© Strojniški vestnik 47(2001)7,300-312
ISSN 0039-2480
UDK 621.57:536.7:004.94
Predhodna objava (1.03)

© Journal of Mechanical Engineering 47(2001)7,300-312
ISSN 0039-2480
UDC 621.57:536.7:004.94
Preliminary paper (1.03)

Model in simuliranje delovanja odprtega hladilnega postopka brez uporabe CFC

Design and Simulation of a CFC-Free Open Air-Conditioning System

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V prispevku sta opisana model in delovanje hladilnega postopka, ki za hladilno sredstvo uporablja zrak. V uvodu so opisana običajna hladiva, ki jih primerjamo z alternativnim hladivom - zrakom. Opisana je sestava sistema in termodinamični model. Model je namenjen za študij delovanja CFC-prostega zračnega klimatizacijskega sistema, pri katerem so ob referenčnih podatkih izbrani parametri spremenljivke, preostali pa konstante. Karakteristične krivulje so definirane s hladilnim številom COP, temperaturo ohlajenega zraka, kompresijskim tlakom in drugimi parametri. Karakteristike sistema so dobljene s dejanskimi obratovalnimi razmerami. Razmerja med specifičnimi parametri in veličinami so predstavljena z diagrami. Obrazložene so izgube sistema zaradi neizentropne kompresije v centrifugalnem kompresorju, ekspanzije v turbinskem ekspanderju in nepovračljivosti postopka prenosa toplote v prenosniku toplote.

(Ključne besede: sistemi hladilni, naprave klimatske, CFC, modeli termodinamični, simuliranje)

In this paper a model is presented and its operation of a cooling process that uses air as the cooling medium. In the introduction, conventional refrigerants are compared with an alternative refrigerant - air. The system construction and the thermodynamic model are presented. The model is used to study the performance of a CFC-free air-conditioning system, in which some parameters of the individual components are variable while the others are constant for the referenced data. The characteristic performance curves of a simple air-conditioning system are defined as the coefficient of performance (COP), the temperature of the cooled outlet air, the compression pressure and other parameters. Using actual data from an air-conditioning system we obtained performance curves of the system. Relations between specific parameters and quantities are presented in diagrams. Losses in the system, resulting from non-isentropic compression in the centrifugal compressor and expansion in the turbine expander and the irreversible losses of the heat transfer in the heat exchanger are explained.

(Keywords: cooling systems, air conditioning systems, CFC free, thermodynamic models, simulation)

0 UVOD

Teoretično bi lahko uporabili katerikoli plin kot hladivo, praktično pa uvrščamo med hladiva samo tiste pline, ki izpolnjujejo določene pogoje: fizikalne, kemične in fiziološke. Ker je hladivo v samem hladilnem krožnem postopku izpostavljeno širokemu območju tlakov in temperatur, so možnosti fizikalnih in kemičnih sprememb zelo velike. Izbira hladiva za posebno uporabo je pogosto odvisna od lastnosti, ki niso neposredno vezane na zmožnost prenašanja toplote. Na primer vnetljivost, strupenost, gostota, viskoznost, dostopnost in druge podobne karakteristike so pogosto odločujoči dejavniki.

0 INTRODUCTION

Theoretically, any gas can be used as a refrigerant, in practice, however, only certain gases that meet various physical, chemical and physiological requirements are used. Every refrigerant is exposed to a wide range of pressures and temperatures in a cooling/refrigerating cycle, so the possibility of physical and chemical changes must be great. The choice of refrigerant for a particular application often depends on properties not directly related to its ability to remove heat, for example, flammability, toxicity, density, viscosity, availability and other similar characteristics are often the most important factors.

0.1 Obstoječa hladiva: Halogeno-ogljikova hladiva

Halogeno-ogljikova hladiva so sintetična, na osnovi ogljika, ter vsebujejo ogljik in halogene atome, kakor so fluor, klor in brom. Lahko vsebujejo tudi vodik. Primeri vključujejo klorfluorogljike (CFCs), fluorogljikovodike (HFCs), halone itn. Halogenirana hladiva se uporabljajo v številnih uporabah: hlajenje in klimatizacija. Halogeni zadevajo domač in mednaroden vidik, ker povzročajo tanjšanje ozonske plasti (ODP), prispevajo pa tudi h globalnemu učinku tople grede (GWP). Slednji je dandanes bolj znan kot sprememba podnebja, ki je povezana s kožnim rakom in zmanjšanjem odpornosti človeškega imunskega sistema. Tanjšanje ozonske plasti lahko vpliva na življenje rastlin do točke, od katere vodi pot do pretrganja celotne prehranjevalne verige. Te kemijske sestavine so znane pod kratico ODS.

Najbolj pogosta hladiva so [1]:

- izpeljanke metana: R11, R12, R13, R14, R22, R23, R50
- izpeljanke etana: R113, R114, R115, R152a
- azeotropne zmesi: R500 (R12+R152a), R512 (R22+R115), R503 (R23+R13)

0.2 Alternativno hladivo: Zrak

Zrak [2] je naravno, okolju popolnoma neškodljivo hladivo. Poleg tega je tudi najcenejše. Okoljska vprašanja o tanjšanju ozonske plasti, učinku tople grede in vedno bolj nujno potrebna zakonodaja so obnovili zanimanje za *alternativne tehnologije hlajenja*.

Zračni hladilni sistemi imajo določene specifične prednosti, ki so značilne za vse potencialne uporabe:

- Delovno sredstvo (zrak) je naravno, okolju neškodljivo, popolnoma nenevarno in nestrupeno.
- Sistemska oprema zračnega kroga je zelo zanesljiva, s tem pa so manjši stroški vzdrževanja in obraba sistema.
- Učinkovitost sistema zračnega kroga se ne zmanjšuje tako kakor pri parno-kompresijskih enotah, če obratujejo zunaj predpisanega območja.
- Sistem zračnega kroga lahko daje toploto pri uporabni temperaturi, če deluje kot hladilni sistem.

Uporaba zraka kot hladiva temelji na načelu plinske izentropne ekspanzije pri določeni temperaturi, končna izstopna temperatura plina pri nekem novem tlaku pa je zato nižja. Ohlajeni plin, v tem primeru zrak, lahko potem uporabimo kot hladivo, in sicer neposredno v odprtem sistemu ali posredno skozi prenosnik toplote v zaprtem sistemu. Izkoristek takih sistemov je omejen z velikostjo izkoristka kompresije in ekspanzije, prav tako z učinkovitostjo prenosa toplote v prenosniku toplote. Prvotno so se uporabljali počasi tekoči kompresorji in turbine. Majhen izkoristek in zanesljivost takih strojev so bili

0.1 Existing refrigerants: Halocarbons

Halocarbons are synthetic, carbon-based compounds that contain carbon and halogen atoms such as fluorine, chlorine and bromine. They may also contain hydrogen. Examples of halocarbons include chlorofluorocarbons (CFCs), hydrofluorocarbons (HFCs), halons, etc. Halocarbons are found in a number of applications including refrigeration and air-conditioning systems. Halocarbons are of both national and international concern, because they cause stratospheric ozone depletion (Ozone Depletion Potential - ODP) and also contribute to global warming (Global Warming Potential - GWP). Today, global warming is better known as climate change, which in turn has been linked to cataracts, skin cancer and human immune-system suppression. Decreased ozone levels can affect plant life to the point where it leads to a disruption of the entire food chain. Such chemicals are known as ozone-depleting substances (ODS).

