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hco := Nuc*k_air(Tc)/od;
w := 0.5*od*(((od+2*fh)/od)-1)*(1+0.35*ln((od+2*fh)/od));
effe := (tanh(sqrt(2*heo/(ft*k))*w)/(sqrt(2*heo/(ft*k))*w));
effc := (tanh(sqrt(2*hco/(ft*k))*w)/(sqrt(2*hco/(ft*k))*w));
End;
If fintype = 3 then
Begin
Afe := ((nc+0.5)*St*Sl-nc*3.14159*od*od*0.25)*2*
(Le*fn/0.0254);
Abe := ((nc*Le*3.14159)/(fs+ft))*od*fs;
Afc := ((nc+0.5)*St*Sl-nc*3.14159*od*od*0.25)*2*
(Lc*10/0.0254);
Abc := ((nc*Lc*3.14159)/(fs+ft))*od*fs;
mfae := 2*nc*Le*(Sd-od-(2*ft*fh)/(fs+ft));
mfac := 2*nc*Lc*(Sd-od-(2*ft*fh)/(fs+ft));
Vmaxe := mh/(den_air(Th)*mfae);
Vmaxc := mc/(den_air(Tc)*mfac);
Ree := Vmaxe*od*den_air(Th)/vis_air(Th);
Rec := Vmaxc*od*den_air(Tc)/vis_air(Tc);
Pre := Cp_air(Th)*vis_air(Th)/k_air(Th);
Prc := Cp_air(Tc)*vis_air(Tc)/k_air(Tc);
heo := 0.14*exp(-0.328*ln(Ree))*exp(-0.502*ln(St/Sl))
*exp(0.031*ln(fs/od))*den_air(Th)*Vmaxe*Cp_air(Th)
*exp((-2/3)*ln(Pre));
hco := 0.14*exp(-0.328*ln(Rec))*exp(-0.502*ln(St/Sl))
*exp(0.031*ln(fs/od))*den_air(Tc)*Vmaxc*Cp_air(Tc)
*exp((-2/3)*ln(Prc));
fp := 1.27*((St/2)/(0.5*od))*sqrt((sqrt((St/2)*(St/2)+Sl*Sl)/2)
/(St/2)-0.3);
w := (fp-1)*(1+0.35*ln(fp));
effe := (tanh(sqrt(2*heo/(ft*k))*w*(od/2))/(sqrt(2*heo/(ft*k))*w
*(od/2)));
effc := (tanh(sqrt(2*hco/(ft*k))*w*(od/2))/(sqrt(2*hco/(ft*k))*w
*(od/2)));
End;
End;
Aeo := effe*Afe+Abe;
Aco := effe*Afc+Abc;
Aei := 3.14159*id*Le*nc;
Aci := 3.14159*id*Lc*nc;
T := (Th+Tc)/2;
If T <= 55 then
Begin
Re := ((Q/Aei)/(Vwl(T)*Lw(T)))*(sqrt(Sw(T)/(9.81*(dwl(T)-dwg(T)))));

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Pr := Cpwl(T)*Vwl(T)/kw(T);
hei := exp(ln(41.27)+0.37*ln(Re)+0.4*ln(Pr))*kw(T)/
      (sqrt(Sw(T)/(9.81*(dwl(T)-dwl(T)))));
hci := exp(1.324*ln(0.94*(exp(0.245*ln(dwl(T)*(dwl(T)-dwl(T))*9.81
      *kw(T)*kw(T)*kw(T)*Lw(T)/(Vwl(T)*Lc*(Q/Aci))))));
End;
If T > 55 then
Begin
x := (1/(Vwl(T)*Lw(T)))*sqrt(Sw(T)/(9.81*(dwl(T)-dwl(T)))));
y := Lw(T)*Vwl(T)/kw(T);
hei := exp(0.9332*(ln(18.688)-ln(x)-1.0716*ln(y)+0.0716*ln(Q/Aei)));
hci := exp(1.30378*(ln(0.943)+0.233*ln(sqrt(dwl(T))*9.81*Lw(T)*
      kw(T)*kw(T)*kw(T)*Aci/(Vwl(T)*Lc*Q))));
End;
UA001 := 1/(1/(heo*Aeo)+1/(hco*Aco)+1/(hei*Aei)+1/(hci*Aci)
      +ln(od/id)/(2*3.14159*k*Le*nc)+ln(od/id)/(2*3.14159*k*Lc*nc));
End;
Function UA002(Th,Tc,mh,mc,Q,nc:real):real; {calculate UA of ethanol}
Const k = 14.9; {thermal conductivity of metal}
Var Ae0,Aei,Aco,Aci,Afe,Afc,Abe,Abc,heo,hei,
    hco,hci,mfac,mfae,Vmaxe,Vmaxc,Ree,
    Rec,Pre,Prc,Nue,Nuc,w,effe,effc,Zc,Zef,Zep,phe2,phe3,T,Z,Re,a : real;
del_Tsat,x,y,heif,heip,Pr,fp : real;
Ve,Vc,Vmaxe1,Vmaxe2,Vmaxc1,Vmaxc2 : real;
Begin
If arrangement = 1 then
Begin
If fintype = 1 then
Begin
Abe := nc*Le*3.14159*od;
Abc := nc*Lc*3.14159*od;
Afe := 0;
Afc := 0;
Ve := mh/(den_air(Th)*cross_section_area);
Vc := mc/(den_air(Tc)*cross_section_area);
Vmaxe := St*Ve/(St-od);
Vmaxc := St*Vc/(St-od);
Ree := den_air(Th)*Vmaxe*od/vis_air(Th);
Rec := den_air(Tc)*Vmaxc*od/vis_air(Tc);
Pre := Cp_air(Th)*vis_air(Th)/k_air(Th);
Prc := Cp_air(Tc)*vis_air(Tc)/k_air(Tc);
Nue := C_aligned(Ree)*exp(m_aligned(Ree)*ln(Ree))
      *exp(0.36*ln(Pre));
Nuc := C_aligned(Rec)*exp(m_aligned(Rec)*ln(Rec))

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        *exp(0.36*ln(Prc));
heo := Nue*k_air(Th)/od;
hco := Nuc*k_air(Tc)/od;
effe := 0;
effc := 0;
End;
If fintype = 2 then
Begin
    Afe := ((nc*Le*3.14159)/(fs+ft))*(0.5*(sqr(od+2*fh)-sqr(od))
        +(od+2*fh)*ft);
    Abe := ((nc*Le*3.14159)/(fs+ft))*od*fs;
    Afc := ((nc*Lc*3.14159)/(fs+ft))*(0.5*(sqr(od+2*fh)-sqr(od))
        +(od+2*fh)*ft);
    Abc := ((nc*Lc*3.14159)/(fs+ft))*od*fs;
    mfae := nc*Le*(St-od-(2*ft*fh)/(fs+ft));
    mfac := nc*Lc*(St-od-(2*ft*fh)/(fs+ft));
    Vmaxe := mh/(den_air(Th)*mfae);
    Vmaxc := mc/(den_air(Tc)*mfac);
    Ree := Vmaxe*od*den_air(Th)/vis_air(Th);
    Rec := Vmaxc*od*den_air(Tc)/vis_air(Tc);
    Pre := Cp_air(Th)*vis_air(Th)/k_air(Th);
    Prc := Cp_air(Tc)*vis_air(Tc)/k_air(Tc);
    Nue := 0.30*exp(0.625*ln(Ree))*exp(-0.375*
        ln((nc*Le*3.14159*od)/(Afe+Abe)))
        *exp(0.333*ln(Pre));
    Nuc := 0.30*exp(0.625*ln(Rec))*exp(-0.375*
        ln((nc*Lc*3.14159*od)/(Afc+Abc)))
        *exp(0.333*ln(Prc));
    heo := Nue*k_air(Th)/od;
    hco := Nuc*k_air(Tc)/od;
    w := 0.5*od*(((od+2*fh)/od)-1)*(1+0.35*ln((od+2*fh)/od));
    effe := (tanh(sqrt(2*heo/(ft*k))*w)/(sqrt(2*heo/(ft*k))*w));
    effc := (tanh(sqrt(2*hco/(ft*k))*w)/(sqrt(2*hco/(ft*k))*w));
End;
End;
If arrangement = 2 then
Begin
    If fintype = 1 then
    Begin
        Abe := nc*Le*3.14159*od;
        Abc := nc*Lc*3.14159*od;
        Afe := 0;
        Afc := 0;
        Ve := mh/(den_air(Th)*cross_section_area);

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Vc := mc/(den_air(Tc)*cross_section_area);
Vmaxe1:= St*Ve/(St-od);
Vmaxc1:= St*Vc/(St-od);
Vmaxe2:= St*Ve/(2*(Sd-od));
Vmaxc2:= St*Vc/(2*(Sd-od));
If Vmaxe1 > Vmaxe2 then Vmaxe := Vmaxe1
else Vmaxe := Vmaxe2;
If Vmaxc1 > Vmaxc2 then Vmaxc := Vmaxc1
else Vmaxc := Vmaxc2;
Ree := den_air(Th)*Vmaxe*od/vis_air(Th);
Rec := den_air(Tc)*Vmaxc*od/vis_air(Tc);
Pre := Cp_air(Th)*vis_air(Th)/k_air(Th);
Prc := Cp_air(Tc)*vis_air(Tc)/k_air(Tc);
Nue := C_staggered(Ree,St,Sl)*exp(m_staggered(Ree)*ln(Ree))
      *exp(0.36*ln(Pre));
Nuc := C_staggered(Rec,St,Sl)*exp(m_staggered(Rec)*ln(Rec))
      *exp(0.36*ln(Prc));
heo := Nue*k_air(Th)/od;
hco := Nuc*k_air(Tc)/od;
effe := 0;
effc := 0;
End;
If fintype = 2 then
Begin
Afe := ((nc*Le*3.14159)/(fs+ft))*(0.5*(sqr(od+2*fh)-sqr(od))
      +(od+2*fh)*ft);
Abe := ((nc*Le*3.14159)/(fs+ft))*od*fs;
Afc := ((nc*Lc*3.14159)/(fs+ft))*(0.5*(sqr(od+2*fh)-sqr(od))
      +(od+2*fh)*ft);
Abc := ((nc*Lc*3.14159)/(fs+ft))*od*fs;
mfae := 2*nc*Le*(Sd-od-(2*ft*fh))/(fs+ft);
mfac := 2*nc*Lc*(Sd-od-(2*ft*fh))/(fs+ft);
Vmaxe := mh/(den_air(Th)*mfae);
Vmaxc := mc/(den_air(Tc)*mfac);
Ree := Vmaxe*od*den_air(Th)/vis_air(Th);
Rec := Vmaxc*od*den_air(Tc)/vis_air(Tc);
Pre := Cp_air(Th)*vis_air(Th)/k_air(Th);
Prc := Cp_air(Tc)*vis_air(Tc)/k_air(Tc);
Nue := 0.242*exp(0.658*ln(Ree))*exp(0.297*ln(fs/fh))
      *exp(-0.091*ln(St/Sl))*exp(0.33333*ln(Pre));
Nuc := 0.242*exp(0.658*ln(Rec))*exp(0.297*ln(fs/fh))
      *exp(-0.091*ln(St/Sl))*exp(0.33333*ln(Prc));
heo := Nue*k_air(Th)/od;
hco := Nuc*k_air(Tc)/od;

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w := 0.5*od*(((od+2*fh)/od)-1)*(1+0.35*ln((od+2*fh)/od));
effe := (tanh(sqrt(2*heo/(ft*k))*w)/(sqrt(2*heo/(ft*k))*w));
effc := (tanh(sqrt(2*hco/(ft*k))*w)/(sqrt(2*hco/(ft*k))*w));
End;
If fintype = 3 then
Begin
Afe := ((nc+0.5)*St*Sl-nc*3.14159*od*od*0.25)*2*
(Le*fn/0.0254);
Abe := ((nc*Le*3.14159)/(fs+ft))*od*fs;
Afc := ((nc+0.5)*St*Sl-nc*3.14159*od*od*0.25)*2*
(Lc*10/0.0254);
Abc := ((nc*Lc*3.14159)/(fs+ft))*od*fs;
mfae := 2*nc*Le*(Sd-od-(2*ft*fh)/(fs+ft));
mfac := 2*nc*Lc*(Sd-od-(2*ft*fh)/(fs+ft));
Vmaxe := mh/(den_air(Th)*mfae);
Vmaxc := mc/(den_air(Tc)*mfac);
Ree := Vmaxe*od*den_air(Th)/vis_air(Th);
Rec := Vmaxc*od*den_air(Tc)/vis_air(Tc);
Pre := Cp_air(Th)*vis_air(Th)/k_air(Th);
Prc := Cp_air(Tc)*vis_air(Tc)/k_air(Tc);
heo := 0.14*exp(-0.328*ln(Ree))*exp(-0.502*ln(St/Sl))
*exp(0.031*ln(fs/od))*den_air(Th)*Vmaxe*Cp_air(Th)
*exp((-2/3)*ln(Pre));
hco := 0.14*exp(-0.328*ln(Rec))*exp(-0.502*ln(St/Sl))
*exp(0.031*ln(fs/od))*den_air(Tc)*Vmaxc*Cp_air(Tc)
*exp((-2/3)*ln(Prc));
fp := 1.27*((St/2)/(0.5*od))*sqrt((sqrt((St/2)*(St/2)+Sl*Sl)/2)
/(St/2)-0.3);
w := (fp-1)*(1+0.35*ln(fp));
effe := (tanh(sqrt(2*heo/(ft*k))*w*(od/2))/(sqrt(2*heo/(ft*k))*w
*(od/2)));
effc := (tanh(sqrt(2*hco/(ft*k))*w*(od/2))/(sqrt(2*hco/(ft*k))*w
*(od/2)));
End;
End;
Aeo := effe*Afe+Abe;
Aco := effe*Afc+Abc;
Aei := 3.14159*id*Le*nc;
Aci := 3.14159*id*Lc*nc;
T := (Th+Tc)/2;
x := (1/(Vel(T)*Lae(T)))*sqrt(Se(T)/(9.81*(del(T)-deg(T))));
y := Lae(T)*Vel(T)/ke(T);
hei := exp(1.01*(ln(17.625)-ln(x)-0.99*ln(y)-0.01*ln(Q/Aei)));
hci := exp(1.35135*(ln(0.93)+0.26*ln(sqr(del(T)))*9.81*Lae(T)*

