment can be solved by only using surfactants in closed transport systems with an indirectly connected heat-generation plant and consumer systems. This can be achieved with the installation of heat-transmission stations and secondary hot-water-pipe network.

2 KOMBINIRANI KATIONSKI POVRŠINSKO AKTIVNI DODATKI

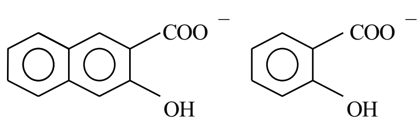
Najboljše rezultate dosežemo s površinsko aktivnimi dodatki Habon G, Obon G in Dobon G v kombinaciji s snovjo NaSal, ki zagotavlja širše temperaturno področje delovanja. Kemijsko strukturo omenjenih dodatkov, ki so uporabni pri koncentracijah do 1500 utežnih ppm in hitrostih toka do 4 m/s, prikazuje slika 4.

2 COMBINED CATIONIC SURFACTANTS

The best results by reduction of friction losses in hot-water pipelines can be achieved with the cationic surfactant substances Habon G, Obon G and Dobon G in combination with the additional counter-ion NaSal, which ensures extended temperature range of operation. The chemical structure of above-mentioned surfactants, which can be used by concentrations up to 1500 wppm and flow velocities up to 4 m/s is shown in Figure 4.



n-Alcyldimethylpolyoxethylammonium – Cation



3-hydroxy-2-naphthoate Salicylate

Sl. 4 *Kemijska struktura površinsko aktivnih dodatkov [3]*
Fig. 4. *Chemical structure of cationic surfactants [3]*

Temperaturno področje, v katerem omenjeni dodatki zagotavljajo znižanje odpora, je odvisno od števila ogljikovih atomov:
n = 16 (tržno ime Habon G / NaSal): od 25 do 105 °C
n = 18 (tržno ime Obon G / NaSal): od 35 do 120 °C
n = 22 (tržno ime Dobon G / NaSal): od 45 do 140 °C

The temperature range for which these surfactants show a friction-reduction effect depends on number of carbon atoms:
n = 16 (trade name Habon G / NaSal): from 25 to 105 °C
n = 18 (trade name Obon G / NaSal): from 35 to 120 °C
n = 22 (trade name Dobon G / NaSal): from 45 to 140 °C

3 DOLOČITEV EMPIRIČNE ENAČBE DARCYJEVEGA KOEFICIENTA TORNIH IZGUB

Dodatek kombiniranega dodatka Dobon G / NaSal v vročevodno cevno mrežo zniža odpor pri pretoku in povzroči zmanjšanje izgube tlaka. To znižanje odpora upoštevamo z enačbo, ki smo jo razvili v Laboratoriju za toplotno tehniko na Fakulteti za strojništvo Univerze v Mariboru [5]. Na temelju eksperimentalnih podatkov smo, z uporabo računalniškega programa Matlab in funkcije FMINS, ki izvaja Nelder-Meadov simpleks algoritem, določili odvisnost koeficienta tornih izgub (l) zaradi dodanega dodatka od Reynoldsovega števila (Re) v obliki potenče funkcije drugega reda z dvema linearnima in dvema nelinearnima koeficientoma v obliki (1):

3 DETERMINATION OF THE EMPIRICAL EQUATION OF DARCY'S FRICTION COEFFICIENT

The addition of the combined surfactant Dobon G / NaSal to hot-water-pipe network decreases the friction losses and reduce pressure drops in pipelines. This reduction of friction can be considered with the equation, which has been developed in our Laboratory for Heat Engineering at the University of Maribor, Faculty of Mechanical Engineering [5]. On the basis of experimental data and the help of The Matlab computer software with the FMINS function which performs Nelder-Meadov's simplex algorithm, the relationship between Darcy's friction coefficient (λ) and Reynolds number (*Re*) has been determined in form of a power function of the second grade with two linear and two non-linear coefficients (1):

$$\lambda = 0{,}17442 \cdot \mathrm{Re}^{-0{,}1989} - 0{,}00603 \cdot \mathrm{Re}^{0{,}001266} \quad (1).$$

4 SKLEP

Uporaba dodatkov v sistemih daljinskega ogrevanja zagotavlja izboljšanje učinkovitosti in gospodarnosti obratovanja. Učinek zmanjšanja viskoznosti in s tem odpora, ki ga povzročajo vodne raztopine kationskih površinsko aktivnih dodatkov temelji na zmanjšanju turbulence in ga lahko pojasnimo s tvorbo in oblikovanjem paličnih micelijev. Pri koncentracijah, večjih od kritične koncentracije CMC_2, se v raztopini dodatka oblikujejo palični miceliji, ki kažejo viskoelastično obnašanje. Takšne celice micelijev se zaradi turbulentnega toka in strižnih sil usmerjajo v smeri toka in oblikujejo viskoelastično prostorsko mrežo, ki razširi prehodni sloj in zmanjša turbulentno jedro glavnega toka.

Zmanjšanje odpora pri pretoku tekočin ima za posledico zmanjšanje tlačnih izgub in zato znižanje stroškov električne energije za pogon črpalk, povečanje kapacitete, manjše stroške plina za ogrevanje vode in manjše toplotne izgube. Znižanje odpora pa se kaže tudi v manjši potrebni moči črpalk in nižji vrednosti naložbe v cevi z manjšimi nazivnimi premeri pri gradnji novega omrežja [6], kar omogoča oskrbo s toploto tudi v primeru večjih pretočnih razdalj. Treba pa je poudariti tudi pozitiven vpliv uporabe dodatkov na smotrno rabo energije in zaradi tega na manjšo obremenitev okolja.

4 CONCLUSION

The use of surfactants in district-heating systems results in an improvement in the system's operation. The effect of friction reduction, which is a result of the surfactants added to hot-water supply, is based on reduction of turbulence intensity and can be explained by the formation of rod-like micelles. At concentrations higher than the critical micelle concentration (CMC_2) the surfactants form rod-like micelles which show viscoelastic behaviour. Such micelle cells become oriented and form viscoelastic network because of the turbulent flow and shear stress. This shear-induced state expands the buffer layer and reduces the layer of turbulent main-stream flow.

The reduction of friction losses and the resulting reduced pressure drops lower the electrical energy costs for pump driving, the gas cost for heating the supply water and the heat losses while increasing the heat capacity. The reduction of friction is also reflected in a decreased pump energy and lower investment costs for hot-water pipelines, as new networks can be designed with smaller pipe diameters [6], or the maximum economic transport length can be increased. Likewise the positive effect of surfactant application is shown in more rational consumption of energy and consecutively lower charge of environment, what has to be mentioned, too.

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Prejeto:
Received: 15.8.2000

Sprejeto:
Accepted: 10.11.2000

© Strojniški vestnik 46(2000)8,532-537
ISSN 0039-2480
UDK 546.212:628.16.04/.09:66.06:537.84
Pregledni znanstveni članek (1.02)
© Journal of Mechanical Engineering 46(2000)8,532-537
ISSN 0039-2480
UDC 546.212:628.16.04/.09:66.06:537.84
Review scientific paper (1.02)

Spremenjeno kristaljenje vodnega kamna pri magnetni obdelavi vode

Modified Scale Crystallization in Magnetic Water Treatment

Andrej Pristovnik - Lucija Črepinšek Lipuš - Jurij Krope

Magnetna obdelava vode (MOV) je alternativna metoda priprave napajalnih vod za nadzor vodnega kamna in prav tako postaja pomembna pri izboljšavah drugih tekočin, ki vsebujejo vodo. Govor je o učinkih naprav MOV, med njimi o spremenjenem kristaljenju vodnega kamna in spremenjeni stabilnosti vodnih disperzij s poudarkom na spremenjeni hidrataciji ionov in trdnih površin zaradi magnetne protonske resonance kot enega izmed možnih mehanizmov. Nadalje je predlagana pojasnitev pospešenega obarjanja aragonita.

(Ključne besede: priprava vode, kristaljenje $CaCO_3$, hidratacija ionska, magnetohidrodinamika)

Magnetic water treatment (MWT) is an alternative method of supplied-water conditioning for scale control and is also important in the amelioration of other water-based fluids. The effects of MWT devices, such as the modified crystallization of the scale–forming components and modified stability of the dispersion are discussed with the emphasis on the modified hydration of ions and solid surfaces due to magnetic proton resonance as one of the possible mechanisms. In addition, a possible explanation for the accelerated aragonite precipitation is proposed.

(Keywords: water conditioning, crystallization, ion hydration, magnetohydrodynamic)

0 UVOD

Nastajanje oblog vodnega kamna je pogosta in draga težava v mnogih industrijskih procesih, ki so napajani z naravnimi vodami. Do oblog na stenah naprav pride zaradi naravne prenasičenosti napajalne vode ali zaradi prenasičenja, ki se vzpostavi med ogrevanjem vode, padcem tlaka ali dvigom *pH* med samo obdelavo vode. Trde obloge zmanjšujejo pretočne zmogljivosti, povečujejo porabo električne energije črpalk in zahtevajo periodično čiščenje.

MOV igra vse pomembnejšo vlogo alternativne metode pri pripravah industrijskih vod. Njene prednosti v primerjavi z znanimi kemijskimi metodami mehčanja vode so: nizki investicijski in obratovalni stroški, preprosta vgradnja, ohranjanje kakovosti vode (npr. organoleptičnih lastnosti) in prispevek k varstvu okolja.

Pri napravah MOV, ki so bile dobro načrtovane za določen vodovodni sistem in dano sestavo vode [1], lahko pričakujemo veliko učinkovitost pri preprečevanju nastanka trdih oblog vodnega kamna, še posebej v ogrevanih in cirkuliranih vodovodnih sistemih.

0 INTRODUCTION

The build up of scale deposit is a common and costly problem in many industrial processes which use natural water supplies. Deposits on the equipment walls result form the natural oversaturation of the supplied water or oversaturation caused by water heating, pressure drop or a *pH* increase during the water processing. The hard-scale deposit reduces water flow capacities, increases the electrical power consumption of pumps and reduces the heat exchanging capabilities of heated surfaces leading to higher operating costs and need to a periodically remove the scale.

MWT is becoming increasingly important as an alternative method of industrial water conditioning. Low investment and operating costs, easy installation, water quality (i.e. organoleptic properties) and ecological benefits are some of the advantages over the well-known chemical methods used for water softening.

MWT devices which are well designed for a particular water composition and industrial process [1], are very effective in preventing hard-scale formation, especially in heated- and circulated-water systems.

Prvi patent tovrstnih naprav je bil vknjižen v Belgiji leta 1945. Praktične izkušnje s to pionirsko napravo so dale zelo različne rezultate: od izjemne učinkovitosti, do popolne neuspešnosti [2]. Od leta 1960 so bili v nekdanji Sovjetski zvezi za nadzor vodnega kamna z veliko ekonomsko koristjo uporabljani močni elektromagneti [3], v USA pa so bile naprave MOV širše sprejete šele po letu 1975 ([4] in [5]).

1 UČINKI NAPRAV MOV NA NARAVNE IN INDUSTRIJSKE VODE

Praktična uporaba naprav MOV je vse bolj razširjena na področju priprave vod za odstranjevanje ali preprečevanje vodnega kamna, prodira pa tudi na nova področja, to so obdelava cementa in goriva ([6] do [12]). V vseh teh primerih gre za obdelavo tekočine, ki vsebuje določen delež vode, da bi se izboljšale njene biokemijske ali fizikalno-kemijske lastnosti.

Iz magnetno obdelane vode se vodni kamen tvorne komponente (predvsem $CaCO_3$ v nizkotemperaturnih sistemih in $CaSO_4$ v pregrevanih sistemih) namesto v obliki težko odstranljivih oblog obarja v suspendirani obliki. Kristali so drugačni po svoji obliki, velikosti in strukturi in so manj adhezivni. V primeru $CaCO_3$ je bilo opaženo, da se lahko z magnetno obdelavo razmerje aragonit/kalcit bistveno zviša ([13] in [14]). Aragonit je kinetično ugodnejša kristalna faza, ki vsebuje slabo adhezivne kristale igličaste oblike. Kalcit je termodinamično ugodnejša kristalna faza, sestavljena iz rombičnih kristalov, ki lahko zaradi svoje velike adhezivnosti tvorijo trde, težko odstranljive obloge.

Zvišano razmerje aragonit/kalcit delno pojasni obarjanje prašnatih oblog iz magnetno obdelane vode. Čeprav je kristaljenje iz magnetno obdelane vode zelo odvisno od same sestave vode in obratovalnih razmer, je v večini primerov opaženo obarjanje zmanjšanega števila kristalov $CaCO_3$, ki so večji in imajo zvišan delež aragonita. Celo kristaljenje iz prenasičene mešanice statično magnetno obdelanih raztopin Na_2CO_3 in $CaCl_2$ v dobro nadziranih laboratorijskih razmerah je dalo podobne rezultate [15].

2 MEHANIZMI DELOVANJA NAPRAV MOV NA PROCESIRANO VODO

Večdesetletne izkušnje na področju MOV so dale nekaj empiričnih osnov za načrtovanje magnetnih naprav, vendar pa še vedno ostaja odprto vprašanje mehanizem, ki bi natančno pojasnil, kako magnetno polje vpliva na obdelovani vodni sistem. Na temelju številnih eksperimentalnih in teoretskih poročil je sklepati, da mehanizem najverjetneje sestoji iz vzporednih, med seboj prepletenih korakov, ki so odvisni od

The first patent refering to a MWT device was registered in Belgium in 1945. Practical experience with these devices showed very different results: from very effective to completely useless [2]. Since 1960, strong electromagnets have been used in the Soviet Union for scale control in high-temperature water systems with significant economic benefits [3]. In the USA, MWT devices have been accepted since 1975 ([4] and [5]).

1 THE EFFECTS OF MWT DEVICES ON NATURAL AND INDUSTRIAL WATERS

The practical use of MWT devices has become increasingly wide spread for descaling or scale prevention and has penetrated into new fields for other purposes such as biochemistry, medicine, agriculture, dispersion separations, concrete and fuel amelioration ([6] to [12]). In all these cases fluids containing some fraction of water were magnetically treated to improve their biochemical or physicochemical properties.

The scale-forming components (mainly $CaCO_3$ in low-temperature systems and $CaSO_4$ in heated systems) precipitate from zhe magnetically treated water in a suspended form rather than by forming hard-scale linings. The crystals are different in terms of their form, size and structure with lowered adhesivity. For $CaCO_3$ it was observed that the ratio of aragonite/calcite crystal phases could be increased ([13] and [14]). The former is a kinetically advanced crystal phase of $CaCO_3$ formed in needle–like crystals which have low adhesion, while the latter is a thermodynamically advanced crystal phase of $CaCO_3$ formed in rhombic crystals which are able to adhere into compact, hard-to-remove scale.

The increased aragonite/calcite ratio partially explains the precipitation of powder deposits resulting from MWT. Although the crystallization in magnetically treated water is strongly dependent on the water composition and working conditions the precipitation of fever and larger $CaCO_3$ crystals with an increased fraction of aragonite was observed in most cases. Even the crystallization in a supersaturated mixture of static magnetically treated solutions of Na_2CO_3 and $CaCl_2$ under well-controlled laboratory conditions gave similar results [15].

2 MECHANISMS OF MWT – ACTING ON PROCESSING WATER

A long history of practical experiences has provided some empirical bases for designing magnetic devices. However, the mechanism which explains how the magnetic field acts on the treated water still remains uncertain. From the many reports relating to laboratory and theoretical research it can be concluded that the mechanism most probably consists of parallel interacting steps, depending on the construction of the MWT devices, the composition of the supplied water as a delovnih razmer (npr. hitrosti pretakanja vode in temperature).

V svetovni literaturi je najti dragocene namige, ki poskušajo pojasniti magnetne učinke, vendar nobeden ni dokončno potrjen in tudi ne pojasni vseh učinkov hkrati.

Vodilne hipoteze so:

- magnetno spremenjena hidratacija ionov in trdnih površin
- magnetohidrodinamični učinek na vodne disperzije
- koncentracijski učinki v delovnih kanalih naprav MOV.

Slednji se štejejo kot zvišanje verjetnosti trkov med ioni ali trdnimi delci v določenih območjih delovnih kanalov zaradi turbulence pretakajoče se vode, visoke magnetnosti suspendiranih korozijskih produktov oz. nehomogenosti magnetnega polja naprave [16]. Učinki tega tipa delno pojasnijo agregacijo drobnih že destabiliziranih delcev v večje, medtem ko je spremenjeno kristaljenje in destabilizacijo dispergiranih komponent, ki tvorijo vodni kamen, laže pojasniti s prvima dvema hipotezama. Kateri mehanizem bo prevladal, je odvisno od sestave vode in samih razmer pri obdelavi. V nadaljevanju bo govor o mogočih vzrokih spremenjenega kristaljenja $CaCO_3$.

2.1 Učinek magnetno spremenjene hidratacije na kristaljenje $CaCO_3$

Eksperimentalna opazovanja vodnih raztopin so med statičnim izpostavljanjem magnetnemu polju pokazala spremembe v nekaterih fizikalno-kemijskih lastnostih, npr.: svetlobni absorbanci [17], viskoznosti [18], topilni entalpiji [19], električni prevodnosti [6], površinski napetosti [20], dielektričnosti [21] in tudi v kristaljenju ter stabilnosti koloidov ([22], [15] in [23]). Opažanja podpirajo hipotezo o magnetno spremenjeni hidrataciji.

dispersion/solution system and on the working conditions (i.e. the water flow velocity and temperature).

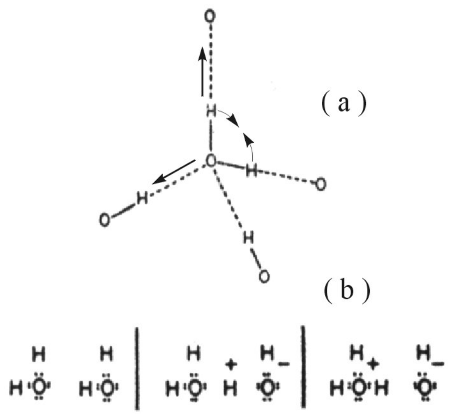
The literature contains some valuable hypotheses explaining magnetic effects on treated water but none is confirmed and none explains all the observed effects simultaneously. The main hypotheses are:

- magnetically modified hydration of ions and solid/solution interfaces;
- magnetohydrodynamic effect on water dispersion systems;
- concentration effects in the working channel of MWT devices.

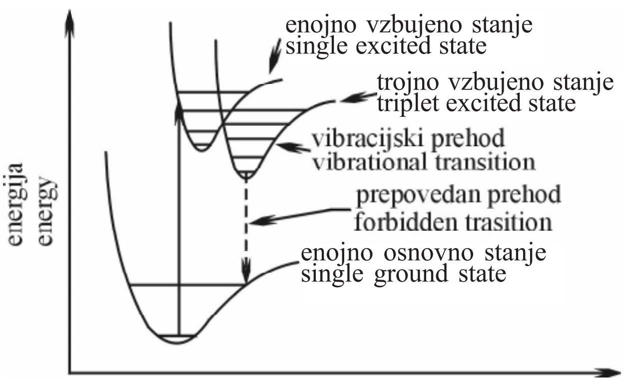
The concentration effects are considered as an increased probability of particle or ion collisions in particular zones of the working channels due to turbulence of the water flow, easy magnetisation of the suspended corrosion products and the inhomogeneity of the applied magnetic field [16]. Effects of this type partially explain aggregation of fine already-destabilized particles into bigger ones, while modified crystallization and destabilization of dispersed scale–forming components are explained by the first two hypotheses. Which mechanism prevails, depends on the water composition and treatment conditions. A possible explanation for the modified crystallization by magnetically modified hydration will be discussed in the next sections of this paper.

2.1 The effect of magnetically modified hydration on $CaCO_3$ crystal growth

Experimentally observed changes in the physicochemical habits of water solutions due to static magnetic field exposure, such as light absorbency [17], viscosity [18], solution enthalpy [19], electrical conductivity [6], surface tension [20], dielectricity [21] and modified crystallization and colloid stability ([22], [15] and [23]), support the hypothesis of magnetically modified hydration.



Sl.1. *Model molekule vode in vodikove vezi [24]*
Fig.1. *The model of water molecule and hydrogen bond [24]*



Sl.2. *Prehodi spinov [25]*
Fig.2. *The spin transitions [25]*

S statističnega vidika namreč voda sestoji iz prostih molekul in ledu podobnih območij, v katerih so tetraedrično oblikovane molekule vode, med seboj povezane z vodikovo vezjo.

Vodikova vez je delno usmerjena zaradi vodikove resonance med kisikovima atomoma (sl. 1/a). Njena moč je odvisna od oscilacij po veznem kotu in dolžini kovalentnih vezi znotraj molekul vode (sl. 1/b). Te kovalentne oscilacije so odvisne od spinskega stanja protona. Magnetna resonanca dveh sosednjih protonov med magnetno obdelavo lahko povzroči spinski preskok (sl. 2) iz enojnega osnovnega stanja (antivzporedna orientacija spinov) v trojno vzbujeno stanje (vzporedna usmeritev spinov), kar lahko posredno privede do oslabitve vodikove vezi.

Neposredna vrnitev v prvotno stanje je po teoriji kvantne fizike prepovedan prehod. Vrnitev po prvotni poti, ki zahteva aktivacijsko energijo, pojasni magnetni spomin - opaženo trajanje učinka magnetne obdelave še nekaj ur ali celo dni po končani obdelavi. Razpoložljiva energija v praktičnih razmerah statične magnetne obdelave je namreč zanemarljiva v primerjavi s termično energijo atomov oz. molekul, ki se sprošča med atomske spremembe v nanosekundah do milisekundah, če te niso povzročene z znotraj atomskimi spremembami v daljših sprostitvenih časih.

Tako oslabljena mreža vode bi pomenila oslabljeno hidratacijo večine ionov v naravnih vodah (z izjemo K^+ in Cl^-), ionov, ki se raztapljajo z nameščanjem v praznine vodne mreže in jo pri tem dodatno krepijo. Po drugi strani pa bodo nekateri ioni (npr. Fe^{2+}), ki se v naravnih vodah pojavljajo v majhnih koncentracijah in se raztapljajo s tvorbo lastne ovojnice iz prostih molekul vode, postali bolj hidratirani. V skladu s to hipotezo so meritve protonske magnetne resonance potrdile magnetno oslabljeno hidratacijo raztopine $Ca(HCO_3)_2$ in ojačano hidratacijo koloidnega železa [26].

Spremembe v hidrataciji delno pojasnijo spremenjeno kristaljenje $CaCO_3$ med končano MOV in po njej. V splošnem kristalna rast poteka skozi več zaporednih korakov, med katerimi najpočasnejši korak določa hitrost rasti kristala. V primeru $CaCO_3$ je bila delna hidratacija kristalotvornih ionov ugotovljena kot najpočasnejši korak [27]. Tako lahko magnetno oslabljena hidratacija Ca^{2+} in HCO_3^- vodi v pospešeno kristalno rast $CaCO_3$ in s tem tvorbo večjih kristalov.

3 SKLEP

V mnogih poročilih raziskav s področja MOV ([28] do [31]) je opaziti pomen komponent, ki vsebujejo železo. Nekatere naprave MOV se lahko štejejo zgolj kot žrtvovane anode (katerih raztapljanje se lahko pospeši z delovanjem magnetnega polja),

From the statistic point of view the liquid water consists of free water molecules and ice-like ranges where the tetrahedral water molecules are bonded with a hydrogen bond.

The hydrogen bond is partially orientated because of the hydrogen resonance between oxygens (Fig. 1/a). Its strength depends on the angle and length oscillation of covalent bonds inside the water molecules (Fig. 1/b). These covalent oscillations depend on the proton spin state. The magnetic resonance of two neighboring protons during the magnetic treatment could cause a spin transition (Fig. 2) from the singled ground state (anti–parallel orientation of spins) into the triplet excited state (parallel orientation of spins) which could indirectly lead to weakening of the hydrogen bond.

A direct return to the ground state is prohibited by the quantum theory. The return through the original transition, which demands an activation energy, explains a magnetic memory which has been observed to last several hours and even days after finishing the magnetic treatment. In other words, the available energy of practical static magnetic treatments is negligible in comparison with the thermal energy of atoms and molecules which relaxes inter–atomic changes in nanoseconds to microseconds if they are not caused by inner–atomic changes with longer relaxation times.

Such a weakened water molecule net would mean a weakened hydration of the main ions in natural waters, with the exception of K^+ and Cl^- ions, which dissolve by placing themselves into a vacancy of the original net with a strengthening of the net. On the other hand, some ions, which in natural waters participate in low concentrations and dissolve by forming their own hydration cover from free water molecules (i.e. Fe^{2+}) become more hydrated. According to this hypothesis, the proton magnetic-resonance measurements have confirmed the magnetically weakened hydration of $Ca(HCO_3)_2$ solutions and the intensified hydration of colloidal iron [26].

The changes in hydration partially explain modified $CaCO_3$ crystallization during and after MWT. In general, the crystal growth is performed with several successive steps, from which the slowest one is the crystal-growing rate-determining step. In the case of $CaCO_3$, the partial dehydration of crystal-forming ions is the slowest growth step [27]. So, the magnetically weakened hydration of Ca^{2+} and HCO_3^- can lead to enhanced crystal growth of $CaCO_3$ and the formation of bigger crystals.

3 CONCLUSION

In many reports about MWT research ([28] to [31]) the importance of iron components has been highlighted. Some MWT devices could be considered only as a sacrificed anode (the dissolving of which is accelerated by a magnetic field), but in most cases vendar pa se v večini primerov vloga komponent z deležem železa ne more pojasniti na ta način. V primerih dinamične MOV pride do magnetohidrodinamičnih premikov kristalotvornih ionov k rastoči površini zaradi Lorentzove sile, kar bi lahko pospešilo in spremenilo kristaljenje vodnega kamna. Ti premiki so večji pri ionih z višjo valenco in manjšim radijem in so različno usmerjeni ob dispergiranih delcih vodnega kamna [32]. Magnetohidrodinamični premiki Fe^{2+}, kot pragovnega zavirala kalcitne rasti, bi lahko delno pojasnili zvišan delež aragonita, medtem ko bi moralo biti pospešeno obarjanje $CaCO_3$ ob navzočnosti mikrokristalov železovega hidroksida nekako povezano s spremembami *pH*, ki imajo močan vpliv na rast $CaCO_3$ in bi lahko bile povzročene z magnetno spremenjeno hidratacijo površin železovega hidroksida.

the role of iron components could not be explained only in this way. In the cases of dynamic MWT, the magnetohydynamic shifts of crystal – forming ions towards the growing crystal surface occur due to Lorentz force and could accelerate and modify the scale crystallization. They are higher for ions with higher ion valence and lower radius, and have different orientation at dispersed scale particles (Lipus [32], 1998). Magnetohydrodynamic shifts of Fe^{2+} ions as threshold inhibitors of calcite growth could partially explain raised part of aragonite. While at the acceleration of $CaCO_3$ precipitation in the presence of iron-hydroxide micro–crystals should be connected with *pH* changes, which have a strong influence on $CaCO_3$ growth and could be caused by magnetically modified hydration of iron-hydroxide surfaces.

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Prejeto:
Received: 15.8.2000

Sprejeto:
Accepted: 10.11.2000

© Strojniški vestnik 46(2000)8,538-548
ISSN 0039-2480
UDK 621.67:534.83:681.892
Pregledni znanstveni članek (1.02)
© Journal of Mechanical Engineering 46(2000)8,538-548
ISSN 0039-2480
UDC 621.67:534.83:681.892
Review scientific paper (1.02)

Analiza obratovalnega hrupa in vibracij okrova radialne črpalke

Radial Pump Operating Noise and Casing-Vibration Analyses

Andrej Predin - Mitja Kastrevc - Ignacijo Biluš

V prispevku je podana teoretična in eksperimentalna študija dinamičnih tekočinskih vibracij in hrupa. Eksperimentalne meritve so izvedene na enostopenjski radialni črpalki, ki obratuje s čisto hladno vodo. Pri različnih obratovalnih režimih (vrtilnih frekvencah rotorja) so izvedene meritve naslednjih obratovalnih karakteristik: dušilna krivulja, pretok – izkoristek, pretok – obratovalni hrup in pretok – amplituda vibracij okrova. Rezultati meritev so podani v časovnem in frekvenčnem prostoru.

(Ključne besede: črpalke radialne, hrup črpalk, meritve vibracij, minimiranje hrupa)

This paper surveys theoretical and experimental studies of fluid-dynamic vibration and noise. The experimental measurements were carried out on a radial one-stage pump which operates with clean cold water. Several experimental measurements on the operating characteristics such as capacity-head, capacity-efficiency, capacity-operating noise and capacity-pump casing vibration amplitudes under different operating regimes (different impeller speed) were performed. Measurement results are given in time and frequency domains.