The most common refrigerants are [1]:

- Methane Series R11, R12, R13, R14, R22, R23, R50
- Ethane Series R-113, R-114, R-115, R152a
- Azeotropic Blends: R500 (R12+R152a), R512 (R22+R115), R503 (R23+R13)

0.2 An alternative refrigerant: Air

Air [2] is by its nature the safest and cheapest refrigerant. Environmental concerns about ozone depletion, global warming and increasingly stringent legislation have renewed interest in *alternative refrigeration technologies*.

Air-cycle systems have specific advantages that apply to all potential applications:

- The working fluid (air) is free, environmentally benign, totally safe and non-toxic.
- Air-cycle equipment is extremely reliable, thereby reducing maintenance costs and system down-time.
- The performance of an air-cycle unit does not deteriorate as much as the performance of a vapour-compression unit when operating away from its design point.
- When an air-cycle unit is operating in a refrigeration cycle it can also produce heat at a useful temperature.

The use of air as a refrigerant is based on the principle that when a gas expands isentropically from a given temperature its final temperature at the new pressure is much lower. The resulting cold gas, in this case air, can then be used as a refrigerant, either directly in an open system, or indirectly by means of a heat exchanger in a closed system. The efficiency of such systems is limited to a great extent by the efficiencies of compression and expansion, as well as those of the heat exchangers employed. The first systems used slow compressors and turbines, however, the low efficiency and reliability of these glavni dejavniki za zamenjavo teh sistemov s parno-kompresijskimi. Klub temu je prišlo do razvoja rotirajočih kompresorjev in turbin ter s tem tudi do višjih izkoristkov in večje zanesljivosti zračnih krogov. Napredek turbinske tehnologije skupaj z razvojem zračnih ležajev in keramičnih komponent ponujajo nadaljnje izboljšave. Skupaj z novimi zgoščenimi prenosniki toplote z boljšimi karakteristikami prenosa toplote postajajo taki sistemi vedno bolj konkurenčni že znanim parno-kompresijskim sistemom in vsekakor tudi sistemom na tekoči dušik.

V nadaljevanju je prikazan potek računalniškega simuliranja zračnega hladilnega sistema brez uporabe CFC, ki lahko simulira dejanski hladilni sistem. Za simuliranje smo vzeli zračni hladilni postopek z odprto zanko, torej odprt sistem s hladivom na zrak.

machines were the main reasons why they were replaced with vapour-compression machines. Despite their lack of popularity there were developments in rotary compressors and turbines that resulted in high efficiencies and higher reliabilities. The development of turbine technologies with air bearings and ceramic components offered further improvements. Together with new compact heat exchangers and their higher heat-transfer characteristics these systems have become more and more competitive with vapour-compression systems.

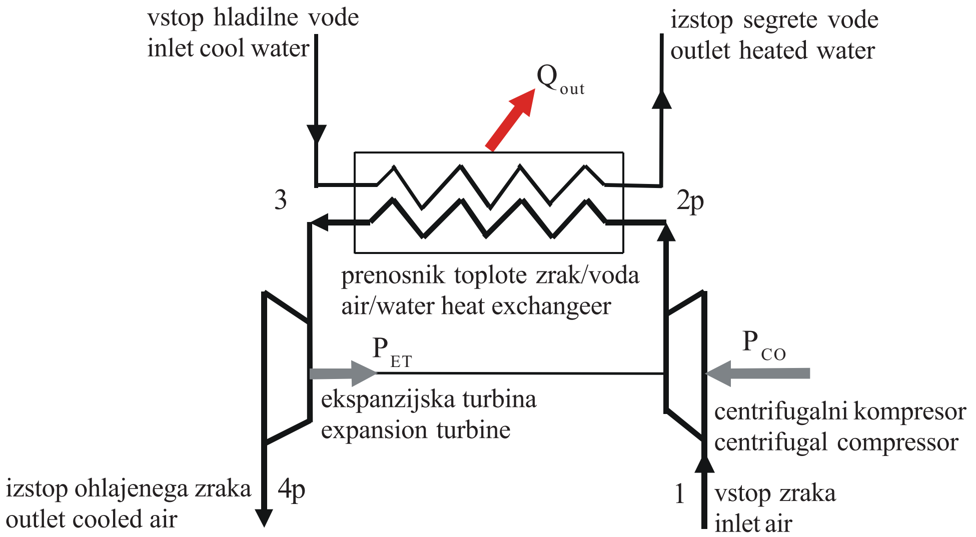
In this paper we present a computer-simulation program of CFC-free air-conditioning system that can simulate the performance of actual systems. For our simulation we chose an open-cycle air-conditioning system with air as the refrigerant.

1 OBLIKOVANJE SISTEMA

Hladilni sistem je sestavljen iz centrifugalnega kompresorja, prenosnika toplote zrak/voda in ekspanzijske turbine.

1 DESIGN OF THE SYSTEM

The cooling system is composed of a centrifugal compressor, on air/water heat exchanger and an expansion turbine.



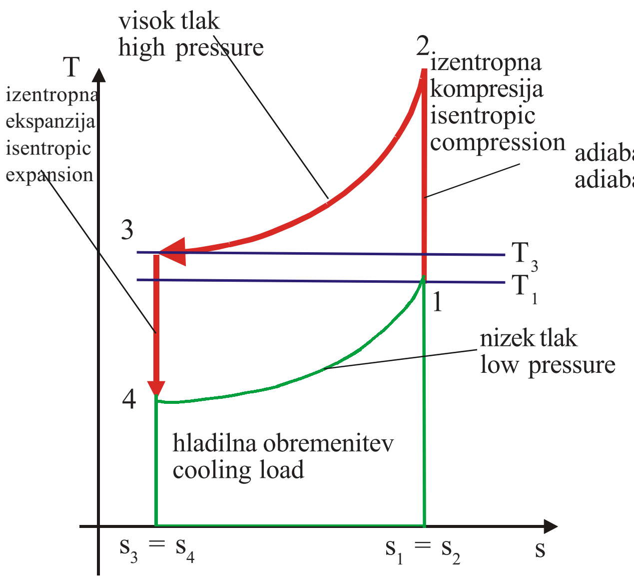
Sl. 1. *Shema preprostega (odprta zanka) zračnega hladilnega sistema*
Fig. 1. *Shematic diagram of a simple (open-loop) air-conditioning system*

Na sliki 2 sta prikazana diagrama T-s in kratek opis za odprti zračni proces [3].