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        ke(T)*ke(T)*ke(T)*Aci/(Vel(T)*Lc*Q)))));
    UA002 := 1/(1/(heo*Aeo)+1/(hco*Aco)+1/(hei*Aei)+1/(hci*Aci)
        +ln(od/id)/(2*3.14159*k*Le*nc)+ln(od/id)/(2*3.14159*k*Lc*nc));
End;
Function  UAmix(Th,Tc,mh,mc,Q,n,wc:real):real;
Begin
    UAmix := UA001(Th,Tc,mh,mc,Q,n)*wc+UA002(Th,Tc,mh,mc,Q,n)*(1-wc);
End;
Begin {main program}
    writeln(' PROGRAM HEAT PIPE 001 ');
    writeln;
    Writeln(' This program is used for calculating performance of the thermosyphon');
    writeln(' heat exchanger operating at low temperature with 2-kinds of working');
    writeln(' fluids (water-ethanol)');
    writeln;
    writeln(' Developed by Atipoang Nuntaphan, Ph.D. student ');
    writeln(' Thermal Technology Division, School of Energy and Materials,');
    writeln(' King Mongkut"s, University of Technology Thonburi');
    writeln;
    Writeln(' *****');
    writeln;
    writeln(' Please enter informations below ');
    writeln(' 1. Heat exchanger informations');
    write(' 1.1 Input number of rows ');
    readln(nr);
    write(' 1.2 Input number of column ');
    readln(nc);
    write(' 1.3 Input outside diameter of tube (m) ');
    readln(od);
    write(' 1.4 Input inside diameter of tube (m) ');
    readln(id);
    write(' 1.5 Input evaporator length (m) ');
    readln(Le);
    write(' 1.6 Input condenser length (m) ');
    readln(Lc);
    writeln(' 1.7 Input type of fin ');
    write(' 1. bare tube 2. circular finned 3. plain plate finned ');
    readln(fintype);
    write(' 1.8 Input fin thickness (m) (input 0 for bare tube) ');
    readln(ft);
    write(' 1.9 Input fin gap (m) (input 0 for bare tube) ');
    readln(fs);
    write(' 1.10 Input fin height (m) (input 0 for bare tube) ');
    readln(fh);

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write(' 1.11 Input number of fins (fin/inch) (input 0 for bare tube) ');
readln(fn);
writeln(' 1.12 Input pipe arrangement ');
write('      1. aligned    2. staggered ');
readln(arrangement);
write(' 1.13 Input transverse pitch (St) (m) ');
readln(St);
write(' 1.14 Input longitudinal pitch (Sl) (m) ');
readln(Sl);
write(' 1.15 Input directional pitch (Sd) (m) (input 0 for aligned) ');
readln(Sd);
write(' 1.16 Input cross section area of duct (m^2) ');
readln(cross_section_area);
writeln;
writeln(' 2. Working conditions ');
write(' 2.1 Input inlet temperature of hot gas (C) ');
readln(Thi);
write(' 2.2 Input inlet temperature of cold gas (C) ');
readln(Tci);
write(' 2.3 Input mass flow rate of hot gas (kg/s) ');
readln(mh);
write(' 2.4 Input mass flow rate of cold gas (kg/s) ');
readln(mc);
writeln(' 2.5 Input flow pattern ');
write('      1. parallel flow  2. counter flow ');
read(flowpattern);
writeln;
If flowpattern = 1 then
  Begin
    Th[1] := Thi;
    Tc[1] := Tci;
    For i := 1 to nr do
      Begin
        write(' row = ',i);
        Q[i] := 1200;
        Q[i] := UA001(Th[i],Tc[i],mh,mc,Q[i]/nc,nc)*(Th[i]-Tc[i]);
        Th[i+1] := Th[i]-Q[i]/(mh*Cp_air(Th[i]));
        Tc[i+1] := Q[i]/(mc*Cp_air(Tc[i]))+Tc[i];
        For j := 1 to 200 do
          Begin
            lmtdd := ((Th[i]-Tc[i])-(Th[i+1]-Tc[i+1]))/ln((Th[i]-Tc[i])/
              (Th[i+1]-Tc[i+1]));
            Q[i] := UA001((Th[i]+Th[i+1])/2,(Tc[i]+Tc[i+1])/2,mh,mc,
              Q[i]/nc,nc)*lmtdd;

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    Th[i+1] := Th[i]-Q[i]/(mh*Cp_air((Th[i]+Th[i+1])/2));
    Tc[i+1] := Tc[i]+Q[i]/(mc*Cp_air((Tc[i]+Tc[i+1])/2));
End;
Qw := Q[i]; Thow := Th[i+1]; Tcow := Tc[i+1];
For j := 1 to 200 do
Begin
    lmtd := ((Th[i]-Tc[i])-(Th[i+1]-Tc[i+1]))/ln((Th[i]-Tc[i])
        /(Th[i+1]-Tc[i+1]));
    Q[i] := UA002((Th[i]+Th[i+1])/2,(Tc[i]+Tc[i+1])/2,mh,mc,
        Q[i]/nc,nc)*lmtd;
    Th[i+1] := Th[i]-Q[i]/(mh*Cp_air((Th[i]+Th[i+1])/2));
    Tc[i+1] := Tc[i]+Q[i]/(mc*Cp_air((Tc[i]+Tc[i+1])/2));
End;
Qe := Q[i]; Thoe := Th[i+1]; Tcoe := Tc[i+1];
If Qw > Qe then
Begin
    Th[i+1] := Thow;
    Tc[i+1] := Tcow;
    Q[i] := Qw;
    writeln(' Th,out = ',Th[i+1]:1:2,' Tc,out = ',Tc[i+1]:1:2,
        ' Q = ',Q[i]:1:2,' water');
End;
If Qe > Qw then
Begin
    Th[i+1] := Thoe;
    Tc[i+1] := Tcoe;
    Q[i] := Qe;
    writeln(' Th,out = ',Th[i+1]:1:2,' Tc,out = ',Tc[i+1]:1:2,
        ' Q = ',Q[i]:1:2,' ethanol');
End;
End;
Qtotal := 0;
for i := 1 to nr do
Begin
    Qtotal := Qtotal+Q[i];
End;
writeln(' Qtotal = ',Qtotal:1:2);
End;
If flowpattern = 2 then
Begin
    for i := 1 to nr do
    Begin
        Q[i] := 1200;
        wc[i] := 1;
    End;
End;

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End;
for i := 1 to nr+1 do
  Begin
    Th[i] := Thi-2*(i-1);
    Tc[i] := Tci+2*(nr+1-i);
  End;
for j := 1 to nr do {start row j}
  Begin
    wc[j] := 1;
    for a := 1 to 200 do
      Begin
        for i := 1 to nr do
          Begin
            Q[i] := UAmix((Th[i]+Th[i+1])/2,(Tc[i]+
              Tc[i+1])/2,mh,mc,Q[i]/nc,nc,wc[i])*
              abs((Th[i]+Th[i+1]-Tc[i]-Tc[i+1])/2));
            Tc[i] := Q[i]/(mc*Cp_air((Tc[i]+Tc[i+1])/2))+Tc[i+1];
            Th[i+1] := Th[i]-Q[i]/(mh*Cp_air((Th[i]+Th[i+1])/2));
          End;
        End;
        for a := 1 to 200 do
          Begin
            for i := 1 to nr do
              Begin
                Q[i] := UAmix((Th[i]+Th[i+1])/2,(Tc[i]+
                  Tc[i+1])/2,mh,mc,Q[i]/nc,nc,wc[i])*
                  ((Th[i]-Tc[i])-(Th[i+1]-Tc[i+1]))/ln((Th[i]-Tc[i])/
                    (Th[i+1]-Tc[i+1])));
                Tc[i] := Q[i]/(mc*Cp_air((Tc[i]+Tc[i+1])/2))+Tc[i+1];
                Th[i+1] := Th[i]-Q[i]/(mh*Cp_air((Th[i]+Th[i+1])/2));
              End;
            End;
          End;
        Qw := Q[j];
        wc[j] := 0;
        for a := 1 to 200 do
          Begin
            for i := 1 to nr do
              Begin
                Q[i] := UAmix((Th[i]+Th[i+1])/2,(Tc[i]+
                  Tc[i+1])/2,mh,mc,Q[i]/nc,nc,wc[i])*
                  abs((Th[i]+Th[i+1]-Tc[i]-Tc[i+1])/2));
                Tc[i] := Q[i]/(mc*Cp_air((Tc[i]+Tc[i+1])/2))+Tc[i+1];
                Th[i+1] := Th[i]-Q[i]/(mh*Cp_air((Th[i]+Th[i+1])/2));
              End;
            End;
          End;
        End;
      End;
    End;
  End;
End;

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End;
for a := 1 to 200 do
  Begin
    for i := 1 to nr do
      Begin
        Q[i] := UAmix((Th[i]+Th[i+1])/2,(Tc[i]+
          Tc[i+1])/2,mh,mc,Q[i]/nc,nc,wc[i])*
          ((Th[i]-Tc[i])-(Th[i+1]-Tc[i+1]))/ln((Th[i]-Tc[i])/
            (Th[i+1]-Tc[i+1])));
        Tc[i] := Q[i]/(mc*Cp_air((Tc[i]+Tc[i+1])/2))+Tc[i+1];
        Th[i+1] := Th[i]-Q[i]/(mh*Cp_air((Th[i]+Th[i+1])/2));
      End;
    End;
    Qe := Q[j];
    If Qw > Qe then wc[j] := 1 else wc[j] := 0;
    for a := 1 to 200 do
      Begin
        for i := 1 to nr do
          Begin
            Q[i] := UAmix((Th[i]+Th[i+1])/2,(Tc[i]+
              Tc[i+1])/2,mh,mc,Q[i]/nc,nc,wc[i])*
              abs((Th[i]+Th[i+1]-Tc[i]-Tc[i+1])/2);
            Tc[i] := Q[i]/(mc*Cp_air((Tc[i]+Tc[i+1])/2))+Tc[i+1];
            Th[i+1] := Th[i]-Q[i]/(mh*Cp_air((Th[i]+Th[i+1])/2));
          End;
        End;
        for a := 1 to 200 do
          Begin
            for i := 1 to nr do
              Begin
                Q[i] := UAmix((Th[i]+Th[i+1])/2,(Tc[i]+
                  Tc[i+1])/2,mh,mc,Q[i]/nc,nc,wc[i])*
                  ((Th[i]-Tc[i])-(Th[i+1]-Tc[i+1]))/ln((Th[i]-Tc[i])/
                    (Th[i+1]-Tc[i+1])));
                Tc[i] := Q[i]/(mc*Cp_air((Tc[i]+Tc[i+1])/2))+Tc[i+1];
                Th[i+1] := Th[i]-Q[i]/(mh*Cp_air((Th[i]+Th[i+1])/2));
              End;
            End;
          End; {end row j}
        for j := 1 to nr do
          Begin
            writeln(' row ',j,' Q = ',Q[j]:1:2);
            If wc[j] = 1 then
              Begin

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        writeln(' working fluid = water');
    End;
    If wc[j] = 0 then
        Begin
            writeln(' working fluid = ethanol');
        End;
        writeln(' Thi = ',Th[j]:1:2,' Tho = ',Th[j+1]:1:2,' Tci = ',Tc[j+1]:1:2,
            ' Tco = ',Tc[j]:1:2);
    End;
    Qtotal := 0;
    for j := 1 to nr do
        Begin
            Qtotal := Qtotal+Q[j];
        End;
    writeln(' Qtotal = ',Qtotal:1:2);
End;
End.

```

A.2 Simulation Program for Selecting the Suitable Mixture Content of TEG-Water for High Operating Temperature Thermosyphon Heat Exchanger

1. Method for Using Program

This program is used to calculate the suitable mixture content of TEG-water in each row of the thermosyphon heat exchanger for reducing the flooding phenomenon of the working fluid inside the thermosyphon. The method for input primary data is similar to that of previous program. The outputs of this program are the suitable mixture content, the heat transfer rate and the inlet and the outlet temperature of the hot and the cold streams in each row.

2. Detail of the Program