(Keywords: radial pumps, pump noise, vibration measurements, noise minimization)

0 UVOD

Obratovalni hrup in vibracije okrova sodobnih črpalk morajo biti minimalni. V ta namen moramo reducirati vse vire. Obratovalni hrup in vibracije okrova so posledica pulzirajočih tokovnih veličin na izstopu iz rotorja. Hrup se sprošča vedno, ko imamo opravka z relativnim gibanjem dveh tekočin (kavitacija – voda in para) ali tekočine in trdne stene (lopatice). Značilni viri hrupa črpalk vsebujejo časovno spremenljiv sistem sil, ki delujejo na eno ali več komponent črpalke. Hrup je rezultat reakcije tekočine na omenjene sile in vsiljenega nihanja teles, ki so v stiku s tokom. Vibracije okrova in njegovih mirujočih ter gibajočih se delov so posledica tokovno vzbujenih vibracij teles v ustaljenem ali pulzirajočem in turbulentnem toku. Pri tokovno vzbujenih vibracijah je prav tako treba upoštevati tudi različne vibracijske oblike kakor pri nihajočih trdnih telesih. Pulzacija toka na izstopu iz rotorja je posledica nepopolnosti rotorja in še posebej relativnega vrtinčnega toka v posameznem rotorskem kanalu, ki je pri obratovanju zunaj preračunske točke (točka največjega izkoristka) še močnejši in povzroča nepravilni natok

0 INTRODUCTION

Modern pumps are expected to have their operating noise and casing vibrations minimized. In this way the noise and vibration pollution of the surroundings are reduced. The operating noise and casing vibrations of radial pumps are a consequence of the pulsating fluid flow properties at the impeller exit. Noise may be emitted whenever there is a relative motion of two fluids (cavitation – water and vapour) or a fluid and a solid surface (blades, vanes). Typical sources of noise from pumps involve a time-varying system of forces affecting one or more components of the pump. Noise results from the fluid's reactions to this force as well as from the forced vibration of structures in contact with the flow. The vibration of the pump casing and its static and dynamic (moving) parts are a consequence of the flow-induced vibration of solid structures in stationary or pulsating and turbulent flow. Therefore, the subject of flow-induced vibrations must also consider the vibration of structures of a single mode or of many modes as well. The pulsating flow at the impeller exit is the result of impeller imperfections, especially, of the relative flow whirl in an individual impeller channel. The whirl increases by shifting the operating mode out of the optimum regime (out of the best v rotorske in vodilne kanale. Tako nastane neustaljeni pulzacijski turbulentni tok, ki povzroča vibracije okrova. Moč opisanih tokovnih pulzacij v posameznih obratovalnih režimih (obratovanje pri majhnih, pod optimalnih pretokih) se spreminja. Odvisna je od natočnega kota na rotorske lopatice, ki se spreminja v odvisnosti od pretoka in/ali vrtilne frekvence.

Študija hidravličnih lastnosti črpalke je zasnovana na modelu črpalke, na katerem so opazovane vibracije okrova in hrup v odvisnosti od vrtilne frekvence. Pri meritvah so določeni tudi različni mehanizmi tokovnih motenj. Za minimizacijo hrupa in vibracij moramo vedeti, da se te povečujejo s povečanjem vrtilne frekvence rotorja in tudi kadar je obratovalni pretok ali obratovalna točka črpalke zunaj točke največjega izkoristka. Hrup in vibracije okrova so minimalne, ko črpalka obratuje z optimalnim pretokom, ki je največkrat kar preračunski pretok. Če pa želimo dosegati zahtevano črpalno višino in pretoke pri manjših vrtilnih frekvencah rotorja, morajo biti izstopne hitrosti toka večje. Te lahko dosežemo le s povečanjem izstopnega kota rotorskih lopatic ali s povečanjem izstopnega premera rotorja. Na žalost sta oba postopka omejena s trdnostjo materiala in kavitacijskimi problemi. Zaradi tega je za minimizacijo hrupa in vibracij priporočena standardna tridimenzionalna metoda optimizacije med vrtilno frekvenco rotorja, izstopnim kotom rotorskih lopatic in izstopnim premerom rotorja.

efficiency point – BEP), and causes the irregular loading of the impeller and guide-vane channels. As a result, the non-stationary pulsating and turbulent flow that causes the casing vibrations is created. The intensity of the flow pulsations in some particular pump regimes (operating with small under-optimum capacities) is changed. It depends on the angle of flow attack to the impeller blade that is changed by the operating capacity or/and by changing the impeller speed.

The study of pump hydraulics behaviour is based on the pump-scale model in which the noise and casing vibrations observed for a given configuration as a function of the impeller speed are tested. With measurements, the particular disturbance mechanisms are determined. For minimizing the pump operating noise and casing vibrations it is necessary to know that the noise intensity and pump-casing vibration amplitudes increase as the impeller speed increases. They also increase when the operating capacity, or pump operating point is out of the BEP. The minimum noise intensity and pump casing vibration amplitudes exist when the pump operates at optimum capacity, that is in most cases the pump design capacity. However, if we want to satisfy the required head and capacity with a smaller impeller speed, the flow velocities at the impeller exit must be larger. This can only be achieved by increasing the impeller-blade exit angle or by increasing the impeller-exit diameter. Unfortunately, both approaches are limited by impeller material strength and by cavitation problems. For these reasons the classic 3D optimisation plan of the impeller speed, the blade's exit angle and the impeller-exit diameter for the minimization of the pump operating noise and casing vibration are recommended.

1 DIMENZIJSKA ANALIZA NASTAJANJA ZVOKA

1 DIMENSIONAL ANALYSIS OF SOUND GENERATION

Za analizo tokovno vzbujenih motenj razdelimo tok na časovno povprečno in oscilirajočo komponento (Reynoldsov postopek). V skladu s tem je lokalna hitrost v določeni točki definirana kot vsota povprečne vrednosti in trenutnega odmika od tega časovnega povprečja. Hitrost v točki tekočinskega toka je tako predstavljena z:

The first feature of flow-induced disturbance to note is that flow is generally usefully regarded as a mean plus a fluctuating part (Reynolds' idea). Therefore, the local velocity at a particular point may be regarded as a superposition of an average value and an instantaneous fluctuating part. Thus the velocity at a point in the fluid flow may be described by:

$$U=\bar{U}+u(x,y,z,t) \quad (1),$$

kje sta - $\bar{U}$ povprečna vrednost in $u(x,y,z,t)$ - njena oscilirajoča vrednost, ki je odvisna od časa in lege v toku in jo poenostavljeno lahko določimo kot:

where $\bar{U}$ is the average value and $u(x,y,z,t)$ is the unsteady value which depends on time and location in the flow, and can be determined in a simplified way as:

$$\overline{u^2}(t)=\frac{1}{t_0}\int_0^{t_0} u^2(t)\,dt \quad (2),$$

kjer je t_0 - opazovan časovni trenutek. V dinamično podobnih tokovih, pri katerih je zahtevana enakost tako amplitude kakor faze, ostane zveza med silami in premiki nespremenjena, kar je pri primerjavi modela s prototipom upoštevamo v razmerju:

where t_0 is the observation time. In similar dynamic flows, this requires the relationship of both the magnitude and the phase, among forces and motions to remain fixed, for example in a model-to-full-size comparison, the ratio:

$$\frac{u(t)}{\bar{U}} \tag{3}$$

ki je nespremenljivo, ne glede na vrednost $\bar{U}$, ta pomeni stopnjo hitrostnih sprememb glede na povprečno hitrost [1] in [2]. Da bi lahko bilo zgornje razmerje nespremenljivo, morajo biti v ravnotežju tudi različne napetosti, ki delujejo na tekočinske delce. Ponavadi je to kombinacija vztrajnostnih in viskoznih sil, katerih razmerje je definirano z Reynoldsovim številom. Zgoraj omenjena podobnost je zelo pomembna, saj je vzbujevalna tlačna napetost, označena s $|p|$, ki vzbuja hrup ali vibracije v danem toku v neposrednem razmerju:

is a constant, regardless of the value of $\bar{U}$, that is, the distribution of velocity fluctuation's scales on the mean velocity, [1] and [2]. For maintenance of this constancy through the flow, the balance of the various types of stress that act on fluid particles must also be maintained. Generally these are the combinations of the inertial and viscous stress and a measure of the ratio of the inertial to viscous stress in the flow is the Reynolds number. The above-mentioned similitude is important because of the exciting stress, denoted here by $|p|$, that produces sound or vibration in a given type of flow which is in direct proportion as:

$$|p| \approx \frac{1}{2}\rho_0 U^2 = p_d \tag{4}$$

kjer pomenita: ρ_0 - gostoto tekočine, p_d - dinamični tlak. Sorazmernost velja tako dolgo, dokler sta sorazmerni tudi povprečna in oscilirajoča komponenta hitrosti. Ker sta napetosti, ki povzročata hrup in vibracije sorazmerni dinamičnemu tlaku p_d, lahko le-tega privzamemo za merilo moči vzbujanja. Merilo ujemanja hidrodinamičnih ali aerodinamičnih gibanj in hitrosti delčkov glede na hitrost širjenja zvoka je Machovo število, ki pomeni razmerje med hidrodinamično hitrostjo in hitrostjo zvoka. Reynoldsovo in Machovo število predstavljata relativen pomen vztrajnostnih, viskoznih in tlačnih napetosti v tekočini. Za dinamično in akustično podobnost modela in prototipa morata biti poleg podobne geometrijske zato enaki tudi vrednosti Reynoldsovega in Machovega števila.

where ρ_0 is the fluid mass density and p_d is the dynamic pressure. The proportionality may hold as long as the fluctuating velocity and the mean velocity are also proportional. Since the sound- and vibration-producing stresses are proportional to p_d, this can be taken as a measure of the intensity of the magnitude of the excitation. A measure of the matching of fluid inertial motions of hydrodynamics or aerodynamics and the particle velocities related to the propagation of sound is the Mach number, which expresses the ratio of the hydrodynamic velocity to the acoustic particle velocity. The Reynolds and Mach numbers express the relative importance of inertial, viscous, and compressive stress in the fluid. Fluid dynamics and acoustic similitude therefore ideally require, in addition to similar geometries, equal values of Reynolds and Mach numbers for model and prototype.

1.1 Stopnja zvočnega tlaka

Osnovna merjena veličina zvoka v določeni točki je tlak *p*. Ker je zvok dinamični pojav je tudi akustično vzbujen tlak časovno spremenljiva veličina. Običajno merilo akustičnega tlaka je njegova časovno povprečena kvadratna vrednost:

1.1 Sound pressure level

The principal measured property of sound is the pressure (*p*) at a point. Since sound is a dynamic phenomenon, the acoustically induced pressure is also a time-varying quantity. The measure of acoustic pressure that is conventionally reported is the time average of a pressure squared, that is:

$$\overline{p^2} = \frac{1}{T}\int_{-T/2}^{T/2} p^2(t)\,dt \tag{5}$$

s časovnim povprečjem enakim nič, $\bar{p} = 0$, ki je v preprosti zvezi z intenziteto in stopnjo jakosti zvoka. Stopnja zvočnega tlaka je določena z:

with the time average equal to zero, $\bar{p} = 0$. This is simply related to sound intensity and power levels. The sound pressure level is determined from the above as:

$$L_S = 10\log\left(\overline{p^2} / p_{ref}^2\right) \tag{6}$$

kjer je $p_{ref} = 2\cdot10^{-5}$ N/m^2, ali 20 μPa za zvok v plinih, in 10^{-6} N/m^2, ali 1 μPa za zvok v kapljevinah. Če obravnavamo širjenje zvoka na močnostni bazi, je stopnja moči zvoka podana z:

where $p_{ref} = 2\cdot10^{-5}$ is N/m^2, or 20 μPa for sound in gases, and 10^{-6} N/m^2, or 1 μPa for a sound in liquids. Generally, if the sound transmission is considered on a power basis, the sound power level is defined as:

$$L_n = 10\log\left(P / P_{ref}\right) \tag{7}$$

kjer sta P - moč zvoka, prenesena prek določene površine in P_{ref} - referenčna veličina, ponavadi enaka 10^{-12} W. Moč zvoka, prenesenega prek krogelne površine A_s tvori točkast izvor, ki je v naslednji zvezi z zvočnim tlakom:

where P is the sound power transmitted across a specified surface and P_{ref} is a reference quantity conventionally taken as 10^{-12} W. The sound power radiated across a spherical surface of the area A_s, forms an omni-directional source that is related to the sound pressure as:

$$L_n = L_S + 10\log\left(\frac{p_{ref}^2 A_S}{\rho_0 c_0 P_{ref}}\right) \quad (8),$$

kjer je c_0 - hitrost zvoka. Intenzivnost zvoka lahko določimo iz:

where c_0 is the acoustic speed. The sound intensity level may be found from:

$$L_I = 10\log\left(\frac{I}{I_{ref}}\right) \quad (9),$$

kjer je zveza akustične intenzivnosti s povprečno kvadratno vrednostjo tlaka definirana kot:

where the acoustic intensity is related to the mean-square pressure by:

$$I = \frac{\overline{p^2}}{\rho_0 c_0} \quad (10)$$

in $I_{ref} = 10^{-12}$ W/m^2. Akustična intenzivnost je dejansko vektorska veličina. Če je dovolj daleč od vira, je njena smer normalna na krogelno površino, ki točkasti izvor obdaja. Smer I je na dovolj veliki oddaljenosti, torej radialno iz središča izvora.

V črpalki veljajo med dimenzijskimi parametri naslednje zveze: tipske hitrosti $U_T \propto D n_s$, tlačna razlika $\Delta p \propto \rho_0 D^2 n_s^2$, prostorninski pretok skozi črpalko $Q \propto D^3 n_s$. n_8 je specifično število vrtljajev rotorja. Tako je za podani medij (npr. zrak) odvisnost skupne nastale moči zvoka od padca tlaka in volumskega pretoka podana kot:

and $I_{ref} = 10^{-12}$ W/m^2. The acoustic intensity is a vector property. However, far enough from the source the acoustic energy intensity across a spherical surface surrounding the source is directed normal to the surface. Therefore in the far field the direction of I is radial from the acoustic centre of the source.

In a pump the following relationships apply between dimensional parameters: the tip speed $U_T \propto D n_s$, the pressure drop across the pump $\Delta p \propto \rho_0 D^2 n_s^2$, and the volumetric flow rate (capacity) through the pump $Q \propto D^3 n_s$. n_8 is the shaft rotation rate or specific impeller speed. Thus for a given working fluid (e. g., air) the overall radiated sound power has the following dependence on the pressure drop and flow rate:

$$P_{rad}[\mathrm{W}] = a_P\left(\Delta p[\mathrm{Pa}]\right)^2 Q\left[\mathrm{m}^3/\mathrm{s}\right] \quad (11),$$

ki jo lahko upoštevamo kot sorazmerno v neki frekvenčni stopnji pa:

and the proportional band levels:

$$P_{rad}(f, \Delta f) = a_P\left(\Delta p\right)^2 Q\, F\left(\frac{fD}{U_T}\right) \quad (12),$$

kjer je a_P - konstanta, odvisna od tipa črpalke. Normni spekter pasovnih stopenj je:

where a_P is a constant that depends on the type of pump. The normalized spectrum of band levels is:

$$\frac{P_{rad}(f, \Delta f)}{P_{rad}} = f\left(\frac{fD}{U_T}\right) \quad (13),$$

iz česar je razvidna odvisnost od tipa črpalke in frekvenčne širine pasu.

Za grobo oceno lahko moč nastalega zvoka določimo sorazmerno P_{rad}, Δp, Q ter preostalim obratovalnim parametrom ([3] in [4]). Preproste enačbe (12) in (13) s podanimi vrednostmi a_P in $F(fD/U_T)$ lahko uporabimo za različne vrste črpalk [5].

which exhibits a dependence on both the type of pump and the frequency bandwidth.

For rough estimations the sound power outputs can be determined with the sizing process, P_{rad}, Δp and Q and and all working parameters ([3] and [4]). Simple formulas such as (12) and (13) with given values of a_P and $F(fD/U_T)$ can thus be used to estimate the sound power for different types of pumps [5].

2 MERILNO POSTROJENJE Z RADIALNO ČRPALKO

2 RADIAL PUMP EXPERIMENTAL SET-UP

Meritve obratovalnega hrupa in vibracij okrova so bile izvedene na radialni črpalki

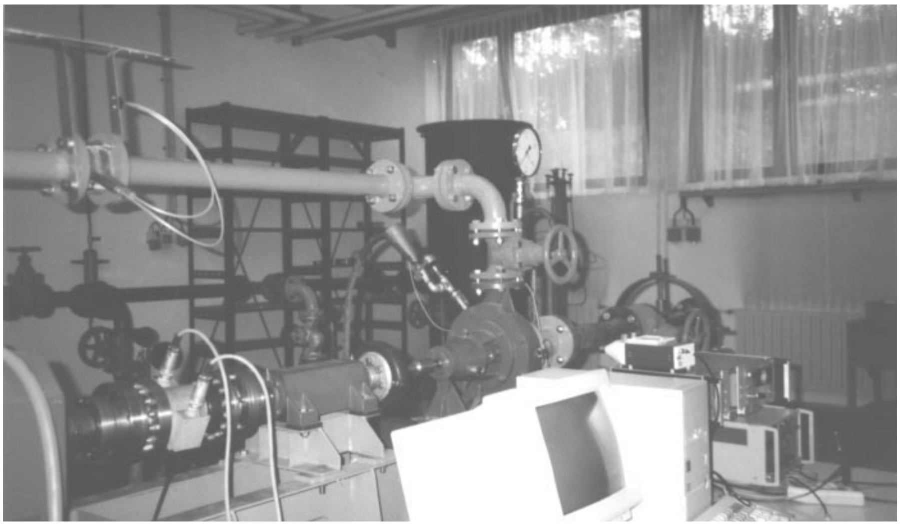
To perform the experimental measurements on the pump operating noise and pump-casing vi-

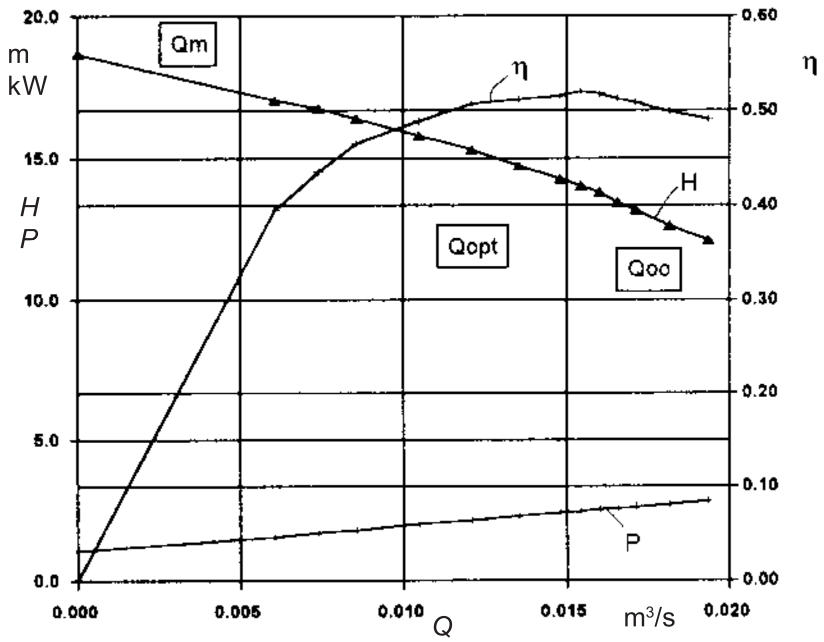
Litostroj CN50/250 (sl. 1). Črpalka ima nizko specifično število vrtljajev n_q = 24 min^{-1}, osem lopatic, spiralni okrov in stabilno dušilno krivuljo (sl. 2).

Vibracije okrova smo merili v treh smereh: pozitivna smer y – osna smer vstopnega cevovoda, pozitivna smer z – radialna smer vstopnega cevovoda, pozitivna smer x – smer tangente na izstopni premer. Za merjenje vibracij smo uporabili merilnik B&K tip 4321 za merjenje hrupa pa merilni sistem RFT 2218 [6]. Za frekvenčni spekter smo uporabili sistem za zbiranje podatkov (DAQ), sestavljen iz osebnega računalnika in več funkcijske kartice Intelligent Instrumentation PCI-2048W. Podatke smo obdelali z Visual Designerjem s frekvenco zajemanja enako 6 kHz na kanal.

brations a radial pump, type CN50/250, manufactured by Litostroj was used (Fig. 1). The pump has a low specific speed (n_q = 24 rpm), eight impeller blades and a volute casing. Its operating characteristics, capacity-head (Q-H) are stable (Fig. 2).

Casing vibrations were investigated in three directions: Positive y–direction in the pump axial direction (in the direction of the intake pipe axis), positive z–direction in the pump radial direction, and positive x–direction in a direction tangentional to the impeller exit diameter. A B&K system type 4321 was used for vibration measurements, and an RFT 2218 measuring system for the noise measurements [6]. For the frequency spectrum (and the power spectrum) a data acquisition system (DAQ) was used. The system consists of a PC with a Multifunctional PCI-2048W board card – Intelligent Instrumentation, and software – Visual Designer. The sampling frequency used in the experiment was 6 kHz per channel.





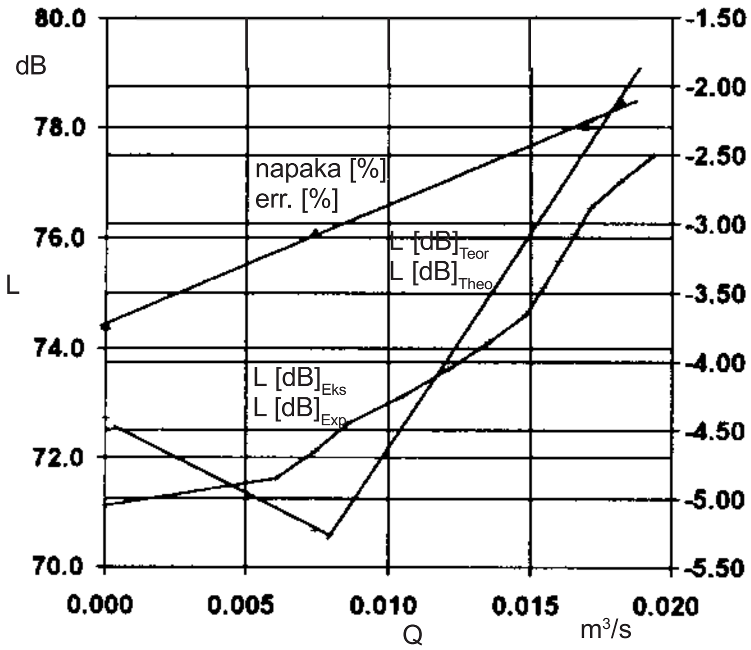
Sl. 2. *Obratovalne karakteristike radialne črpalke*
Fig. 2. *Tested radial pump operating characteristics*

2.1 Obratovalni hrup črpalke pri delni obremenitvi

Povprečne vrednosti stopnje hrupa so razvidne iz rezultatov meritev, prikazanih na sliki 3, od koder je razvidno, da se hrup zvečuje s pretokom. Domnevamo lahko, da sta za to dva vzroka: prvič: dušilna krivulja je stabilna z majhno spremembo črpalne višine (od H = 23 m pri najmanjšem pretoku, do H = 15 m pri največjem pretoku), zato je majhna tudi sprememba hitrosti na izstopu iz rotorja; drugič: absolutna sprememba pretoka je razmeroma majhna (v razponu od 0 do 0,02 m^3/s), zato so hitrosti toka skozi črpalko tudi majhne, kar povzroča spremembo jakosti zvoka.

2.1 Pump operating noise during part-load operation

The mean values of noise level are evident from the common-noise measurement results (Figure 3). The noise level increases with capacity increase. The are two possible reasons for this. First, the pump characteristic *Q-H* is stable and the change of the head by the increase in capacity is small (from H=23 m at zero capacity, up to H = 15 m at maximum capacity). Therefore, the flow velocity changes at the impeller exit are small. Second, the absolute capacity changes are also relatively small (in the range 0 up to 0,02 m^3/s), so the flow velocities across the pump are also small, which causes the change in the noise level.



Sl. 3. *Povprečni karakteristični obratovalni hrup radialne črpalke*
Fig. 3. *Mean-common operating noise of the radial pump*

2.1.1 Močnostni spekter

V nizkofrekvenčnem močnostnem spektru vibracij črpalke do 2000 Hz pri obratovanju z najmanjšim pretokom (sl. 4a) sta v obodni smeri (x) opazni frekvenca vrtenja rotorja (I1) – prvi vrh z leve in prva višja harmonska frekvence lopatice (B2) – drugi vrh z leve. V zgornjem intervalu (do 3000 Hz) je razvidno večje število vrhov, ki pripadajo toku, povzročajo jih tokovno vzbujene vibracije. S porastom pretoka do optimalnega (sl. 4b) in nad optimalnega (sl. 4c) ostane frekvenčni spekter nespremenjen. V vseh posnetkih (sl. 4a, b,c) prevladuje prva višja harmonska frekvence lopatic oz. pulzacije toka (B2), [7] in [8].

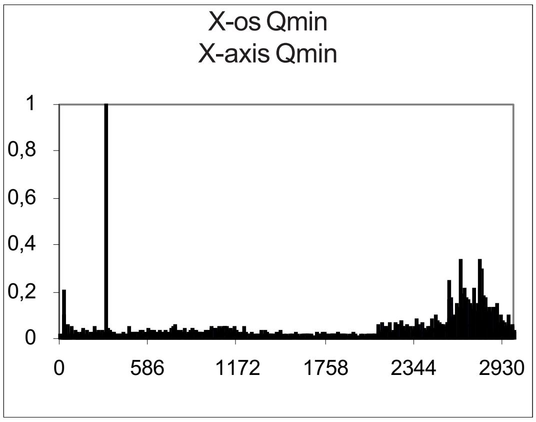
Pri obratovanju z najmanjšim pretokom prevladuje v osni smeri (y) frekvenca vrtilne frekvence rotorja (I1) – prvi vrh z leve proti desni. Iz spektra je razvidna tudi frekvenca lopatic (B1) in

2.1.1 Power spectra records

In the power-spectra record of pump-casing vibrations with operation at minimum capacity the impeller rotation frequency (I1), first left peak (Figure 4.a), and the first higher harmonics of the blade frequency (B2), second left peak (Figure 4.a), are evident in the tangentional *x*–direction of the pump-casing vibration in the lower frequency range up to 2000 Hz. In the upper frequency range (up to 3000 Hz) a larger crowd of higher peaks are evident. These vibration frequencies belong to the flow. The vibration may be caused by the flow-induced vibrations. With a capacity increase up to the optimum capacity (Figure 4.b) and a further capacity increase up to over-optimum capacities (Figure 4.c), practically the same frequency situation as observed for minimum capacity is evident. In all three records, Figure 4.a,b and c, the first higher harmonic of the flow pulsation or blade frequency (B2) is dominant, [7] and [8].

njenih sedem višjih harmonskih (B2-B8). Posebej prevladujoča je prva višja harmonska (B2) – tretji vrh z leve. Amplitude drugih vibracij v višjem frekvenčnem intervalu so višje kakor v primeru obodne smeri. To dokazuje, da vplivajo vstopne tokovne vibracije na vibracije okrova v osni smeri, [9]. S povečanjem pretoka do optimalnega (sl. 4e) in nadoptimalnega (sl. 4f) zasledimo podobni frekvenčni spekter. Edina večja razlika je v osnovni pulzacijski frekvenci toka (frekvenca lopatice (B1)), katere amplituda je najmanjša pri optimalnem pretoku skozi črpalko.

In the axial direction (*y*–direction) at minimum pump capacity operation the impeller speed frequency (I1) amplitude (first peak from left to right) in the power spectrum's record (Figure 4.d) is dominant. The blade frequency (B1), second peak from left to right, and its seven higher harmonics (B2 – B8) are present in the record. This proves that the intake flow vibration has an influence on the pump casing vibration in the axial direction, [9]. With capacity increases up to the optimum (Figure 4.e) and with over-optimum (Figure 4.f), a similar frequency situation as with the minimum capacity is evident. The only difference is the basic flow pulsation frequency (blade frequency B1), the amplitude of which is a minimum at the optimum pump capacity.