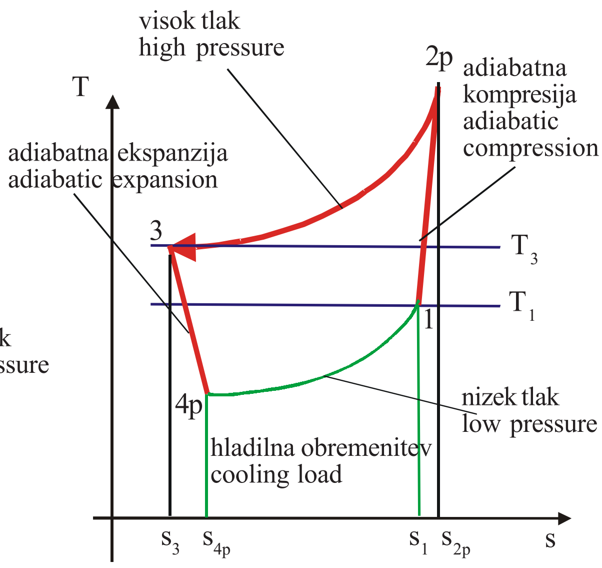
Obrnjen Jouleov proces je sestavljen iz treh korakov (sl. 2a). Prvi korak je kompresija delovnega medija, drugi korak je izmenjava toplote med delovnim (zrak) in hladilnim (voda) sredstvom in tretji korak ekspanzija delovnega medija. Izvedba obrnjenega Jouleovega procesa je lahko mogoča z različnimi kompresorji, prenosniki toplote in ekspanzijskimi turbinami. Delo, pridobljeno pri ekspanziji, se uporabi za kompresijo (sl. 1). Če sta delo kompresije in ekspanzije enakega reda velikosti, bo že majhno povečanje dela kompresije ali majhen padec dela ekspanzije poguben za ves proces. Zaradi tega je eden od odločilnih problemov doseganje

In Fig. 2 T-s diagrams and a short description of the open-air process are presented [3].

The reversed Joule cycle is composed of three steps (Fig. 2a). First, a working medium is compressed, then heat is exchanged between the working medium (air) and the cooling medium (water), finally, the medium is expanded. The reversed Joule process can be realised with many different compressors, heat exchangers and expanders. The work gained by the expansion is used for the compression (Fig. 1). As they are both of the same order of magnitude, a small increase in the compressor work or a small decrease in the expander work will be detrimental to the process. One of the crucial problems is, therefore, to achieve sufficiently high efficiencies in



Sl. 2a. *Izentropna kompresija in ekspanzija (idealna kapljevina brez trenja)*
Fig. 2. *Isentropic compression and expansion (ideal fluid with no friction)*



Sl. 2b. *Adiabatna kompresija in ekspanzija (realna kapljevina s trenjem)*
Fig. 2b. *Adiabatic compression and expansion (real fluid with friction)*

primerno visokih izkoristkov kompresorja in ekspanderja. To še posebno drži pri majhnih napravah, pri katerih imajo tradicionalni turbo-stroji majhne izkoristke.

Tehtnost izkoristkov je prikazana v diagramu T-s (sl. 2b). T je absolutna temperatura in *s* je entropija. Če se entropija med kompresijo in ekspanzijo močno zvečuje, je to ekvivalentno majhnemu izkoristku procesa. Hladilna moč se zelo zmanjša.

Izentropen proces ni nikoli 100-odstotno izvedljiv. Če bi bilo to mogoče, bi pomenilo, da v procesu ni spremembe entropije (proces kompresije in ekspanzije). Dejansko je proces kompresije, prenosa toplote in ekspanzije vedno adiabaten. Torej je delo kompresorja in ekspanderja omejeno z izentropnim izkoristkom. Proces prenosa toplote poteka pri nespremenljivem tlaku. Toda to je le idealna predpostavka, kajti dejansko pride do najmanjšega znižanja tlaka.

Glavna kriterija pri izbiri komponent zračnega hladilnega sistema sta določitev primerne velikosti enote prostora, ki ga moramo ohladiti, in razmerje energijske učinkovitosti (EER) – hladilno število (COP) sistema. EER (ali COP) je izračunan z deljenjem hladilne moči (BTU/h – British Thermal Unit ali W-moč v vatih) in vložene električne energije (W-moč v vatih). Višji ko je rezultat, večji je izkoristek hladilne naprave.

2 TERMODINAMIČNI MODEL ZRAČNEGA HLADILNEGA SISTEMA

Termodinamični model, ki simulira delovanje preprostega zračnega hladilnega sistema je bil razvit za določevanje učinkovitosti sistema. Model se tudi

the compressor and expander. This is especially true for small machines, where traditional turbo-machines have a low efficiency.

The importance of the efficiencies is best shown in a T-s diagram (Fig. 2b). T is the absolute temperature and s is the entropy. If the entropy increases significantly during the compression or expansion, this is equivalent to low efficiency. The refrigeration power decreases immensely.

The isentropic process cannot be 100% practicable. If it were, this would mean that there is no change of entropy during the process of compression and expansion. In fact the process of compression, heat exchange and expansion is always adiabatic (real fluid). So the compressor's and expander's mechanical work (power) is limited by the isentropic efficiency. The process of heat exchange in a heat exchanger flows at constant pressure. But that is just an ideal supposition, because in reality there is always some pressure drop.

The main criteria when choosing the components of an air-conditioning system are to determine the correct size of the unit for the area to be cooled and to look for a unit with a high energy-efficiency ratio (EER) or coefficient of performance (COP). An EER (or COP) is computed by dividing the cooling output (BTU/h or W) by the electrical input (W). The higher the result, the more efficient is the appliance.

2 THERMODYNAMIC MODEL OF THE SYSTEM

A thermodynamic model that simulates the working of a simple air-conditioning system has been developed for determining the performance of a sysuporablja za napovedovanje optimalnih vrednosti potrebnih parametrov.

Vsaka komponenta sistema mora biti pravilno izbrana in povezana z drugo komponento. Med seboj povezane komponente pomenijo sistem. V našem primeru imamo opravka s hladilnim sistemom. Delovno sredstvo je zrak, zato sta v sistemu zračni masni in toplotni tok.

Model simuliranja prikazuje, kaj se dogaja z določenimi parametri znotraj hladilnega sistema in kako ti vplivajo med seboj. V ta namen uporabljamo računalniški program, ki je napisan v programskem jeziku FORTRAN. To je preprost model, kjer sodeluje veliko spremenljivih parametrov, ki jih poda uporabnik simulirnega progama. Nespremenljivi parametri, kakršna sta atmosferski tlak (p_1) in plinska konstanta zraka (R) ter drugi, so vstavljeni v program. Prilagodljivost je ena od glavnih karakteristik simuliranja.