```

Program Heat_Pipe_002; {heat transfer rate at high operating temperature}
Uses Wincrt;
Var Thi,Tci,Qtotal,factor,Bo,Xvteg,T : real;
    mh,mc,Le,Lc,fs,fh,ft,St,Sd,Sl,od,fn,id,Qc : real;
    Q : array[1..100] of real;
    Th : array[1..100] of real;
    Tc : array[1..100] of real;
    Qcc : array[1..100] of real;
    Xvtegc : array[1..100] of real;
    lmtd,Thow,Tcow,Thoe,Tcoe : real;
    fntype,arrivalment,flowpattern,cross_section_area: real;
    a,nr,nc,i,j,k : integer;
    wc : array[1..100] of real;
    mixture : string[10];
Function den_air(T:real):real; {calculate density of air}
Begin
    den_air := 0.000005*sqr(T+273.15)-0.0063*(T+273.15)+2.6043;
End;
Function vis_air(T:real):real; {calculate viscosity of air}
Begin
    vis_air := (-0.0003*sqr(T+273.15)+0.6361*(T+273.15)+17.349)/100000000;
End;
Function Cp_air(T:real):real; {calculate Cp of air}
Begin
    Cp_air := (0.0000004*sqr(T+273.15)-0.0002*(T+273.15)+1.0324)*1000;
End;

```

```

Function k_air(T:real):real; {calculate thermal conductivity of air}
Begin
  k_air := (-0.00003*sqr(T+273.15)+0.0955*(T+273.15)+0.24)/1000;
End;
Function Pw(T:real):real; {vapor pressure of water}
Var Pc,Tc,b1,b2,b3,b4,Tr : real;
Begin
  Pc := 22.093*1000000;  Tc := 647.25;
  b1 := -7.78747;      b2 := 1.50255;
  b3 := -2.81152;      b4 := -1.22268;
  Tr := (T+273.15)/Tc;
  Pw := exp(ln(Pc)+(1/Tr)*(b1*(1-Tr)+b2*exp((3/2)*ln(1-Tr))
    +b3*exp(3*ln(1-Tr))+b4*exp(6*ln(1-Tr))));
End;
Function Dwl(T:real):real; {density of liquid water}
Var dc,Tc,b1,b2,b3,b4,Tr : real;
Begin
  dc := 315.5;  Tc := 647.25;
  b1 := 2.24670;  b2 := -2.09405;
  b3 := 2.73700;  b4 := -1.74750;
  Tr := (T+273.15)/Tc;
  Dwl := dc*exp(b1*exp((1/3)*ln(1-Tr))+b2*exp((2/3)*ln(1-Tr))
    +b3*(1-Tr)+b4*exp((4/3)*ln(1-Tr)));
End;
Function Dwg(T:real):real; {density of vapor water}
Var dc,Tc,b1,b2,b3,b4,Tr : real;
Begin
  dc := 315.5;  Tc := 647.25;
  b1 := -1.38200;  b2 := -6.06253;
  b3 := 5.91090;  b4 := -6.68477;
  Tr := (T+273.15)/Tc;
  Dwg := dc*exp(b1*exp((1/3)*ln(1/Tr-1))+b2*exp((2/3)*ln(1/Tr-1))
    +b3*(1/Tr-1)+b4*exp((4/3)*ln(1/Tr-1)));
End;
Function Lw(T:real):real; {latent heat of water}
Var Tc,b1,b2,b3,b4,Tr : real;
Begin
  Tc := 647.25;
  b1 := 1.7035*1000000;  b2 := 1.12332*10000000;
  b3 := -1.47041*10000000;  b4 := 6.35750*10000000;
  Tr := (T+273.15)/Tc;
  Lw := b1*exp((1/3)*ln(1-Tr))+b2*exp((2/3)*ln(1-Tr))
    +b3*(1-Tr)+b4*exp((4/3)*ln(1-Tr));
End;

```

```

Function Cpw(T:real):real; {density of liquid water}
  Var Tc,b1,b2,b3,b4,Tr : real;
  Begin
    Tc := 647.25;
    b1 := -1.4995*10000;  b2 := 8.8*0.01;
    b3 := -6.82*0.1;     b4 := -7.05*0.1;
    Tr := (T+273.15)/Tc;
    Cpw := b1*(1+b2*exp((-2/3)*ln(1-Tr))+b3*exp((-1/3)*ln(1-Tr))
           +b4*exp((1/3)*ln(1-Tr)));
  End;

Function Vwl(T:real):real; {dynamic viscosity of liquid water}
  Var Tc,b1,b2,b3,b4,Tr : real;
  Begin
    Tc := 647.25;
    b1 := -1.01083*10;  b2 := 1.39621;
    b3 := 4.8431*0.1;   b4 := 7.1019*0.1;
    Tr := (T+273.15)/Tc;
    Vwl := exp(b1+b2*exp((1/3)*ln(1/Tr-1))+b3*exp((4/3)*ln(1/Tr-1))
              +b4*exp((7/3)*ln(1/Tr-1)));
  End;

Function Vwg(T:real):real; {dynamic viscosity of vapor water}
  Var Tc,b1,b2,b3,b4,Tr : real;
  Begin
    Tc := 647.25;
    b1 := -1.0373*10;  b2 := -8.6737*0.1;
    b3 := -2.9699*0.1; b4 := 9.051*0.01;
    Tr := (T+273.15)/Tc;
    Vwg := exp(b1+b2*exp((1/3)*ln(1/Tr-1))+b3*exp((4/3)*ln(1/Tr-1))
              +b4*exp((7/3)*ln(1/Tr-1)));
  End;

Function kw(T:real):real; {thermal conductivity water}
  Var Tc,b1,b2,b3,b4,b5,Tr : real;
  Begin
    Tc := 647.25;
    b1 := -1.63975;    b2 := 1.11421*10;
    b3 := -2.00805*10; b4 := 1.67447*10;
    b5 := -5.78763;
    Tr := (T+273.15)/Tc;
    kw := b1+b2*Tr+b3*Tr*Tr+b4*Tr*Tr*Tr+b5*Tr*Tr*Tr*Tr;
  End;

Function tanh(x:real):real; {calculate hyperbolic tangent}
  Begin
    tanh := (exp(x)-exp(-x))/(exp(x)+exp(-x));
  End;

```

```

Function Sw(T:real):real; {surface tension water}
  Var  Tc,b1,b2,b3,b4,Tr : real;
  Begin
    Tc := 647.25;
    b1 := 2.358*0.1;  b2 := -6.25*0.1;
    b3 := 1.256;
    Tr := (T+273.15)/Tc;
    Sw := b1*exp(b3*ln(1-Tr))*(1+b2*(1-Tr));
  End;
Function Pt(T:real):real; {vapor pressure of TEG}
  Var  a,b : real;
  Begin
    a := (T+273.15)/710.15;
    b := -3.2121*(1/exp(2.3086*ln(a))-1)+1.1106*(exp(7*ln(a))-1);
    Pt := 101325*33.2*exp(b);
  End;
Function Dtl(T:real):real; {density of liquid TEG}
  Begin
    Dtl:= -1.2979*T+1212.3;
  End;
Function Dtg(T:real):real; {density of vapor TEG}
  Begin
    Dtg:= -4*0.000001*T*T*T+0.0012*T*T-0.0712*T+1.5854;
  End;
Function Lt(T:real):real; {latent heat of TEG}
  Var  x,Tr,Tbr : real;
  Begin
    Tr := (T+273.15)/710.15;
    Tbr := 0.7857;
    x := (Tbr/Tr)*(1-Tr)/(1-Tbr);
    Lt := 502046*(Tr/Tbr)*(x+exp(0.35298*ln(x)))/(1+exp(0.13856*ln(x)));
  End;
Function Cpt(T:real):real; {density of liquid TEG}
  Begin
    Cpt := 4.5756*T+2037.2;
  End;
Function Vtl(T:real):real; {dynamic viscosity of liquid TEG}
  Begin
    Vtl:= 1.3994*exp(-1.159*ln(T));
  End;
Function kt(T:real):real; {thermal conductivity TEG}
  Begin
    kt := -0.00004*T+0.1975;
  End;

```

```

Function Ste(T:real):real; {surface tension TEG}
Begin
  Ste := 0.001*(126.314+exp((11/9)*ln(1-(T+273.15)/710.15)));
End;
Function Dml(T,Xvteg:real):real; {density of liquid water-TEG mixture}
Var x:real;
Begin
  x := dtl(T)*Xvteg/(dtl(T)*Xvteg+dwl(T)*(1-Xvteg));
  Dml:= dtl(T)*x+dwl(T)*(1-x);
End;
Function Dmg(T,Xvteg:real):real; {density of vapor water-TEG mixture}
Begin
  If Xvteg = 0.25 then
    Begin
      Dmg := 0.0071*exp(1.3375*ln(T));
    End;
  If Xvteg = 0.5 then
    Begin
      Dmg := 0.0004*exp(1.9077*ln(T));
    End;
  If Xvteg = 0.75 then
    Begin
      Dmg := 0.00006*exp(2.3188*ln(T));
    End;
End;
Function Lm(T,Xvteg:real):real; {latent heat of water-TEG mixture}
Var x:real;
Begin
  x := Dtl(T)*Xvteg/(Dtl(T)*Xvteg+Dwl(T)*(1-Xvteg));
  Lm := Lt(T)*x+Lw(T)*(1-x);
End;
Function Sm(T,Xvteg:real):real; {surface tension of water-TEG mixture}
Var Ww,Wt,B,W,Ww1,Ww2,xm,x : real;
Begin
  xm := (1-Xvteg)*Dwl(T)/(Xvteg*Dtl(T)+(1-Xvteg)*Dwl(T));
  x := (xm/18)/((xm/18)+((1-xm)/150));
  Ww := x*(150/Dtl(T))/(x*(150/Dtl(T))+(1-x)*(18/Dwl(T)));
  Wt := (1-x)*(150/Dtl(T))/(x*(150/Dtl(T))+(1-x)*(18/Dwl(T)));
  B := ln(exp(6*ln(Ww))/Wt)/ln(10);
  W := (0.441*6/(T+273.15))*((Ste(T)*exp(0.667*ln(150/Dtl(T)))/6)
    -Sw(T)*exp(0.667*ln(18/Dwl(T))));
  Ww1 := 0.1;
  For i := 1 to 100 do
    Begin

```

```

    Ww2 := 1-exp(6*ln(Ww1))/(exp((B+W)*ln(10)));
    Ww1 := Ww2;
  End;
  Sm := exp(4*ln(Ww1*sqrt(sqrt(Sw(T)))+(1-Ww1)*sqrt(sqrt(Ste(T)))));
End;
Function C_staggered(Re,St,Sl:real):real; {Zhukauskas factor}
  Var s : real;
  Begin
    If Re<100 then C_staggered := 0.9
    Else
      Begin
        If Re<1000 then C_staggered := 0.51
        Else
          Begin
            s := St/Sl;
            If Re<200000 then
              Begin
                If s < 2 then C_staggered := 0.35*exp(0.2*ln(s))
                else C_staggered := 0.4;
              End
            Else
              Begin
                If Re<1000000 then C_staggered := 0.022;
              End;
            End;
          End;
        End;
      End;
    End;
  End;
Function m_staggered(Re:real):real; {Zhukauskas factor}
  Begin
    If Re<100 then m_staggered := 0.4
    Else
      Begin
        If Re<1000 then m_staggered := 0.5
        Else
          Begin
            If Re<200000 then m_staggered := 0.6
            Else
              Begin
                If Re<1000000 then m_staggered := 0.84;
              End;
            End;
          End;
        End;
      End;
    End;
  End;
Function C_aligned(Re:real):real; {Zhukauskas factor}

```

```

Begin
  If Re<100 then C_aligned := 0.8
  Else
    Begin
      If Re<1000 then C_aligned := 0.51
      Else
        Begin
          If Re<200000 then C_aligned := 0.27
          Else
            Begin
              If Re<1000000 then C_aligned := 0.21;
            End;
          End;
        End;
      End;
    End;
  End;
Function m_aligned(Re:real):real; {Zhukauskas factor}
Begin
  If Re<100 then m_aligned := 0.4
  Else
    Begin
      If Re<1000 then m_aligned := 0.5
      Else
        Begin
          If Re<200000 then m_aligned := 0.63
          Else
            Begin
              If Re<1000000 then m_aligned := 0.84;
            End;
          End;
        End;
      End;
    End;
  End;
Function UA003(Th,Tc,mh,mc,Q,nc,Xvteg:real):real;{calculate UA of water-TEG}
  Const k = 14.9; {thermal conductivity of metal}
  Var
    Aeo,Aei,Aco,Aci,Afe,Afc,Abe,Abc,heo,hei,hco,hci,mfac,mfae,Vmaxe,Vmaxc,Ree,
    Rec,Pre,Prc,Nue,Nuc,w,effe,effc,Zc,Zef,Zep,phe2,phe3,T,Z,Re,a : real;
    del_Tsat,x,y,heif,heip,Pr,fp,P,xw,xt,yw,yt : real;
    Ve,Vc,Vmaxe1,Vmaxe2,Vmaxc1,Vmaxc2,heiw,heit,hciw,hcit : real;
  Begin
    If arrangement = 1 then
      Begin
        If fintype = 1 then
          Begin
            Abe := nc*Le*3.14159*od;

```