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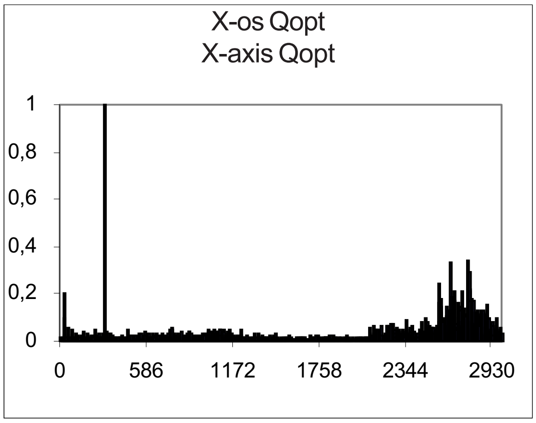

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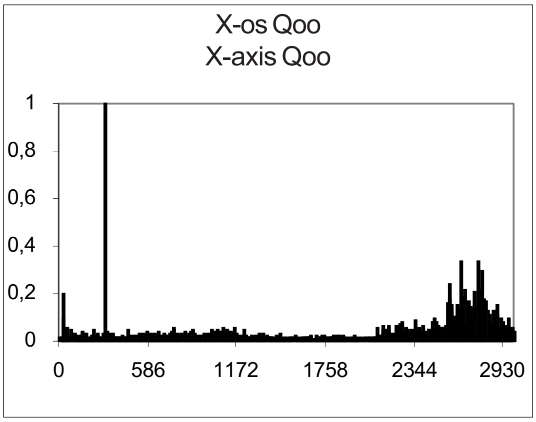

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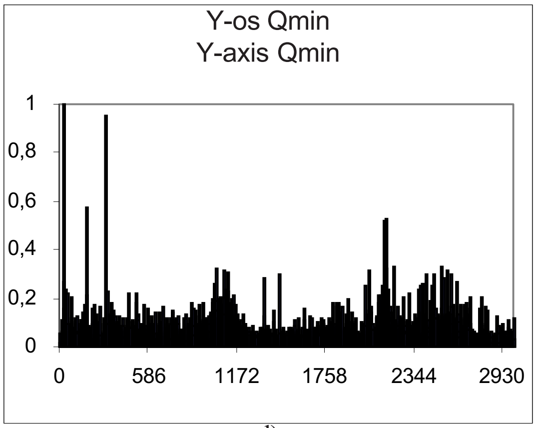

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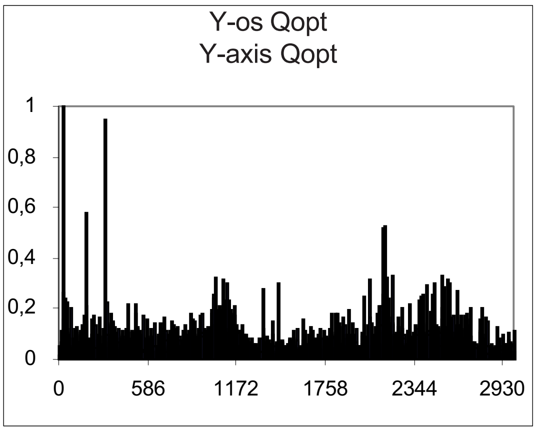

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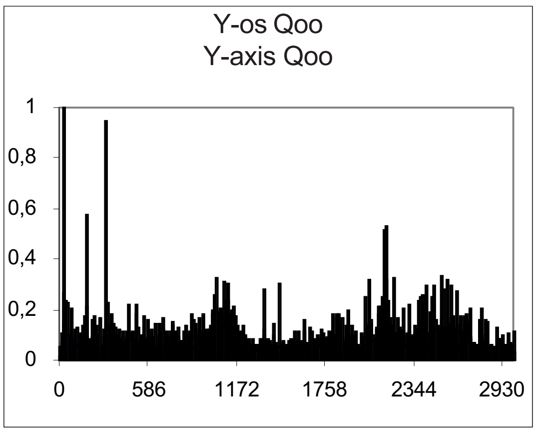

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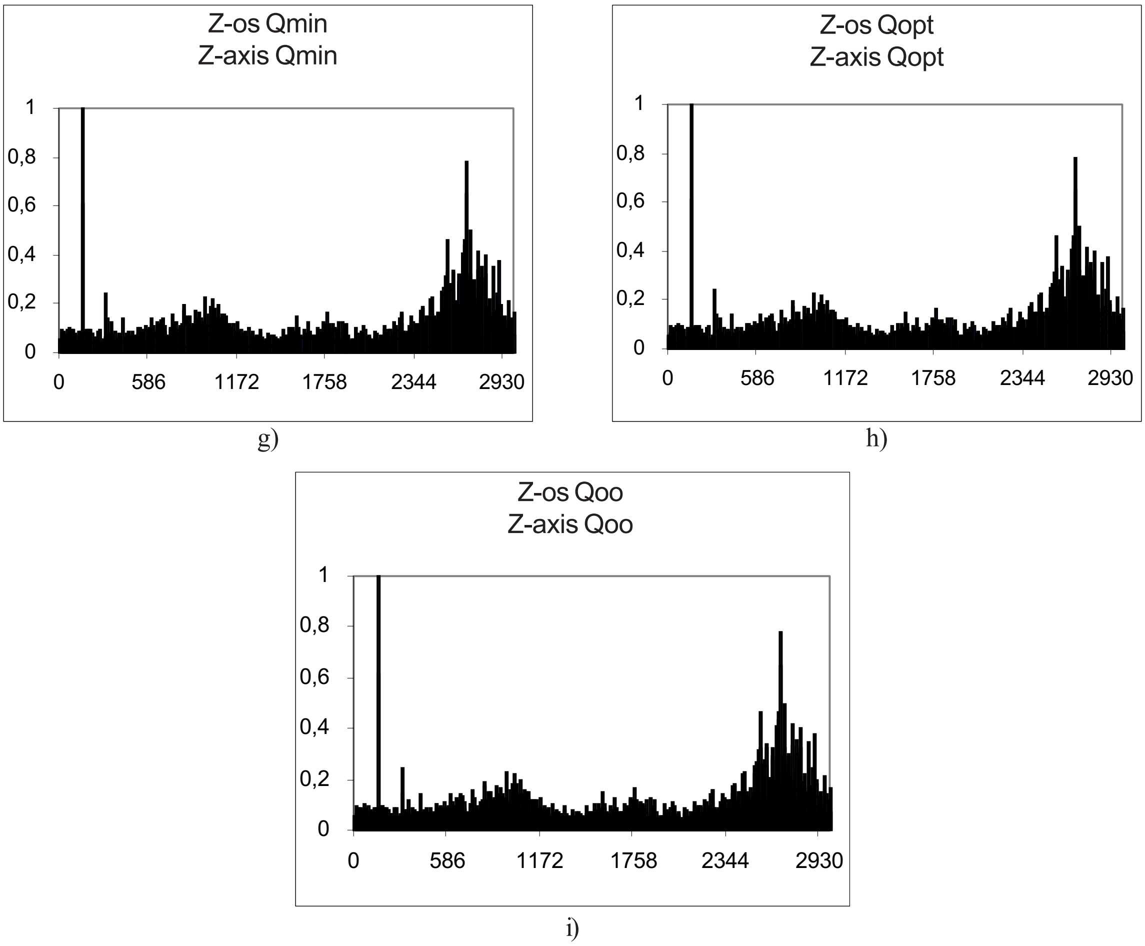


Sl. 4. *Močnostni spektri vibracij okrova v treh koordinatnih smereh*
Fig. 4. *Power spectrum records of the pump casing vibrations in three directions*

V radialni smeri (x) je pri obratovanju z najmanjšim pretokom (sl. 4g) prevladujoča frekvenca lopatice (B1). V močnostnem spektru so opazne tudi njene višje harmonske (B2-B8). V zgornjem frekvenčnem območju je večja množica frekvenc, ki pripadajo tokovno vzbujenim vibracijam. S povečevanjem pretoka skozi črpalko do optimalnega (sl. 4h) in nadoptimalnega (sl. 4i) dobimo podobne rezultate, [10].

Močnostni spekter obratovalnega hrupa, prikazanega na sliki 5 je šibkejši od močnostnega spektra vibracij okrova. V vseh treh koordinatnih smereh (sl. 5a, b, c) prevladuje prva višja harmonska frekvence lopatic (B2). Amplituda te frekvence je pri spremembi pretoka od najmanjšega do največjega domala konstantna. Zato sta frekvenca pulzacije toka oziroma lopatična frekvenca (B1) in njena prva višja harmonska (B2) glavna vira zvoka.

Zdaj je problem ustrezno določiti amplitudo prvega višjega harmonika pulzacije toka. Obratovalni hrup lahko določimo z enačbo:

In the radial direction (*x*–direction) during minimum pump operating capacity the blade frequency (B1) amplitude is dominant in the power spectrum's record (Figure 4.g). Also all higher harmonics of the blade frequency (B2 – B8) are present in the record. In the upper frequency range, the larger frequency amplitude crowd that belongs to flow-induced casing vibrations in the radial direction is evident. By increasing the capacity up to the optimum (Figure 4.h) and up to over-optimum (Figure 4.i) a similar frequency situation is evident [10].

From the power spectra's records (Figure 5) of pump operating noise, a much weaker record in comparison with casing vibration is evident. For all three records (Figure 5.a,b,c) the first higher harmonic's amplitude of the blade frequency (B2) is dominant. The amplitude of this highest peak in the record is practically constant during the capacity change from minimum up to maximum capacity. Therefore, the flow pulsation frequency, or blade frequency (B1), and its first higher harmonic (B2) are the main noise sources.

However, the problem is how to estimate the first higher harmonic's amplitude of the basic flow pulsation frequency amplitude properly. The pump operating noise can be evaluated by the equation:

$$P_{rad} = a_P \cdot \rho\, c_2^3 \left(\frac{c_2^2}{c_0}\right)^2 D_2^2 \qquad (14),$$

kjer so c_2 - absolutna hitrost toka na izstopnem premeru rotorja D_2, in a_P funkciji tipa črpalke in vrtilne frekvence rotorja, podana z:

where c_2 is the absolute flow velocity at the pump impeller exit diameter D_2, and a_P is a function of the pump type and impeller speed in the following form:

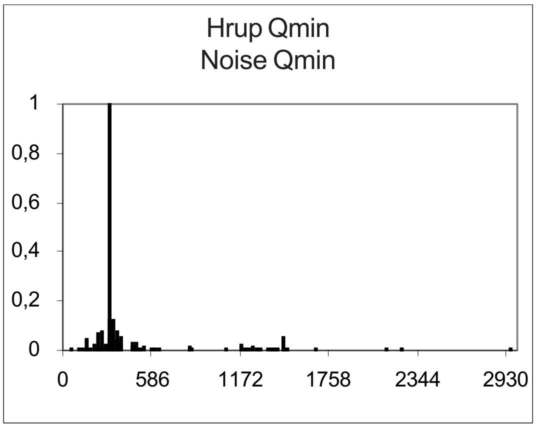
$$a_P = 4.8 \cdot 10^{-5} \left[\frac{n}{n_{des}}\right]^{\frac{n}{n_{des}}} \qquad (15),$$

kjer sta n - število vrtljajev rotorja, n_{des} - preračunsko število vrtljajev rotorja. Številčna konstanta v enačbi je eksperimentalno določena srednja vrednost merilnih rezultatov.

S slike 3 je razvidno dobro ujemanje med teoretično določenim obratovalnim hrupom in eksperimentalnimi rezultati.

where n is the impeller speed and n_{des} is the impeller design speed. The numerical constant in equation is experimentally determined as the mean-common value of the measuring results.

There is a good agreement between the theoretically determined pump operating noise and that of the experimental measurements (Figure 3).



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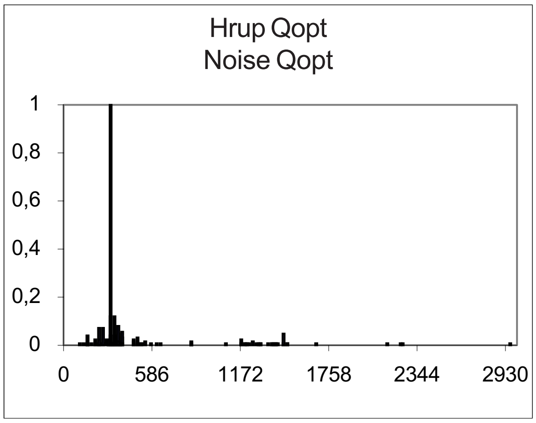

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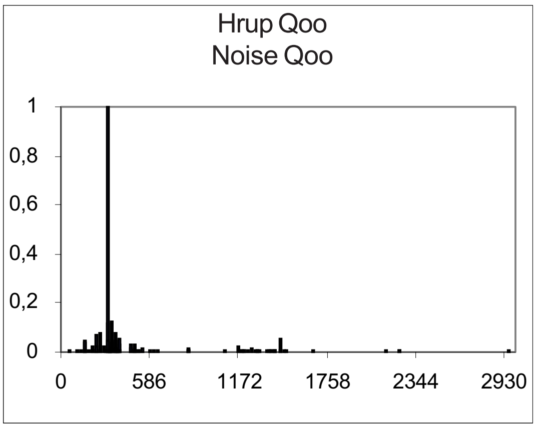

c)

Sl. 5. *Močnostni spekter obratovalnega hrupa radialne črpalke*
Fig. 5. *Power spectrum records of the radial pump operating noise*

3 SKLEP IN RAZLAGA

V večini testiranih obratovalnih režimov sta prevladujoči frekvenci vrtenja rotorja (I1) in prva višja harmonska frekvence lopatice (B2). Tako lahko sklenemo, da sta prevladujoča vira obratovalnega hrupa in vibracij okrova vrtilna frekvenca rotorja in pulzacija toka na izstopu rotorja s frekvenco lopatic.

3 CONCLUSIONS AND COMMENTARY

In almost all tested operating regimes the first higher harmonic of the impeller speed frequency (I1) and the first higher harmonic of the blade frequency (B2), are dominated by the amplitude in the power-spectra records. Therefore, the dominant sources of the operating noise and casing vibrations are the impeller speed and pulsating flow with the blade frequency at the impeller exit diameter.

Amplituda obratovalnega hrupa in vibracij okrova pada z manjšanjem pretoka skozi črpalko in manjšanjem vrtilne frekvence rotorja.

Ker v močnostnem spektru prevladujeta prva višja harmonika vrtilne frekvence in frekvence lopatice, lahko povzamemo, da je za zmanjšanje obratovalnega hrupa treba znižati vrtilno frekvenco rotorja. Če želimo dosegati zahtevano črpalno višino pri znižani vrtilni frekvenci, moramo povečati izstopno hitrost. To lahko dosežemo na dva načina: prvič: s povečanjem izstopnega kota rotorske lopatice, ki pa je na žalost omejena s trdnostjo materiala na 25° do 40°; drugič: s povečanjem izstopnega premera, ki pa je omejen z optimalnim razmerjem vstopnega in izstopnega premera. Tako moramo poiskati najugodnejšo rešitev z optimizacijskim procesom.

With a capacity and impeller speed decrease the noise and casing-vibration amplitudes also decrease.

Finally, in almost all power-spectra records the first higher harmonics of the impeller speed frequency and of the blade frequency dominate. Therefore, for minimizing operating noise the impeller speed must be decreased. However, if the required head and the capacity at lower impeller speed is to be satisfied, the flow velocities at the impeller exit must be increased. There are two possibilities to achieve this: first, by increasing the impeller-blade exit angles, but unfortunately they are limited in the range of 25 to 40 degrees by material strength; and second, by an exit diameter increase, which is also limited by an optimum intake / exit diameter ratio. So, for the best solution the optimising process must be used.

4 SIMBOLI
4 SYMBOLS

konstanta (funkcija tipa črpalke)	a_p	-	constant (function of the type of the pump)
sferična površina	A_s	m^2	spherical surface area
hitrost zvoka	c_0	m/s	acoustic speed
absolutna hitrost toka na izstopnem premeru	c_2	m/s	absolute flow velocity at exit diameter
premer rotorja	D	m	impeller diameter
izstopni premer rotorja	D_2	m	impeller exit diameter
frekvenca	f	Hz	frequency
črpalna višina	H	m	pump head
zvočna intenzivnost	I	W/m^2	acoustic intensity
referenčna zvočna intenzivnost	I_{ref}	W/m^2	reference acoustic intensity
stopnja zvočne intenzivnosti	L_I	-	sound intensity level
stopnja zvočne jakosti	L_n	-	sound power level
stopnja zvočnega tlaka	L_S	-	sound pressure level
število vrtljajev rotorja	n	l/s	impeller speed
preračunsko število vrtljajev rotorja	n_{des}	l/s	impeller design speed
zvočni tlak	p	Pa	acoustic pressure
vzbujevalna tlačna napetost	$\|p\|$	Pa	exciting stress
dinamični tlak	p_d	Pa	dynamic pressure
referenčni tlak	p_{ref}	Pa	reference pressure
zvočna moč	P	W	sound power
sevajoča zvočna moč	P_{rad}	W	radiated sound power
referenčna zvočna moč	P_{ref}	W	reference sound power
prostorninski pretok	Q	m^3/s	capacity
najmanjši prostorninski pretok	Q_{min}	m^3/s	minimum capacity
optimalni prostorninski pretok	Q_{opt}	m^3/s	optimum capacity
nadoptimalni prostorninski pretok	Q_{00}	m^3/s	over optimum capacity
čas	t	s	time
opazovan časovni trenutek	t_0	s	observing time
čas periode	T	s	period time
neustaljena, oscilirajoča komponenta hitrosti	u	m/s	unsteady, fluctuating part of velocity
hitrost v točki tekočinskega toka	U	m/s	velocity at a point in fluid flow
povprečna hitrost	$\bar{U}$	m/s	average value of velocity
vršna hitrost	U_T	m/s	tip speed
padec tlaka	Δp	Pa	pressure drop
gostota	ρ_0	kg/m^3	density

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Prejeto:
Received: 15.8.2000

Sprejeto:
Accepted: 10.11.2000

© Strojniški vestnik 46(2000)8,549-556
ISSN 0039-2480
UDK 697.3:621.694:620.92(430)
Pregledni znanstveni članek (1.02)
© Journal of Mechanical Engineering 46(2000)8,549-556
ISSN 0039-2480
UDC 697.3:621.694:620.92(430)
Review scientific paper (1.02)

Kombinirano daljinsko ogrevanje in hlajenje v mestu Gera (Nemčija) s tehnologijo parnih kotlov

A Combined District Heating and Cooling Network in City of Gera (Germany) using Steam-Jet Ejector Technology

Peter Noeres - Daniel Hölder - Wilhelm Althaus

Prispevek predstavlja tehnologijo parnega ejektorja in njegovo uporabo v primeru pilotnega projekta oskrbe z energijo Gera v Nemčiji (ustanovitelj Nemško ministrstvo za znanost in izobraževanje). S tem projektom je bila izdelana hladilna naprava s parno ejektorsko hladilno enoto in kompresorskim hladilnikom, ki oskrbuje daljinsko hlajenje v središču mesta Gera. V prispevku bo prikazan tehnični koncept hladilnega obrata, karakteristike parnega ejektorja, prejšnje izkušnje iz obratovanja in področja uporabe v prihodnosti.

(Ključne besede: ogrevanje daljinsko, hlajenje daljinsko, ejektorji parni, stroški obratovalni)

The paper introduces steam-jet ejector technology and its use in a pilot project of Energieversorgung Gera GmbH in Germany (funded by the German Ministry of Science and Education). Within this project a chiller plant with a steam-jet ejector chiller unit and a compression chiller supplies a district cooling network in downtown Gera, Germany. The technical concept of the chiller plant, performance characteristics of the steam-jet ejector, previous operational experiences and future areas of application will be described.

(Keywords: district heating, district cooling, steam ejectors, operating costs)

0 UVOD

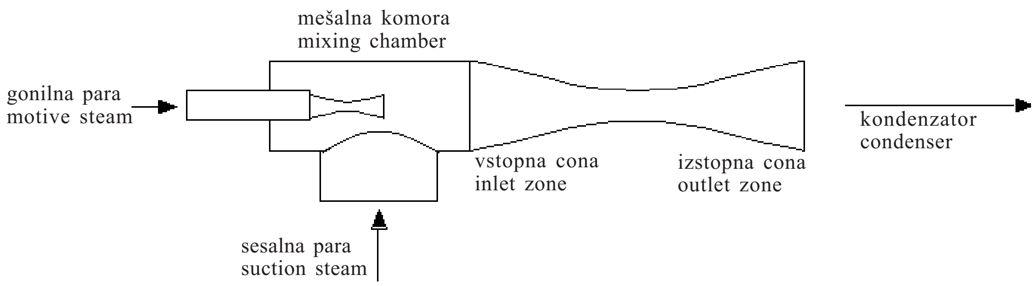
Kot del integralne energijske oskrbe toplotno vodeni hladilniki vode nudijo veliko prednosti v primeru kombinirane proizvodnje toplote, hladu in moči (KPTHM - CHCP) v povezavi z daljinskim ogrevanjem ali kombinirano proizvodnjo toplote in elektrike. Izboljšana uporaba in s tem tudi gospodarnko obratovanje je mogoče s toplotno gnanimi hladilniki še posebej v poletnih mesecih. V nasprotju z električno gnanimi kompresorji hladilnikov je v primeru kombinirane proizvodnje elektrike in toplote (KPET - CHP) mogoče zmanjšanje uporabe primarne energije. Nadalje je mogoče izbrati naravna hladiva kot sta voda in amoniak, od katerih nobeden ne prispeva k učinku tople grede. Visoki investicijski stroški hladilnih obratov s toplotno gnanimi hladilnimi procesi in njihove karakteristike so glavni problem KPTHM v Nemčiji. Začetek obratovanja hladilnega obrata je bil v letu 1997. Dobava hladu je bila začeta januarja leta 1998.

0 INTRODUCTION

As part of an integrated energy supply thermally driven chilled water generation offers many advantages within the context of a combined heat, cold and power supply (CHCP) in connection with district heating or block heat and power plants. An improved utilisation, and thus economical operation of CHP plants becomes possible with thermally driven chillers, particularly in the summer months. In contrast to chilled-water generation with electrically driven compression chillers, a reduction of primary energy demand can be realised by the use of heat from CHP plants. Furthermore, natural refrigerants such as water or ammonia can be chosen, none of them contributing to the greenhouse effect. However, the high investment costs for chiller plants with thermally driven refrigeration processes and their performance characteristics are a problem regarding the realisation of CHCP in Germany. The operation of the chiller plant started in early 1997. The cold supply started in January 1998.

1 NASTAJANJE HLADU S SISTEMOM PARNEGA EJEKTORSKEGA HLAJENJA

Manj znani hladilni proces je parni ejektorski hladilnik. Njegova uporaba v kemijskih procesih kot hladilna naprava ali vakuumska črpalka je že dolgo poznana. Do sedaj je njegova uporaba v kombinaciji s KPTHM realizirana le v primeru dveh demonstracijskih projektov na Danskem in v Nemčiji. V tej tehnologiji je mehanski kompresor nadomeščen s parnim ejektorjem (sl.1).



Sl. 1. *Model parnega ejektorja*

Proces je sestavljen iz dveh krogov, srednjega gonilnega kroga in hladilnega kroga. Tlačna energija iz gonilnega kroga se spremeni v kinetično energijo v gonilni šobi parnega ejektorja. Para hladiva je vodena vanj iz uparjalniške mešalne komore. Tlak pare v uparjalniku se znižuje, dokler se hladivo uparja in hladna voda se hladi glede na potrebno hladilno moč. S to tehnologijo se lahko dosežejo tudi temperature pod 0 °C.

V difuzorju se gibalna energija gonilne pare in hladilne pare spremeni nazaj v tlačno energijo. Celotni masni tok je stisnjen na tlačni nivo glede na tlak uparjanja vode pri hladilnih temperaturah. Para kondenzira v pršilnem kondenzatorju. Toplota kondenzacije se odvaja v hladilnem ciklusu z uporabo ploščnega prenosnika toplote, kondenzat pa vodi nazaj v uparjalnik ali v zbiralnik daljinskega ogrevalnega sistema.

Ker je hladivo voda, lahko proces razdelimo v parni, hladno vodni in hladilno vodni sistem. Pršilni kondenzator in uparjalnik (odprt ali pol odprt proces) se lahko uporabita namesto prenosnikov toplote. Zaprt proces lahko uporabljamo tam, kjer je delitev narejena s prenosniki toplote.

Karakteristike delovanja parnega ejektorskega hladilnega sistema so drugačne od drugih toplotnih hladilnih sistemov. Hladilno število (HŠ - COP) je močno odvisno od tlaka kondenzatorja in s tem od temperature hladilne vode. To definira temperatura zraka in delovanje hladilnih stolpov. Poleti pri veliki vlagi v zraku in visokih zunanjih temperaturah

1 COLD GENERATION WITH STEAM-JET REFRIGERATING SYSTEMS

The steam-jet ejector chiller (SJEC) is a little-known refrigeration process. Its use in the area of chemical processes as a cooling device or vacuum pump has long been known. Until now, its use in combination with CHCP, however, has only been realised within the context of two demonstration projects in Denmark and Germany. In SJEC technology the mechanical compressor of a compression chiller is replaced by a steam-jet ejector (see figure 1).

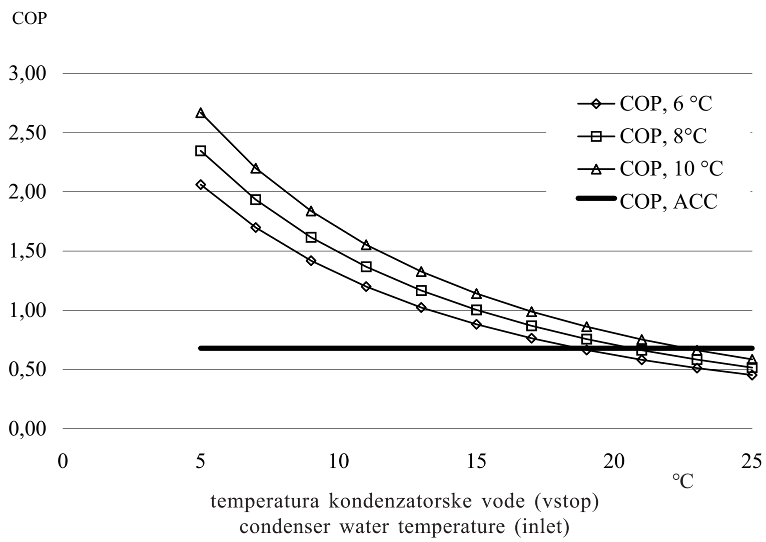
Fig. 1. *Model of a steam-jet ejector*

The process consists of two cycles, the motive medium cycle and the refrigerant cycle. The pressure energy of the motive steam is converted into kinetic energy in the motive steam nozzle of the a steam-jet ejector. Steam of the refrigerant is drawn in from the evaporator of the adjoining mixing chamber by momentum exchange. The steam pressure in the evaporator is lowered as long as the refrigerant evaporates and the cold water cools down according to the required chiller capacity. Temperatures below 0 °C can be realised with this technology.

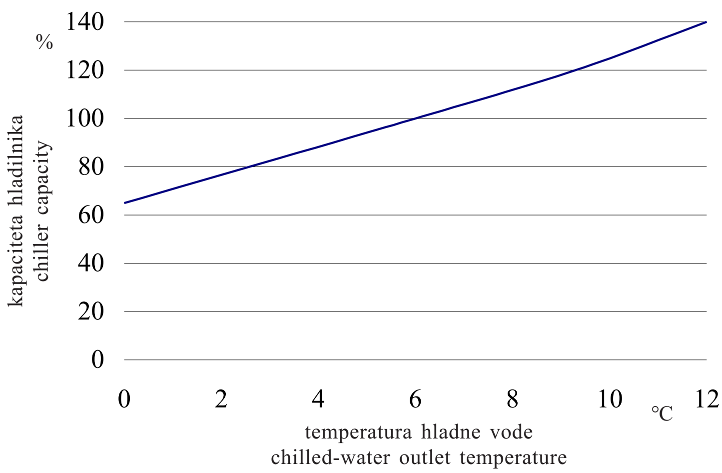
In the diffusor the motive energy of the motive steam and the refrigerant steam is converted back into pressure energy. The complete mass flow is compressed to the pressure level according to the saturation pressure of water at the re-cooling temperatures. Vapor is condensed in a spray condenser. The heat of condensation is transferred to the re-cooling cycle by a plate heat exchanger and the condensate is led back into the evaporator or into the condensate collecting pipe of the district heating system.

Since the refrigerant is water, one can omit a separation of the process into vapor, cold water, and re-cooling water systems. Spray condensers and flash evaporators (open or half-open processes) can be applied instead of heat exchangers. However, a closed SJEC process can be conceived as well, where a separation from the periphery is done by the use of a heat exchanger.

The performance characteristics of steam-jet refrigerating systems are different from that of other thermally driven chiller systems. The coefficient of performance (COP) depends very strongly on the condenser back-pressure and thus on the re-cooling water temperature. This is determined by the state of the ambient air and by the performance of the cooling towers. In mid-



Sl. 2. *Hladilno število (HŠ)*
Fig. 2. *Coefficient of performance (COP)*



Sl. 3. *Hladilna moč parnega ejektorskega hladilnega procesa*
Fig. 3. *Chiller capacity of a steam-jet ejector chiller*

je hladilno število HŠ parnega ejektorskega sistema (PES - SJEC) nižji od običajnih absorpcijskih hladilnih sistemov (AHS - ACC). Prek leta pa temperatura hladilne vode leži pod temi vrednostmi, tako da lahko dosežemo HŠ > 1. Posledica tega je, da z nižjo energijo lahko pozitivno vplivamo na obratovalne stroške takega hladilnega sistema. Slika 2 kaže ta dejstva za različne razmere delovanja (podatki za hladilni sistem Brückenski ulici).

2 HLADILNI OBRAT BRÜCKEN V MESTU GERA

Da bi bolje uporabili sedanjo parno mrežo in nov obrat za proizvodnjo električne energije in toplote (KPET), je bil Fraunhofski UMSICHT določen, naj razišče prednosti kombinirane dobave toplotne energije in hladu. Od leta 1996 je v obratovanju nov postroj KPET (P_{el} = 76 MW, Q_t = 140 MW). Omrežje

summer with high air humidity and high outside temperatures, the coefficient of performance of an SJEC is lower than the COP of ACCs. Over the period of a year the recooling water temperatures, however, lie well below these conditions of the design case so that an average COP > 1 can be reached. As a consequence, the lower demand of driving energy positively effects the running costs of the chiller plant. Figure 2 shows these facts for different operating conditions (data for the Brückenstraße chiller plant).

2 THE CHILLER PLANT BRÜCKENSTRASSE IN CITY OF GERA

In order to use the existing steam network and the new CHP plant (gas turbine combined cycle) of EGG in an improved way, Fraunhofer UMSICHT was engaged to investigate the benefits of a combined district heating and cooling supply. Since 1996 the new CHP plant (P_{el} = 76 MW_{el}, Q_{th} = 140 MW_{th})

daljinskega sistema povezuje 244 MW in ga sestavlja parni del (primarni del) in vročevodni del (sekundarni del). Obe omrežji sta povezani z 12 postajami. Glede na sedanje stanje (parni sistem v središču mesta) je tehnologija parnega ejektorskega hlajenja zelo obetajoča.

Uvodne raziskave so obsegale izračun potrebe po hladu in ocenitev posledic daljinskega ogrevanja in hlajenja na proizvodnjo električne energije in toplote (KPET). Z uporabo teh podatkov je bila izračunana potreba po hladu v mestu Gera.

Na podlagi teh podatkov so na Fraunhofer UMSICHT začeli z načrtovanjem hladilnega sistema s hladilno močjo 1,2 MW. Na podlagi rezultatov je Fraunhofski UMSICHT prišel do sklepov, da je uporaba parnega hladilnega sistema tehnično mogoča, stroškovno učinkovitejša v primerjavi z drugimi hladilnimi sistemi in tudi gospodarnejša.

Celotna hladilna zmogljivost hladilnega sistema se doseže s parnim ejektorjem in kompresijskim hladilnikom, vsakim s po 600 kW_t. Ta delitev na eni strani zagotavlja optimalno delovanje in na drugi visoko zanesljivost delovanja. Ejektorski hladilni sistem (PES) sestoji iz dveh delov, od katerih vsak vsebuje tri parne ejektorje. Za zmanjšanje količine gonilne pare je vstavljen parni ventil, ki krmili tlak pare in kot posledico porabo pare glede na tlak v kondenzatorju. Preglednica 1 podaja povzetek podatkov sistema PES:

has been in operation. The district heating network has a connected load of 244 MW and consists of a steam net (primary net) and a hot-water net (secondary net). Both nets are connected by 12 transfer stations. Due to the favorable prerequisites (a steam net up to the downtown area) the use of steam-jet ejector chiller technology was very promising.

The preliminaries for the development of a strategy included a cold demand survey and an estimation of the consequences of a combined district heating and cooling supply for the operation of the CHP plant. With this data the cold demand in downtown Gera was determined.