2.1 Vrednosti parametrov

Nespremenljivi parametri (vstavil jih je programer modela simuliranja in so fiksni):
– plinska konstanta zraka: R = 287 J/kgK,
– specifična toplota zraka: $c_{\mathrm{p\,zrak}}$ = 1005 J/kgK,
– atmosferski/začetni tlak: $p_1 = p_4 = p_4p$ = 0,101325 MPa (opomba: tlak pred kompresijo in po ekspanziji je enak atmosferskemu tlaku),
– razmerje specifičnih toplot za zrak: χ = 1,4,
– specifična toplota vode: $c_{\mathrm{p\,v}}$ = 4200 J/kgK,
– entropija in entalpija je enaka nič pri (zrak): p_0 =1 atm in T_0 = 78,8 K;

Spremenljivi parametri (poda jih uporabnik modela simuliranja):
– vstopna temperatura zraka: T_1 = 303 K,
– prostorninski tok zraka: V_1 = 0,1 m^3/s,
– tlak kompresije: $p_2 = p_2p = p_3$ = 0,2 MPa,
– (opomba: tlačni padec v prenosniku toplote je zanemarjen),
– izkoristek kompresije: η_{CO} = 0,85,
– masni tok hladilne vode: m_{v} = 0,083 kg/s,
– vstopna temperatura hladilne vode: T_{IN} = 288 K,
– toplotna prehodnost prenosnika toplote: k = 1342 W/m^{2}K,
– površina prenosnika toplote: A = 0,08 m^2,
– stopnja izmenjave temperature v prenosniku toplote: ϕ = 0,554 [6],
– izkoristek ekspanzije: η_{ET} = 0,85,

Ti parametri so vrednosti simuliranja referenčnega hladilnega procesa.

2.2 Zaporedje izračuna parametrov

A) Prvi korak:
Proces ***kompresije*** od točke 1 do točke 2p

Izračun specifične entropije in specifične entalpije za točko 1:

tem. The model is also used for predicting the optimum formation of included parameters.

Each component of a system has to be defined and connected properly. Furthermore, these connected components comprise a composition, which is called the system. In our case we are dealing with an air-conditioning system. The working medium is air, therefore, an air flow takes place through the system.

The simulation shows the relations between different parameters and how they impact on the performance of this system. The system was simulated with a computer program written in the FORTRAN language. Constant parameters like atmospheric pressure (p_1), the gas constant for air (R) and the others are fixed in the program. Flexibility is one of the main characteristics of this program.

2.1 Values of the parameters

Constant parameters in the model:
– gas constant for air: R = 287 J/kgK,
– specific heat for air: $c_{\mathrm{p\,air}}$ = 1005 J/kgK,
– atmospheric/initial pressure: $p_1 = p_4 = p_4p$ = 0,101325 MPa (note: pressure before compression and after expansion – atmospheric pressure),
– ratio of specific heat capacity for air: χ = 1.4,
– specific heat for water: $c_{\mathrm{p\,w}}$ = 4200 J/kgK,
– entropy and enthalpy zero for liquid boiling (air) at: p_0 =1 b and T_0 = 78.8 K.

Variable parameters (inserted by the user of the simulation model):
– temperature of intake air: T_1 = 303 K,
– volume flow of air: V_1 = 0.1 m^3/s,
– compression pressure: $p_2 = p_2p = p_3$ = 0,2 MPa;
– (note: pressure drop in heat exchanger is omitted),
– compression efficiency: η_{CO} = 0.85,
– mass flow of cooling water: m_w = 0.083 kg/s,
– temperature of cool inlet water: T_{IN} = 288 K,
– thermal conductance of heat exchanger: k = 1342 W/m^{2}K,
– surface of heat exchanger: A = 0.08 m^2,
– degree of reciprocity of temperature in heat exchanger: ϕ = 0.554 [6],
– expansion efficiency: η_{ET} = 0.85.

These parameters are simulation values of the referenced air-conditioning process.

2.2 Computation sequence of parameters and quantities

A) First step:
Process of ***compression*** from point 1 to point 2p

Calculation of specific enthalpy and specific entropy for point 1:

$$h_1 = c_{p\ \mathrm{zrak}} T_1 \quad (1),$$

$$s_1 = c_{p\ \mathrm{zrak}} \left(\ln \frac{T_1}{T_0} \right) - R \left(\ln \frac{p_1}{p_0} \right) \quad (2).$$

Masni tok zraka:

Mass flow rate of air:

$$m_{zrak} = \frac{p_1 \cdot V_1}{R \cdot T_1} \quad (3).$$

Temperatura zraka po izentropni kompresiji:

Temperature of air after isentropical compression:

$$T_2 = e^{\ln T_0 + \frac{s_1 + R \cdot \ln \frac{p_2}{p_0} - s_1}{c_{p\ \mathrm{zrak}}}} \quad (4).$$

Specifična entalpija in specifična entropija za točko 2:

Specific enthalpy and specific entropy for point 2:

$$h_2 = c_{p\ \mathrm{zrak}} T_2 \quad (5),$$

$$s_2 = s_1 \quad (6).$$

Specifična entalpija, temperatura in entropija po politropni kompresiji (točka 2p):

Specific enthalpy, temperature and entropy after polytropic compression (point 2p):

$$h_{2p} = \frac{h_2 - h_1}{\eta_{CO}} + h_1 \quad (7),$$

$$T_{2p} = \frac{h_{2p}}{c_{p\ \mathrm{zrak}}} \quad (8),$$

$$s_{2p} = c_{p\ \mathrm{zrak}} \left(\ln \frac{T_{2p}}{T_0} \right) - R \left(\ln \frac{p_{2p}}{p_0} \right) \quad (9).$$

Eksponent politrope:

Polytropic exponent:

$$n_{CO} = \frac{-\left(\dfrac{\log \dfrac{p_1}{p_{2p}}}{\log \dfrac{T_1}{T_{2p}}} \right)}{1 - \left(\dfrac{\log \dfrac{p_1}{p_{2p}}}{\log \dfrac{T_1}{T_{2p}}} \right)} \quad (10).$$

Volumski tok zraka po kompresiji:

Volume flow rate of air after compression:

$$V_2 = V_1 \cdot \left(\frac{p_1}{p_{2p}} \right)^{\frac{1}{n_{CO}}} \quad (11).$$

Moč kompresije in izguba energije (trenje):

Compression power and losses of heat energy (friction):

$$P_{CO} = m_{zrak} \cdot \left(h_1 - h_{2p} \right) \quad (12),$$

$$Q_{IZGUBE} = \frac{\chi - n_{CO}}{\chi - 1} \cdot \frac{P_{CO}}{n_{CO}} \quad (13),$$

odstotno:

percentage:

$$\mathrm{X} = \frac{\mathrm{Q_{IZGUBE}}}{\mathrm{P_{CO}}} \cdot 100 \quad (14)$$