```

Abc := nc*Lc*3.14159*od;
Afe := 0;
Afc := 0;
Ve := mh/(den_air(Th)*cross_section_area);
Vc := mc/(den_air(Tc)*cross_section_area);
Vmaxe := St*Ve/(St-od);
Vmaxc := St*Vc/(St-od);
Ree := den_air(Th)*Vmaxe*od/vis_air(Th);
Rec := den_air(Tc)*Vmaxc*od/vis_air(Tc);
Pre := Cp_air(Th)*vis_air(Th)/k_air(Th);
Prc := Cp_air(Tc)*vis_air(Tc)/k_air(Tc);
Nue := C_aligned(Ree)*exp(m_aligned(Ree)*ln(Ree))
      *exp(0.36*ln(Pre));
Nuc := C_aligned(Rec)*exp(m_aligned(Rec)*ln(Rec))
      *exp(0.36*ln(Prc));
heo := Nue*k_air(Th)/od;
hco := Nuc*k_air(Tc)/od;
effe := 0;
effc := 0;
End;
If fintype = 2 then
Begin
  Afe := ((nc*Le*3.14159)/(fs+ft))*(0.5*(sqr(od+2*fh)-sqr(od))
        +(od+2*fh)*ft);
  Abe := ((nc*Le*3.14159)/(fs+ft))*od*fs;
  Afc := ((nc*Lc*3.14159)/(fs+ft))*(0.5*(sqr(od+2*fh)-sqr(od))
        +(od+2*fh)*ft);
  Abc := ((nc*Lc*3.14159)/(fs+ft))*od*fs;
  mfae := nc*Le*(St-od-(2*ft*fh)/(fs+ft));
  mfac := nc*Lc*(St-od-(2*ft*fh)/(fs+ft));
  Vmaxe := mh/(den_air(Th)*mfae);
  Vmaxc := mc/(den_air(Tc)*mfac);
  Ree := Vmaxe*od*den_air(Th)/vis_air(Th);
  Rec := Vmaxc*od*den_air(Tc)/vis_air(Tc);
  Pre := Cp_air(Th)*vis_air(Th)/k_air(Th);
  Prc := Cp_air(Tc)*vis_air(Tc)/k_air(Tc);
  Nue := 0.30*exp(0.625*ln(Ree))*exp(-0.375*
        ln((nc*Le*3.14159*od)/(Afe+Abe)))
        *exp(0.333*ln(Pre));
  Nuc := 0.30*exp(0.625*ln(Rec))*exp(-0.375*
        ln((nc*Lc*3.14159*od)/(Afc+Abc)))
        *exp(0.333*ln(Prc));
  heo := Nue*k_air(Th)/od;
  hco := Nuc*k_air(Tc)/od;

```

```

w := 0.5*od*(((od+2*fh)/od)-1)*(1+0.35*ln((od+2*fh)/od));
effe := (tanh(sqrt(2*heo/(ft*k))*w)/(sqrt(2*heo/(ft*k))*w));
effc := (tanh(sqrt(2*hco/(ft*k))*w)/(sqrt(2*hco/(ft*k))*w));
End;
End;
If arrangement = 2 then
Begin
  If fintype = 1 then
  Begin
    Abe := nc*Le*3.14159*od;
    Abc := nc*Lc*3.14159*od;
    Afe := 0;
    Afc := 0;
    Ve := mh/(den_air(Th)*cross_section_area);
    Vc := mc/(den_air(Tc)*cross_section_area);
    Vmaxe1:= St*Ve/(St-od);
    Vmaxc1:= St*Vc/(St-od);
    Vmaxe2:= St*Ve/(2*(Sd-od));
    Vmaxc2:= St*Vc/(2*(Sd-od));
    If Vmaxe1 > Vmaxe2 then Vmaxe := Vmaxe1
    else Vmaxe := Vmaxe2;
    If Vmaxc1 > Vmaxc2 then Vmaxc := Vmaxc1
    else Vmaxc := Vmaxc2;
    Ree := den_air(Th)*Vmaxe*od/vis_air(Th);
    Rec := den_air(Tc)*Vmaxc*od/vis_air(Tc);
    Pre := Cp_air(Th)*vis_air(Th)/k_air(Th);
    Prc := Cp_air(Tc)*vis_air(Tc)/k_air(Tc);
    Nue := C_staggered(Ree,St,Sl)*exp(m_staggered(Ree)*ln(Ree))
          *exp(0.36*ln(Pre));
    Nuc := C_staggered(Rec,St,Sl)*exp(m_staggered(Rec)*ln(Rec))
          *exp(0.36*ln(Prc));
    heo := Nue*k_air(Th)/od;
    hco := Nuc*k_air(Tc)/od;
    effe := 0;
    effc := 0;
  End;
  If fintype = 2 then
  Begin
    Afe := ((nc*Le*3.14159)/(fs+ft))*(0.5*(sqrt(od+2*fh)-sqrt(od))
          +(od+2*fh)*ft);
    Abe := ((nc*Le*3.14159)/(fs+ft))*od*fs;
    Afc := ((nc*Lc*3.14159)/(fs+ft))*(0.5*(sqrt(od+2*fh)-sqrt(od))
          +(od+2*fh)*ft);
    Abc := ((nc*Lc*3.14159)/(fs+ft))*od*fs;

```

```

mfae := 2*nc*Le*(Sd-od-(2*ft*fh)/(fs+ft));
mfac := 2*nc*Lc*(Sd-od-(2*ft*fh)/(fs+ft));
Vmaxe := mh/(den_air(Th)*mfae);
Vmaxc := mc/(den_air(Tc)*mfac);
Ree := Vmaxe*od*den_air(Th)/vis_air(Th);
Rec := Vmaxc*od*den_air(Tc)/vis_air(Tc);
Pre := Cp_air(Th)*vis_air(Th)/k_air(Th);
Prc := Cp_air(Tc)*vis_air(Tc)/k_air(Tc);
Nue := 0.242*exp(0.658*ln(Ree))*exp(0.297*ln(fs/fh))
      *exp(-0.091*ln(St/Sl))*exp(0.33333*ln(Pre));
Nuc := 0.242*exp(0.658*ln(Rec))*exp(0.297*ln(fs/fh))
      *exp(-0.091*ln(St/Sl))*exp(0.33333*ln(Prc));
heo := Nue*k_air(Th)/od;
hco := Nuc*k_air(Tc)/od;
w := 0.5*od*(((od+2*fh)/od)-1)*(1+0.35*ln((od+2*fh)/od));
effe := (tanh(sqrt(2*heo/(ft*k)))*w)/(sqrt(2*heo/(ft*k))*w);
effc := (tanh(sqrt(2*hco/(ft*k)))*w)/(sqrt(2*hco/(ft*k))*w);
End;
If fintype = 3 then
Begin
  Afe := ((nc+0.5)*St*Sl-nc*3.14159*od*od*0.25)*2*
        (Le*fn/0.0254);
  Abe := ((nc*Le*3.14159)/(fs+ft))*od*fs;
  Afc := ((nc+0.5)*St*Sl-nc*3.14159*od*od*0.25)*2*
        (Lc*10/0.0254);
  Abc := ((nc*Lc*3.14159)/(fs+ft))*od*fs;
  mfae := 2*nc*Le*(Sd-od-(2*ft*fh)/(fs+ft));
  mfac := 2*nc*Lc*(Sd-od-(2*ft*fh)/(fs+ft));
  Vmaxe := mh/(den_air(Th)*mfae);
  Vmaxc := mc/(den_air(Tc)*mfac);
  Ree := Vmaxe*od*den_air(Th)/vis_air(Th);
  Rec := Vmaxc*od*den_air(Tc)/vis_air(Tc);
  Pre := Cp_air(Th)*vis_air(Th)/k_air(Th);
  Prc := Cp_air(Tc)*vis_air(Tc)/k_air(Tc);
  heo := 0.14*exp(-0.328*ln(Ree))*exp(-0.502*ln(St/Sl))
        *exp(0.031*ln(fs/od))*den_air(Th)*Vmaxe*Cp_air(Th)
        *exp((-2/3)*ln(Pre));
  hco := 0.14*exp(-0.328*ln(Rec))*exp(-0.502*ln(St/Sl))
        *exp(0.031*ln(fs/od))*den_air(Tc)*Vmaxc*Cp_air(Tc)
        *exp((-2/3)*ln(Prc));
  fp := 1.27*((St/2)/(0.5*od))*sqrt((sqrt((St/2)*(St/2)+Sl*Sl)/2)
        /(St/2)-0.3);
  w := (fp-1)*(1+0.35*ln(fp));
  effe := (tanh(sqrt(2*heo/(ft*k)))*w*(od/2))/(sqrt(2*heo/(ft*k))*w)

```

```

        *(od/2)));
    effc := (tanh(sqrt(2*hco/(ft*k))*w*(od/2))/(sqrt(2*hco/(ft*k))*w
        *(od/2)));
    End;
End;
Aeo := effe*Afe+Abe;
Aco := effe*Afc+Abc;
Aei := 3.14159*id*Lc*nc;
Aci := 3.14159*id*Lc*nc;
T := (Th+Tc)/2;
If Xvteg > 0 then
    Begin
        If Xvteg < 1 then
            Begin
                P := (-8.6856*sqr(Xvteg)+4.3062*Xvteg+10.711)*T*T
                    +(1561.8*sqr(Xvteg)-1564.4*Xvteg-101.65)*T
                    +(-5337.6*sqr(Xvteg)-7864.8*Xvteg+16019);
                xw := (P-Pt(T))/(Pw(T)-Pt(T));
                xt := 1-xw;
                yw := ((1/P)-(1/Pt(T)))/((1/Pw(T))-(1/Pt(T)));
                yt := 1-yw;
                {calculate heat transfer coefficient of water}
                x := (1/(Vwl(T)*Lw(T)))*sqrt(Sw(T)/(9.81*(Dwl(T)-Dwg(T))));
                y := Lw(T)*Vwl(T)/kw(T);
                heiw := exp(0.9332*(ln(18.688)-ln(x)-1.0716*ln(y)+0.0716*ln(Q/Aei)));
                hciw := exp(1.30378*(ln(0.943)+0.233*ln(sqr(Dwl(T))*9.81*Lw(T)*
                    kw(T)*kw(T)*kw(T)*Aci/(Vwl(T)*Lc*Q))));
                {calculate heat transfer coefficient of TEG}
                x := (1/(Vtl(T)*Lt(T)))*sqrt(Ste(T)/(9.81*(Dtl(T)-Dtg(T))));
                y := Lt(T)*Vtl(T)/kt(T);
                heit := exp(0.910249*ln(20.565*exp(0.0986*ln(Q/Aei))/
                    (x*exp(1.0986*ln(y)))));
                hcit := exp(1.2195*ln(0.943*exp(0.18*ln(sqr(Dtl(T))*9.81*Lt(T)*
                    kt(T)*kt(T)*kt(T)*Aci/(Vtl(T)*Lc*Q))));
                {calculate heat transfer coefficient of mixture}
                hei := xw*(heiw)+xt*(heit);
                hci := yw*(hciw)+yt*(hcit);
            End;
        If Xvteg = 1 then
            Begin
                x := (1/(Vtl(T)*Lt(T)))*sqrt(Ste(T)/(9.81*(Dtl(T)-Dtg(T))));
                y := Lt(T)*Vtl(T)/kt(T);
                hei := exp(0.910249*ln(20.565*exp(0.0986*ln(Q/Aei))/
                    (x*exp(1.0986*ln(y)))));
            End;
        End;
    End;

```