On the basis of this data Fraunhofer UMSICHT was entrusted with the planning of the first chiller plant with an installed chiller capacity of 1.2 MW_{th}. As a result of the basic engineering Fraunhofer UMSICHT came to the conclusion that the use of a steam jet refrigerating system is technically possible and cost-effective compared to other chiller systems.

The total chiller capacity of the chiller plant is covered by a steam-jet ejector chiller unit and a compression chiller with 600 kW_{th} each. This partitioning ensures optimal operating conditions for the SJEC and a high supply guarantee for the chiller. The SJEC consists of two stages, each of them with three steam-jet ejectors. To reduce motive-steam demand a motive-steam control valve is used to control the motive-steam pressure and, as a consequence, the steam consumption, depending on the condenser back-pressure. Table 1 gives a summary of the design data of the SJEC:

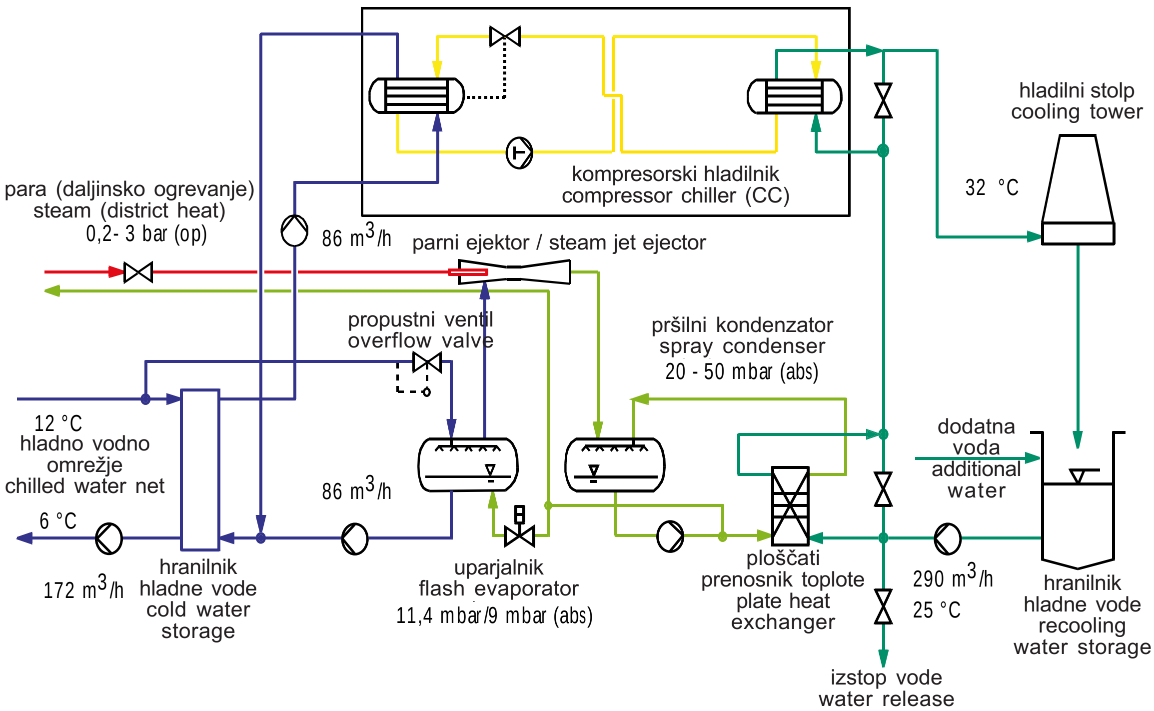
Preglednica 1. *Imenski podatki delovanja Parne ejektorske hladilne enote v mestu Gera*
Table 1. *Nominal Operation Data of the Steam-Jet Ejector Chiller Unit in City of Gera*

hladilna moč PES chiller capacity SJEC	600 kW_{th}
temperatura hladne vode chilled water temperature	6 / 12 °C
temperatura hladilne vode (dotok odtok) re-cooling water temperatures (supply line / return line)	25 / 30 °C
temperatura okolice wet bulb temperature (design)	21 °C
parametri gonilne pare motive steam parameter (district heat)	143 °C, 3 bar
HŠ (imensko, povprečje) COP (nominal, assumed average)	0,55 / 1

Hladilni postroj je shematično prikazan na sliki 4. Dva hladilnika sta povezana vzporedno na strani hladne vode. Hidravlično sta povezana s hladnovodnim omrežjem in s hranilnikom hladne vode, tako da je prostorninski tok skozi uparjalnik kompresorskega cikla (KC - CC) in skozi uparjalnik parnega ejektorskega cikla (PES) nadzorovan neodvisno od prostorninskega toka do hladilnega obrata. Kondenzatorja PES in KC sta vezana zaporedno na hladno vodo, tako da dosežemo najmanjši pretok hladne vode. Parni ejektorski hladilni

The chiller plant is shown schematically in Figure 4. The two chillers are connected in parallel on the cold-water side. They are decoupled hydraulically from the chilled water net by a cold water storage so that the volume flows through the evaporator of the CC and the flash evaporator of the SJEC can be controlled independently of the net volume flow to the chiller plant. The condensers of the SJEC and the CC are switched in series on the re-cooling water side to obtain a re-cooling water volume flow as low as possiproces (PES) je neposredno povezan z omrežjem EGG daljinskega hladilnega sistema in omrežjem parnega sistema. Hladilni krog, ki vsebuje hladilne stolpe, je bil hidravlično povezan s kondenzatorjem s ploščnimi prenosniki toplote.

ble. The SJEC is linked directly to EGG's district cooling system and district heating steam network. The re-cooling cycle containing the cooling towers was decoupled hydraulically from the condenser water cycle of the SJEC by a plate heat exchanger.



Sl. 4. *Shema procesov hladilnega obrata*

Fig. 4. *Process-Flow Scheme of the Chiller Plant*

3 EKONOMSKA OCENA

Za ekonomsko oceno toplotno gnanih hladilnih procesov je treba upoštevati tako investicijske kakor tudi obratovalne stroške. Specifični investicijski stroški parnega ejektorskega hladilnega procesa (PES), vključno z dodatki za pilotni obrat v Geri so – za hladilni obrat v Brückenski ulici – malo višji kot stroški za absorpcijski hladilni sistem pri istih pogojih delovanja. V tej točki je treba upoštevati, da so za pilotni objekt stroški za proizvodnjo in načrtovanje razmeroma visoki. Poleg tega je bilo potrebno na pilotnem objektu izvesti številne meritve in nadzora. Izbran PES je samo en izveden hladilni obrat. V nadaljevanju je pričakovati nižje investicijske stroške v primerjavi z absorpcijskih sistemom (AHS) za enake robne pogoje, ki jih lahko dosežemo z masovno proizvodnjo in nadaljnjim optimiranjem hladilnega sistema.

Za ekonomsko oceno je treba upoštevati tudi obratovalne stroške. V tem primeru je prednost PESa zaradi večjega letnega deleža delovanja (HŠ). Specifično vrednost lahko izračunamo iz podatkov hladilne obremenitve in stanja zraka (vlažnosti in temperature). Za primer Gera naj bi letni delež delovanja dosegel vrednost približno 1. Kot

3 ECONOMIC EVALUATION

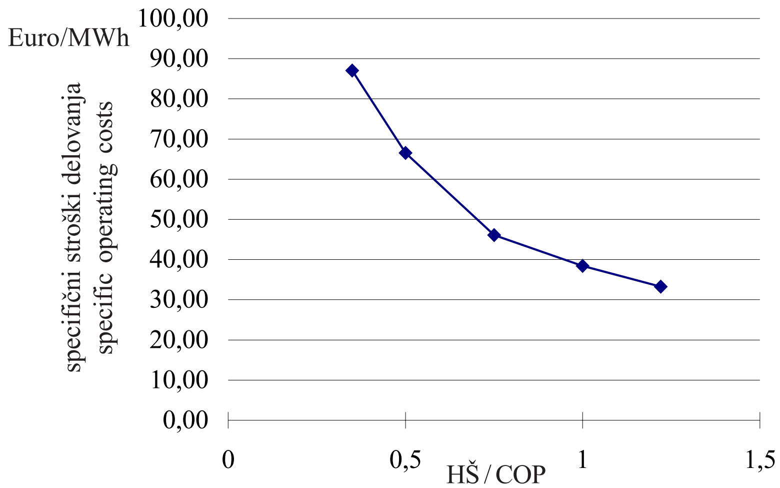
For an economical evaluation of thermally driven refrigerating processes both the investment costs and the running costs have to be considered. The specific investment costs of the SJEC including auxiliaries for the pilot plant in Gera are, for the chiller plant Brückenstraße, a little higher than the costs of an absorption chiller under the same operating conditions. At this point it has to be taken into account that for a pilot plant the expenditures for production (single manufacturing) and planning were still relatively high. Additionally, a lot of measuring and control techniques were equipped in the pilot plant. Furthermore, the chosen SJEC is a single manufactured chiller. In the future, lower investment costs compared to an ACC have to be expected for equal boundary conditions, which could be made possible by series production and further optimization in chiller-plant design).

For the economic evaluation of cold generation the running costs have to be taken into account as well. Here, advantages for the SJEC technology arise due to the larger annual mean COP. This specific value can be calculated from data on cooling-load duration curves and the state of the ambient air (humidity and temperature). For the supply case in Gera the annual mean COP was supposed to achieve a value of rezultat pilotnega objekta in obratovalnih izkušenj je HŠ od 0,9 do 1 in je ugodna ocena za nadaljnje uporabe (število HŠ je za absorpcijske klimatske naprave (AHS) pogosto slabši od 0,6). Za izračun obratovalnih stroškov toplotnih hladilnih procesov uporabljajo naslednje predpostavke, ki so tipične za pridobivanje hladu v Nemčiji.

approximately 1. As a result of the pilot plant and the operational experiences an annual mean COP of 0.9 to 1 is a suitable estimation for further application (in reality the COP number of ACCs is often worse than 0.6). To calculate the running costs of thermally driven refrigeration processes the following assumptions are chosen, as these are typical for a cold supply in Germany.

Preglednica 2: *Podatki za izračun obratovalnih stroškov (samo kot primer)*
Table 2. *Preliminaries to calculate operating costs (only as an example)*

količnik f (razmerje med vodovodno in odpadno vodo) factor f (ratio of tap water to waste water)	3
poraba elektrike na kW hladilne moči electricity demand per kW re-cooling demand	0,025 kW_{el}/kW_{th}
voda (vključno s pripravo, kemični stroški) water (including make-up, chemical costs)	1,02 Euro/m^2
odpadna voda waste water	1,53 Euro/m^3
energija, elektrika energy rate, electricity	0,071 Euro/kWh
cena elektrike demand price tariff, electricity	102 Euro/kW/a
ure dobave hladu pri polni obremenitvi full load hours cold supply	1000 h/a
obratovalne ure dobave hladu operating hours cold supply	5000 h/a
specifični stroški toplote specific costs of heat	20,47 Euro/MWh



Sl. 5. *Specifični stroški obratovanja toplotno gnanih hladilnikov*
Fig. 5. *Specific operation costs of thermally driven chillers*

Ugotovimo močno odvisnost obratovalnih stroškov od HŠ. S parametri za EU-AHS (HŠ = 0,6) lahko napovemo specifične obratovalne stroške približno 60 Euro/MW_t. Specifični stroški obratovanja za PES z letnim HŠ 0,9 (imensko delovanje 0,55) so okoli 25% nižji. Če povzamemo, lahko ugotovimo nižje stroške (obratovalne in investicijske) za dane karakteristike porabnikov.

We see the strong dependence of running costs from the COP. With the parameters for an SE-ACC (COP=0.6) given here we have to assume specific operating costs of approximately 60 Euro/MWh_{th}. The specific operating costs of an SJEC with an annual mean COP of 0.9 (nominal operating case 0.55), however, are around 25% lower. In summary, a lower total cold-supply cost (operating and capital costs included) can be realized for the given supply characteristics of the customers.

4 IZKUŠNJE DELOVANJA

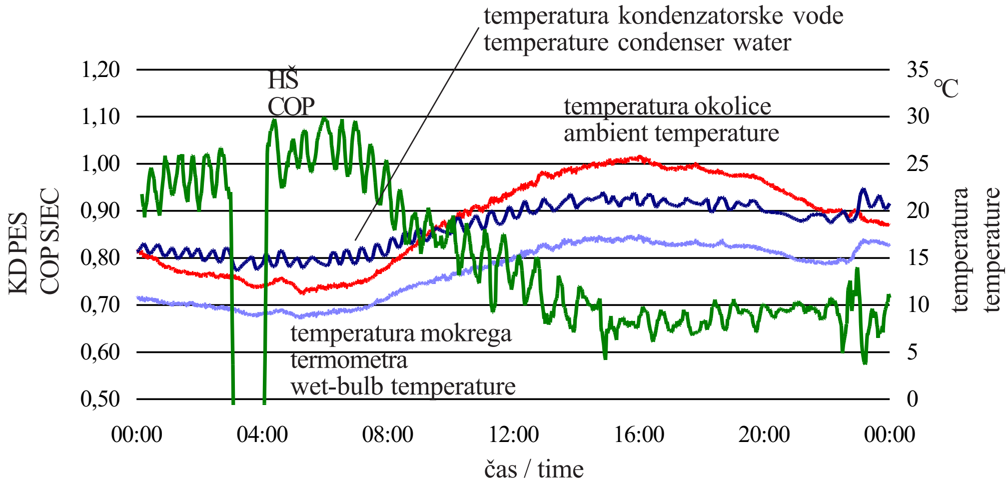
Hladilni obrat deluje dve leti. V tem času ni prišlo do nobene nezgode ali motnje pri dobavi hladne vode. Poraba hladne vode je bila pričakovana gledano predvsem letno (sl. 6). Karakteristike delovanja parnega ejektorja so bile zelo dobre, predvsem kakovost nadzora hladne vode. Zadnje leto je bilo HŠ 0,62, kar je malo slabše od pričakovanega 1. Razlog za manjše HŠ je nižja temperatura povratne hladne vode, vmesno odpravljanje ovir krmiljenja gonilnega parnega toka in ne optimalna konstrukcija ejektorjev.

Z instaliranim sistemom krmiljenja gonilne pare ni bilo mogoče znižati tlaka pare pod 0,2 bar (sl. 7). Zaradi tega do tedaj ni bilo mogoče uporabiti odličnih pogojev delovanja parnega ejektorja v zimskem času in ob spremembah. Za to leto pričakujemo boljše obratovalne rezultate zaradi boljše (oz. višje) temperature hladne vode, ker so

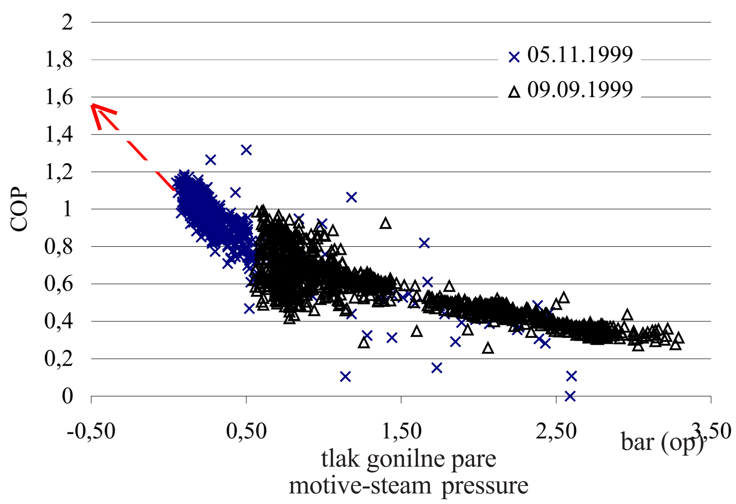
4 OPERATIONAL EXPERIENCES

The chiller plant has been in operation for two years. During this period no accidents or interruptions of the chilled-water supply occured. The chilled-water demand corresponds to the expectations, especially concerning the year-round base demand for chilled water (Fig. 6). The operational characteristics of the steam-jet ejector are very good, especially the quality of the chilled-water supply control. For the last year the mean COP was 0.62, which is a little worse than the expected value of about 1. The reason for the reduced COP was the lower chilled-water return temperatures, in the meantime cleared constraints of motive steam flow control and a non-optimal design of ejectors are suspected.

With the installed motive-steam control systems it was not possible to lower the motive-steam pressure below 0.2 bar(op) (Fig. 7). Because of this it was not possible, until now, to use the excellent operation conditions of a steam-jet ejector in winter time and transfer period in a complete matter. For this year we expect better performance data due to a better (or higher) chilled-water return-line temperature, a new



Sl. 6. *Hladilni obrat v Brückenski ulici – HŠ in različne obratovalne temperature (podatki 19.5.1999)*
Fig. 6. *Chiller Plant Brückenstraße – COP and different operation temperatures (data 19.05.1999)*



Sl. 7. *Hladilnik v Brückenski ulici – HŠ v odvisnosti od tlaka pare (podatki 19.5.1999 in 09.09.1999)*
Fig. 7. *Chiller Brückenstraße – COP as function of motive steam pressure (data 05.11.1999 and 09.09.1999)*

bili januarja letos vgrajeni nov krmilni sistem gonilne pare in parne šobe.

5 SKLEP

Izkušnje delovanja obrata iz prvih dveh let so zelo ugodne. Med prvo fazo delovanja in prvo fazo optimiranja so bili začetni problemi zelo hitro rešeni. Neučinkovito delovanje, ki je zelo znano pri večini toplotno gnanih hladilnih procesih, ni bilo ugotovljeno pri parnem hladilnem procesu. Dinamično obnašanje PES je primerljivo z drugimi običajnimi kompresorskimi hladilniki. V tem trenutku poteka optimizacija delovanja hladilnika predvsem HŠ in krmilnega sistema izklopa.

motive-steam control system and modified motive-steam nozzles, which were changed in January of this year.

5 CONCLUSION

The experience from the first two years of the plant's operation are very positive. During the primary phase of operation and the first optimization phase the initial operational problems of the pilot plant could be solved very quickly. The inert operating response, well known from many heat-driven refrigerating processes was not observed for the steam-jet refrigeration system. In their dynamic behaviour SJECs are comparable to conventional compression chillers. At the moment the optimization of chiller operation, especially COP and switch-off control systems is in progress.

6 SIMBOLI
6 SYMBOLS

absolutno	abs	absolute
absorpcijski hladilni krog	AHS - ACC	absorption chiller cycle
kompresorski hladilni krog	KHK - CCC	compression chiller cycle
kombinirano daljinsko ogrevanje in daljinsko hlajenje	KDODH - - CDHDC	combined district heating and district cooling
kombinirana proizvodnja toplote, hladu in elektrike	KPTHM - - CHCP	combined heating, cold, and power
kombinirana proizvodnja toplote in elektrike	KPET - CHP	combined heat and power
hladilno število	HŠ - COP	coefficient of performance
daljinsko ogrevanje	DO - DH	district heating
razmerje dodatne vode	f	ratio additional water to released water
nadtlak	op	over-pressure
enojni učinek	EU - SE	single effect
parni ejektorski hladilnik	PES - SJEC	steam jet ejector chiller
temperatura	T (°C)	temperature

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Prejeto:
Received: 15.8.2000

Sprejeto:
Accepted: 10.11.2000

© Strojniški vestnik 46(2000)8,557-563
ISSN 0039-2480
UDK 697.97:621.574.013
Pregledni znanstveni članek (1.02)
© Journal of Mechanical Engineering 46(2000)8,557-563
ISSN 0039-2480
UDC 697.97:621.574.013
Review scientific paper (1.02)

Energetska učinkovitost daljinskega hlajenja za klimatizacijo prostorov

The Energy Efficiency of District Cooling for Space Conditioning

Alojz Poredoš

Oskrba uporabnikov z grelno in hladilno energijo iz daljinskih energetskih sistemov bistveno prispeva k smotrni porabi energije in varovanju okolja. V preteklih nekaj letih se je uporaba daljinskega hlajenja v nekaterih državah pomembno zvečala.

Uporaba absorpcijskih hladilnikov za daljinsko hlajenje se v zadnjem desetletju prav tako povečuje. Te hladilne naprave potrebujejo vir toplote za svoje delovanje. To je lahko plin, kurilno olje, para ali vroča voda. V članku so podani rezultati raziskave vplivnih parametrov na specifične izgube toplote v vročevodnem omrežju in celotni učinek daljinskega hladilnega sistema.

(Ključne besede: hlajenje daljinsko, naprave hladilne, učinkovitost energijska, izgube toplotne)

Supplying customers with heating and cooling energy from district energy systems contributes to the rational use of energy and to environmental protection. In the past few years, the use of district cooling has significantly increased in some countries.

In the last decade, the use of absorption chillers for district cooling has increased. These chillers need a heat source for their operation. It can be gas, fuel oil, steam or hot water. This paper presents the results of research on the parameters which influence the specific heat losses in a district heating network and the overall efficiency of the system.

(Keywords: district cooling, absorption chillers, energy efficiency, heat losses)

0 UVOD

Dvig ravni delovnih in bivalnih pogojev ljudi je nujno povezan s hlajenjem v okviru klimatizacije. V te namene se uporablja različna naključno izbrana oprema različnih proizvajalcev brez ustreznega zagotavljanja učinkovitosti ter tehnične in okoljne neoporečnosti. Posledica tega je povečana poraba elektrike kot najbolj kvalitetne energije.

Rešitev tovrstnih problemov se ponuja z daljinskimi hladilnimi sistemi. Rezultati številnih študij v svetu kažejo, da imajo v primerjavi z lokalnim hlajenjem z uporabo električne energije daljinski hladilni sistem številne prednosti.

Daljinski hladilni sistemi so dokazano okolju bolj prijazni kot posamezne manjše hladilne enote. Z njimi zmanjšujemo oddajo škodljivih snovi, z uporabo sorpcijske tehnike hlajenja pa lahko povsem izločimo ozonu škodljiva hladiva. Kadar so hladilne naprave gnane s toploto iz kogeneracije, neposredno zmanjšu-

0 INTRODUCTION

An increase in the quality of working and living conditions is inevitably connected with cooling as part of air-conditioning systems. Various types of equipment produced by different manufacturers are used for this purpose, they are selected at random and do not provide any assurance as to efficiency and compliance with technical and environmental standards. This has resulted in an increase in the consumption of electrical energy: the energy of the highest quality.

A solution tor such problems is offered by district cooling systems. The results of numerous studies in the world have shown that in comparison with local cooling using electrical energy, district cooling systems have numerous advantages.

District cooling systems have been proven to be more environmentally friendly than individual smaller cooling units. They reduce the emissions of hazardous substances and, with the use of a sorption cooling technique, ozone-unfriendly refrigerants can be eliminated jemo toplotno obremenitev okolja, z uporabo plina za njihov pogon pa odpadejo vmesne pretvorbe energije in dodatne izgube [1].

Odločitev za uporabo absorpcijskih ali kompresorskih hladilnih naprav je odvisna predvsem od investicijskih in obratovalnih stroškov. Investicijski stroški so nekoliko večji za absorpcijske hladilne naprave in so odvisni od tipa hladilnika ter od vrste toplotnega vira. Eden od odločilnih kriterijev izbire vrste hladilnika je lahko tudi ekološka primernost. Absorpcijske hladilne naprave delujejo skoraj neslišno. Uporabljajo tudi hladiva, kot npr. amoniak, litijev bromid in vodo ($NH_3 – H_2O$, $LiBr – H_2O$), ki imajo vpliv na nastanek tople grede (GWP) in vpliv na tanjšanjeozonske plasti (ODP) enak nič. Če upoštevamo porabo primarne energije (določeno z ekvivalentom električne energije, ki je potrebna za pogon kompresorskih hladilnih naprav), imajo nekatere absorpcijske hladilne naprave manjšo porabo, kakor prikazuje sl. 1.

Z vročo vodo in s paro gnani absorpcijski hladilniki lahko uporabljajo toploto odvzeto iz kogeneracijskega sistema. Zaradi odvzema toplote iz parne turbine v kogeneracijskem sistemu se zmanjša pridobljena električna energija.

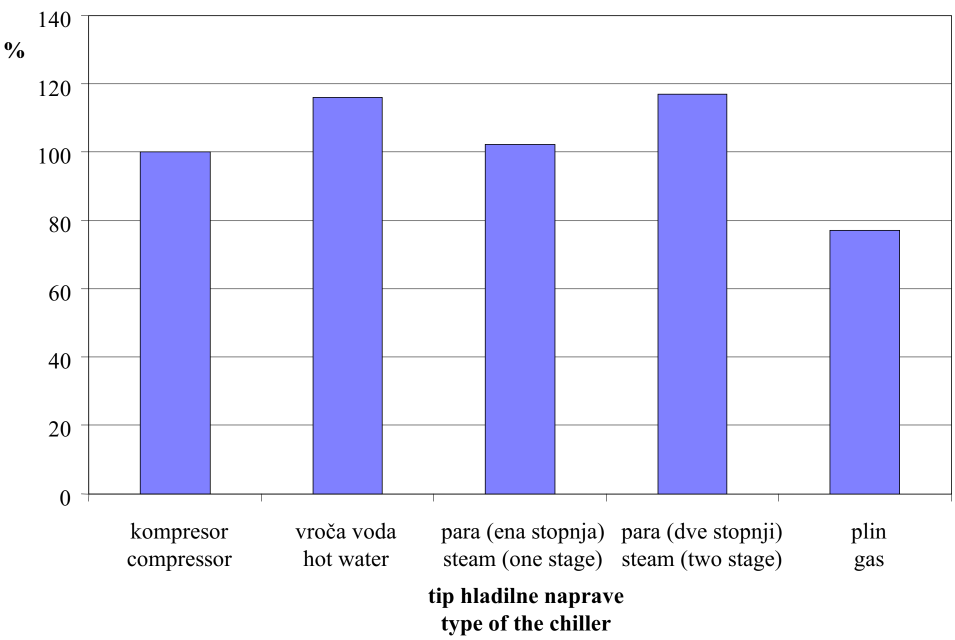
V primeru plinsko gnanih absorpcijskih hladilnih naprav je upoštevana učinkovitost plinsko gnanih turbin, ki je potrebna za določitev ustrezne električne moči. Če primerjavo različne vrste hladilnih naprav, vidimo, da predstavljajo z vročo vodo in s paro gnani dvostopenjski absorpcijski hladilniki najmanjšo porabo ekvivalenta električne energije na enoto hladu, kar je lahko tudi eden izmed načinov zmanjšanja emisij CO_2.

completely. When cooling devices are driven by heat from cogeneration, the thermal burden on the environment is reduced directly, while using gas to drive them means the intermediary stages of energy conversion and additional energy losses are avoided as well [1].

The decision, whether to use absorption chillers or compressor chillers basically depends on the investment and operational costs. The investment costs for absorption chillers can be a little higher, depending on the type of the absorption chiller and the heat source in use. One of the criteria for the decision could be also the ecological aspect. Absorption chillers are almost noiseless during operation. They also use the refrigerants, such as amonia (NH_3 - H_2O) and water ($LiBr$ - H_2O), which have a global warming potential (GWP) and ozone depleting potential (ODP) equal to zero. When the primary energy consumption is considered (defined by the equivalent electrical power which is needed for driving the compressor chillers, and which could be produced by the heat source for absorption chillers), some of the absorption chillers have a reduced consumption as shown in Figure 1.

Hot-water and steam-driven absorption chillers use heat extracted from a cogeneration plant. Because of the heat extracted from a steam turbine in a cogeneration plant, less electrical power will be produced.

In the case of a gas-driven absorption chiller, the efficiency of a gas-driven turbine is considered necessary to determine the equivalent electric power. Comparing different types of chillers, hot-water and two-stage steam-absorption chillers present the lowest equivalent electric power per unit of cold pro-



Sl. 1. *Delna hladilna obremenitev hladilnika na enoto ekvivalenta električne moči [2]*
Fig.1. *Cooling capacity share per unit of equivalent electrical power [2]*

Plinsko gnane absorpcijske hladilne naprave predstavljajo največjo porabo ekvivalenta električne energije na enoto hladu. V primerjavi z drugimi hladilnimi napravami so tudi investicijski stroški največji. Če upoštevamo, da lahko plinsko gnana absorpcijska hladilna naprava nadomesti tudi vročevodni kotel v nekem objektu, je njihova uporaba upravičena.

1 SPECIFIČNE TOPLOTNE IZGUBE

Najboljša izvedba sistemov daljinskega hlajenja je v kombinaciji z absorpcijskimi hladilniki, povezanimi v sistem daljinskega ogrevanja. Ti sistemi morajo delovati tudi v poletnem obdobju zaradi zagotovitve toplote, potrebne za ogrevanje tople sanitarne vode. To pa predstavlja razmeroma velike relativne izgube glede na količino dobavljene toplote.

Z vročo vodo gnani absorpcijski hladilniki potrebujejo za svoje normalno delovanje višje temperature, kot so običajno pri obratovanju sistema daljinskega ogrevanja v poletnem obdobju. S povišanjem temperature dovoda in povratka vroče vode se povečujejo toplotne izgube omrežja, s tem pa tudi specifične toplotne izgube ($kW_{izgube}/MW_{dobava\ toplote}$). S povečanjem odjema za potrebe hlajenja se lahko specifične toplotne izgube zmanjšajo (sl. 2).

Temperaturna razlika med dovodom in povratkom vroče vode v omrežju daljinskega ogrevanja nima vpliva samo na toplotne izgube, ampak tudi na porabo električne energije, ki je potrebna za pogon obtočnih črpalk.

Za dosego večje učinkovitosti z vročo vodo gnanih absorpcijskegih hladilnikov mora biti temperaturna razlika med dovodom in povratkom čim manjša. Povečanje temperature dovoda vroče vode

duced. This could also be a way of introducing state support in a sense of reduced CO_2 emissions.

Gas absorption chillers have the biggest equivalent electrical power consumption. Their investment costs are also higher when compared to the other chillers. As they can also replace the boiler for heat production, there is no doubt about their advantages over compressor chillers.