B) Drugi korak:

Postopek ***prenosa toplote*** iz točke 2p na točko 3 (postopek izračuna za prenos toplote v protismernem prenosniku toplote [4])

B) Second step:

Process of ***heat exchange*** from point 2p to point 3 (calculation procedure for heat exchange is developed by [4])

Kapaciteta zraka in vode:

Capacitance rate for air and water:

$$W_{zrak} = m_{zrak} \cdot c_{p\ \mathrm{zrak}} \quad (15),$$

$$W_{voda} = m_{voda} \cdot c_{p\ \mathrm{voda}} \quad (16).$$

Razmerje kapacitet:

Ratio of capacitance:

$$\tau = \frac{W_{zrak}}{W_{voda}}; \qquad 0 \le \tau \le 1 \tag{17}$$

Karakteristično število kapacitete (prenosnik toplote):

Characteristic number of capacity (heat exchanger):

$$KSK = \frac{k \cdot A}{W_{zrak}} \tag{18}$$

Temperaturna razlika med vstopnim zrakom in vstopno vodo:

Temperature difference between inlet air and inlet cooling water:

$$\vartheta = T_{2p} - T_{IN} \tag{19}$$

Temperaturna razlika med vstopnim in izstopnim zrakom:

Temperature difference between inlet and outlet air:

$$\Delta t_1 = \phi \cdot \vartheta \tag{21}$$

Temperatura zraka iz prenosnika toplote (točka 3):

Temperature of cooled air from heat exchanger (point 3):

$$T_3 = T_{2p} - \Delta t_1 \tag{22}$$

Prostorninski tok zraka po prenosu toplote:

Volume flow of air after heat exchange:

$$V_3 = V_2 \left(\frac{T_3}{T_{2p}} \right) \tag{23}$$

Temperaturna razlika med vstopno in izstopno vodo:

Temperature difference between inlet and outlet water:

$$\Delta t_2 = \tau \Delta t_1 \tag{24}$$

Temperatura izstopne segrete vode:

Temperature of outlet heated water:

$$T_{OUT} = T_{IN} + \Delta t_2 \tag{25}$$

Izmenjava toplote med zrakom in vodo:

Heat exchange between air and water:

$$Q_{OUT} = m_{zrak} c_{p\ \mathrm{zrak}} \Delta t_1 \tag{26}$$

C) Tretji korak:
Postopek ***ekspanzije*** od točke 3 do točke 4p

C) Third step:
Process of ***expansion*** from point 3 to point 4p

Specifična entalpija in specifična entropija točke 3:

Specific enthalpy and specific entropy for point 3:

$$h_3 = c_{p\ \mathrm{zrak}} T_3 \tag{27}$$

$$s_3 = c_{p\ \mathrm{zrak}} \left(\ln \frac{T_3}{T_0} \right) - R \left(\ln \frac{p_3}{p_0} \right) \tag{28}$$

Temperatura zraka po izentropni ekspanziji:

Temperature of air after isentropical expansion:

$$T_4 = e^{\ln T_0 + \frac{s_4 + R \cdot \ln \frac{p_4}{p_0} - s_4}{c_{p\ \mathrm{zrak}}}} \tag{29}$$

Specifična entalpija in specifična entropija točke 4:

Specific enthalpy and specific entropy for point 4:

$$h_4 = c_{p\ \mathrm{zrak}} T_4 \tag{30}$$

$$s_4 = s_3 \tag{31}$$

Specifična entalpija, temperatura in entropija po politropni ekspanziji (točka 4p):

Specific enthalpy, temperature and entropy after polytropic expansion (point 4p):

$$h_{4p} = h_3 - \eta_{ET}\left(h_3 - h_4\right) \quad (32),$$

$$T_{4p} = \frac{h_{4p}}{c_{p\ \mathrm{zrak}}} \quad (33),$$

$$s_{4p} = c_{p\ \mathrm{zrak}}\left(\log\frac{T_{4p}}{T_0}\right) - R\left(\log\frac{p_{4p}}{p_0}\right) \quad (34).$$

Eksponent politrope: Polytropic exponent:

$$n_{ET} = \frac{-\left(\dfrac{\log\dfrac{p_3}{p_{4p}}}{\log\dfrac{T_3}{T_{4p}}}\right)}{1-\left(\dfrac{\log\dfrac{p_3}{p_{4p}}}{\log\dfrac{T_3}{T_{4p}}}\right)} \quad (35).$$

Moč ekspanzije: Expansion power:

$$P_{ET} = m_{zrak}\left(h_3 - h_{4p}\right) \quad (36).$$

Potrebna moč kompresije: Required compression power:

$$W = P_{CO} - P_{ET} \quad (37).$$

Hladilno število: Coefficient of performance:

$$COP = \frac{Q_{IN}}{\left|W\right|} \quad (38).$$

Slika 3 prikazuje diagram poteka delovanja simulirnega preprostega zračnega hladilnega sistema, ki je sestavljen iz treh korakov:

A. Prvi korak je kompresija, v katerem so vhodni podatki temperatura in prostorninski tok vstopnega zraka, tlak kompresije in izkoristek kompresije. Izhodni podatki so masni tok zraka, specifična entalpija in specifična entropija, temperatura zraka v točki 1, 2 in 2p, kompresijski eksponent politrope, prostorninski tok zraka po kompresiji, moč kompresije in izgube energije med kompresijo (trenje).

B. Drugi korak je prenos toplote, v katerem so vhodni podatki masni tok vode, vstopna temperatura hladilne vode, toplotna prehodnost prenosnika toplote, površina prenosnika toplote in stopnja izmenjave temperature v prenosniku toplote. Izhodni podatki so kapaciteta zraka in vode, razmerje kapacitet, karakteristično število kapacitete, temperaturna razlika med vstopnim zrakom in vstopno vodo, temperaturna razlika med vstopnim in izstopnim zrakom, temperatura zraka iz prenosnika toplote, prostorninski tok zraka po prenosu toplote, temperaturna razlika med vstopno in izstopno vodo, temperatura izstopne segrete vode in izmenjava toplote med zrakom in vodo.