```

        hci := exp(1.2195*ln(0.943*exp(0.18*ln(sqr(Dtl(T))*9.81*Lt(T)*
            kt(T)*kt(T)*kt(T)*Aci/(Vtl(T)*Lc*Q)))));
    End;
End;
If Xvteg = 0 then
    Begin
        {calculate heat transfer coefficient of water}
        x := (1/(Vwl(T)*Lw(T)))*sqr(Sw(T)/(9.81*(Dwl(T)-Dwg(T))));
        y := Lw(T)*Vwl(T)/kw(T);
        hei := exp(0.9332*(ln(18.688)-ln(x)-1.0716*ln(y)+0.0716*ln(Q/Aei)));
        hci := exp(1.30378*(ln(0.943)+0.233*ln(sqr(Dwl(T))*9.81*Lw(T)*
            kw(T)*kw(T)*kw(T)*Aci/(Vwl(T)*Lc*Q)))));
    End;
    UA003 := 1/(1/(heo*Aeo)+1/(hco*Aco)+1/(hei*Aei)+1/(hci*Aci)
        +ln(od/id)/(2*3.14159*k*Le*nc)+ln(od/id)/(2*3.14159*k*Lc*nc));
End;
Begin {main program}
    writeln(' PROGRAM HEAT PIPE 002 ');
    writeln;
    Writeln(' This program is used for calculating performance of the thermosyphon');
    writeln(' heat exchanger operating at high temperature with binary working');
    writeln(' fluids (water-TEG)');
    writeln;
    writeln(' Developed by Atipoang Nuntaphan, Ph.D. student ');
    writeln(' Thermal Technology Division, School of Energy and Materials,');
    writeln(' King Mongkut's, University of Technology Thonburi');
    writeln;
    Writeln(' *****');
    writeln;
    writeln(' Please enter informations below ');
    writeln(' 1. Heat exchanger informations');
    write(' 1.1 Input number of rows ');
    readln(nr);
    write(' 1.2 Input number of column ');
    readln(nc);
    write(' 1.3 Input outside diameter of tube (m) ');
    readln(od);
    write(' 1.4 Input inside diameter of tube (m) ');
    readln(id);
    write(' 1.5 Input evaporator length (m) ');
    readln(Le);
    write(' 1.6 Input condenser length (m) ');
    readln(Lc);
    writeln(' 1.7 Input type of fin ');

```

```

write('      1. bare tube 2. circular finned 3. plain plate finned ');
readln(fintype);
write(' 1.8 Input fin thickness (m) (input 0 for bare tube) ');
readln(ft);
write(' 1.9 Input fin gap (m) (input 0 for bare tube) ');
readln(fs);
write(' 1.10 Input fin height (m) (input 0 for bare tube) ');
readln(fh);
write(' 1.11 Input number of fins (fin/inch) (input 0 for bare tube) ');
readln(fn);
writeln(' 1.11 Input pipe arrangement ');
write('      1. aligned 2. staggered ');
readln(arrangement);
write(' 1.12 Input transverse pitch (St) (m) ');
readln(St);
write(' 1.13 Input longitudinal pitch (Sl) (m) ');
readln(Sl);
write(' 1.14 Input directional pitch (Sd) (m) (input 0 for aligned) ');
readln(Sd);
write(' 1.15 Input cross section area of duct (m^2) ');
readln(cross_section_area);
writeln;
writeln(' 2. Working conditions ');
write(' 2.1 Input inlet temperature of hot gas (C) ');
readln(Thi);
write(' 2.2 Input inlet temperature of cold gas (C) ');
readln(Tci);
write(' 2.3 Input mass flow rate of hot gas (kg/s) ');
readln(mh);
write(' 2.4 Input mass flow rate of cold gas (kg/s) ');
readln(mc);
writeln(' 2.5 Input flow pattern ');
write('      1. parallel flow 2. counter flow ');
read(flowpattern);
writeln;
{PARALLEL FLOW}
If flowpattern = 1 then
  Begin
    Th[1] := Thi;
    Tc[1] := Tci;
    For i := 1 to nr do
      Begin
        write(' row = ',i);
        Q[i] := 1200;
      End
    End
  End

```

```

Xvteg := 0;
Q[i] := UA003(Th[i],Tc[i],mh,mc,Q[i]/nc,nc,Xvteg)*(Th[i]-Tc[i]);
Th[i+1] := Th[i]-Q[i]/(mh*Cp_air(Th[i]));
Tc[i+1] := Q[i]/(mc*Cp_air(Tc[i]))+Tc[i];
For j := 1 to 200 do
  Begin
    lmtdd := ((Th[i]-Tc[i])-(Th[i+1]-Tc[i+1]))/ln((Th[i]-Tc[i])/
      (Th[i+1]-Tc[i+1]));
    Q[i] := UA003((Th[i]+Th[i+1])/2,(Tc[i]+Tc[i+1])/2,
      mh,mc,Q[i]/nc,nc,Xvteg)*lmtdd;
    Th[i+1] := Th[i]-Q[i]/(mh*Cp_air((Th[i]+Th[i+1])/2));
    Tc[i+1] := Tc[i]+Q[i]/(mc*Cp_air((Tc[i]+Tc[i+1])/2));
  End;
{calculate critical heat flux of water}
T := 0.5*((Th[i]+Th[i+1])/2+(Tc[i]+Tc[i+1])/2);
Bo := id*sqrt(9.81*(Dwl(T)-Dwg(T))/Sw(T));
factor := exp(0.14*ln(Dwl(T)/Dwg(T)))*sqr(tanh(exp(0.25*ln(Bo))));
Qc := 0.69*factor*Lw(T)*(3.14159*id*id/4)*
  exp(0.25*ln(9.81*Sw(T)*(Dwl(T)-Dwg(T))))*
  exp(-2*ln(exp(-0.25*ln(Dwg(T)))+exp(-0.25*ln(Dwl(T))))));
If Qc > Q[i] then
  Begin
    Q[i] := Qc;
  End;
If Qc < Q[i] then
  Begin
    Repeat
      Xvteg := Xvteg+0.25;
      For j := 1 to 200 do
        Begin
          lmtdd := ((Th[i]-Tc[i])-(Th[i+1]-Tc[i+1]))/
            ln((Th[i]-Tc[i])/(Th[i+1]-Tc[i+1]));
          Q[i] := UA003((Th[i]+Th[i+1])/2,(Tc[i]+Tc[i+1])/2,
            mh,mc,Q[i]/nc,nc,Xvteg)*lmtdd;
          Th[i+1] := Th[i]-Q[i]/(mh*Cp_air((Th[i]+Th[i+1])/2));
          Tc[i+1] := Tc[i]+Q[i]/(mc*Cp_air((Tc[i]+Tc[i+1])/2));
        End;
      {calculate critical heat flux of mixture}
      T := 0.5*((Th[i]+Th[i+1])/2+(Tc[i]+Tc[i+1])/2);
      Bo := id*sqrt(9.81*(Dml(T,Xvteg)-Dmg(T,Xvteg))/
        Sm(T,Xvteg));
      factor := exp(0.14*ln(Dml(T,Xvteg)/Dmg(T,Xvteg)))*
        sqr(tanh(exp(0.25*ln(Bo))));
      Qc := 0.69*factor*Lm(T,Xvteg)*(3.14159*id*id/4)*

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exp(0.25*ln(9.81*Sm(T,Xvteg)*(Dml(T,Xvteg)-
Dmg(T,Xvteg)))*exp(-2*ln(exp(-0.25*ln(Dmg(T,Xvteg)))+
exp(-0.25*ln(Dml(T,Xvteg))))));
Until Q[i] < Qc;
End;
Writeln(' row ',i,' Q = ',Q[i]:1:2,' Tho = ',Th[i+1]:1:2,' Tco = ',
Tc[i+1]:1:2,' Xteg = ',Xvteg:1:2);
End;
Qtotal := 0;
for i := 1 to nr do
Begin
Qtotal := Qtotal+Q[i];
End;
writeln(' Qtotal = ',Qtotal:1:2);
End;
{COUNTER FLOW}
If flowpattern = 2 then
Begin
for i := 1 to nr do
Begin
Q[i] := 1200;
End;
for i := 1 to nr+1 do
Begin
Th[i] := Thi-2*(i-1);
Tc[i] := Tci+2*(nr+1-i);
End;
for a := 1 to 200 do
Begin
for i := 1 to nr do
Begin
Q[i] := UA003((Th[i]+Th[i+1])/2,(Tc[i]+Tc[i+1])/2
,mh,mc,Q[i]/nc,nc,0)*
abs((Th[i]+Th[i+1]-Tc[i]-Tc[i+1])/2);
Tc[i] := Q[i]/(mc*Cp_air((Tc[i]+Tc[i+1])/2))+Tc[i+1];
Th[i+1] := Th[i]-Q[i]/(mh*Cp_air((Th[i]+Th[i+1])/2));
End;
End;
for a := 1 to 200 do
Begin
for i := 1 to nr do
Begin
Q[i] := UA003((Th[i]+Th[i+1])/2,(Tc[i]+Tc[i+1])/2,
mh,mc,Q[i]/nc,nc,0)*

```

```

        ((Th[i]-Tc[i])-(Th[i+1]-Tc[i+1]))/
        ln((Th[i]-Tc[i])/(Th[i+1]-Tc[i+1]));
    Tc[i] := Q[i]/(mc*Cp_air((Tc[i]+Tc[i+1])/2))+Tc[i+1];
    Th[i+1] := Th[i]-Q[i]/(mh*Cp_air((Th[i]+Th[i+1])/2));
End;
End;

{Calculate critical heat flux}
for i := 1 to nr do
Begin
    T := 0.5*((Th[i]+Th[i+1])/2+(Tc[i]+Tc[i+1])/2);
    {water}
    Bo := id*sqrt(9.81*(Dwl(T)-Dwg(T))/Sw(T));
    factor := exp(0.14*ln(Dwl(T)/Dwg(T)))*sqrt(tanh(exp(0.25*ln(Bo))));
    Qcc[1] := 0.69*factor*Lw(T)*(3.14159*id*id/4)*
        exp(0.25*ln(9.81*Sw(T)*(Dwl(T)-Dwg(T)))*
        exp(-2*ln(exp(-0.25*ln(Dwg(T)))+exp(-0.25*ln(Dwl(T))))));
    {25% TEG}
    Xvteg := 0.25;
    Bo := id*sqrt(9.81*(Dml(T,Xvteg)-Dmg(T,Xvteg))/Sm(T,Xvteg));
    factor := exp(0.14*ln(Dml(T,Xvteg)/Dmg(T,Xvteg)))*
        sqrt(tanh(exp(0.25*ln(Bo))));
    Qcc[2] := 0.69*factor*Lm(T,Xvteg)*(3.14159*id*id/4)*
        exp(0.25*ln(9.81*Sm(T,Xvteg)*(Dml(T,Xvteg)-
        Dmg(T,Xvteg)))*
        exp(-2*ln(exp(-0.25*ln(Dmg(T,Xvteg)))+
        exp(-0.25*ln(Dml(T,Xvteg))))));
    {50% TEG}
    Xvteg := 0.50;
    Bo := id*sqrt(9.81*(Dml(T,Xvteg)-Dmg(T,Xvteg))/Sm(T,Xvteg));
    factor := exp(0.14*ln(Dml(T,Xvteg)/Dmg(T,Xvteg)))*
        sqrt(tanh(exp(0.25*ln(Bo))));
    Qcc[3] := 0.69*factor*Lm(T,Xvteg)*(3.14159*id*id/4)*
        exp(0.25*ln(9.81*Sm(T,Xvteg)*(Dml(T,Xvteg)-
        Dmg(T,Xvteg)))*
        exp(-2*ln(exp(-0.25*ln(Dmg(T,Xvteg)))+
        exp(-0.25*ln(Dml(T,Xvteg))))));
    {75% TEG}
    Xvteg := 0.75;
    Bo := id*sqrt(9.81*(Dml(T,Xvteg)-Dmg(T,Xvteg))/Sm(T,Xvteg));
    factor := exp(0.14*ln(Dml(T,Xvteg)/Dmg(T,Xvteg)))*
        sqrt(tanh(exp(0.25*ln(Bo))));
    Qcc[4] := 0.69*factor*Lm(T,Xvteg)*(3.14159*id*id/4)*
        exp(0.25*ln(9.81*Sm(T,Xvteg)*(Dml(T,Xvteg)-

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    Dmg(T,Xvteg))) *
    exp(-2*ln(exp(-0.25*ln(Dmg(T,Xvteg))))+
    exp(-0.25*ln(Dml(T,Xvteg)))));
{TEG}
Bo := id*sqrt(9.81*(Dtl(T)-Dtg(T))/Ste(T));
factor := exp(0.14*ln(Dtl(T)/Dtg(T)))*sqr(tanh(exp(0.25*ln(Bo))));
Qcc[5] := 0.69*factor*Lt(T)*(3.14159*id*id/4)*
    exp(0.25*ln(9.81*Ste(T)*(Dtl(T)-Dtg(T)))*
    exp(-2*ln(exp(-0.25*ln(Dtg(T)))+exp(-0.25*ln(Dtl(T))))));
If Q[i] < Qcc[1] then Xvtegc[i] := 0
else
    Begin
    If Q[i] < Qcc[2] then Xvtegc[i] := 0.25
    else
        Begin
        If Q[i] < Qcc[3] then Xvtegc[i] := 0.50
        else
            Begin
            If Q[i] < Qcc[4] then Xvtegc[i] := 0.75
            else
                Begin
                If Q[i] < Qcc[5] then Xvtegc[i] := 1.0
                else writeln(' critical heat flux higher than that of TEG
                in row ',i);
                End;
            End;
        End;
    End;
End;
End;
{Re-calculate heat transfer rate}
for a := 1 to 200 do
    Begin
    for i := 1 to nr do
        Begin
        Q[i] := UA003((Th[i]+Th[i+1])/2,(Tc[i]+Tc[i+1])/2,
            mh,mc,Q[i]/nc,nc,Xvtegc[i])*
            ((Th[i]-Tc[i])-(Th[i+1]-Tc[i+1]))/
            ln((Th[i]-Tc[i])/(Th[i+1]-Tc[i+1])));
        Tc[i] := Q[i]/(mc*Cp_air((Tc[i]+Tc[i+1])/2))+Tc[i+1];
        Th[i+1] := Th[i]-Q[i]/(mh*Cp_air((Th[i]+Th[i+1])/2));
        End;
    End;
End;
{Show data}
for j := 1 to nr do

```