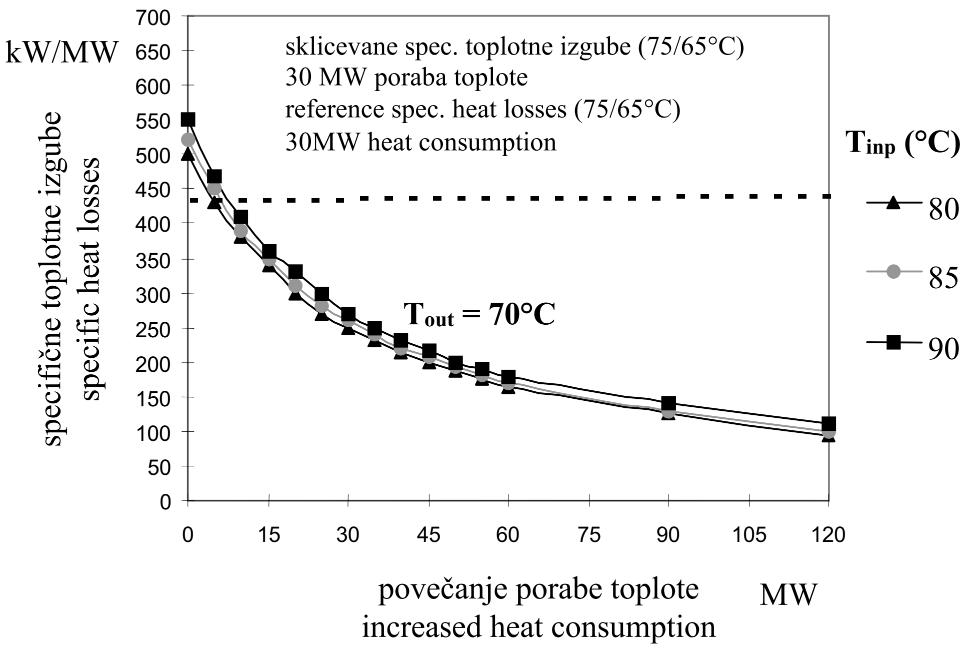
1 SPECIFIC HEAT LOSSES

District cooling systems with sorption chillers are the best when dsigned in combination with district heating systems, the majority of which also have to operate in the summer in order to ensure the supply of sanitary water. But this means a large relative loss with regard to the amount of supplied heat.

Absorption chillers, driven with hot water, need heat with higher temperatures for their normal operation as the district heating network. By increasing the supply and return temperatures of hot water, the heat losses of the district heating network are higher and so also are the specific heat losses ($kW_{losses}/MW_{heat\ supply}$). But with increasing heat consumption, the specific heat losses decrease (Fig. 2).

The supply- and return-temperature difference for a district heating network does not have an influence only on heat losses, but also on the consumption of electrical energy needed to drive network pumps.

To achieve high chiller efficiencies, the supply and return hot-water temperature difference should be low. Increased hot-water supply tempera-



Sl. 2. *Specifične toplotne izgube vročevodnega omrežja[3]*
Fig. 2. *Specific heat losses of a district heating network [3]*

in zmanjšanje temperaturne razlike pa poveča toplotne izgube v omrežju daljinskega ogrevanja.

2 UČINKOVITOST ABSORPCIJSKE HLADILNE NAPRAVE

Analiza posameznih učinkovitosti je temeljila na obratovalnih parametrih vročevodnega absorpcijskega hladilnika Carrier (16 JB 032 /036) z enojnim efektom in delovno snovjo Li-Br. Parametri obratovanja so bili:

a) temperatura hlajene vode (6/12°C)
b) temperatura hladilne vode (27/32°C)

Rezultati kažejo, da se hladilno število hladilnika povečuje z višanjem temperature dovoda vroče vode in z manjšanjem temperaturne razlike med dovodom in povratkom. Iz slike 3 je razvidno, da je razlika med največjo in najmanjšo vrednostjo hladilnega števila samo okrog 5%.

Temperatura dovedene vroče vode je omejena z najnižjo temperaturo. Pri nižji temperaturi dovoda (< 85 °C), je učinkovitost absorpcijskih hladilnikov zelo majhna in so zato specifični investicijski stroški previsoki. Pri teh pogojih se pojavijo tudi problemi v samem delovanju absorpcijskega hladilnika.

Bistvo naše analize je bila določitev temperaturnega območja obratovanja omrežja daljinskega ogrevanja, v katerem dosežemo ustrezno učinkovitost tako absorpcijskega hladilnika kot tudi celotnega omrežja daljinskega ogrevanja.

tures and a small temperature difference, however, increase the heat losses of a district heating network.

2 ABSORPTION-CHILLER EFFICIENCY

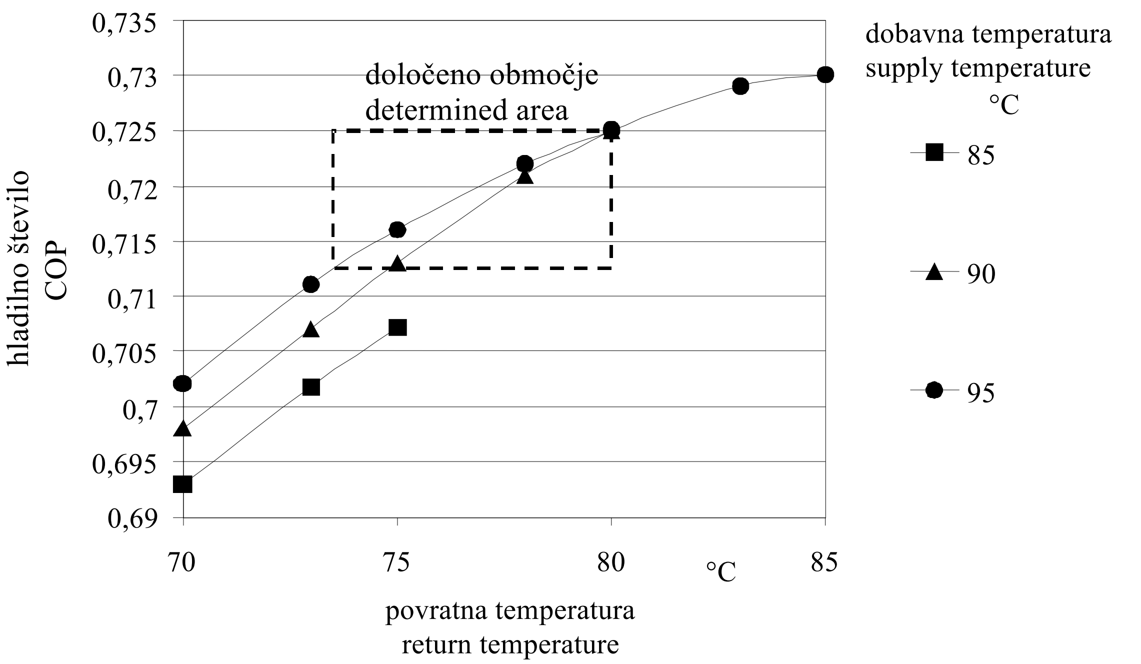
An analysis of individual efficiencies using the data from a commercially available Carrier's single-effect Li-Br absorption chiller, which is arranged for use with hot water (16 JB 032 / 036) was made. We chose further parameters:

a) chilled water temperatures (6/12°C)
b) cooling water temperatures (27/32°C)

The COP (coefficient of preformance) increases by increasing the hot-water supply temperatures and by decreasing the differences between the hot-water supply and return temperatures. Referring to Fig. 3, the difference between the highest and the lowest COP is only about 5%.

The hot-water supply temperatures were limited by the lowest temperature. At lower supply temperatures (<85°C), the efficiencies of absorption chillers are very low and the specific investment costs are too high. Besides this, the operational problems at lower hot-water supply temperatures occur.

The return hot-water temperatures were also limited by the determined temperature range. The purpose of our analysis was to determine the temperature range of a district heating network operation, at which the corresponding efficiency of both an absorption chiller and the district heating network could be achieved.



Sl. 3. *Hladilno število absorpcijskega hladilnika v odvisnosti od temperatur dovoda in povratka vroče vode [4]*
Fig. 3. *COP of the absorption chiller depending on different hot-water supply and return temperatures [4]*

3 UČINKOVITOST OMREŽJA DALJINSKEGA OGREVANJA

Izkoristek vročevodnega sistema smo definirali kot razmerje med odjemnim toplotnim tokom na strani odjemalca, ter celotnim transportiranim toplotnim tokom in električno močjo črpalk (za transport vroče vode). Toplotni ekvivalent električne moči smo definirali kot trikratno vrednost električne moči.

$$\eta = \frac{\dot{Q}_{absorpcija}}{\dot{Q}_{absorpcija} + \dot{Q}_{izgub} + 3 \cdot P_{električna}} \qquad (1)$$

$\dot{Q}_{absorpcija}$ - toplotni tok, doveden v absorpcijski hladilnik
$\dot{Q}_{izgub}$ - izgube toplotnega toka pri transportu skozi vročevod
$P_{električna}$ - električna moč črpalk za transport vroče vode

Izkoristek vročevodnega sistema smo opazovali pri različnih izgubah toplotnega toka ter različnih toplotnih močeh na strani odjemalca. Slika 4 prikazuje izkoristek vročevodnega sistema pri izgubah toplotnega toka 0,04 MW/K in odjemni toplotni moči 30 MW. Temperatura okolice je bila izbrana kot povpreček srednje dnevne temperature za Ljubljano v mesecih junij, julij, avgust in znaša 18,8°C.
Izgube v MW/K predstavljajo v enačbi :

$$\dot{Q}_{izgub} = k \cdot A \cdot \Delta T \qquad (2)$$

3 DISTRICT-HEATING-NETWORK EFFICIENCY

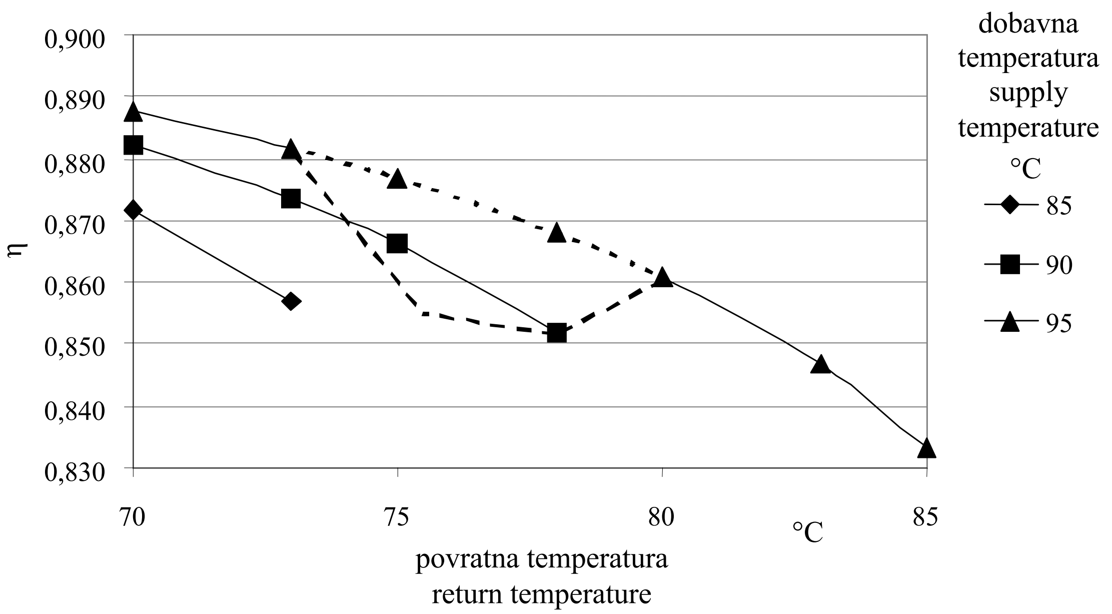
We determined the overall district-heating-network efficiency as the ratio of consumed heat, and the heat transported through the network, added to the electrical power of the network pumps (used to transport the hot water). The heat equivalent of the electrical power was defined by the three times increased value of the electrical power.

$$\eta = \frac{\dot{Q}_{absorption}}{\dot{Q}_{absorption} + \dot{Q}_{losses} + 3 \cdot P_{electric}} \qquad (1)$$

$\dot{Q}_{absorption}$ - heat flow, supplied to the absorption chiller
Q_{losses} - heat flow losses due to hot-water transport through district heating pipes
$P_{electric}$ - electric power for the pumps, which are used for transporting the hot water through the district heating network

The efficiencies of a district heating system at different heat losses and for different heat consumptions of the customers were observed. Figure 4 shows the efficiency of the district heating system at heat flow losses of 0,04 MW/K and at 30 MW of heat consumption. An ambient temperature of 18,8°C was defined as an average of the medium daily temperatures in Ljubljana in June, July and August.
In the equation:

$$\dot{Q}_{losses} = k \cdot A \cdot \Delta T \qquad (2)$$



Sl. 4. *Učinkovitost omrežja daljinskega ogrevanja v odvisnosti od različnih temperatur dovoda in povratka vroče vode pri toplotnih izgubah 0,04 MW/K in odjemu 30 MW [5]*
Fig. 4. *The efficiency of the district heating system depending on different hot-water supply and return temperatures at heat flow losses of 0.04 MW/K and the 30 MW of heat consumption [5]*

člen $k{\cdot}A$. Temperaturna razlika predstavlja razliko med temperaturo vroče vode v vročevodu in temperaturo okolice.

Ker izkoristek vročevodnega sistema z naraščanjem temperature povratka vroče vode pada (slika 4), je skoraj nemogoče določiti točko optimalnega obratovanja, saj so v interesu odjemalca vroče vode čim nižji investicijski ter obratovalni stroški absorpcijskega hladilnika, interes distributerja vroče vode pa čim nižji obratovalni stroški vročevodnega sistema.

Pri visokih temperaturah povratka vroče vode se investicijski stroški absorpcijskega hladilnika znižajo, vendar prične v tem območju izkoristek vročevodnega sistema naglo padati, kot je to prikazano na sliki 4.

Iz slike 4 je razvidno, da dosegamo najboljše izkoristke vročevodnega sistema ob čim višjih temperaturah dovoda vroče vode in čim večjih temperaturnih razlikah med temperaturama dovoda in povratka vroče vode (pri določeni temperaturi povratka). Velike temperaturne razlike omogočajo manjše pretoke vroče vode, to pa zmanjša električno moč črpalk.

Na sliki 4 je črtkano označeno področje, v katerem naj bi vročevodni sistem obratoval.

Krivulje, ki predstavljajo temperature dovoda vroče vode, se pri večjih temperaturnih razlikah približujejo neki skupni vrednosti. Iz tega lahko sklepamo, da je izbrano področje ustrezno, saj je padec izkoristka med temperaturama povratka 73 do 80°C veliko manjši v primerjavi z desno stranjo diagrama. Če povečujemo toplotno moč na strani odjemalca, se tudi izkoristek vročevodnega sistema povečuje, tendenca in razvrstitev krivulj pa ostajata zelo podobna.

Iz primerjave dveh režimov obratovanja pri enakem odjemu (30MW) in različnih izgubah (0,04 MW/K in 0,12 MW/K), dobimo razlike v hladilnih številih le okrog 3%, razlike v hladilnih močeh pa tudi največ 3%. Ta padec hladilne moči ter hladilnega števila bi lahko korigirali, če bi npr. povišali temperaturo hlajene vode absorpcijskega hladilnika s sistema 6/12°C na sistem 7/13°C.

the factor $k{\cdot}A$ represents determined values in MW/K. The temperature difference (ΔT) represents the difference between the temperature of the hot water in the pipe and of the ambient temperature.

By increasing the hot-water return temperature, the efficiency of the district heating network decreases (Fig.4), as a result, it is almost impossible to determine the optimum temperatures for the operation of a district heating network. The customer demands the lowest investment and operating costs for the absorption chiller, and the hot-water distributor demands the lowest operating costs for the district heating network.

At very high hot-water return temperatures the specific investment costs of an absorption chiller are lower, but the efficiency of the district heating network is rapidly decreasing (Fig. 4).

At the highest hot-water supply temperatures and at the highest temperature differences between the hot-water supply and return temperatures (at the determined return temperature), the highest efficiency of the district heating system can be achieved. Higher temperature differences enable smaller hot-water mass flow through the district heating network and therefore reduced electrical power consumption for the network pumps.

In Figure 4 the temperature range in which the district heating system should operate is marked (dashed line).

At higher hot-water temperature differences, the curves in Figure 4 approach a determined value. The decrease in the efficiency is much lower at hot-water return temperatures between 73 and 80°C when compared to the right side of the chart. Therefore, we can make a conclusion, that the selected temperature range is suitable. By increasing the heat consumption, the efficiency of the district heating system is increasing and the trend and arrangement of the curves remain very similar.

A comparison of the operation of two district heating sytems at the same consumption (30 MW) but with different heat losses (0,04 MW/k and 0,12 MW/K), shows a decrease in the COP and cooling capacities of the absorption chiller by approximately 3%. The drop in the cooling capacity can be compensated for by increasing (for instance) the chilled-water temperature from 6/12°C to 7/13°C.

4 SKLEPI

Sistemi daljinskega hlajenja omogočajo možnost sistematične in nadzorovane uvedbe hlajenja. Omenjena možnost je izražena predvsem na področjih, na katerih že obstajajo ti sistemi daljinskega ogrevanja, ki omogočajo povezavo daljinskega hlajenja z lokalnim generiranjem hladu s pomočjo absorpcijskih hladilnikov, ki so gnani z vročo vodo iz vročevodnega omrežja.

Če se količina toplote iz sistema daljinskega

4 CONCLUSIONS

District cooling systems offer the possibility of a systematic and controlled introduction of cooling. This is particularly true for areas in which district heating systems already exist and can be used to generate cooling with refrigerators driven by heat from the hot-water supply system.

If the amount of heat supplied by a district heating system for sanitary water preparation in the summer was increased to drive cooling devices with ogrevanja, ki je v poletnem obdobju potrebna za pripravo tople sanitarne vode, poveča zaradi pogona absorpcijskih hladilnikov, bi se relativne izgube toplote znatno zmanjšale, prav tako pa tudi poraba elektrike za potrebe hlajenja.

Rezultati analize učinkovitosti absorpcijskega hladilnika prikazujejo, da lahko dosežejo ti hladilniki največjo učinkovitost pri čim višjih temperaturah dovoda vroče vode in čim manjših temperaturnih razlikah med temperaturama dovoda in povratka. Učinkovitost sistema daljinskega ogrevanja s povečanjem odjema vroče vode ter zmanjšuje z višanjem temperatur dovoda in povratka. Primeren odgovor o optimalnih temperaturah obratovanja absorpcijskega hladilnika dobimo z upoštevanjem investicije in obratovalnih stroškov celotnega sistema daljinskega hlajenja.

absorption chillers, the relative heat loss would be reduced considerably and the consumption of electricity for cooling would be reduced.

The results of the presented efficiency of an absorption chiller depending on different hot-water supply and return temperature show that at the highest hot-water supply temperatures and temperature differences, the best efficiencies for the absorption chiller can be achieved. On the other hand, with higher hot-water supply and return temperatures, the efficiency of the district heating and also district cooling system decreases.

The answer to the question of the optimum temperature for the absorption chiller supplying heat can be achieved by taking into account the investment and operating costs of the complete district cooling system.

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Prejeto:
Received: 15.8.2000

Sprejeto:
Accepted: 10.11.2000

© Strojniški vestnik 46(2000)8,564-572
ISSN 0039-2480
UDK 621.313.12:536.2:621.574.013
Strokovni članek (1.04)
© Journal of Mechanical Engineering 46(2000)8,564-572
ISSN 0039-2480
UDC 621.313.12:536.2:621.574.013
Speciality paper (1.04)

Možnosti za znižanje temperature toplotnega vira z absorpcijskimi hladilnimi napravami

The Possibilities of Reducing the Temperature of the Heat Source for Absorption Chillers

Janko Remec - Andrej Arhar

Povečanje pomembnosti s simultano proizvodnjo elektrike sprožijo toplota in mraz stroge zahteve po uporabi absorpcijskih hladilnikov. Naprave bi morale biti manjše, imeti manjšo hladilno zmogljivost in uporabljati nizko temperaturno ogrevano vodo pri 90°C ali nižjo. Zaradi tega so potrebne spremembe v konstrukciji generatorja. Padajoči sloj omogoča generatorjem uporabo vroče vode pri nižjih temperaturah. Omenjen tip generatorjev se izogiba visokim hidrostatičnim tlakom, potrebuje nižje pregretje in zmanjšuje mešanje raztopin. Zaradi tega lahko uporabimo za približno 7°C nižjo temperaturo vroče vode za dosego enakih hladilnih zmogljivosti.

(Ključne besede: generatorji, sloj padajoči, litijev bromid, prenos toplote)

The increasing importance of the simultaneous production of electricity, heat and cold results in increasingly rigorous demands on absorption chillers. These devices should be smaller, having a low cooling capacity and they have to be able to use low-temperature heating water at 90°C or below. As a result, changes in generator construction are needed. Falling-film generators allow the use of hot water with lower temperatures. This type of generator avoids the need for high hydrostatic pressure, requires lower superheating and diminishes solution mixing. Consequently, we can use hot water at a temperature 7°C lower and reach the same cooling capacity.

(Keywords: generators, falling-film, lithium bromide, heat transfer)

0 UVOD

Sistemi za soproizvodnjo toplotne in električne energije (STEE - CHP) imajo visoke energijske izkoristke, v primeru kadar uporabimo vso pridobljeno toplotno in električno energijo. Temperaturni nivoji toplote so odvisni od izbrane naprave. Plinske turbine omogočajo proizvodnjo pare, medtem ko plinski dizelski motorji proizvajajo samo vročo vodo na temperaturnem nivoju pod 100°C, običajno 90°C. V primeru manjših izvedb (pod 1 MW električne moči), so v uporabi dizelski motorji. Njihova prednost je v prilagajanju obremenitvam in dobrem izkoristku v širokem območju obremenitev. Para iz plinske turbine se lahko uporabi v industrijske namene. Toploto dizelskih motorjev odvajamo z vodo, ki se lahko uporabi samo za ogrevanje. V obeh primerih se pojavi presežek toplote v poletnem obdobju. Toploto moramo odvesti v okolico, za kar potrebujemo dodatno energijo. Hkrati so v poletnem času pričakovane večje potrebe po hladu. Presežek toplote iz STEE lahko uporabimo v

0 INTRODUCTION

Systems for combined heat and power (CHP) have a high energy efficiency when they use all the heat and power produced. The temperature levels of the heat depend on the selected device. Gas turbines are able to produce steam, but gas diesel engines only produce hot water at temperatures below 100°C, usually about 90°C. When the facilities are small (under 1 MW of electrical power), diesel engines are the preferred choice. Their advantages are operating flexibility and good efficiency over a wide range of loads. The steam from gas turbines can be used in industrial applications. Waste heat from diesel engines is rejected with the water which can only be used for heating. In both cases there is a surplus of heat in summer time which has to be put into the environment with the use of additional energy. At the some time in summer a greater demand for cooling is expected. Excess heat from CHP systems should absorpcijskih hladilnikih za proizvodnjo hladu. Za pogon absorpcijskih hladilnikov najpogosteje ostane le nizkotemperaturna toplota.

Na področju absorpcijskih hladilnih naprav z majhno hladilno močjo, ki jih poganja nizkotemperaturna vroča voda, je trg sorazmerno majhen. Eden glavnih vzrokov je visoka cena, ki je povezana z majhnim povpraševanjem po absorpcijskih hladilnih naprav. Trg je bolje razvit v Aziji in še predvsem na Japonskem. Uvoz z daljnjega vzhoda je onemogočen zaradi stroškov prevoza in neusklajenosti s predpisi v ES, kar je pomembno predvsem za hladilne naprave z majhno hladilno močjo.

Na trgu obstajata dve vrsti absorpcijskih hladilnih naprav, ena temelji na krožnem procesu z enojnim učinkom in druga z dvojnim učinkom, obe pa kot delovni par uporabljata vodno raztopino litijevega bromida. Pri obeh vrstah lahko za pogon uporabljamo plin ali paro, medtem ko nizkotemperaturna vroča voda zadošča le za pogon naprav z enojnim učinkom. Hladilne naprave, ki delujejo z delovnim parom amoniak/voda, so manj primerne, zaradi manjših vrednosti hladilnega števila (HŠ - COP) v enakih razmerah.

1 VPLIV NA HŠ IN HLADILNO ZMOGLJIVOST

Krožni procesi z enojnim učinkom so najprimernejši za absorpcijske hladilne naprave, ki so gnane z nizkotemperaturno vročo vodo. Spremembe vstopne in izstopne temperature povzročijo spremembo delovne točke absorpcijske hladilne naprave. Spreminjanje delovne točke lahko ublažimo s spremembo nivojev in koncentracij v napravi. Če so spremembe prevelike, se spremeni mehanizem prenosa toplote in snovi. Teh sprememb ne moremo več kompenzirati.

Toploto za pogon dovajamo v generator absorpcijske hladilne naprave. Nizka temperatura dovedene tople vode zahteva nizek tlak v generatorju. Tlak je odvisen od koncentracije raztopine in temperature v kondenzatorju. Znižanje koncentracije zniža učinkovitost absorberja zato ne moremo doseči nizke temperature na izstopu. Nižjo temperaturo v kondenzatorju pa dosežemo z nižjo temperaturo hladilne vode.

Delovanje absorpcijskih hladilnih naprav ni odvisno samo od vstopne temperature vroče vode, ampak tudi od temperature na izstopu. Izbrali smo primer s temperaturo hlajene vode na vstopu 12°C in 7°C na izstopu ter vstopno temperaturo vroče vode 90°C. Slika 1 prikazuje vpliv izstopne temperature vroče vode in hladilne vode na hladilno moč naprave pri nespremenjenem volumskem pretoku. Komercialne absorpcijske hladilne naprave so bile konstruirane za pogon s paro in kasneje prilagojene za vročo vodo pod 100 °C (več cevi v generatorju in kondenzatorju).

be used in absorption chillers to produce cold. Only low-level hot water is usually available for these applications.

There is a relatively small market for absorption chillers with low cooling capacities driven by low-temperature hot water. One of the main reasons for the small market is the high price which is the result of the low sales of these chillers. The market is, however, more developed in Japan and SE Asia. Importation of these chillers from the far East is obstructed by transport costs and harmonization with EU regulations. This is especially important for low-capacity chillers.

Two types of absorption chiller exist on the market, one based on single and another on double-effect cycles, both using aqueous lithium-bromide solution as the working pair. Both types could be driven by steam or gas, but for low-temperature hot-water applications only the single-effect chillers are apropriate. Chillers with amonia-water as the working pair are less convenient because of the lower COP for the same conditions.

1 INFLUENCES ON COP AND COOLING CAPACITY

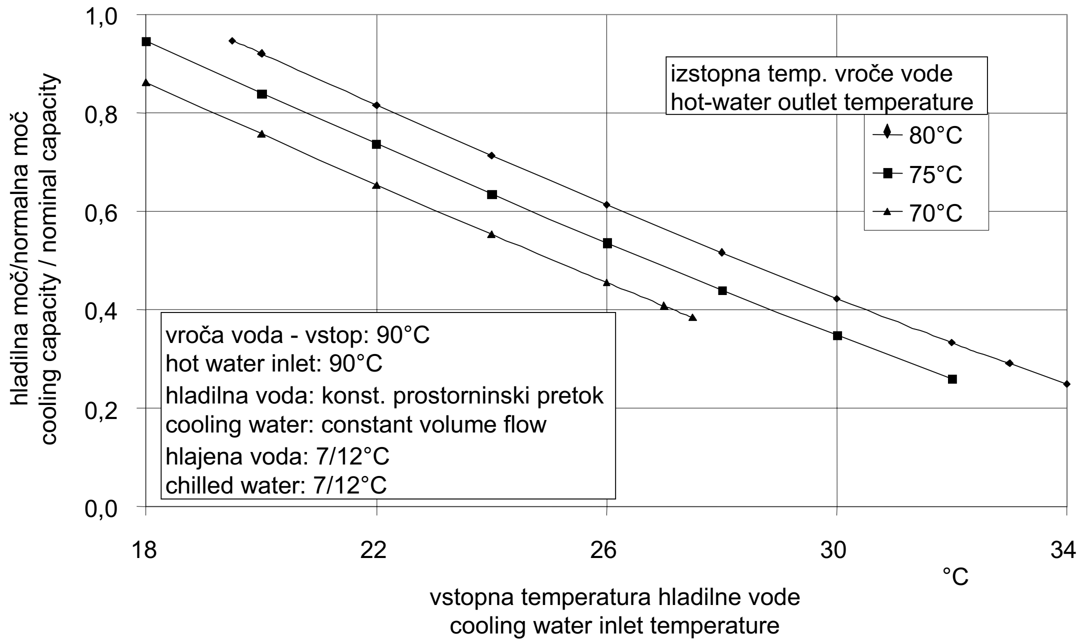
Single-effect chillers are the most suitable for absorption chillers driven by low- temperature hot water. Changes of any inlet or outlet temperature cause a different working point of the absorption chiller. Construction can compensate for these changes with different liquid levels and concentrations in the elements of the chiller. If the changes are too large, heat- and mass-transfer mechanisms can be changed too. These changes cannot be compensated any more.

Driving heat is put into the generator of the absorption chiller. A low hot-water input temperature requires a low pressure in the generator. Pressure depends on the concentration of the solution and the temperature in the condenser. The decrease in concentration reduces the absorber efficiency and the outlet chilled-water temperature cannot be reached. A lower temperature in the condenser is reached with a lower temperature of the cooling water. But both possibilities are limited.

The operation of absorption chillers depends not only on the inlet temperature of the hot water but also on its outlet temperature. We have selected an example with the temperature of chilled water at 12°C for the inlet and 7°C for the outlet and inlet hot-water temperature of 90°C. Figure 1 shows the influence of different outlet temperatures of hot water and inlet cooling-water temperature with a constant volume flow on the cooling capacity. It is a commercial absorption chiller designed to be driven by steam and adapted for hot water under 100°C (more tubes in the generator and condenser).