C. Tretji korak je ekspanzija, v katerem se prične postopek z izračunom specifične entalpije in

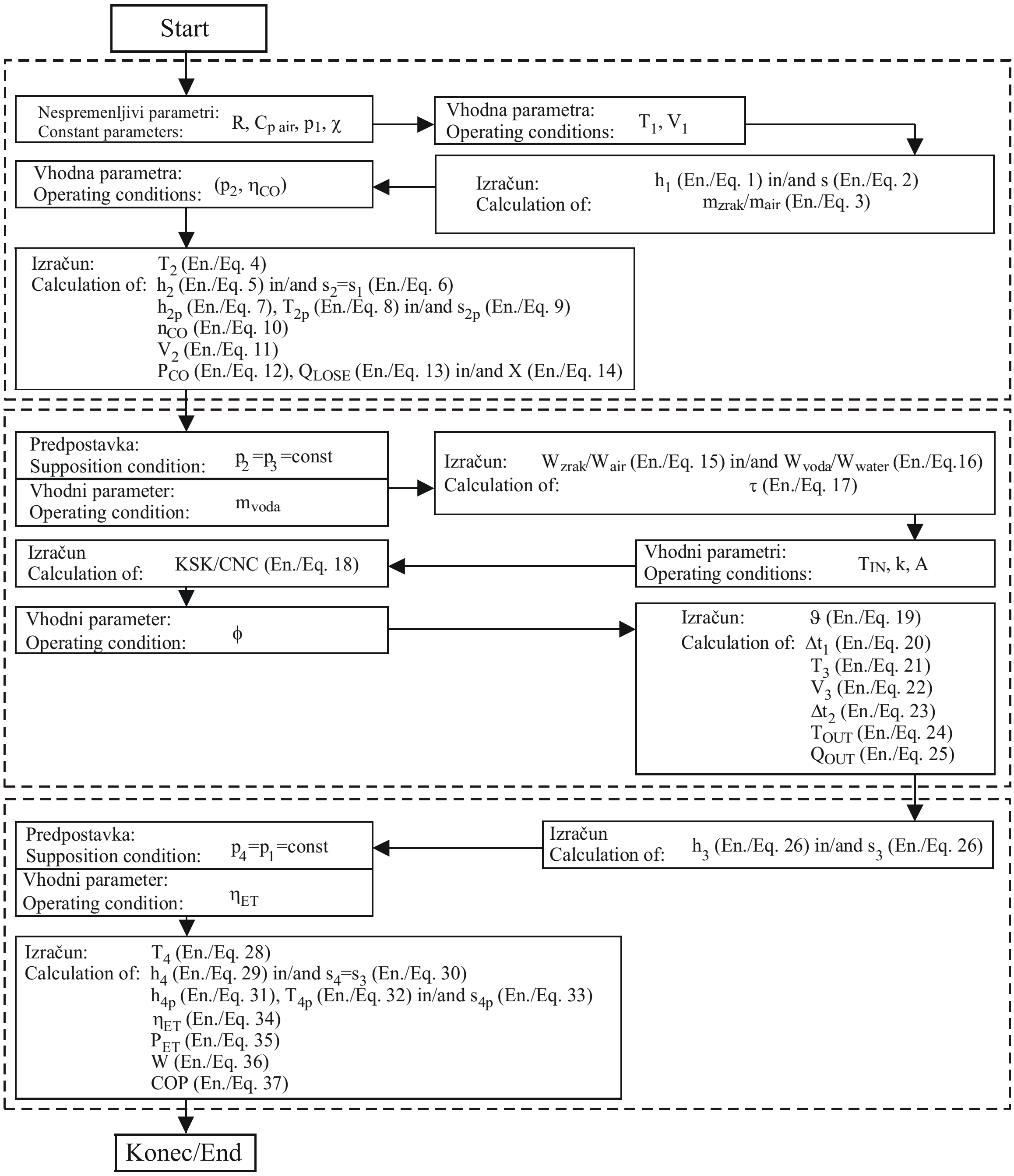
Figure 3. shows the flow chart of the performance simulation program of a simple air-conditioning system that consists of three steps:

A. The first step is compression, where the input data are the temperature and volume flow of the inlet air, the compression pressure and the compression efficiency. The output data are the mass flow of the air, the specific enthalpy and specific entropy, the air temperature for points 1, 2 and 2p, the polytropic exponent of compression, the volume flow after compression, the compression power and the heat losses during compression.

B. The second step is heat exchange, where the input data are the mass flow of water, the temperature of the inlet cooling water, the heat conductance of the heat exchanger, the surface of the heat exchanger and the degree of reciprocity of the temperature in the heat exchanger. The output data are the capacitance rate for the air and water, the ratio of the capacitance, the characteristic number of capacity, the temperature difference between the inlet air and the inlet cooling water, the temperature difference between the inlet and outlet air, the temperature of the cooled air from the heat exchanger, the volume flow of air after heat exchange, the temperature difference between the inlet and outlet water, the temperature of the outlet heated water and the heat exchange between the air and water.

C. The third step is expansion, where the operation starts with a calculation of the specific enthalpy specifične entropije točke 3. Vhodni podatek je izkoristek ekspanzije, izhodni podatki pa so temperatura, specifična entalpija in specifična entropija točke 4 in 4p, ekspanzijski eksponent adiabate, moč ekspanzije, potrebna moč kompresije in hladilno število. Ta korak je na blokovni shemi označen z zelenim okvirom (sl. 3).

and the specific entropy for point 3. The input data is the efficiency of expansion, the output data are the temperature, the specific enthalpy and the specific entropy for points 4 and 4p, the adiabatic exponent of expansion, the expansion power, the required compression power and the coefficient of performance.



Sl. 3. *Blokovna shema simuliranja termodinamičnega modela*
Fig. 3. *Flow chart for the thermodynamic model*

3 REZULTATI SIMULIRANJA

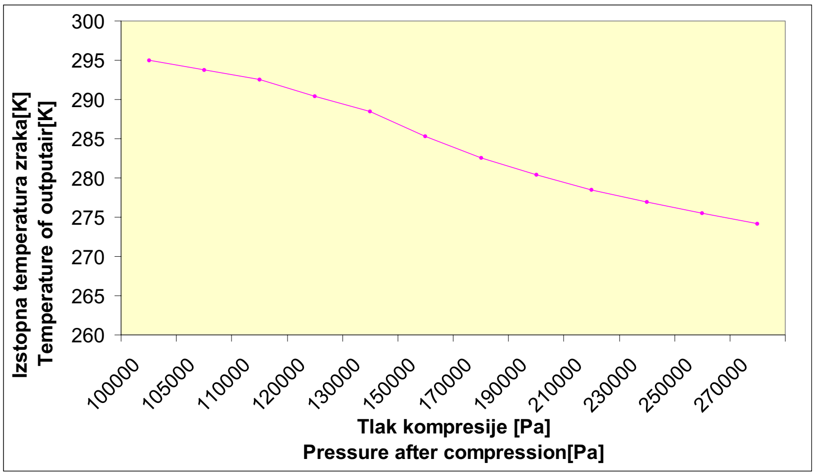
Spodnji diagrami prikazujejo, kako določeni parametri sistema vplivajo med seboj. Podobno so rezultate predstavili tudi drugi avtorji ([5] in [6]):