```

Begin
  If Xvtegc[j] = 0 then mixture := 'water 100%';
  If Xvtegc[j] = 0.25 then mixture := 'TEG 25%';
  If Xvtegc[j] = 0.50 then mixture := 'TEG 50%';
  If Xvtegc[j] = 0.75 then mixture := 'TEG 75%';
  If Xvtegc[j] = 1.0 then mixture := 'TEG 100%';
  writeln(' row ',j,' Q = ',Q[j]:1:2,' %TEG = ',mixture);
  writeln(' Thi = ',Th[j]:1:2,' Tho = ',Th[j+1]:1:2,' Tci = ',
          Tc[j+1]:1:2,' Tco = ',Tc[j]:1:2);
End;
Qtotal := 0;
for j := 1 to nr do
  Begin
    Qtotal := Qtotal+Q[j];
  End;
writeln(' Qtotal = ',Qtotal:1:2);
End;
End.

```

APPENDIX B

PUBLICATIONS

Kiatsiriroat T., Nuntaphan A. and Tiansuwan J., 2000, Thermal Performance Enhancement of Thermosyphon Heat Pipe with Binary Working Fluids, **Experimental Heat Transfer**, 13:137-152.



THERMAL PERFORMANCE ENHANCEMENT OF THERMOSYPHON HEAT PIPE WITH BINARY WORKING FLUIDS

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Thermal performance of a thermosyphon heat pipe using ethanol-water and TEG-water with variations of parameters such as the mixture content, the pipe diameter, and the working temperature have been studied in this research work. From the experiments, it is found that at low temperature of heat source (less than 80°C), the ethanol-water mixture has a higher heat transfer rate than that of water and close to that of pure ethanol. In the case of TEG-water mixture, the heat transfer rate of the thermosyphon varies with the content of TEG in the mixture, and it is found that TEG in the mixture can increase the critical heat flux due to the flooding limit in a small thermosyphon.

The boiling equation of Rohsenow and the condensation equation of Nusselt are modified to predict the heat transfer coefficients inside the thermosyphon. For the mixtures, the weighted average of the heat transfer coefficient of each component can be used to predict the total heat transfer coefficient. Furthermore, it is found that Faghri's equation can be used to predict the critical heat flux due to the flooding limit of the thermosyphon with the binary mixtures.

At present, development of heat exchangers for energy recovery from industrial waste heat has become an interesting topic for energy conservation programs. The thermosyphon heat pipe, one type of heat exchanger, has been used in many industrial processes because of its advantages, such as high thermal conductivity, low cost, and ease of construction.

The thermosyphon heat pipe shown in Figure 1 can be divided into three parts, evaporator, adiabatic, and condenser. When heat is added at the evaporator section, the working fluid inside the heat pipe vaporizes and carries heat from the heat source to the condenser section where heat is rejected to the heat sink. The working fluid condensate returns to the evaporator section by gravity.

The thermosyphon normally uses a single component, usually water, as a working fluid, because water has a high working temperature range (50–200°C) [1] and high latent heat of vaporization. However, in some working conditions, the

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NOMENCLATURE

		Subscripts
A	area, m^2	
Bo	Bond number	
C_p	heat capacity, $J/kg\ K$, $J/mol\ ^\circ C$	1 high-volatile component
D	diameter of pipe, m	2 low-volatile component
g	gravitational acceleration ($= 9.81\ m/s^2$)	ave average
h	heat transfer coefficient, $W/m^2\ K$	c condenser
k	thermal conductivity, $W/m\ K$	cs cross section
L	length, m	e evaporator
$\dot{m}$	mass rate of flow, kg/s	i inlet, inside
Pr	Prandtl number, $(C_p \mu / k)$	id ideal
Q	heat transfer rate, W	l liquid
T	temperature, $^\circ C$, K	max maximum
x	mole fraction	o outlet
Z	thermal resistance, K/W	s surface
λ	latent heat of vaporization, J/kg	v vapor
μ	viscosity, kg/ms	w water
ρ	density, kg/m^3	
σ	surface tension, N/m	

temperature of the heat source may be lower or higher than that of the normal range and the heat transfer rate of the thermosyphon heat pipe tends to decrease. Some experiments [2] showed that some binary mixtures could give better performance than that of a single working fluid.

In the case of a lower-temperature heat source (lower than $80^\circ C$), ethanol-water is proposed as a working fluid because ethanol has a lower boiling point than water. Consequently, it can be boiled easily and higher performance of the thermosyphon should be obtained. In the case of a higher-temperature heat source (more than $200^\circ C$), for water, the transfer of heat may be hindered by a critical limit, particularly the flooding or dryout limit. To extend these limits, triethylene glycol (TEG)-water is proposed as a working fluid. Because TEG has a high

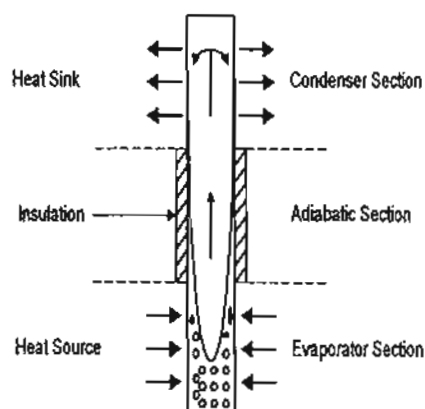


Figure 1. The thermosyphon heat pipe.

boiling point (278°C at normal pressure), TEG in the mixture may retard the limits [3].

The aim of this research work is to study the performance of a thermosyphon using binary working fluids, ethanol-water and TEG-water, and find a heat transfer model which can predict the heat transfer coefficients of the mixtures.

THE EXPERIMENT AND TEST PROCEDURE

Figure 2 shows a schematic diagram of the experimental apparatus. It consists of a thermosyphon heat pipe, a heat source, and a heat sink. In this experiment, three diameter sizes of thermosyphon, 25.40, 19.05, and 12.70 mm with 2-mm wall thickness, have been tested. All of the pipes are made of 304 stainless steel. The total length of each is 100 cm, of which 40 cm is for the evaporator, 40 cm is for the condenser, and 20 cm is for the adiabatic section. There is a pressure gauge at the top of each pipe for measuring the inside pressure. The evaporator section of the thermosyphon is dipped in a hot oil bath and the condenser is cooled by water from a constant-head tank.

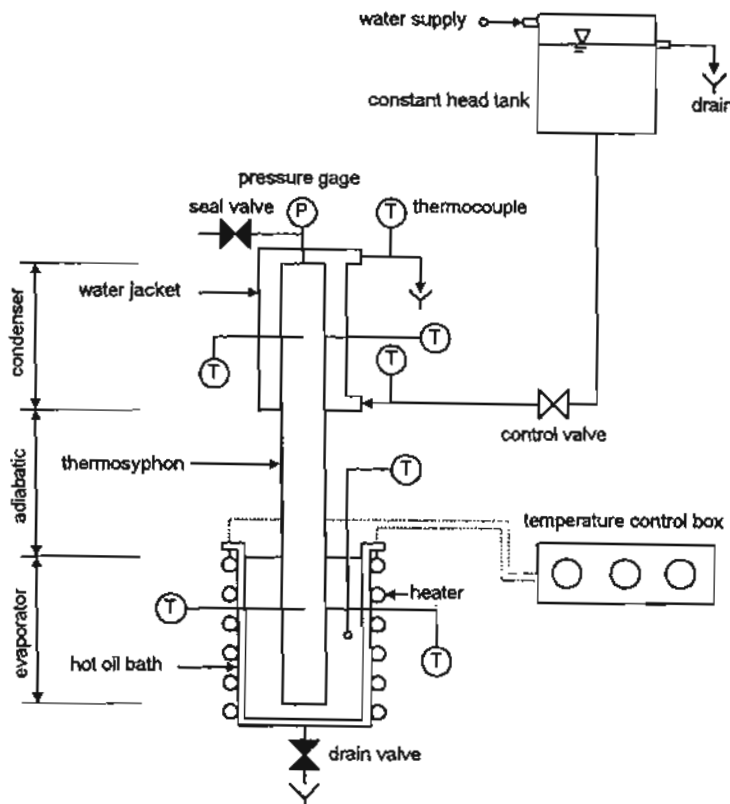


Figure 2. The experiment apparatus.

Seven K-type thermocouples with $\pm 0.5\%$ accuracy are attached to the outer surface of the thermosyphon at 5, 20, 35, 50, 65, 80, and 95 cm from the evaporator end. Moreover, for the biggest tested thermosyphon, seven thermocouples are inserted into the pipe to measure the inside temperatures, and they are located at the same distances as the outer surface measurement.

A hot paraffin oil bath, which is a cylinder of 8.89 cm in diameter and 50 cm in length, is used to generate heat to the thermosyphon. To achieve a uniform oil temperature, air bubbles are supplied to perturb the oil.

The heat sink is a well-insulated water jacket having water flowing in and out to absorb heat at the condenser section. The water jacket is a 5-cm cylindrical tube, 45 cm long. Two thermocouples are inserted into the inlet and outlet ports of the water jacket to measure the water temperature. The flow rate of the cooling water is controlled by a constant-head tank between 6 and 19 g/s.