Znižanje temperature vroče vode do 80°C ne povzroči izrazitega zmanjšanja vrednosti HŠ, kar pa ne velja za hladilno moč. Moč je odvisna tudi od hladilne vode (sl. 1). V izbranem območju pričakovanih parametrov je zmanjšanje hladilne moči premosorazmerno z izstopno temperaturo vroče vode in temperaturo hladilne vode. Znižanje izstopne temperature za 5°C ima za posledico zmanjšanje hladilne moči za 8 %. Izstopne temperature pod 70°C so manj primerne zaradi velikih notranjih izgub v generatorju in premajhnem pregretju. To povzroča nepravilno delovanje hladilne naprave [1].

By reducing the temperature of hot water to about 80°C the COP does not decrease drastically, however the cooling capacity is significantly decreased. The capacity depends on the cooling water too (Figure 1). In the selected range of anticipated parameters a decrease of the cooling capacity is linearly proportional to the hot-water outlet temperature and the cooling-water temperature. A reduction of the outlet temperature by 5°C results in a decrease in the cooling capacity by 8%. Outlet temperatures below 70°C are unfavorable because of high internal losses in the generator and too low superheating. This causes incorrect chiller operation [1].



Sl. 1. *Vpliv zmanjšanja temperature hladilne vode na hladilno moč [1]*
Fig. 1. *Influence of cooling water temperature decrease on cooling capacity [1]*

Na trgu dostopne absorpcijske hladilne naprave so bile konstruirane tako, da jih poganja para ali vroča voda nad 100°C, ločitev hladiva in absorpcijskega sredstva je z vrenjem vodne raztopine litijevega bromida v posodi. Omenjeni način vrenja potrebuje visoko pregretje zlasti, kadar uporabljamo vodne raztopine soli. Za dosego toplotnega toka 50 kW/m^2 je potrebno pregretje za 12 °C [2]. Pri tem toplotnem toku je v generatorju konvektivno mehurčkasto vrenje. Manjši toplotni tok močno poveča površino generatorja in ceno hladilne naprave. Za dosego 50 kW/m^2 v generatorju z gladkimi cevmi (baker-nikel), je potrebna 18°C temperaturna razlika med grelno vodo in raztopino.

Existing commercial absorption chillers were designed to be driven by steam or hot water over 100°C and the separation is carried out by pool boiling of aqueous lithium-bromide solution. This type of boiling needs high superheating, especially when using aqueous salt solutions. To reach a heat flux of 50 kW/m^2 more than 12°C of superheating is needed [2]. At this heat flux, free convectional pool boiling occurs. A lower heat flux would strongly increase the surface of the generator and the price of the chiller. To reach 50 kW/m^2 in a generator with smooth copper-nickel tubes a temperature difference between the hot water and the solution of about 18°C is needed.

2 TEHNIČNE MOŽNOSTI

2 TECHNICAL POSSIBILITIES

Nizkotemperaturno ogrevno vodo lahko uporabimo brez sprememb temperature hlajene in

We can use low-temperature heating water, without changing the cooling and chilled water hladilne vode, če v generatorju: povečamo površino cevi (i), povečamo koeficient toplotne prestopnosti (ii) in zmanjšamo hidrostatični tlak (iii). Spremembe delovnega para ne bomo obravnavali.

Prva možnost (i) je najpogosteje uporabljena rešitev. Cilj lahko dosežemo z večjim številom cevi ali/in z uporabo orebrenih, ožlebljenih ali posebno obdelanih cevi. Če uporabimo več cevi, se močno povečata velikost hladilne naprave in cena. Prav tako se poveča količina raztopine v napravi in hidrostatični tlak. Z uporabo cevi z večjo površino se poveča samo cena.

Koeficient toplotne prestopnosti (ii) je mogoče povečati na strani vode in/ali na strani raztopine. Uporaba nizko temperaturne vroče vode omogoča nizek tlak v ceveh. Zaradi tega lahko uporabimo bakrene cevi, ki imajo trikrat boljšo toplotno prevodnost, kot običajno uporabljene Cu-Ni cevi. Z zmanjšanjem debeline stene cevi in spremembo materiala ne vplivamo bistveno (do 3 %) na koeficient toplotne prestopnosti. Pri nizko temperaturnem viru toplote lahko uporaba cevi z povečano ali izboljšano površino poviša koeficent toplotne prestopnosti na strani raztopine. Koeficient toplotne prestopnosti je močno odvisen od pregretja. Kadar v generatorju uporabljamo vrenje v sloju, je koeficient toplotne prestopnosti odvisen tudi od gostote masnega toka. Povečanje gostote masnega toka do 0,15 kg/s·m ima za posledico povečanja koeficenta toplotne prestopnosti. Z uporabo orebrenih in ožlebljenih cevi je koeficient prenosa toplote skoraj dvakrat večji [3]. Oba načina povečane površine omogočata boljšo razlitje kapljevine in s tem zmanjšanje suhih površin, ki zmanjšajo povprečni koeficient toplotne prestopnosti. Celotni koeficient toplotne prestopnosti je odvisen tudi od lete na vodni strani. Omejitve hitrosti na strani vroče vode so povezane z dopustnim tlačnim padcem. Pri nizkih hitrostih vroče vode je primerna uporaba cevi s povečano površino na notranji strani.

Slika 2 prikazuje vpliv na koeficient toplotne prestopnosti. Izbrali smo hitrost vroče vode med 0,5 in 2 m/s v cevi z notranjim premerom 18,5 mm. Majhen premer in relativno velike hitrosti omogočajo dober koeficient toplotne prestopnosti med vročo vodo in cevjo. Možna je tudi uporaba večjih hitrosti, kar daje celo boljše rezultate, pri čemer pa se močno poveča tlačni padec. Na strani raztopine je koeficient toplotne prestopnosti manjši.Koeficient toplotne prestopnosti je pri vrenju vodne raztopine LiBr slabši kot pri vrenju čiste vode. Pri vrenju v posodi dosežemo vrednost 5000 W/m^{2}K pri pregretju 20°C. Vrenje raztopine v padajočem sloju na gladke vodoravne cevi da slabše eksperimentalne rezultate [3]. To je v nasprotju z vodo, pri kateri opazimo povečanje koeficineta toplotne prestopnosti pri nizkih toplotnih tokovih tako na gladkih kot na povečanih površinah cevi [6].

Vrednost koeficienta toplotne prestopnosti 2500 W/m^{2}K dosežemo pri vrenju v posodi pri

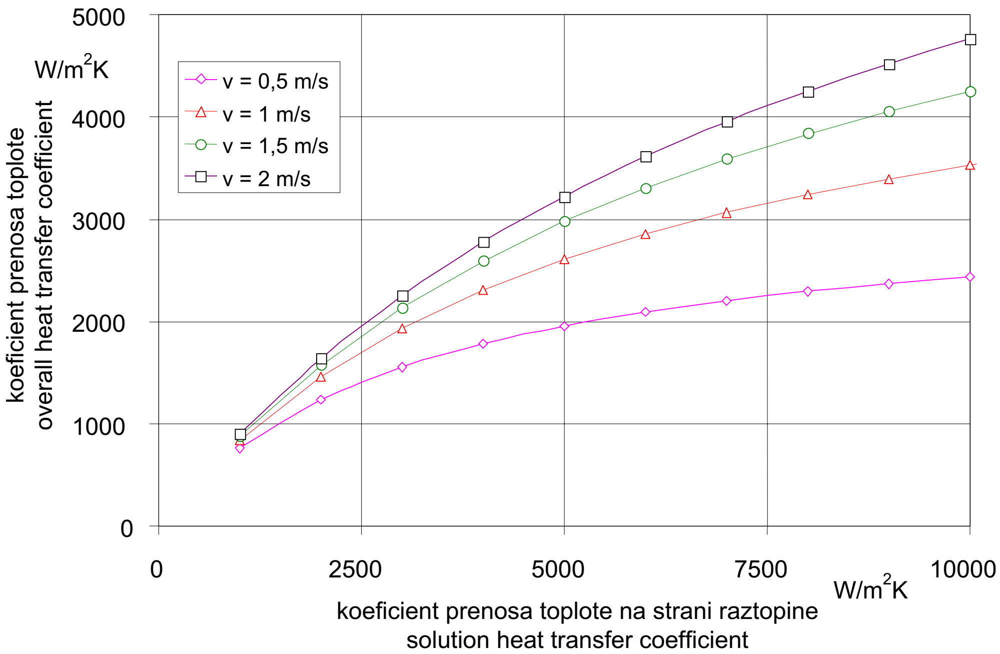
temperature, with the following changes to the generator: increasing the tube surface (i), increasing the heat-transfer coefficient (ii) and decreasing the hydrostatic pressure (iii). Changing the working pair will not be discussed.

The first possibility (i) is the most common solution. We can reach this goal with more tubes or/and using finned, grooved or specially treated tubes. If we use more tubes the chiller size and price will increase significantly. It will also cause an increase of the depth of the solution in a vessel and hydrostatic pressure. In case of non smooth tubes, only the price will be higher.

The heat-transfer coefficient (ii) could be increased on thewater side and/or the solution side. Using a low hot-water temperature enables a lower pressure in the tubes. As a consequence we can use Cu tubes with three times better conductivity rather than Cu-Ni tubes which are used in normal situations. The tube-wall thickness decreases and the material changes do not influence significantly (up to 3%) the overall heat-transfer coefficient. With a low-temperature heat source the use of extended or enhanced surfaces can increase the boiling heat-transfer coefficient of the solution. The heat-transfer coefficient depends strongly on superheating. When using thin-film boiling in a generator, the heat-transfer coefficient also depends on the mass-flow density. The increase of mass-flow density up to 0.15 kg/s·m increases the heat-transfer coefficient. Using finned or grooved tubes the heat transfer coefficient is almost twice a high [3]. Both types of extended surface enable a better spread of liquid and avoid dry surfaces which decrease the average heat-transfer coefficient. The overall heat-transfer coefficient also depends on the water side. The limits on the hot-water side are due to the water velocity being restricted by the maximum pressure drop. By using a low hot-water velocity, extended surfaces inside the tube are also a possibility.

Figure 2 shows the influence on the overall heat-transfer coefficient. Hot-water velocities between 0.5 and 2 m/s in a tube with a 18.5 mm inner diameter are selected. A small diameter and a relatively high velocity enables good heat-transfer coefficient between the hot water and the tube. The use of a higher velocity is also possible and gives even better results, but the pressure drop increases significantly. On the other hand, heat transfer on the solution side is low. In comparison with water the lithium-bromide boiling heat-transfer coefficient is weak. Using pool boiling, 5000 W/m^{2}K can be reached for only 20°C of superheating. Solution boiling in a falling film on smooth horizontal tubes gives poorer experimental results [3]. This is in contrast to water, where a significant increase in the low-heat flux is measured on smooth and structured surfaces [6].

The overall heat-transfer coefficient of 2500 W/m^{2}K by pool boiling at a temperature difference of 20°C and a hot-water velocity of 1.5 m/s can be



Sl. 2. *Vpliv koeficienta prenosa toplote na strani raztopine na celotni prenos toplote pri različnih hitrostih v ceveh*

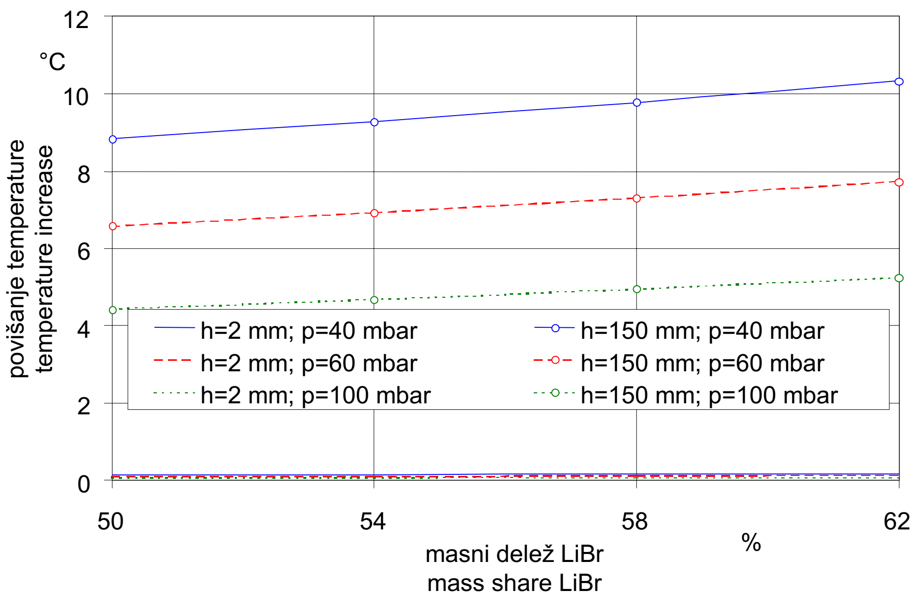
Fig. 2. *Influence of the solution heat-transfer coefficient on the solution side on the overall heat transfer with a different velocity in the tubes*

temperaturni razliki 20°C in hitrosti vroče vode 1,5 m/s. Podvojitev hitrosti vroče vode zmanjša potrebno temperaturno razliko za 2,5 °C ali zmanjša površino cevi za 20%.

Tlak (iii) v generatorju je odvisen od koncentracije raztopine in temperature kondenzacije. Vrenje je lahko v posodi ali v sloju. Cevi pri vrenju v posodi so potopljene v raztopini. Pri generatorjih s

reached. Doubling the hot-water velocity decreases the required temperature difference by 2.5°C or decreases the tubes' surface area by 20%.

Pressure (iii) in the generator depends on the solution concentration and the condensation temperature. We can use pool boiling or thin-film boiling in the generator. Tubes for pool boiling are immersed in the solution and boiling occurs on the tubes. With



Sl. 3. *Učinek hidrostatičnega tlaka na povišanje temperature*

Fig. 3. *Effect of hydrostatic pressure on temperature increase*

pršenjem se na ceveh oblikuje sloj raztopine. Nad snopom cevi je cev, ki razprši raztopino. Uporaba vrenja v sloju zahteva vodoravno postavitev naprave in s tem tudi cevi.

Absolutni tlak v generatorju je nizek (~ 70 mbar), zato ima hidrostatični tlak močan vpliv. V tankem sloju je povečanje temperature vrenja zaradi hidrostatičnega tlaka zanemarljiv. Pri vrenju v posodi pa je 150 mm pod nivojem raztopine je tlak približno 25 mbar višji in zaradi tega se poviša temperatura vrenja. Povišanje temperature vrenja v tankem sloju (2 mm) na cevi in cev, ki je 150 mm pod nivojem raztopine, je prikazana na sliki 3.

Hidrostatični tlak ima pri zelo nizkem tlaku v generatorju (40 mbar) in temperaturi v kondenzatorju (29°C) velik vpliv. Uporaba hladilne vode s temperaturo 31°C omogoča, da doseže temperaturo v kondezatorju okoli 36°C in absolutni tlak 60 mbar. S prehodom iz vrenja v posodi na vrenje v sloju lahko znižamo temperaturo vroče vode za 6,5 do 7,7 °C. Potrebna temperatura vroče vode se zniža zaradi znižanja hidrostatičnega tlaka. Absolutni tlak 100 mbar je neprimeren za generatorje z nizko temperaturo vroče vode, saj je ravnotežna temperatura približno 90°C (pri 58 mas. %) brez upoštevanja temperaturne razlike za prenos toplote in vrenja. Pri generatorjih gretih z nizko temperaturno vročo vodo je pri enakih pogojih vrenje v sloju kapljevine učinkovitejše od vrenja v posodi.

Padajoči sloj kapljevine v generatorju onemogoča mešanje raztopine. Zaradi tega imamo določeno porazdelitev temperature in koncentracij. Ravnotežna temperatura se močno poviša pri ceveh v spodnjem delu generatorja. Zaradi omenjenega razloga je priporočljivo večje število prehodov vroče vode skozi snop cevi. Vroča voda vstopa na spodnjem delu snopa cevi in raztopini z že zvišano temperaturo omogoča vrenje. V zgornjem delu generatorja je ogrevana voda že nekoliko ohlajena, pri čemer pa ima tudi raztopina nižjo koncentracijo. To omogoča največjo temperaturno razliko med vstopom in iztopom vroče vode ter zmanšanja eksergijskih izgub.

Na ceveh nastala para teč skozi snop cevi v kondenzator. Naloga generatorja je tudi onemogočiti vstop raztopine v kondenzator v primeru nepravilnega delovanja. Zato je kondenzator nameščen nad generatorjem. Hitrost pare je velika, ker je specifični volumen velik. Omejitve so pri tlačnem padcu in odnašanju sloja ali kapljic raztopine. Nespremenljivo hitrost in tlačni padec lahko dosežemo s povečanjem razmakov med cevmi.

Slika 4 prikazuje spremembo temperature in koncentracije na ceveh. Za zmanjšanje prostornine generatorja izberemo izmenično razvrstitev cevi. Prenosniki toplote s fazno spremembo imajo spremenljiv masni in volumski pretok kapljevine in pare. Zaradi tega se pogosto

falling-film generators the layer of the solution is produced by spraying it on the tubes. They require an additional tube, or tubes, for solution distribution above the tube bundle. Additional attention should be paid to setting up the device horizontally, because not only the absorber and evaporator but also the generator is of the falling-film type.

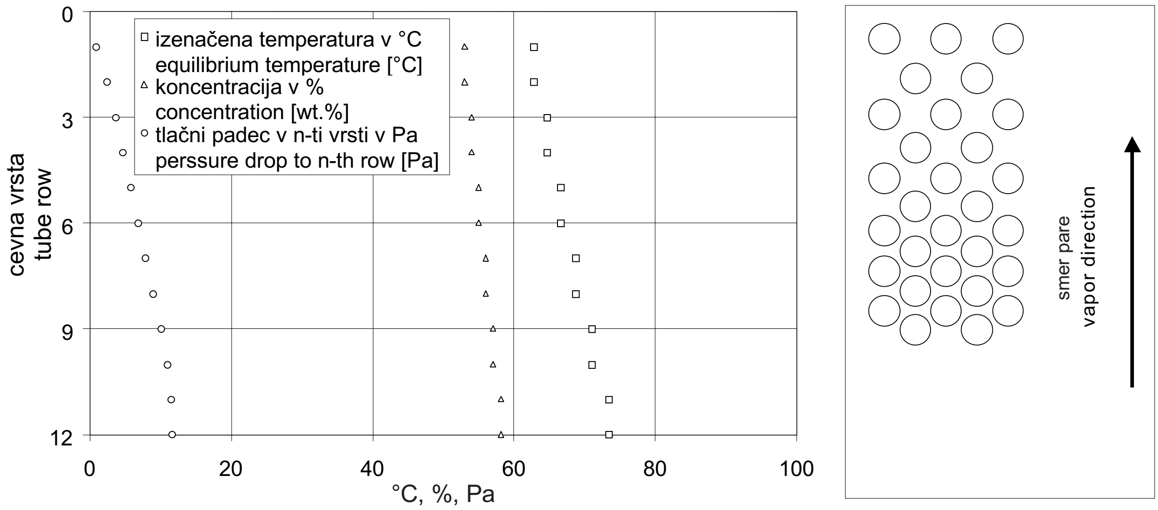
Since the absolute pressure in the generator is very low (~70 mbar) the hydrostatic pressure has a strong influence. There is a negligible boiling temperature increase in the thin film because of the hydrostatic pressure. But 150 mm under the solution level the pressure is about 25 mbar higher and consequently the boiling temperature increases. The boiling temperature increases in the case of a thin film (2 mm) on the tube, and the tube which is 150 mm under the solution level can be seen in Figure 3.

At a very low pressure (40 mbar) and temperature in the condenser (29°C) the hydrostatic pressure has the greatest influence. The use of cooling water at 31°C causes an internal temperature in the condenser of about 36°C and an absolute pressure of 60 mbar. The decrease in the required hot-water temperature by changing from boiling on immersed tubes to boiling in a falling film is between 6.5 and 7.7°C. This is a realistic hot-water temperature decrease with a reduced hydrostatic pressure. An absolute pressure of 100 mbar is not appropriate for low-temperature hot-water applications because the equilibrium temperature is about 90°C (at 58 wt.%) without any consideration of the temperature difference for heat transfer and boiling. For a low-temperature hot-water generator, thin-film boiling is much more efficient than pool boiling.

Falling-film generators also prevent mixing of the containing solution. Therefore, we have a clear concentration and temperature distribution. The equilibrium temperature for tubes with a lower position increases strongly. For this reason, more passes of the hot water through the tube bundle is recommended. The hottest water should inlet at the lower part of the tube bundle and meet with the hottest solution. On the top part, cooled hot water hits the solution with the lowest concentration. This enables the biggest temperature difference between hot-water inlet and outlet and decreases the exergy losses.

Vapor generated on the tubes goes through the bundle in the condenser. This is usually above the generator to prevent entry of the solution to the condenser in the case of faulty operation. Vapor velocities should be high because the specific volume is also high. The limitations are pressure drop and the solution-film or droplets carry off. A constant velocity and pressure drop can be maintained with an increased distance between the tubes.

Figure 4 presents thetemperature and concentration changes on the tubes. To reduce the volume of the generator an alternating tube arrangement is usually selected. Heat exchangers with a phase change have changeable mass and volume-

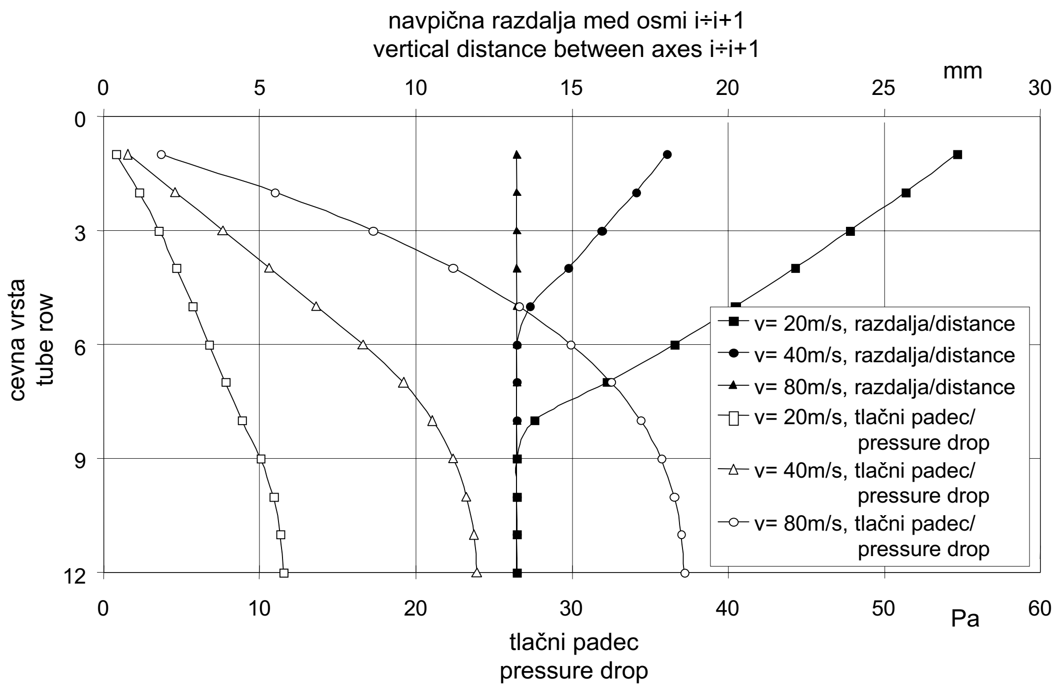


Sl. 4. *Ravnotežna temperatura na površini raztopine, povprečna temperatura in celotni tlačni padec do n-te vrste cevi (levo) in cevni porazdeljenosti (desno)*

Fig. 4. *Equilibrium surface temperature, average concentration and total pressure drop to the n-th tube row (left) and tube distribution (right)*

uporablja spremenljiva razdalja med cevmi. Pri izračunu smo zanemarili mešanje raztopine in konstanten toplotni tok na vseh ceveh (70 kW/m^2). Navpične razdalje med cevmi so izbrane z uporabo nespremenljive hitrosti pare (20m/s), med stolpci je vseskozi enaka razdalja, ki znaša 35 mm. Tlačni padec je majhen, njegovo linearnost pa dosežemo z nespremenljivo hitrostjo. Povečanjem hitrosti pare na 100 m/s povzroči tlačni padec 140 Pa [4]. Koncentracijsko razlika 5% je dosežemo pri gostoti masnega toka 0,1 kg/(s·m).

flow of liquid and vapor so varying spaces between tubes are often used. No solution mixing and constant heat flux on all tubes (70 kW/m^2) have been assumed. Vertical distances between tubes are selected on the basis of constant vapor velocity (20 m/s) and horizontal distances are fixed at 35 mm. No significant pressure drop can be seen and linearity is reached because the velocity is constant. The increase in velocity to 100 m/s causes a pressure drop to 140 Pa [4]. A concentration difference of 5% is reached with a mass flux density of 0.1 kg/(s·m).



Sl. 5. *Tlačni padec in navpična razdalja med cevnimi vrstami z različnimi največjimi hitrostmi*

Fig. 5. *Pressure drop and vertical distances between tube rows with different maximum velocities*

Najmanjša razdalja med cevmi je 6,5 mm [5]. To omogoča izdelavo z uvaljanjem cevi in mehansko čiščenje. Tlačni padec v cevnem snopu je majhen. Slika 5 prikazuje navpično razdaljo med cevnimi vrstami. Ko pri nespremenljivi vodoravni (40 mm) razdalji med osmi cevi omogočimo povečanje hitrosti, se bo navpična razdalja zmanjšala. Pri izbrani največji dovoljeni hitrosti (80 m/s) je razdalja nespremenljiva in omejena z najmanjšo razdaljo med cevmi. Gradient tlačne razlike se zmanjša zaradi manjše hitrosti. Izbrana srednja hitrost (40 m/s) zahteva večjo navpično razdaljo med cevmi v zgornjem delu prenosnika. Omenjeni učinek je še značilnejši pri nižji hitrosti (20 m/s), pri kateri samo zadnje tri cevi dosežejo najmanjšo razdaljo (omejena z najmanjšo razdaljo med cevmi). Tlačni padec med cevmi je nespremenljiv, dokler je razdalja določena s hitrostjo. V teh razmerah padec tlaka povzroči povišanje temperature za približno 0,1°C (pri 60 mbar in 55 %). Hitrost in tlačni padec padata, ko je omejitev razdalja med cevmi. Z večjo hitrostjo je mogoče zmanjšati višino cevnega snopa iz 240 mm (20 m/s) na 180 mm (80 m/s). Najmanjša razdalja med cevnimi vrstami v generatorju je določena s tehnologijo izdelave generatorja. Velika hitrost pare ne povzroči izrazitejšega tlačnega padca in povišanja temperature. Zaradi tega je primerno zmanjšati izmere generatorja.

The minimum gap between tubes is set to 6.5 mm [5]. This enables tube end attachment with roller expansion and mechanical cleaning. The pressure drop in the tube bundle shell be low. Figure 5 shows the vertical distances between tube rows. When at a constant (40 mm) horizontal distance, the allowed velocity is increased the vertical distance will be decreased. At the highest selected velocity (80 m/s) the distance is constant and limited with a minimum gap. The gradient of pressure difference decreases because of the lower velocities. A medium velocity (40 m/s) requires a higher vertical distance between the tubes in the upper half of the exchanger. This effect is more significant at the lowest velocity (20 m/s) where only the last three tubes reach the minimum distance (limited with a minimum gap). The pressure drop between tubes (one above another) is constant as long as the distance is defined by velocity. Under these conditions pressure drop causes a temperature increase of about 0.1°C (at 60 mbar and 55 wt.%). When the gap is at the limit, the velocity and pressure drop decrease. With a higher velocity the height of the tube bundle can be reduced from 240 mm (20 m/s) to 180 mm (80 m/s). With a small number of tube rows in the generator the distance between them is defined by production technology. High vapor velocities have not caused a significant pressure drop and temperature increase. As a result, it is reasonable to decrease the dimensions of the generator.

3 SKLEP

Večina absorpcijskih hladilnih naprav uporablja v generatorju vrenje v posodi. Njihova konstrukcija je robustna. Pri vrenju v posodi je potrebna visoka vstopna temperatura grelne vode, ki omogoča pregretje in tudi mešanje raztopine. Znižanje vstopne in/ali izstopne temperature bistveno ne poslabša HŠ, ampak povzroči padec hladilne moči v absorpcijskih hladilnih napravah.

Pomemben napredek pri uporabi nizkotemperaturne tople vode so dosegli, s spremembo načina vrenja v generatorju - prehod iz vrenja v posodi na vrenje v sloju. Zaradi znižanja hidrostatičnega tlaka lahko uporabimo za približno 7 °C hladnejšo grelno vodo. Pri nižjem tlaku (nižji temperaturi hladilne vode) je vpliv še mnogo večji. Učinek vrenja v sloju je večji, če je več prehodov vroče vode, vstop vroče vode pa je na spodnjem delu generatorja. Dodatno povečanje lahko dosežemo z večjo hitrostjo vroče vode, rezultat tega je višji tlačni padec na strani vroče vode. S povečanjem hitrosti vroče vode iz 1,5 na 3 m/s lahko znižamo njeno temperaturo za 2,5°C. Z uporabo razširjene površine (orebrene ali ožlebljene cevi) se poveča koeficient toplotne prestopnosti raztopine skoraj za 100 %. Zmanjšanje debeline stene cevi in sprememba materiala nimata izrazitega vpliva (do 3 %) na koeficient toplotne prestopnosti. Generirana vodna para ima majhen vpliv na tlak v generatorju. Povišanje temperature zaradi padca tlaka je najvišje na spodnjih ceveh in pri hitrosti pare 80 m/s znaša samo 0,1°C.

3 CONCLUSION

Most absorption chillers use pool boiling of the solution in a generator. Their construction is robust. On the other hand, a high temperature of the input heat is required not only for superheating but also for mixing of the solution. Lowering input and/ or output temperatures does not make the COP significantly worse but causes a reduction in the cooling capacity of the absorption chillers.