3 SIMULATION RESULTS

In the diagrams below we show how certain parameters of the system impact on each other. Similar results were presented by other authors ([5] and [6]):

1) Diagram *Izstopna temperatura zraka / Tlak kompresije*

1) Diagram *Temperature of outlet Air / Compression Pressure*

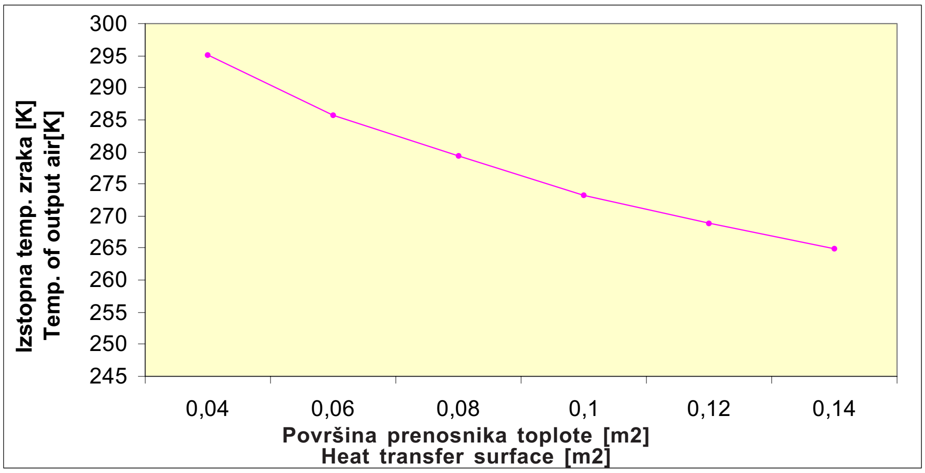


Bolj ko zrak komprimiramo, nižjo izstopno temperaturo zraka dobimo. Karakteristika ni linearna. Obstaja točka, kjer je tlak kompresije zelo nizek in izstopna temperatura zraka ostane enaka kakor temperatura zraka pred komprimiranjem. Če hočemo torej zrak ohladiti, ga moramo komprimirati na določen nadtlak.

The more the air is compressed, the lower the temperature of the outlet air. The characteristic is non-linear. There is a point where the compression pressure is very low and the temperature of the outlet air is the same as the temperature of the inlet air. So, if the air must be cooled some compression work has to be done.

2) Diagram *Izstopna temperatura zraka / Površina prenosnika toplote*

2) Diagram *Temperature of outlet Air / Surface of Heat Exchanger*

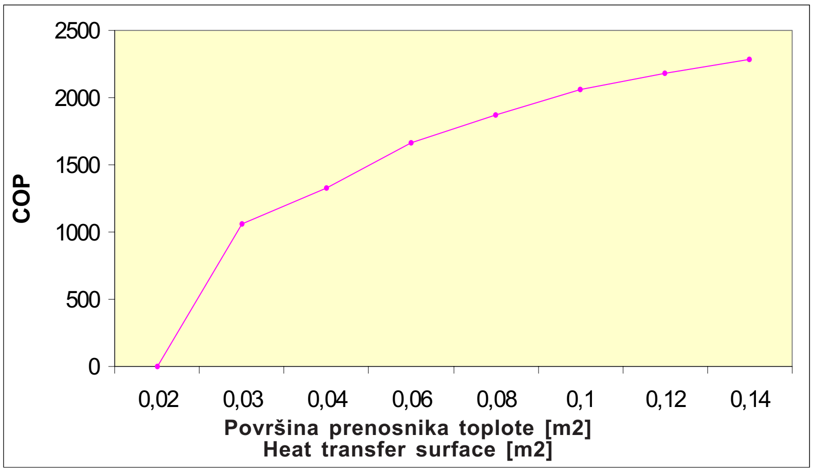


Temperatura izstopnega zraka je podobno odvisna od površine prenosnika toplote, kakor je tlak kompresije odvisen od temperature izstopnega zraka. Tudi tu obstaja točka, kjer je površina prenosnika toplote premajhna, da bi se zrak ohladil. Karakteristika je približno linearna.

The temperature of the outlet air is similarly dependent on the surface of the heat exchanger as the compression pressure affects the temperature of the outlet air. There is a point where the surface of the heat exchanger is too small to cool the air. The characteristic is approximately linear.

3) Diagram *COP / Površina prenosnika toplote*

3) Diagram *COP / Surface of Heat Exchanger*

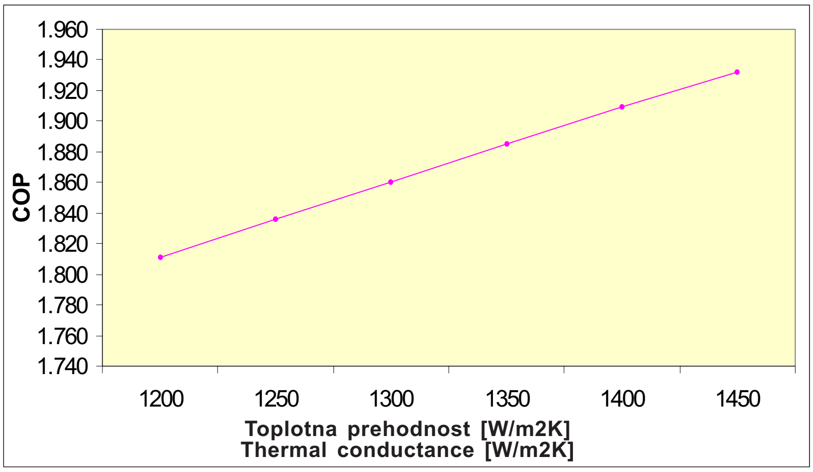


Iz diagrama ugotovimo, da se COP s povečano površino prenosnika toplote povečuje.

From the diagram we can see that the COP increases with the surface of the heat exchanger.

4) Diagram *COP / Toplotna prehodnost prenosnika toplote*

4) Diagram *COP / Thermal Conductance*

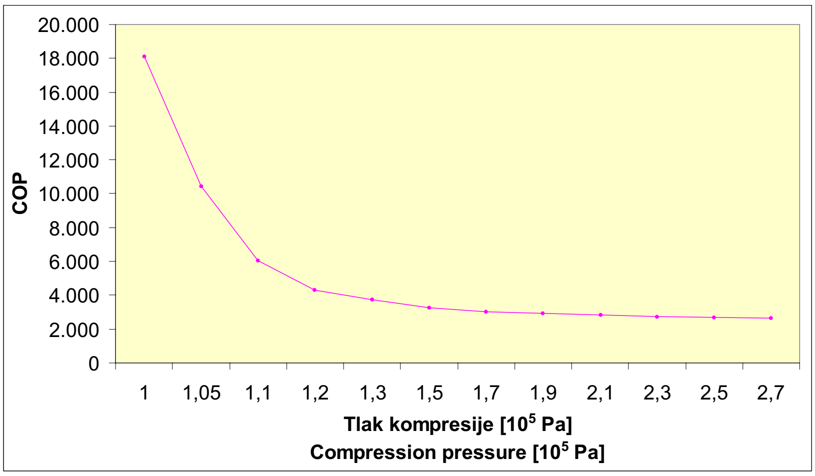


Vidimo lahko, da je COP linearno sorazmeren s toplotno prehodnostjo prenosnika toplote.