Ethanol-water and TEG-water at 0, 25, 50, 75, and 100% by volume of higher-volatile component are used as working fluids. The filling ratio for all tests is 50% of the evaporator section's volume. The temperature of the hot oil is controlled within a range of 50 to 225°C. The experiments have been carried out under steady-state conditions.

RESULTS AND DISCUSSION

Temperature Distributions Inside the Thermosyphon

The temperature distributions inside the thermosyphon are shown in Figures 3a and 3b for ethanol-water and TEG-water, respectively.

In Figure 3a it is found that the temperature difference of the evaporator and the condenser sections for the low-temperature heat source (60°C) are dependent on the amount of ethanol in water. At this heat source temperature, a small amount of water could be vaporized, so the transfer of heat is hindered and a high temperature difference between the evaporator and the condenser sections is obtained. At the higher heat source temperature (150°C), both components in the binary mixtures can be boiled and heat transfer rates are high, so the temperature distributions of the fluids inside the thermosyphon are nearly uniform.

The temperature distributions of TEG-water are shown in Figure 3b. Since TEG has a very high boiling point (278°C) so even the heat source temperature is 150°C, a small amount of TEG could be vaporized, which results in low heat transfer rate and high temperature difference between the evaporator and the condenser sections. But when water is mixed with the TEG, better heat transfer is achieved.

Heat Transfer Rate

The heat transfer rate can be calculated by

$$Q = \dot{m}_w C_p (T_{wo} - T_{wi}) \quad (1)$$

where $\dot{m}_w$ is the flow rate of the cooling water, and T_{wi} and T_{wo} are the average temperatures at the inlet and the outlet of the cooling water at the water jacket, respectively.

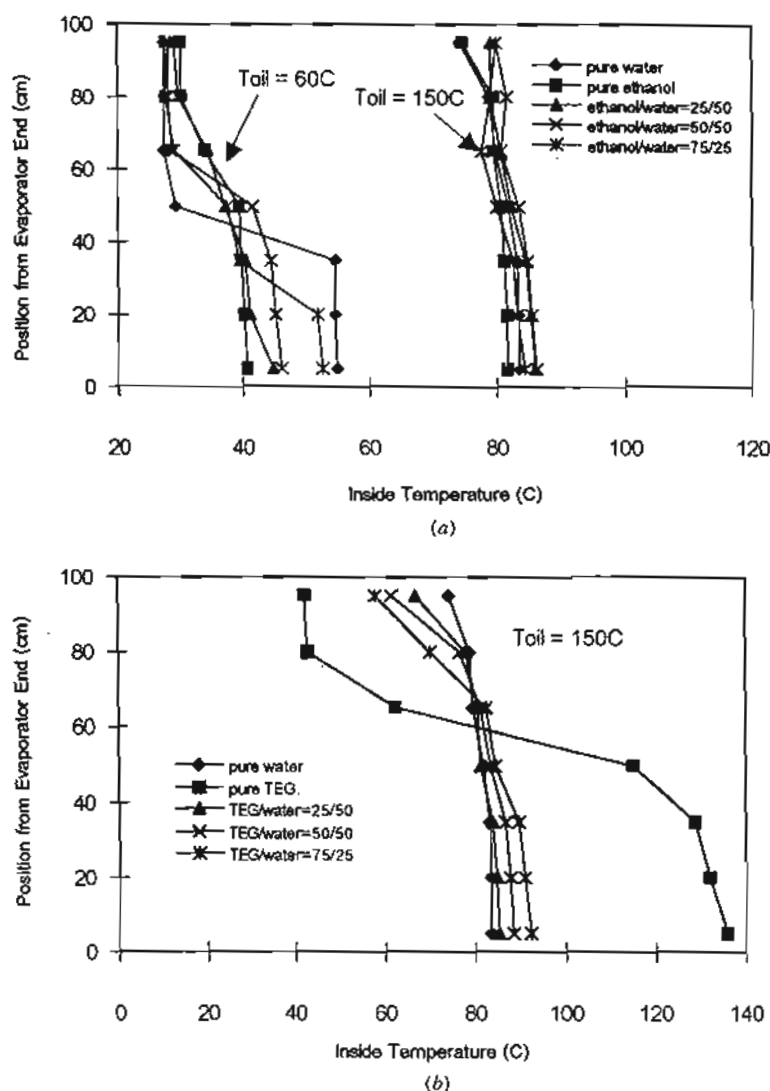


Figure 3. Temperature distributions along the thermosyphon at various conditions: (a) working fluid ethanol-water, tube diameter 25.4 mm; (b) working fluid TEG-water, tube diameter 25.4 mm.

The heat transfer rates at various hot oil temperatures are shown in Figures 4a-4f. In the case of ethanol-water mixtures (Figures 4a-4c), it is found that, at low hot-oil temperature, the heat transfer rates of ethanol-water mixtures are higher than for pure water and close to that of pure ethanol. But when the hot-oil temperature is increased, the heat transfer rate for pure water increases more rapidly than those of the mixtures and overcomes the mixtures at high temperature. This phenomenon occurs because at low temperature, ethanol can be boiled more easily than water and a better heat transfer rate is obtained. At high

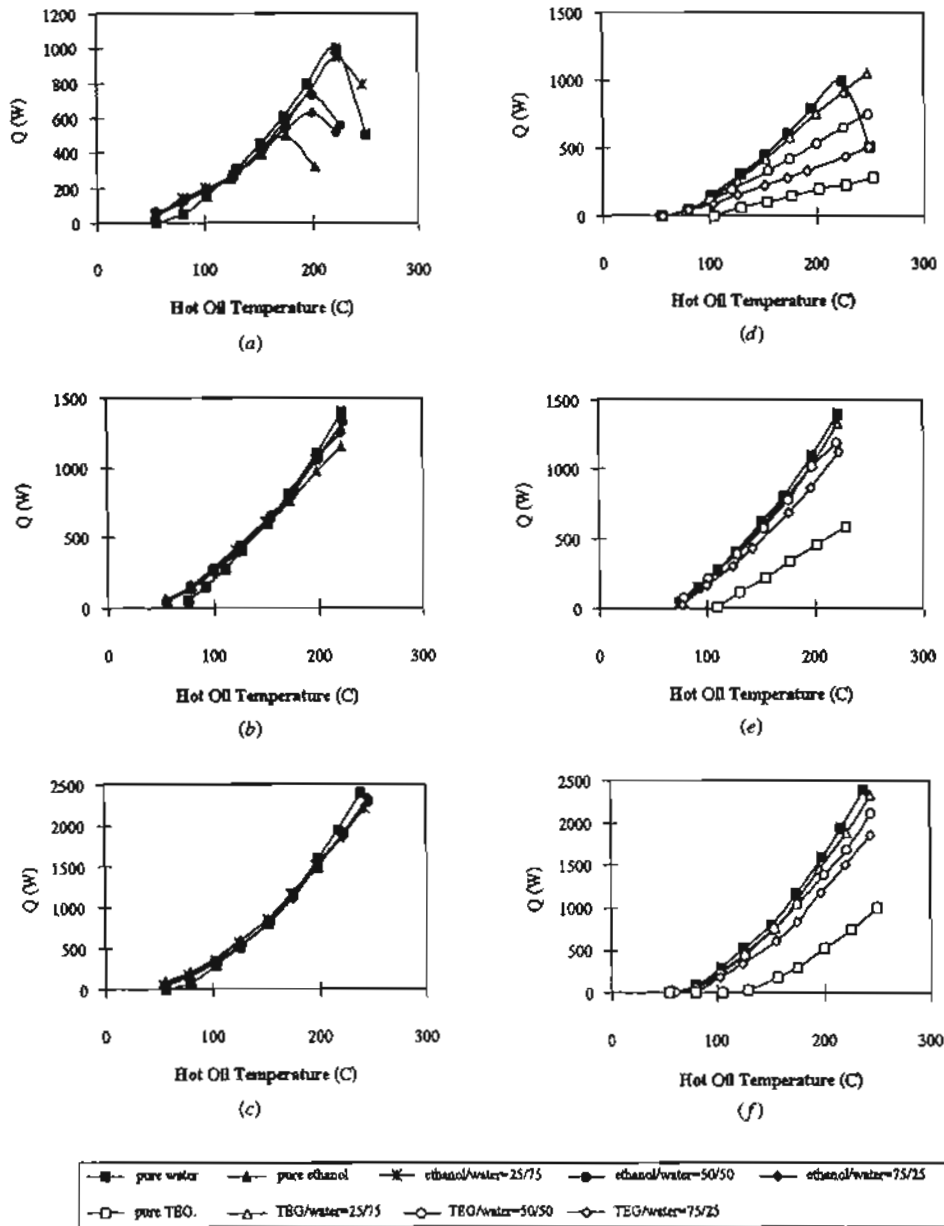


Figure 4. The relations between heat transfer rate of the thermosyphon heat pipe with binary mixtures and the hot oil temperature at various conditions: (a) working fluid ethanol-water, tube diameter 12.7 mm; (b) working fluid ethanol-water, tube diameter 19.05 mm; (c) working fluid ethanol-water, tube diameter 25.4 mm; (d) working fluid TEG-water, tube diameter 12.7 mm; (e) working fluid TEG-water, tube diameter 19.05 mm; (f) working fluid TEG-water, tube diameter 25.4 mm.

temperature, both ethanol and water can be boiled, and since the latent heat of vaporization of water is higher than that of ethanol, at this condition the thermosyphon with pure water can transfer more heat than those of ethanol or mixtures.

Flooding of working fluid in a small thermosyphon (12.70 mm diameter) has been found in the experiment. This phenomenon is due to the interaction between the countercurrent liquid and vapor flows occurring at the liquid-vapor interface in the thermosyphon. The viscous shear force at the interface may retard the return of liquid to the evaporator [4]. From Figure 4a, as the hot-oil temperature increases, the heat transfer rate of each working fluid also increases until the flooding limit has been obtained. After that the heat transfer rate drops drastically. This phenomenon always occurs when using a small thermosyphon with high temperature [4]. The critical heat flux at the flooding limit also depends on the composition of the mixtures.

The latent heat of vaporization of the working fluid plays an important role in controlling the flooding limit [4], and higher latent heat results in higher critical heat flux. In the case of ethanol-water mixtures, the latent heat of vaporization of water is very high compared to that of pure ethanol. Therefore, higher water content in the mixtures results in higher critical heat flux at the flooding limit.

TEG has a high boiling point, therefore the TEG-water mixtures could be vaporized well at high temperature. The heat transfer rate increases as the content of TEG in the mixture decreases. The results are shown in Figures 4d-4f. It was also found that no flooding occurs or the flooding limit is retarded in the working temperature range. From Figure 4d, it is found that with a small amount of TEG (25%) content in the mixture, the heat transfer rate is nearly the same as with pure water and no flooding occurs at high temperature even if the syphon diameter is small.

Thermal Resistance

The thermal resistance, Z , of the thermosyphon is calculated from

$$Z = \frac{T_{se,ave} - T_{sc,ave}}{Q} \quad (2)$$

where $T_{se,ave}$ and $T_{sc,ave}$ are the average outside surface temperatures at the evaporator and the condenser, respectively.

For ethanol-water mixtures (Figures 5a-5c), it is found that at low heat source temperature the resistance is very high, especially for water because a low amount of vapor could be generated. The thermal resistance of pure ethanol and the ethanol-water mixtures are lower than that of pure water because ethanol has a lower boiling point than water. For the high-temperature heat source, a large amount of both ethanol and water could be vaporized and transfer heat at the condenser, resulting in low thermal resistance. The resistances for all the working fluids are close to the values at high temperature. However, the resistance for pure water is lower than the others because of its high latent heat. Moreover, it is found

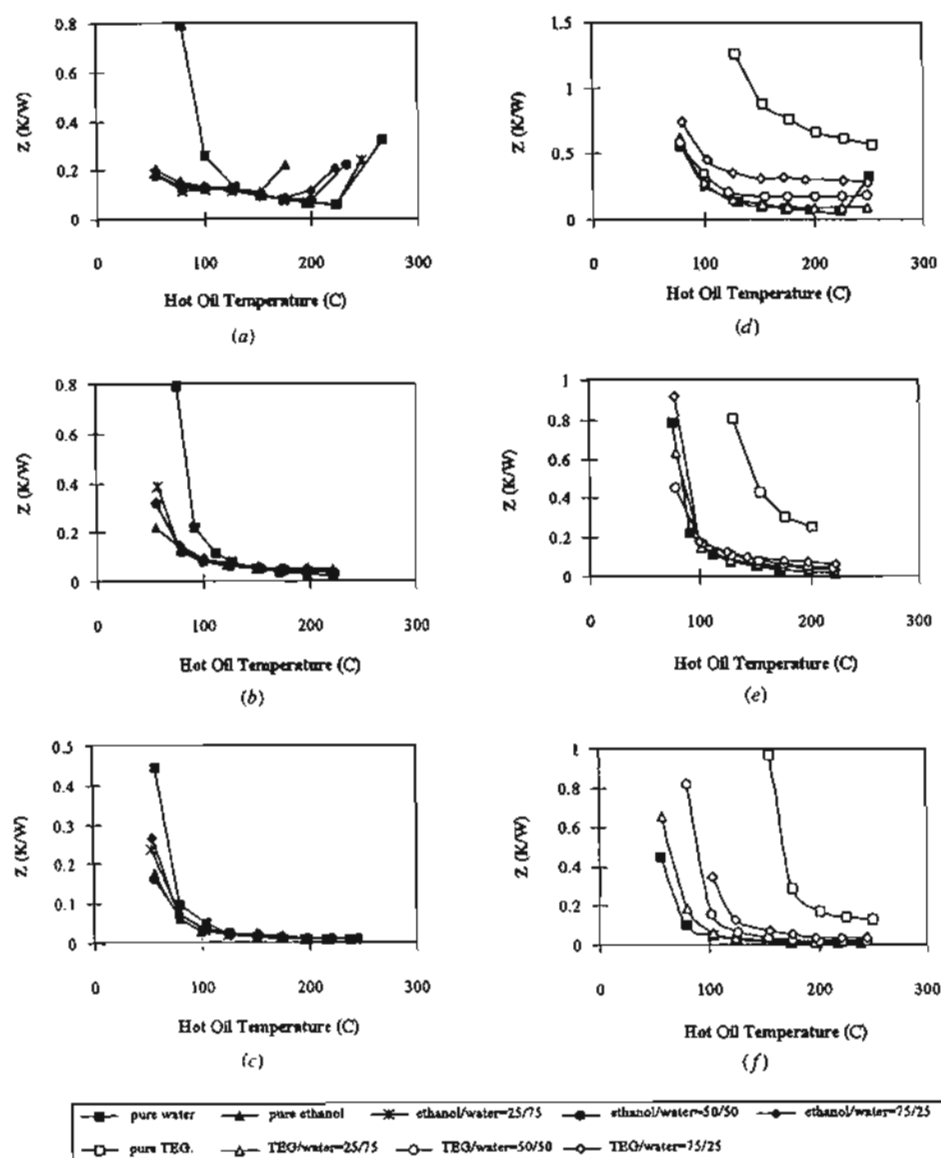


Figure 5. The relation between thermal resistance of the thermosyphon heat pipe with binary mixtures and the hot oil temperature at various conditions: (a) working fluid ethanol-water, tube diameter 12.7 mm; (b) working fluid ethanol-water, tube diameter 19.05 mm; (c) working fluid ethanol-water, tube diameter 25.4 mm; (d) working fluid TEG-water, tube diameter 12.7 mm; (e) working fluid TEG-water, tube diameter 19.05 mm; (f) working fluid TEG-water, tube diameter 25.4 mm.