Using low-temperature hot water the greatest improvement is achieved when changing from a pool-boiling generator to a falling-film generator. About 7°C cooler driving water can be used and the hydrostatic pressure is reduced. At low pressure (low cooling-water temperature) the influence is even stronger. The effect is greater with more passes of hot water with the inlet at the bottom of the tube rows of the generator. Improvements can be obtained with higher hot-water velocity but the result is a higher pressure drop too. We can reduce the hot-water temperature by 2.5°C if we increase its velocity from 1.5 to 3 m/s. Using extended surfaces (finned or grooved tubes) the solution heat-transfer coefficent is increased by almost 100%. The tube-wall thickness decreases and material changes do not significantly influence (up to 3%) the overall heat-transfer coefficient. The pressure drop of the generated steam through the tube bundle has a negligible effect. It causes a temperature of increase for only 0.1°C on the lower tubes by using vapor velocities of about 80 m/s.

Generatorji z vrenjem v posodi niso primerni za pogon z nizkotemperaturno toplo vodo. V takšnih primerih uporabljamo vrenje v sloju, kar omogoča optimalno konstrukcijo in razmeroma nizke proizvodne stroške.

We can conclude that generators with pool boiling are not convenient for low hot-water temperatures. Falling-film generators should be used for such applications, because they enable optimum design and production costs.

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Prejeto:
Received: 15.8.2000

Sprejeto:
Accepted: 10.11.2000

© Strojniški vestnik 46(2000)8,573-579
ISSN 0039-2480
UDK 621.643:621.564:536.21
Strokovni članek (1.04)

© Journal of Mechanical Engineering 46(2000)8,573-579
ISSN 0039-2480
UDC 621.643:621.564:536.21
Speciality paper (1.04)

Določitev optimalne debeline izolacije cevnih sistemov za transport hladilnega sredstva

The Determination of the Optimum Insulation Thickness of Pipe Systems for Transporting Cooling Media

Darko Goričanec - Jurij Krope - Igor Tičar

V prispevku je opisan postopek določitve optimalne debeline izolacije cevovodov za transport hladilnega sredstva. Izračunana optimalna (ekonomska) debelina cevne izolacije je odvisna od izbrane ekonomske metode ter tehnično-ekonomskih podatkov. Na podlagi znanih postopkov za izračun ustaljenega toplotnega toka skozi izolacijo cevnih sistemov za transport hladilnega sredstva in v prispevku opisanega ekonomskega izračuna je bila izdelana računalniška aplikacija. Numerični algoritem, ki je bil izdelan v ta namen, omogoča izbiro med statično in dinamično ekonomsko metodo. Izračun optimalne (ekonomske) debeline toplotne izolacije hladilnih cevnih sistemov se je z izdelavo uporabniško prijaznega računalniškega postopka poenostavil, vendar se od uporabnika kljub temu zahteva osnovno znanje prenosa toplote in ekonomike.

(Ključne besede: mediji hladilni, izolacije cevovodov, prevodnost toplotna, debeline izolacije)

In this paper we present a procedure for determining the economic pipeline-insulation thickness for the transport of a cooling medium. The calculated economic (optimum) pipe-insulation thickness depends on the chosen economic method and the technical-economic data. A computer software application has been developed using the known procedures for the calculation of stationary heat flow through the insulation of pipe systems for cooling medium transportation together with the economic calculations described in the paper. The numerical algorithm developed for this application enables a choice between static and dynamic economic methods. The calculation of the economic heat-insulation thickness of cooling pipe systems has been simplified with the development of a user-friendly computer application, but in spite of this, the user of this application must have a basic knowledge in the field of heat transfer and economics.

(Keywords: cooling media, pipelines insulation, heat conductivity, insulation thickness)

1 IZRAČUN TOPLOTNIH IZGUB SKOZI TOPLOTNO IZOLACIJO

Med trdnimi telesi, tekočinami in plini poteka prenos toplote, če so različnih temperatur. Smer prenosa toplote poteka vedno s področja z višjo temperaturo na področje z nižjo temperaturo, dokler se temperature ne izenačijo. Ustaljeni prenos toplote skozi izoliran cevovod lahko poteka na tri načine:
- s prevajanjem,
- s konvekcijo,
- s sevanjem.

Pri izračunu toplotnega toka skozi izolacijo je treba upoštevati tudi vpliv toplotnih mostov nosilnih elementov izolacije, cevovoda in armatur na povečanje toplotnega toka skozi izolacijo izoliranega cevovoda za transport hladnega ali vročega sredstva. Pri določevanju optimalne debeline izolacije se

1 CALCULATION OF HEAT LOSSES THROUGH HEAT INSULATION

Heat transfer occurs between solids, liquids and gases whilst they are at different temperatures. The direction of the heat transfer is always from the higher temperature region to the lower temperature region until the temperatures become equal. Steady heat transfer through an insulated pipeline can run in three ways:
- by conduction,
- by convection,
- by radiation.

When calculating heat flow through insulation we have to consider the influence of the heat bridges (heat conductors which support the elements of insulation, pipeline and armatures) on the increase of heat transfer through the insulation of an insulated pipeline for the transport of a cooling or heating medium. When determining the optimal insulaupoštevajo samo toplotni mostovi nosilnih elementov izolacije. Za določitev toplotnega toka skozi cevno izolacijo pri naravni ali prisilni konvekciji so v izdelanem numeričnem postopku uporabljene enačbe, ki so podane v [1] in [2] in drugi podobni literaturi.

1.1 Določitev toplotne prevodnosti

V praksi se določa naslednje toplotne prevodnosti izolacijskega materiala:

- laboratorijska toplotna prevodnost
- nominalna toplotna prevodnost
- praktična toplotna prevodnost
- operativna toplotna prevodnost.

Te štiri definicije [1] predstavljajo vse stopnje določitve toplotne prevodnosti od laboratorijskih izračunov pa vse do določitve operativne toplotne prevodnosti, ta skupaj s korekcijskimi faktorji upošteva vse določljive fizikalne zakonitosti, ki vplivajo na prenos toplote v praksi.

Pri izračunih toplotnega toka skozi izolacijo uporabljamo praktično toplotno prevodnost takrat (operativna toplotna prevodnost je v tem primeru enaka praktični), kadar ni toplotnih mostov. Toplotne mostove, ki jih povzročajo enakomerno razporejeni pritrdilni elementi izolacije, je treba upoštevati pri določitvi operativne toplotne prevodnosti izolacije.

2 IZRAČUN EKONOMSKE DEBELINE IZOLACIJE

Določitev ekonomske (optimalne) debeline izolacije ([1] in [3]) temelji na iskanju čim nižjih investicijskih stroškov in stroškov izgubljene toplote, ki jo izgubimo zaradi toplotnega toka skozi izolacijo. Ekonomsko debelino izolacije lahko določimo z določitvijo:

- ekonomske debeline izolacije s statično ekonomsko metodo ali
- ekonomske debeline izolacije z dinamično ekonomsko metodo.

Z naraščajočo debelino toplotne izolacije se zmanjšujejo toplotne izgube in s tem stroški izgubljene toplote, naraščajo pa stroški investicije v izolacijo, amortizacija, obresti in vzdrževalni stroški. Vsota vseh stroškovnih postavk doseže minimum pri določeni debelini izolacije. To debelino imenujemo ekonomska (optimalna) debelina izolacije.

Specifične stroške za izračun optimalne debeline cevne izolacije podajamo v USD/m

tion thickness only the heat bridges of the supporting elements of the insulation are considered. When determining the heat flow through the pipeline insulation by natural or forced convection the equations in references [1] and [2] and other similar literature are used in developing the numerical algorithm.

1.1 Determination of heat conductivity

The following heat conductivities of the insulation material are determined in practice:

- laboratory heat conductivity,
- nominal heat conductivity,
- practical heat conductivity,
- operative heat conductivity.

These four definitions [1] represent all stages of heat-conductivity determination from laboratory calculations to the determination of the operative heat conductivity. When these four, together with the correction factors are considered, this determines the physical lawfulness which influences heat transfer in practice.

When calculating the heat flow through insulation, practical heat conductivity is used when there are no heat bridges (operative heat conductivity is in this case equal to practical heat conductivity). Heat bridges, which are the cause of uniformly disposed fixable elements of insulation, have to be considered by a determination of the heat conductivity of insulation.

2 CALCULATION OF ECONOMIC INSULATION THICKNESS

Determination of the economic insulation thickness ([1] and [3]) is based on a search for the lowest investment costs and the lowest cost of heat losses due to heat flow through the insulation.
The economic insulation thickness can be determined by:

- dynamic economic method,
- static economic method.

Heat losses decrease with growing heat insulation thickness and therefore so does the cost of heat, lost by heat transfer through the insulation. The investment costs of insulation, amortization, interest and maintenance costs rise with greater insulation thickness. The total cost reaches a minimum at a certain insulation thickness. This thickness is called the economic (optimum) insulation thickness.

The specific costs for the calculation of the optimal pipeline insulation thickness are given in USD/m

$$C_S = 3,6 \cdot 10^{-6} \cdot \Phi \cdot C_T \cdot t + b \cdot C_M \quad (1).$$

Da bi bil prehod med statično in dinamično ekonomsko računsko metodo čim lažji, se lahko za poenostavitev izračuna uvede faktor f:

In order to clearly distinguish between static and dynamic economic methods, the factor f is introduced to simplify the calculation:

$$f = \frac{S_1}{S_2} \quad (2)$$

$$S_1 = \frac{1-\left(\frac{1+p/100}{1+z/100}\right)^n}{1-\frac{1+p/100}{1+z/100}} \quad (3)$$

in

and

$$S_2 = \frac{1-\left(\frac{1}{1+z/100}\right)^n}{1-\frac{1}{1+z/100}} \quad (4).$$

Pri uporabi dinamične ekonomske metode za določitev minimuma stroškov C_S, ter s tem optimalno debelino izolacije, se v enačbo (1) uvede faktor f:

Using the minimum C_S of the dynamic economic method for the determination of costs the optimum insulation-thickness factor f is introduced into equation (1):

$$C_S = 3,6 \cdot 10^{-6} \cdot f \cdot \Phi \cdot C_T \cdot t + b \cdot C_M \quad (5).$$

2.1 Statična ekonomska metoda

Statična ekonomska metoda izračuna se pogosto uporablja za določitev minimuma stroškovnih funkcij. Stroški toplotnih izgub, ki nastajajo letno, se prištevajo k letnim stroškom vzdrževanja in investicije v toplotno izolacijo. Ekonomsko debelino izolacije določimo z iskanjem minimuma stroškovne funkcije z uporabo statične ekonomske metode, ob predpostavki, da so stroški in obrestna mera konstantni čez vso dobo trajanja. Izračunan minimum stroškovne funkcije po statični metodi je sprejemljiv samo, če so v izračunu predpostavljenem časovnem obdobju ekonomski pogoji stabilni in ostajajo konstantni ves čas uporabe izolacije.

2.1 Static economic method

The static economic method is often used for the determination of minimum cost functions. Annual costs of heat losses are added to annual costs of maintenance and investment in heat insulation. The economic insulation thickness is determined by searching for the minimum cost function using the static economic method, and a supposition that the costs and interest rates remain constant during the service life of the insulation. The static method for calculating the minimum cost function is acceptable only when the economic conditions are favorable during the presumed calculation time period and remain constant during the use of the heat insulation.

2.2 Dinamična ekonomska metoda

Pri določevanju minimuma stroškovne funkcije z dinamično ekonomsko metodo se upošteva večanje letnih stroškov izgubljene toplote (zaradi vpliva inflacije) kljub temu, da so toplotne izgube skozi izolacijo konstantne.

Za določitev minimuma stroškovne funkcije po dinamični ekonomski metodi se uporablja metoda sedanje vrednosti, ki temelji na predpostavljenih stroških, ki nastajajo med dobo trajanja izolacije.

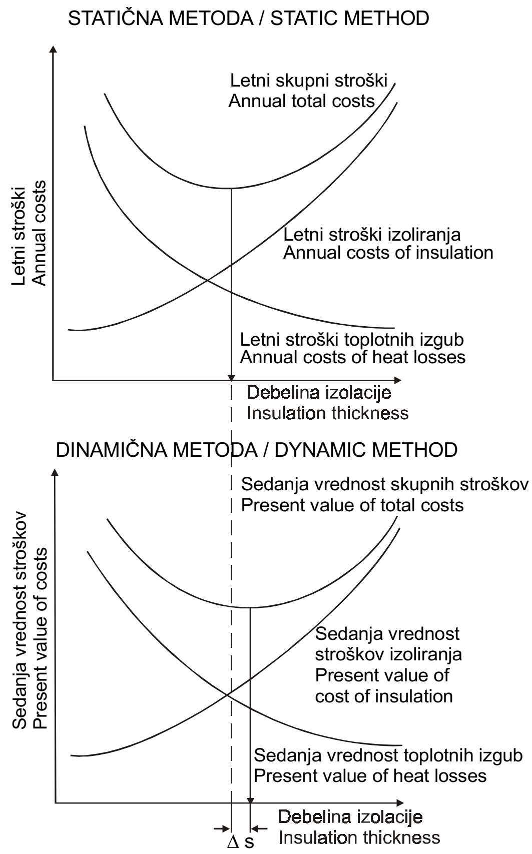
Pri dinamični ekonomski metodi se upošteva tudi inflacija (letno povečevanje cene) različnih vrst stroškov, npr. cena proizvedene toplote, vzdrževanje itn. Ker se stroški toplotnih izgub vsako leto povečujejo, je minimum stroškovne funkcije, izračunan z dinamično ekonomsko metodo pri večji debelini toplotne izolacije kot minimum stroškovne funkcije, določen s statično ekonomsko metodo (sl. 1). Z enačbo (5) izračunana optimalna debelina izolacije je glede na enačbo (1) večja približno za $\sqrt{f}$-krat.

2.2 Dynamic economic method

When determining the minimum cost function using the dynamic method, the rise in the annual costs of heat losses (due to inflation) is considered in spite of the constancy of heat losses through the insulation.

When determining the minimum of the cost function using the dynamic method the present value method is used. This method is based on supposed costs which may appear during the service life of the insulation.

The effect of inflation on different kinds of costs such as the price of produced heat, maintenance, etc., is considered when using this economic method. Because the costs of heat losses rise every year the minimum cost function, as calculated using the dynamic method, is placed at a greater-heat insulation thickness than determined by the static economic method (Figure 1). Using equation (5) the calculated optimal insulation thickness when compared to equation (1) is greater by about $\sqrt{f}$-times.



Sl. 1. *Razlika pri določitvi ekonomske debeline toplotne izolacije cevnih sistemov z uporabo statične ali dinamične ekonomske metode ([1] in [3])*

Fig. 1. *The differences between the economic heat-insulation thickness of pipe systems using the static and dynamic economic methods ([1] and [3])*

2.3 Določitev diskontnega faktorja

Diskontni faktor "b" upošteva dobo trajanja izolacije, obresti, stopnjo povečevanja stroškov vzdrževanja in drugih stroškov.

Odvisno od metode izračuna se uporabljajo različni diskontni faktorji, ki pa v vsakem primeru upoštevajo obrestno stopnjo, inflacijo, stopnjo amortizacije itn.

V praksi se uporabljajo naslednji diskontni faktorji:

a) osnovni diskontni faktor:

2.3 Determination of discount factor

The discount factor "b" considers the service life of the insulation, interest, maintenance costs and other cost rises.

Depending on the calculation method, different discount factors are used but in all cases they consider interest rates, inflation, amortization rate, etc.

The following discount factors are used in practice:

a) basic discount factor:

$$b_1 = \frac{1}{n} + \frac{z + r + g}{100} \tag{6}$$

b) enako kakor a) in z upoštevanjem amortizacije med dobo trajanja izolacije:

b) same as a) and with consideration of amortization during the service life of the insulation:

$$b_2 = \frac{1}{n} + \frac{1}{100} \cdot \left(\frac{z+1}{2} + r + g \right) \tag{7}$$

c) diskontni faktor, ki je anuitetno - dinamično ovrednotena funkcija:

c) discount factor which is an annuity-dynamic valued function:

$$b_3 = \frac{\frac{z}{100}}{1 - \left(\frac{1}{1 + z/100} \right)^n} + \frac{r+g}{100} \tag{8}.$$

3 DOLOČITEV INVESTICIJSKIH STROŠKOV

Investicijski stroški v toplotno izolacijo [1] vsebujejo stroške nakupa izolacijskega materiala, stroške izoliranja in indirektne stroške izoliranja.

Enačbo nelinearne stroškovne funkcije poenostavimo tako, da jo zapišemo v linearni - zvezni obliki:

3 DETERMINATION OF INVESTMENT COSTS

The investment costs of heat insulation [1] include the costs of the insulation material, costs of insulation work and indirect insulating costs.

The equation of the nonlinear cost function can be simplified and written in linear form:

$$C = C_0 + C' \cdot s \qquad (9).$$

Dejanska stroškovna funkcija ni linearna zaradi prehodov k naslednji debelini izolacije, ki se proizvaja ali v primeru dodatnih stroškov montaže.

Indirektni stroški izoliranja so stroški, ki npr. nastajajo zaradi: zavzetja prostora, zmanjšanje uporabnega prostora, cevnih povezav, kanalov itn.

Stroškovno funkcijo C_M investicije v cevno izolacijo lahko zapišemo z enačbo:

The real cost function is not linear when moving to the next available insulation thickness or in the case of additional assembly costs.

Indirrect costs of insulating are costs which rise with e.g. occupation of place, reduction of available space, intersections of pipe segments, channels etc.

The cost function of the investment in heat insulation C_M is written as in equation:

$$C_M = \left(C_0 + C' \cdot s\right) \cdot \pi \cdot \left(d_i + 2 \cdot \frac{s}{100}\right) + \Delta C_R(s) \qquad (10).$$

V nekaterih primerih proizvajalec izolacije podaja ceno izolacije C_0 v USD/m, takrat enačbo (10) spremenimo tako, da dobi investicijska stroškovna funkcijo C_M naslednjo obliko:

In some cases when the producer of the heat insulation gives the price of the insulation C_0 in USD/m equation (10) has to be modified and the investment-cost function C_M is written in the form:

$$C_M = C_0 + C' \cdot s \cdot \pi \cdot \left(d_i + 2 \cdot \frac{s}{100}\right) + \Delta C_R(s) \qquad (11).$$

4 PRIMER

Določiti je treba ekonomsko debelino cevne izolacije za pretok hladilnega medija po cevi zunanjega premera 89 mm. Podatki, potrebni za izračun, so podani v preglednici 1 in 2. Rezultat izračuna z določitvijo ekonomske debeline cevne izolacije je podan v preglednici 3.

4 EXAMPLE

In this example the determination of economic heat-insulation thickness for a flow of cooling medium through a pipeline with an outside diameter of 89 mm is presented. The technical and economic data are shown in tables 1 and 2. The results of the calculation are shown in table 3.

Preglednica 1. *Komercialno razpoložljiva debelina in cena izolacije*
Table 1. *Commercially available thickness and price of heat insulation*

debelina izolacije v mm insulation thickness (mm)	20	30	40	50	60	70	80
cena izolacije v USD/m price of insulation (USD/m)	2,89	3,42	4,43	5,54	6,16	8,54	10,42

5 SKLEP

Izračun ekonomske debeline izolacije je odvisen od velikega števila tehničnih in ekonomskih parametrov. Prav tako je za običajnega uporabnika izolacije ali projektanta postopek izračuna ekonomske debeline izolacije zamuden in zahteven. Zaradi tega smo na Univerzi v Mariboru, razvili programsko opremo za izračun toplotnih izgub skozi izolacijo in ekonomsko debelino izolacije. Osnovni podatki proizvajalca izolacije (komercialno razpoložljive debeline izolacije, cena izolacije, toplotna prevodnost izolacije itn.), ki so potrebni za izračun, so zapisani v bazi podatkov.

5 CONCLUSION

The calculation of an economic insulation thickness depends on several technical and economic parameters. The procedure for the optimal insulation-thickness calculation is difficult and exacting for the normal user of insulation or a project engineer. Because of this our team at the University of Maribor has developed a computer software application for the calculation of heat losses through heat insulation and an economic insulation thickness. Basic data on heat-insulation material (commercially available insulation thickness, price of insulation, heat conductivity of insulation, etc.) which are required for the calculation are written and stored in our data base.

Preglednica 2. *Drugi tehnično-ekonomski podatki*
Table 2. *Other technical and economic data*

temperatura hladnega sredstva temperature of cooling medium	-20,0°C
temperatura zunanjega zraka temperature of outside air	25,0°C
zunanji premer cevi outside diameter of pipe	0,089 m
hitrost toka zrak (prisilna konvekcija) velocity of air flow (forced convection)	1,00 m/s
koeficient emisije coefficient of emission	0.44
korekcija λ (zaradi toplotnih mostov) correction of λ (due to heat bridges - conductors)	0,0 W/mK
ekonomski izračun statični/dinamični economic calculation static/dynamic	dinamični dynamic
diskontni faktor - enačba (7) discount factor - equation (7)	b_2
doba trajanja izolacije service life of insulation	30 let 30 years
diskontna stopnja discount rate	8,00 %
letno povečanje stroškov hlajenja annual rise of cooling costs	5,00 %
letno povečanje stroškov vzdrževanja (r + g) annual rise of maintenance costs (r + g)	2,00 %
cena hlajenja price of cooling	12,22 USD/GJ
letno število obratovalnih ur number of operating hours per year	3000 h/leto 3000 h/year
stroški izoliranja costs of insulating	1,94 USD/(m^2.cm)
posredni stroški izoliranja indirect cost of insulating	0,0 USD/m

Preglednica 3. *Rezultat izračuna ekonomske debeline izolacije*
Table 3. *Results of economic insulation-thickness calculation*

ekonomska debelina izolacije economic insulation thickness	70,0 mm
srednja temperatura izolacije middle temperature of insulation	1,3°C
temperatura zunanje površine izolacije temperature of exterior insulation surface	22,6°C
toplotna prestopnost heat transfer coefficient	10,18 W/m^2K
toplotna prevodnost izolacije thermal conductivity of insulation	0,0342 W/mK
toplotna prehodnost izolacije overall heat transfer coefficient of insulation	0,221 W/m^2K
toplotni tok na meter cevi heat flow per meter of pipe	9,9 W/m
gostota topotnega toka heat flux	13,8 W/m^2
cena izolacije price of insulation	8,54 USD/m
stroški izoliranja costs of insulating	9,78 USD/m
indirektni stroški izoliranja indirect costs of insulating	0,0 USD/m
zunanji obseg izolacije size of insulation	0,719 m

6 OZNAKE
6 NOMENCLATURE

diskontni faktor	b		discount factor
skupni stroški investicije izolacijskega sistema	C_M	USD/m	total costs of investment in insulation system
skupni stroški toplotnih izgub in cevne toplotne izolacije	C	USD/m	total costs of heat losses and pipe heat insulation
minimum stroškov po statični ali dinamični metodi	C_S	USD/m	minimum of costs by static or dynamic method
cena toplotne izolacije	C_0	USD/m	price of heat insulation
stroški izoliranja	C'	USD/m^{2}cm	costs of insulating
posredni stroški izoliranja	C_R	USD/m	indirect costs of insulating
cena toplote	C_T	USD/GJ	price of heat
zunanji premer izolacije	d	m	outside diameter of insulation
faktor za uvedbo dinamične metode	f		factor for the dynamic method introduction
letno povečanje preostalih stroškov	g	%	annual rise of other costs
doba trajanja izolacije	n		service life of insulation
letno povečevanje stroškov hlajenja	p	%	annual rise of cooling costs
letno povečevanje stroškov vzdrževanja	r	%	annual rise of maintenance costs
debelina izolacije	s	m	insulation thickness
število obratovalnih ur na leto	t	h	number of operating hours per year
obrestna mera	z	%	interest rate
toplotni tok	Φ	W	heat flow

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Prejeto:
Received: 15.8.2000

Sprejeto:
Accepted: 10.11.2000

© Strojniški vestnik 46(2000)8, 580-588
ISSN 0039-2480
UDK 621.352.6:621.311.23
Strokovni članek (1.04)

© Journal of Mechanical Engineering 46(2000)8, 580-588
ISSN 0039-2480
UDC 621.352.6:621.311.23
Speciality paper (1.04)

Kogeneracija z gorilnimi celicami v stanovanjskih poslopjih

Residential Co-Generation Using Fuel Cells

Rodolfo Taccani

Kombinirana proizvodnja električne energije in toplote (KPETH - CHP) ali kogeneracija je idealna uporaba gorilnih celic. V tem prispevku so predstavljena načela delovanja in različni tipi gorilnih celic. Podrobno je predstavljeno delovanje kogeneracije z gorilnimi celicami. Predstavljeni so nekateri komercialno uporabni sistemi in prednosti gorilnih celic. Podrobno je analiziran obrat s trdnimi oksidnimi gorilnimi celicami in plinsko turbino (TOGC in PT - SOFC in GT). Nekateri preliminarni rezultati kažejo, da lahko dosežemo električni izkoristek okoli 65%, medtem ko je izkoristek po prvem glavnem zakonu nad 80%.

(Ključne besede: kogeneracija, celice gorilne, zgradbe stanovanjske, izkoristek električni)

Combined heat and power (CHP) or co-generation is an ideal application for the fuel cell. In this paper the working principle and the different types of fuel cells are briefly presented. The typical layout of a fuel cell co-generation system is described. Some of the commercially available systems are considered and the advantages of fuel cells are discussed. In particular an integrated plant with solid-oxide fuel cells and a gas turbine (SOFC+GT) is analysed. Some preliminary results obtained using a simulation program show that an electric efficiency of approximatively 65% can be obtained, while the First Law efficiency is over 80%.

(Keywords: cogeneration, fuel cells, residential buildings, electrical efficiency)

0 UVOD

Velika zanesljivost, majhen vpliv na okolje in prilagodljiva velikost so nekatere značilnosti, ki dajo gorilnim celicam lastnost, da so idealna tehnologija za kombinirano proizvodnjo električne energije in toplote (KPET) [1]. Glavni cilj tega prispevka je podati splošne informacije o delovanju različnih gorilnih celic in se osredotočiti na izdelavo celic, ki bi bile primerne za kogeneracijo.

Nadaljnje izboljšanje izkoristka gorilnih celic je mogoče doseči, ko delujejo pri višjih temperaturah (~1000 °C) in pod tlakom, tako da spreminjajo izhodne pline v elektriko v plinski turbini. Ta izboljšava je mogoča samo danes zaradi sedanjih izboljšav v oblikovanju mikroplinskih turbin, ki omogočajo precejšnji izkoristek v manjših enotah (50 do 100 kW). Z namenom, da bi analizirali delovanje takšnega sistema gorilnih celic in plinske turbine, je bil razvit simulirni program. Zadnji del prispevka poroča o nekaterih predhodnih rezultatih z uporabo programa, ki prikazuje, da lahko dosežemo zelo visok izkoristek.

0 INTRODUCTION

High reliability, low environmental impact and size flexibility are some of the characteristics that make fuel cells one of the ideal technologies for combined heat and power (CHP) generation [1]. The aim of this paper is to provide some general information about the working principle of different types of fuel cells and to focus on the aspects that make them suitable for co-generative applications.

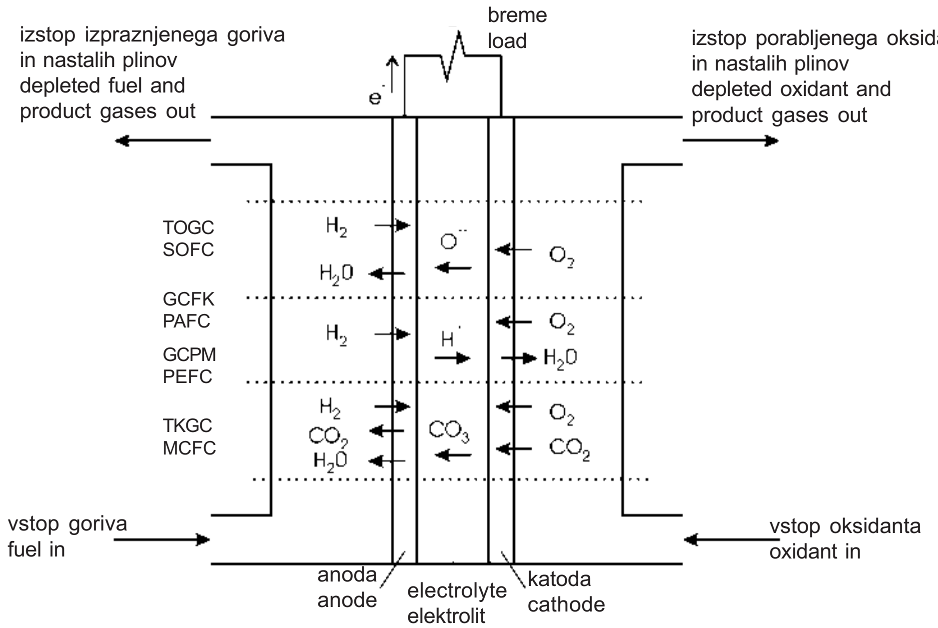
A further improvement in the efficiency of the fuel-cell co-generative system can be achieved when operating at high temperature (~1000 °C) and in high pressures, converting the exhaust stream to electricity via a gas turbine. This improvement has only recently been achieved because of advances in the design of micro gas turbines allowing an appreciable efficiency to be achieved even in small-size units (50 to 100 kW). In order to analyse the performance of these fuel-cell gas-turbine integrated systems a simulation program has been developed. The last part of the paper reports some of the preliminary results provided by the program, showing that a very high efficiency can be obtained.

1 GORILNE CELICE

Gorilne celice so elektrokemične naprave, ki spreminjajo kemično energijo neposredno v električno ([2] do [4] in [W1]). Osnovni del gorilne celice sestavlja elektrolit v stiku s porozno anodo in katodo na drugi strani. Shematična predstavitev gorilne celice je prikazana na sliki 1.