We can see that the COP is directly proportional to the thermal conductance of the heat exchanger.

5) Diagram *COP / Tlak kompresije*

5) Diagram *COP / Compression Pressure*



Zgornji diagram prikazuje, kako je hladilno število odvisno od tlaka kompresije. Manj ko dovedemo kompresijskega dela v sistem, večje je hladilno število. Hkrati pa to pomeni, da pri nižjih tlakih zrak tudi manj ohladimo. Teoretično obstaja točka, kjer je optimalno razmerje mod COP in temperaturo izstopnega zraka.

4 SKLEP

Razvili smo preprost simulirni program delovanja odprtega zračnega hladilnega procesa brez uporabe CFC, v kar je usmerjen razvoj klimatizacijske tehnike z namenom ohranjanja okolja. Računalniški program, napisan v programskem jeziku FORTRAN, temelji na termodinamičnih enačbah. Ta specifičen hladilni sistem je sestavljen iz treh osnovnih stopenj: kompresijska enota, prenosnik toplote in ekspanzijska enota. Veliki izkoristki centrifugalnih kompresorskih in ekspanzijskih turbin ter učinkovit prenos toplote med obema tekočinama so bistveni za izkoristek celotnega sistema. Kompresorska in ekspanzijska turbina morata biti narejeni tako, da so torne izgube (toplotne izgube) na stene in izgube zaradi spremembe hitrosti zmanjšane. V prenosniku toplote lahko opazimo dva glavna vira izgube eksergije: temperaturni in tlačni padec. Posledica prvega vira izgub je nižja temperatura hladilnega sredstva (vode), delovnega sredstva (zraka) ali kot višje razmerje T_3/T_1, kar zmanjša hladilno število COP (sl. 2). Tlačni padec je vzrok za nižje pridobljeno delo ekspanzije, čeprav nismo upoštevali tlačnega padca v prenosniku toplote. Iz rezultatov simuliranja lahko ugotovimo, da ne moremo poiskati celovite točke optimalnega delovanja. Le to lahko določimo za vsak primer posebej.

Pravilnost in natančnost tega simulirnega programa bomo potrdili z izvedbo preskusa odprtega zračnega hladilnega cikla, katerega demonstracijsko progo postavljajo na Inštitutu za okoljske znanosti in energijske raziskave - TNO, Nizozemska [7]. Izmerjene vrednosti določenih parametrov sistema bodo pokazale odstopanja med preskusom, torej realnim postopkom, in računalniškim simuliranjem delovanja hladilnega zračnega kroga.

The diagram shows how the cooling number depends on the compression pressure. The less compression work is lead to the system, the lower is the cooling number. At the same time this means that air is less effectively cooled. Theoretically, there exists a point that is the best comprimise between the COP and the temperature of the outlet air.

4 CONCLUSION

A performance simulation program for a CFC-free air-conditioning system has been developed. Such air-conditioning systems are popular because of their low environmental impact. A computer program was written in the FORTRAN language and based on thermodynamic equations. The cooling system consisted of three basic units: compression unit, heat-exchanger unit and expansion unit. High efficiencies of the centrifugal compressors and expanders and an effective heat exchange between fluids are essential for the efficiency of the system. Both compressor and expander must be built so that frictional losses (power-heat losses) to the walls and the velocity-change losses are reduced. Two main losses of exergy can be identified in a heat exchanger: the temperature drop and the pressure drop. The first loss will result in a lower temperature of the cooling heat-transfer fluid (water) or a higher than necessary T_3/T_1 that will reduce the COP (Fig. 2). A pressure drop will result in a lower regained power in the expander, though we did not observe that a pressure drop in the heat exchanger occurred. From the results of the simulation we found that a global optimum cannot be determined. It can only be found for each individual case.

The correctness and accuracy of this simulation program will be confirmed by performing an experiment on an open-cycle air-conditioning system that has been made at the Institute of Environmental Sciences, Energy Research and Process Innovation TNO, Netherlands [7]. The measured values of certain parameters of the system will show any differences between the simulation performance and an actual CFC-free air-conditioning system.

5 SIMBOLI
5 SYMBOLS

specifična entalpija stanja i	h_i	J/kg	specific enthalpy at state point i
entalpijska razlika	Δh	J/kg	enthalpy difference
plinska konstanta zraka	R	J/kgK	gas constant for air
specifična toplota posameznega fluida	$c_{p\,i}$	J/kgK	specific heat for an individual fluid
tlak	p_i	Pa	pressure
razmerje specifičnih toplot zraka	χ	-	ratio of the specific heat capacity for air
temperatura stanja i	T_i	K	temperature at state point i
volumski tok stanja i	V_i	m^3/s	volume flow rate at state i
masni tok za posamezen fluid	m_i	kg/s	mass flow rate for individual fluid

toplotna prehodnost	k	W/m^{2}K	thermal conductance
površina	A	m^2	surface
stopnja izmenjave temperature v prenosniku toplote	ϕ		degree of reciprocity of temperature in heat exchanger
izkoristek stanja i	η_i		efficiency at state point i
specifična entropija stanja i	s_i	J/kgK	specific entropy at state point i
eksponent politrope stanja i	n_i		polytropic exponent at state point i
kompresijska moč	P_{CO}	W	compression power
moč ekspanzije	P_{EX}	W	expansion power
izgube toplotne energije v kompresorju	Q_{LOSE}	W	losses of heat energy in compressor
odstotne izgube toplotne energije v kompresorju	X		percentage losses of heat energy in compressor
toplotna vrednost pretoka za posamezen fluid	W_i	W/K	capacitance rate for individual fluid
razmerje toplotnih vrednosti	τ		ratio of capacitance
karakteristično število kapacitete prenosnika toplote	CNC		characteristic number of capacity for heat ex changer (NTU)
temperaturna razlika med vstopnim zrakom in izstopno hladilno vodo	ϑ	K	temperature difference between inlet air and inlet cooling water
temperaturna razlika	Δt	K	temperature difference
odvedena toplota s prenosnikom toplote	Q_{OUT}	W	heat transferred by heat exchanger
hladilna obremenitev	Q_{IN}	W	refrigerating power, also Q_R
hladilno število	W	W	cooling load
	COP		coefficient of performance

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Prejeto:
Received: 26.6.2001

Sprejeto:
Accepted: 12.10.2001