1 FUEL CELLS

Fuel cells are electrochemical devices that convert the chemical energy of a reaction directly into electrical energy ([2] to [4] and [W1]). The basic physical structure of a fuel cell consists of an electrolytic layer in contact with a porous anode and cathode on either side. A schematic representation of a fuel cell is shown in Figure 1.



Sl. 1. *Shematični prikaz posamezne gorilne celice*
Fig. 1. *Schematic of an individual fuel cell*

Osnovno načelo za vse gorilne celice je takšno, kakršno uporabljajo elektrokemične baterije, ki jih poznamo v številnih področjih vsakdanjega življenja. Največja razlika je v tem, da je kemična energija shranjena v sami bateriji. Ko se kemična energija spremeni v električno, moramo baterijo zavreči (navadna baterija) ali ponovno napolniti (obnovljiva baterija). V gorilni celici se kemična energija stalno dovaja k anodi (negativna elektroda), oksidant pa h katodi (pozitivna elektroda). Plinasto gorivo je shranjeno zunaj celice, kjer tečejo kemične reakcije. Dve glavni značilnosti gorilnih celic, ki sta pomembni z vidika današnjih interesov, sta razmeroma velik izkoristek in majhen vpliv na okolje. Ker gorilna celica ne dela kot termodinamični krožni proces, zanjo ne velja pojem mejnega toplotnega izkoristka, izkoristek pa je v mejah od 40 do 55% v primeru majhne kurilnosti goriva (SKV - LHV) in vode, kot izstopnega produkta (če uporabljamo vodik). Razvrstitev gorilnih celic po tipu elektrolita in po nekaterih tehničnih lastnostih je prikazana v preglednici 1.

The basic principles of a fuel cell are those of electrochemical batteries, which are involved in many activities of our everyday life. The big difference is that, in the case of batteries the chemical energy is stored in substances located inside them. When this energy has been converted to electrical energy, the battery must be thrown away (primary batteries) or recharged (secondary batteries). In a fuel cell, the chemical energy is provided by a fuel that feeds continuosly to the anode (negative electrode) and an oxidant that feeds continuosly to the cathode (positive electrode). These gaseous fuels are stored outside the cell in which the chemical reaction takes place. Two major fuel-cell characteristics that have been important in driving the recent interest are the combination of a relatively high efficiency and a very low environmental impact. Since a fuel cell does not operate as a thermodynamic power cycle, the notion of a limiting thermal efficiency imposed by the Second Law is not applicable. The efficiency of typical fuel-cell plants are in the range of 40 to 55%, based on the lower heating value (LHV) of the fuel, and the exhaust stream of water (if hydrogen is used). A classification of fuel cells by the type of electrolyte used and some technical specifications are reported in Table1.

Preglednica 1. *Lastnosti gorilnih celic*
Table 1. *Attributes of fuel cells*

	GCPM-PEMFC	AGC-AFC	GCFK-PAFC	TKGC-MCFC	TOGC-SOFC	MGC-DMFC
elektrolit electrolyte	polimerna membrana polymer membrane	KOH	fosforna kislina phosforic acid	staljen karbonat molten carbonate	keramika ceramic	polimerna membrana polymer membrane
temp. (°C)	70-80	80-100	200-220	600-650	800-1000	70-120
gostota current dens.	velika high	velika high	srednja moderate	srednja moderate	velika high	majhna low
izvajalec reformer	zunanji external	zunanji external	zunanji external	zunanji / notranji external / internal	zunanji / notranji external / internal	notranji internal
CO_2	da / yes	ne / no	da / yes	da / yes	da / yes	da / yes
CO	ne / no	ne / no	ne / no	da / yes	da / yes	da / yes
področje uporabe application area	vesoljske postaje space station	vesoljske uporabe space applications	električne uporabe power applications	pridobivanje elektrike power generation	pridobivanje elektrike power generation	transport transportation
izkoristek efficiency	50%	50%	50%	60%	60%	ni poznan N.A.

GCPM: gorilna celica s protonsko membrano
PEMFC: proton exchange membrane fuel cell
AGC: alkalna gorilna celica
AFC: alkaline fuel cell
GCFK: gorilna celica s fosforno kislino
PAFC: phosforic acid fuel cell
Za izkoristek so navedene samo okvirne vrednosti.
Efficiency values are only indicative values.
TKGC: taljena karbonatna gorivna celica
MCFC: molten carbonate fuel cell
TOGC: trdno oksidna gorilna celica
SOFC: solid oxide fuel cell
MGC: metanolna gorilna celica
DMFC: direct methanol fuel cell

2 SISTEMI GORILNIH CELIC

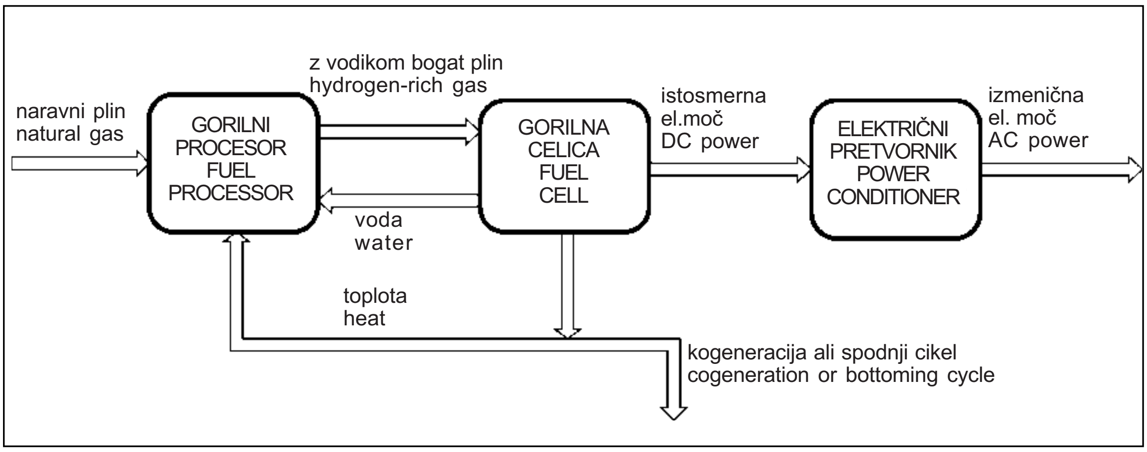
Gorilna celica povezuje vodik iz goriva in kisik iz zraka za pridobivanje električne energije (istosmernega toka), vode in toplote. Ta reakcija mora biti izvedena pri ustrezni temperaturi in tlaku. Okoli gorilne celice je treba izdelati sistem, ki celico oskrbuje z zrakom in svežim gorivom in pretvarja energijo v bolj uporabno obliko in odstranjuje izpraznjene reaktante in toploto, ki nastaja pri reakcijah v gorilni celici. Na sliki 2 je prikazan shematični pregled gorilne celice.

Posebno tedaj, ko uporabljamo nizkotemperaturne gorivne celice (t.j. AGC, GCPM, GCFK) in ko vodika ni na voljo, je treba preoblikovati (izločiti vodik) gorivo. Ta proces se izvaja v napravi, ki se

2 FUEL-CELLS SYSTEMS

The fuel cell combines hydrogen produced from the fuel and oxigen from the air to produce electrical power (DC), water and heat. This reaction must be carried out at a suitable temperature and pressure. A system must be built around the fuel cells to supply air and clean fuel, convert the power to a more usable form and remove the depleted reactants and heat that are produced by reactions in the fuel cell. In Figure 2 a schematic view of a fuel cells plant is shown.

Expecially when using low temperature fuel cell (i.e. AFC, PEM, PAFC) and hydrogen is not available it is necessary to reform (extract hydrogen) a hydrocarbon fuel. This process is carried out in an



Sl. 2. *Preprost blokovni diagram proizvodnje elektrike*
Fig. 2. *Power plant simplified block diagram*

imenuje *reformer*. Pridobi-vanje goriva je odvisno od goriva in tehnologije gorilnih celic. Slednji določa, katere sestavine so ustrezne in sprejemljive v gorivu. Na primer gorivo , ki se uporablja v GCPM, mora biti bogato z vodikom in mora imeti nizko koncentracijo CO (<20ppm). Medtem pa v TOGC-ju lahko uporabljamo metan in CO v celici. Če uporabljamo naravni plin, enega najpomembnejših goriv za kogeneracijo, moramo odstraniti žveplo in spremeniti plin v vodik v parnem reformerju. Koncentracijo CO moramo znižati z uporabo kemične preobrazbe in selektivnega katalitičnega oksidatorja.

Ko je gorivo pridobljeno, vstopi v elektrokemični del: v gorilno celico. Čeprav gorilna celica ni toplotni stroj, se gorivo vseeno proizvaja in mora biti odstranjeno. Glede na velikost sistema, temperature toplote in drugih zahtev, je treba toplotno energijo zavreči, uporabiti za pridobivanje tople vode ali spremeniti v elektriko v plinski turbini. Če se gorilna celica uporablja za pridobivanje izmenične električne moči, je potrebno, da sistem vsebuje razsmernik ter krmilnik toka, napetosti in nadzor frekvence.

appropriate deviced normally called *reformer*. The fuel processing depends on both the raw fuel and the fuel-cell technology. The latter determines what constituents are desirable and acceptable in the processed fuel. For example, fuel used in PEMFC needs to be hydrogen rich and have a very low CO concentration (<20 ppm), while SOFCs are capable of utilizing methane and CO within the cell. When using natural gas, one of the most desirable fuels for residential co-generation, sulfur has to be removed and the gas is converted to hydrogen in a steam-reforming reactor. CO concentration may be reduced using a shift conversion and a selective catalytic oxidizer.

When the fuel has been processed it enters the electrochemical section: the fuel cell. Although fuel cells are not heat engines, heat is still produced and must be removed. Depending upon the size of the system, the temperature of available heat and the requirements of the particular site, the thermal energy can either be rejected, used to produce hot water or converted to electricity via a gas turbine. If a fuel cell is used to supply AC, the system should include at least a DC to AC conversion unit, current, voltage and frequency control.

Preglednica 2. *Karakteristike gorilnih celic*
Table 2. *Pros and cons of fuel cells*

Karakteristike, ki jih ponuja gorilna celica   ***Characteristics that fuel cells plants offer***
mirujoči deli v energetskem pretvorniku   no moving parts in the energy converter
tiho delovanje   quiet
velika možna razpoložljivost / zanesljivost / trajnost nizkotemperaturnih enot   demonstrated high availability / reliability / endurance of lower temperature units
prilagodljivost za gorivo   fuel flexibility
dobro obnašanje pri delovanju pri neračunski obremenitvi   good performance at off-design load operation
modularna sestava za pokrivanje bremena   modular installation to match load
(nanadzorovano) delovanje na daljavo   remote / unattended operation
hitro prilagajanje na obremenitev   rapid load following capacity
Splošne negativne lastnosti gorivne celice   ***General negative features of fuel cells***
visoka cena na trgu   market entry cost high
zanesljivost / trajanje visokotemperaturnih delov neizkazano   reliability / endurance of higher temperature units not demonstrated
v nergetiki nepoznana tehnologija   unfamiliar technology to the power industry
ni infrastrukture   no infrastructure

3 KOGENERACIJSKI SISTEMI

Številna podjetja, General Electric, American Power Corp., Northwest Power Systems, Avista Labs, Ballard ([W2] do [W4]) se ukvarjajo z raziskavami in razvojem kompletnih kogeneracijskih sistemov v obsegu od nekaj kW do 250 kW in več.

3 CO-GENERATIVE SYSTEMS

Several companies, General Electric, American Power Corp., Northwest Power Systems, Avista Labs, Ballard ([W2] to [W4]) are involved in the research and development of complete co-generative systems, ranging from a few kW to 250 kW and over.

Najbolj uspešen doslej je bil PC 25 izdelan v podjetju ONSI (184 postrojenj prodanih v 14 državah). PC 25 [5] je paketna, samooskrbovalna fosforno kislinska gorilna celica z električno močjo 200 kW. Pri polni moči sistem pridobi 223 kW_t uporabne toplote pri temperaturah med 40 °C in 80 °C. Izkoristek pridobivanja elektrike je 40% glede na kurilnost (naravni plin). Izkoristek ostaja skoraj na istem nivoju med četrtinsko in polno obremenitvijo. Celotni izkoristek je 84% pri 100% obremenitvi. Raven zvočnega tlaka je pod 60 dBA pri 10m od naprave. Elektrarna vsebuje vse potrebne komponente: za preobrazbo naravnega plina v (izmenično) električno moč, pridobivanje uporabne toplote za stranke in oddajanje odvečne toplote v zrak. Cena je približno 3000 $/kW. Obstajajo številni demonstracijski projekti, ki bazirajo na GCPM, toda njihova pojava na trgu je stvar bodočnosti. Preglednica 3 prikazuje nekatere tehnične podrobnosti elektrarne, ki bo v kratkem na trgu.

The most successful to date has been the PC 25 produced by ONSI Corporation (184 plants sold in 14 countries).The PC 25 [5] is a packaged, self-contained, phosforic acid fuel-cell power plant with a continuous electrical rating of 200 kW. At full load the system provides 223 kW_t of useful heat at temperatures between 40°C and 80°C. The electrical generation efficiency is 40% on a lower heat value (natural gas LHV) basis. The efficiency remains almost at the same level at loads between one quarter and full load. The total efficiency is 84% at 100% load. The sound pressure level is below 60 dBA at 10 meters from the plant. The power plant includes all components required to convert natural gas into utility AC power, provide useful heat to the customer and reject excess heat to air. The cost is approximatively $3000/kW. There are many other demonstrative co-generative systems based on PEMFC technology, but these have yet to enter the market. Table 3 shows some technical specifications of a power system which will soon be available.

Preglednica 3. *Karakteristike skoraj komercialne elektrarne, ki temelji na PEMFC*
Table 3. *Specification of a near commercial power system based on PEMFC*

izhod output	7 kW zvezno 15 kW vrhunec 7 kW continuous 15 kW peak
napetost voltage	120/240 @60Hz 100/230V @50Hz
izkoristek cikla simple cycle efficiency	40% @2 kW izhod / output 29% @7 kW izhod / output
izkoristek kogeneracije cogen efficiency	> 75%
uporabljena odpadna toplota recoverable waste heat	> 2 kWh @2 kW izhod / output > 11 kWh @7 kW izhod / output
gorivo fuel	naravni plin utekočinjen propan natural gas liquid propane
intervali servisiranja maintenance intervals	9000 h
projektna doba trajanja design life	15 let 15 years

4 TRDNO OKSIDNA GORILNA CELICA V KOMBINACIJI S PLINSKO TURBINO

Plinske turbine v kombinaciji s trdno oksidno gorilno celico (TOGC - PT) so bile obravnavane v številnih študijah ([6] do [9]), toda številne so obravnavale MW postrojenja. Dandanes so na voljo številne nove možnosti, ker je interes za proizvodnjo elektrike v mikro turbinah (30 do 200 kW). Številna podjetja, ki se ukvarjajo z mikro turbinskimi generatorji, sedaj oglašajo svoje izdelke za končne uporabnike, družbe in proizvajalce energije ([10] in [1]). To pospešuje uporabo TOGC s PT celo v stanovanjskih objektih, kjer 200 kWe plinska turbina daje izhodno moč 30kWe.

4 SOLID OXIDE FUEL CELLS IN COMBINATION WITH A GAS TURBINE

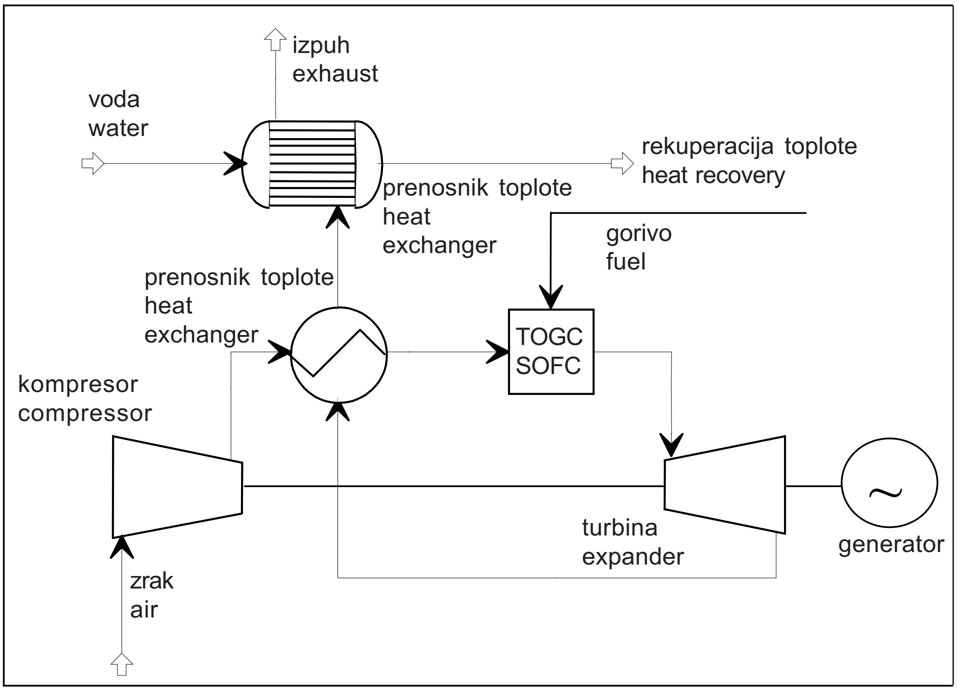
Studies of a gas-turbine cycle with a solid-oxide fuel cell (SOFC-GT) have been carried out by several researchers ([6] to [9]), but most of them have been considering MW sized power plants. Recently, new opportunities have arisen as there has been a sustained interest in power applications for microturbines (30 to 200 kW), and several microturbine-generator manufacturers are now announcing commercial availability of their products, targetting end-users, utilities and energy service providers ([10] and [1]). This context facilitates the SOFC-GT integration even for residential co-generation, where, for example, the gas-turbine output of a 200 kWe system is calculated to be approximatively 30 kWe.

V TOGC visoke temperature (~1000ºC) zagotavljajo, da vse sestavine goriva, v kombinaciji s potrebno količino vodne pare, oksidirajo v trenutku in dosežejo termodinamično ravnotežje, če dovedemo zadostno količino zraka. Zaradi visokih temperatur so drage reakcije nepotrebne kar omogoča neposredno porabo goriva v sami celici. Ker je trdni elektrolit normalno zelo stabilen, ni premikanja elektrolita. Raziskujejo dva različna modela: cevni model in ravninski model. V našem delu smo uporabili cevni model.

Shematična predstavitev krožnega procesa je prikazana na sliki 3. Prej obdelano gorivo (metan) in oksidant (zrak) vstopata v gorilno celico po kompresiji. Oksidacija poteka večinoma v gorilni celici. Celotna reakcija pa se konča v zgorevalni komori. Zgoreli plini pod tlakom odtekajo skozi turbino. Izstopna para iz turbine zagotavlja toploto ne samo za pripravo goriva, ampak tudi za pridobivanje tople vode.

In SOFCs the high temperatures (~1000 °C) ensure that all fuel compositions, when combined with the necessary amount of water vapor, will oxidize rapidly and reach thermodynamic equilibrium if sufficient air is provided. The high temperature makes expensive reactions unnecessary and permits direct processing of fuel in the fuel cell itself (i.e. internal reforming). Because the solide-oxide electrolyte is normally very stable, no electrolyte migration problems exist. Basically two different designs are under development: the tubular design and the planar design. In our work we have been considering the tubular design.

A schematic view of the considered cycle is presented in Figure 3. The preprocessed fuel (methane) and the oxidant (air) enter the fuel cell after being compressed. The fuel oxidation reaction occurs predominantly within the fuel cell. The reaction is completed in a combustion chamber. The pressurized-fuel combustion products are exhausted through a turbine.



Sl. 3. *Obtočni diagram krožnega procesa s TOGC*
Fig. 3. *SOFC-GT cycle flowsheet diagram*

Za raziskavo procesa smo uporabili simulirni računalniški program, ki je vseboval tudi simuliranje trdno oksidne gorilne celice, napisane v jeziku Fortran ([11] in [12]). Simulirni model je bil razvit za raziskovanje:
- delovne temperature in tlaka,
- sestav plinov reaktantov,
- izkoristka uporabe goriva.

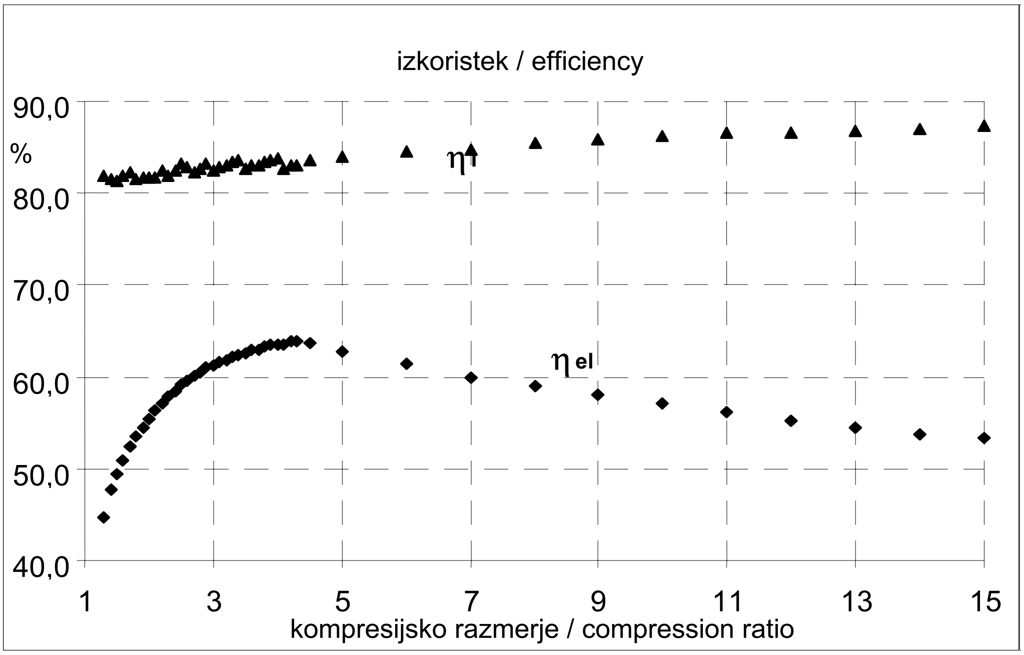
Načrtovani izkoristki za glavne komponente so: kompresor in turbina (izentropno) 79,5% in 84,5%; generator in razsmernik: 92%.

The cycle was studied using commercial process-simulation software integrated with a solid oxide fuel cell steady-state operation simulator that has been implemented using Fortran ([11] and [12]). The simulation model has been developed with the objectives of evaluating the performance of the system when varing:
- operating temperature and pressure,
- reactant gases composition,
- fuel utilization coefficient.

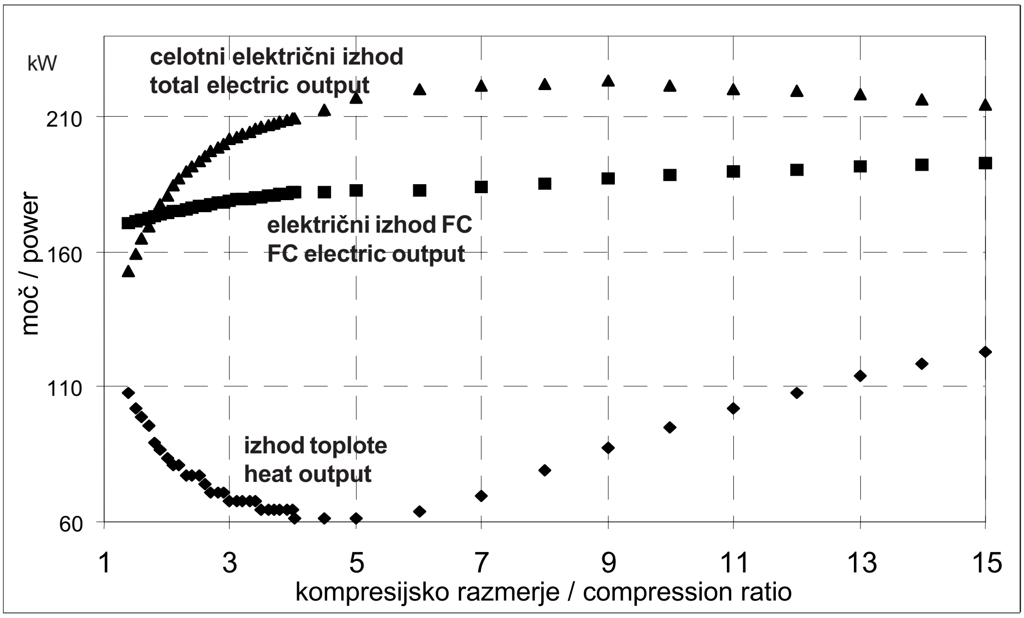
The assumed efficiency for the major components are: compressor and turbine (isoentropic) 79.5% and 84.5% respectively; generator and DC/AC conversion: 92%.

Predhodni rezultati so predstavljeni na sliki 4 in sliki 5. Slika 4 prikazuje električni izkoristek in izkoristek po prvem glavnem zakonu v odvisnosti od kompresijskega razmerja. Največji električni izkoristek dobimo pri kompresijskem razmerju 4,3, in sicer je izkoristek 63,8%, medtem ko je izkoristek po prvem glavnem zakonu 83,1%. Ta električni izkoristek je zelo visok, če ga primerjamo z običajnimi sistemi proizvodnje elektrike, tudi večjimi. Na sliki 5 je prikazana izhodna moč v odvisnosti od kompresijskega razmerja. Kjer je električni izkoristek višji, je izhod toplote na minimumu. Nato se zopet

Preliminary results are reported in Figure 4 and Figure 5. Figure 4 shows the electric and First Law efficiency as a function of the compression ratio. The maximum electric efficiency is obtained when operating at a compression ratio of 4.3 and it is 63.8%, while the First Law efficiency is 83.1%. This electric efficiency is very high when compared with any conventional power-generation systems, even those of larger size. In Figure 5 the system power output is plotted versus compression ratio. Where the electric efficiency is higher the heat output is at a minimum, and then increases again because the power system



Sl. 4. *TOGC - PT: Električni izkoristek, izkoristek po prvem glavnem zakonu v odvisnosti od kompresijskega razmerja*
Fig. 4. *SOFC - GT: Electric efficiency, First Law efficiency as a function of the compression ratio*



Sl. 5. *SOFC - GT: Električni izhod v odvisnosti od kompresijskega razmerja*
Fig. 5. *SOFC - GT: Power outputs as a function of the compression ratio*

zvišuje, ker je sistem opremljen s pomožnim gorilnikom, ki skrbi za stalno temperaturo vhodnih plinov v gorilno celico. Električna izstopna moč turbine je največja pri eni petini celotne moči.

is provided with an auxiliary burner in order to keep constant the temperature of the reactant gases entering the fuel cell. The turbine electrical output is at maximum of one fifth of the total power.

5 SKLEP

V prispevku je predstavljeno delovanje gorilne celice predvsem z vidika, da so gorilne celice ugodne za kogeneracijo v stavbah. Predstavljeni so nekateri sistemi. Predlagan je bil sistem s trdno oksidno gorilno celico in plinsko turbino, ki je bil simuliran z matematičnim modelom. Glavni rezultati tega dela so:

- gorilna celica je idealna tehnologija za kogeneracijo v stavbah,
- danes je samo en sistem kogeneracije na voljo na trgu, toda številni proizvajalci zatrjujejo, da bodo njihovi sistemi kmalu na voljo,
- trdno oksidna gorilna celica se teoretično lahko poveže z mikroplinsko turbino, tako da proizvaja elektriko z visokim izkoristkom tudi pri manjših močeh,
- izračunani električni izkoristek sistema je 63,8%.

Sistemi TOGC in PT so zelo obetajoči tudi za stanovanjske enote, čeprav je treba ugotoviti možnost in trpežnost v primerjavi z običajnimi sistemi.

Široka uporaba bo dala gorivnim celicam cenovno primerljivost z drugimi sistemi. Za gorilne celice že desetletja trdijo, da se bodo pocenile, toda čas za to še ni prišel. Potreba po gorilnih celicah v avtomobilski industriji in nizkoemisijskih vozilih naj bi vplivala na razvoj in možnost, da gorilne celice postanejo cenovno ugodne.

5 CONCLUSION

In this paper the working principle of fuel cells has been briefly described, focusing on those aspects that make this technology attractive for residential co-generation. Some of the existing systems have been presented. Then a system based on solid-oxide fuel cells and a gas turbine has been proposed and analysed using a mathematical model. The main conclusions of this work are:

- fuel cells seem to be one of the ideal technology for residential co-generation;
- to date only one co-generative system has reached commercial maturity, but many manufacturers are now announcing the commercial availability of their products;
- Solid-oxidef fuel cells can be well integrated, theoretically, with a micro gas turbine to yield high-efficiency power-generation cycles, even in the sub-MW power range;
- the calculated electrical efficiency of the system is 63.8%.

SOFC-GT systems seem to be very attractive even for residential-size power units, although reliability and durability comparable with conventional power plants and lower cost, essential to market entry, have still to be proved.

However, their use will become widespread when they become cost-competitive. Fuel-cell advocates have been promising reductions in price for decades, but the time might actually be at hand. The need in the automotive industry for fuel cells in zero-emission vehicles may fuel an explosion in the technology development and manufacturing capability, finally bringing to reality the time of low-cost fuel cells.

Zahvala

Hvaležen sem inž. Riccardu Valente-ju za sodelovanje in razvoj računalniškega programa za modeliranje sistema z TOGC in PT.

Acknowledgement

I am very grateful to ing. Riccardo Valente for his decisive contribution to developing the computer program for modelling the SOFC-GT system.

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6 REFERENCES

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Medmrežje
In the Web
[W1] http://www.fuelcells.org
[W2] http://www.northwestpower.com
[W3] http://www.gefuelcell.com/homegen_prod_desc.html
[W4] http://www.ballard.com

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Prejeto:
Received: 15.8.2000

Sprejeto:
Accepted: 10.11.2000

© Journal of Mechanical Engineering 46(2000)8,589-590
ISSN 0039-2480
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