that, as the diameter of the thermosyphon increases, the thermal resistance decreases, since a bigger thermosyphon results in a larger heat transfer area.

From Figure 5a, it is found that for the high-temperature heat source, the thermal resistance increases again because of flooding inside the thermosyphon.

In the case of TEG-water mixtures (Figures 5d-5f), it is found that the thermal resistance of pure water is the lowest, since water can be vaporized more easily than TEG. The effects of diameter and temperature on the thermal resistance are similar to those of ethanol-water mixtures. Again, for 25% volume content of TEG in the mixture, the performance is close to that of pure water but flooding does not occur even when the thermosyphon is operated at high temperature and the pipe diameter is small.

Heat Transfer Models

The heat transfer model of heat transfer coefficient of pure working fluids has been correlated. The pool boiling equation of Rohsenow [5] is modified to predict the boiling heat transfer coefficient inside the thermosyphon. The equation is

$$h_e = \frac{C_1}{\Delta T_e A} \left[\left(\frac{\Delta T_e}{B} \right)^3 \right]^{C_2} \quad (3)$$

where

$$A = \frac{1}{\mu_1 \lambda} \sqrt{\frac{g_c \sigma}{g(\rho_l - \rho_v)}}$$

and

$$B = \frac{\lambda \text{Pr}_l}{C_{p_l}}$$

where h_e is the boiling heat transfer coefficient of the thermosyphon, ΔT_e is the temperature difference between the fluid inside and the surface at the evaporator section, g_c is a conversion factor (in SI units, g_c is equal to 1 kg m/Ns^2), and C_1 and C_2 are empirical constants. Rohsenow's equation is developed for pure pool boiling, but in a thermosyphon, parts of the liquid return from the condenser sections also evaporate.

The factors C_1 and C_2 of each pure substance in Eq. (3) modified for the thermosyphon can be found from our experiments and are shown in Table 1.

Table 1. Factors C_1 and C_2 from the experiments

Working fluid	C_1	C_2
Water	18.688	0.3572
Ethanol	17.625	0.3300
TEG	20.565	0.3662

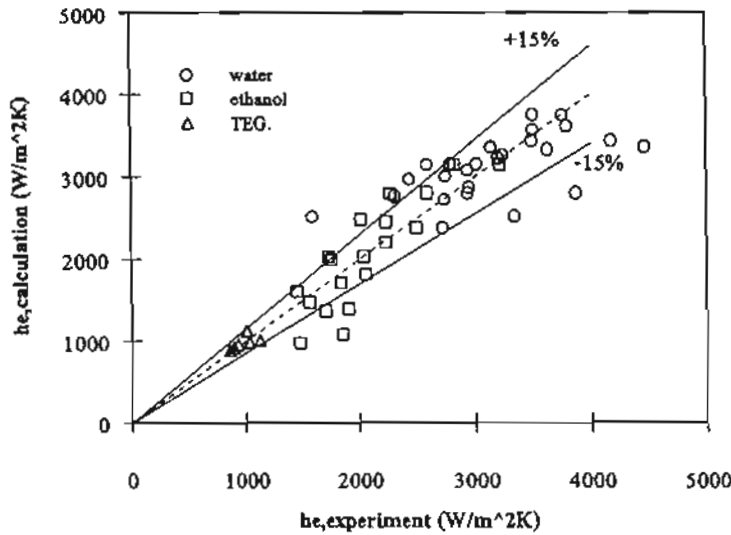


Figure 6. Comparisons of boiling heat transfer coefficients from the experiments and Eq. (3).

Figure 6 shows the results of the heat transfer coefficients of the pure working fluids, calculated from Eq. (3) and compared with those of the experimental data. Most of the calculated values are within $\pm 15\%$ of the experimental data.

At the condenser section, Nusselt's equation of condensation [5] is modified to predict the heat transfer coefficient of the condenser section. The equation is

$$h_c = C_3 \left(\frac{\rho_1^2 g \lambda k_1^3}{\mu_1 L_c \Delta T_c} \right)^{C_4} \quad (4)$$

where h_c is the condensation heat transfer coefficient, L_c is the length of condenser section, ΔT_c is the temperature difference between the fluid inside and the surface at the condenser section, and C_3 and C_4 are empirical constants.

With our experimental results, the constants C_3 and C_4 for the pure working fluids are shown in Table 2. Comparison of the calculated condensation heat transfer coefficients with those of the experiments are shown in Figure 7. It is found that the boiling and condensation heat transfer coefficients of pure water and pure ethanol are close to those of Shiraishi et al.'s experiment [6].

Table 2. Factors C_3 and C_4 from the experiments

Working fluid	C_3	C_4
Water	0.943	0.233
Ethanol	0.930	0.260
TEG	0.943	0.180

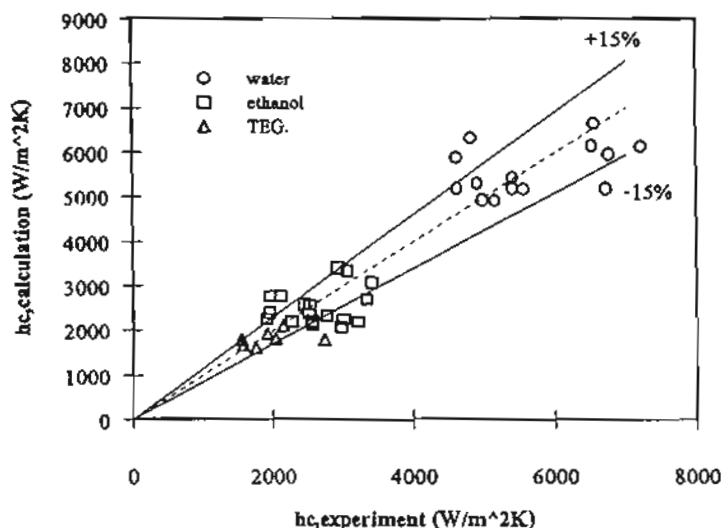


Figure 7. Comparisons of condensation heat transfer coefficients from the experiments and Eq. (4).

Heat Transfer Model of Binary Working Fluids

For a binary mixture, a method to evaluate the mixture heat transfer coefficient from each component [7] has been proposed as

$$h_m = x_1 h_1 + (1 - x_1) h_2 \quad (5)$$

where h_m is the total heat transfer coefficient of the mixture, h_1 and h_2 are the heat transfer coefficients of high- and low-volatile components, respectively, and x is the mole fraction of the high-volatile component.

Normally the heat transfer coefficient of the binary mixture calculated from Eq. (5) could be predicted precisely for very low working pressure (ideal case) [7–10]. In the case of the experimental thermosyphon, the partial pressure of each working fluid inside the pipe is found to be very low, thus Eq. (5) can be used to predict the heat transfer coefficient very well. The results are shown in Figures 8 and 9 for boiling and in Figures 10 and 11 for condensation.

The composition of each component in liquid and vapor phases can be evaluated by Raoult's law and Dalton's law [11], and from the heat transfer coefficient of each pure component the heat transfer coefficient of the binary mixtures can be evaluated. The comparison of the heat transfer coefficients from the experiments and those from Eq. (5) are shown in Figures 8–11 for boiling and condensation at the evaporator and the condenser, respectively. It is found that the weighted-average method can be used to predict both boiling and condensation heat transfer coefficient within $\pm 15\%$ of the experimental values. Moreover, it is found that the heat transfer coefficients of those mixtures are between those of the pure working fluids.

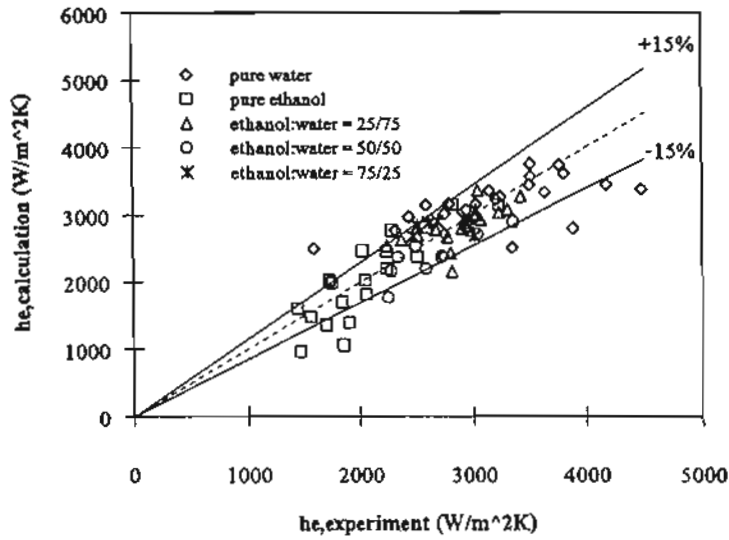


Figure 8. Comparisons of boiling heat transfer coefficients of ethanol-water mixtures from the experiments and Eq. (5).

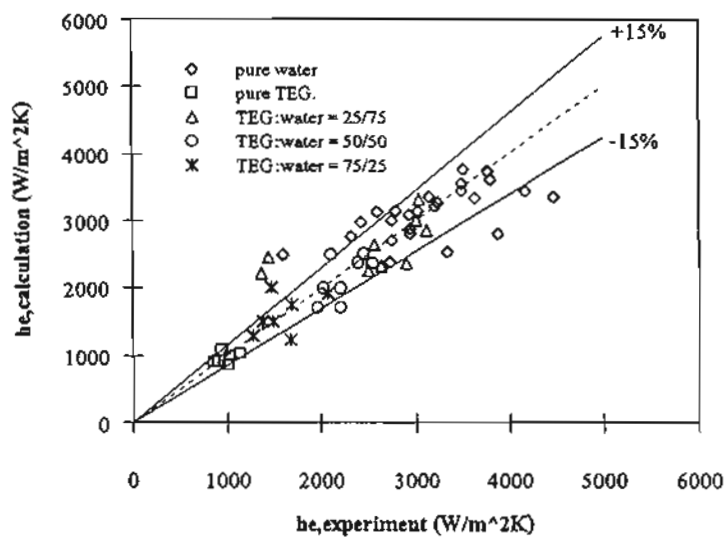


Figure 9. Comparisons of boiling heat transfer coefficients of TEG-water mixtures from the experiments and Eq. (5).

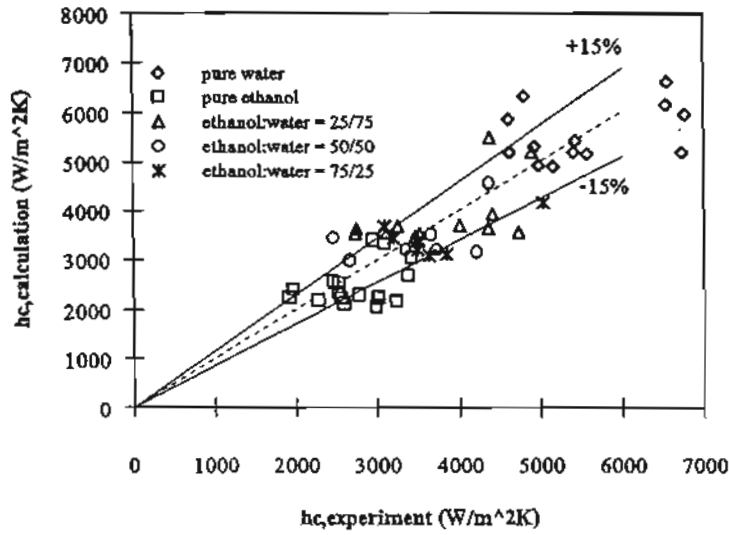


Figure 10. Comparisons of condensation heat transfer coefficients of ethanol-water mixtures from the experiments and Eq. (5).

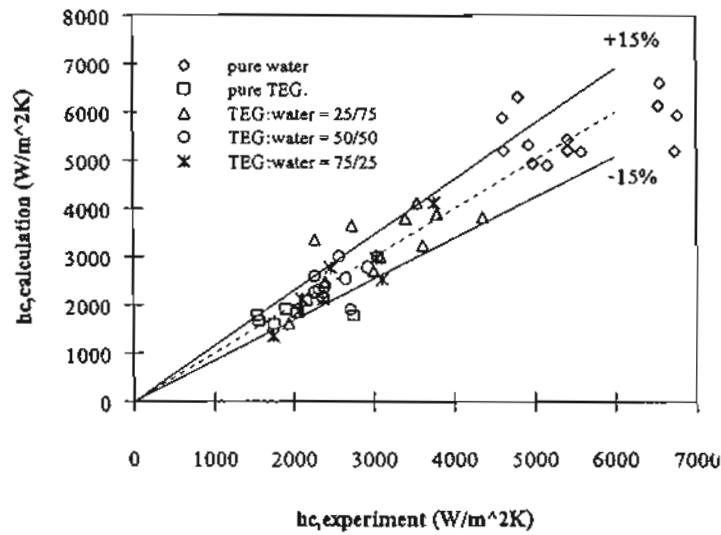


Figure 11. Comparisons of condensation heat transfer coefficients of TEG-water mixtures from the experiments and Eq. (5).

Flooding Limit of the Thermosyphon

Flooding always occurs in small thermosyphons with high temperature. Faghri's equation is normally used to calculate the critical heat flux due to the flooding limit [1] and is of the form

$$Q_{\max} = K \lambda A_{cs} [g \sigma (\rho_l - \rho_v)]^{-0.25} (\rho_v^{-0.25} + \rho_l^{-0.25})^{-2} \quad (6)$$

where

$$K = \left(\frac{\rho_l}{\rho_v} \right)^{0.14} \tanh^2 \text{Bo}^{0.25}$$

and

$$\text{Bo} = D_i \left[\frac{g (\rho_l - \rho_v)}{\sigma} \right]^{0.5}$$

where A_{cs} is the cross-sectional area of the thermosyphon, D_i is the inside diameter of the pipe, and Bo is the Bond number.

Faghri's equation has been applied for single working fluids. In this research work, for a binary mixture, ethanol-water, the properties of the mixture [12-17] have been used and it could be found that the prediction of the maximum heat transfer evaluated from the equation could be applied quite well (Figure 12).

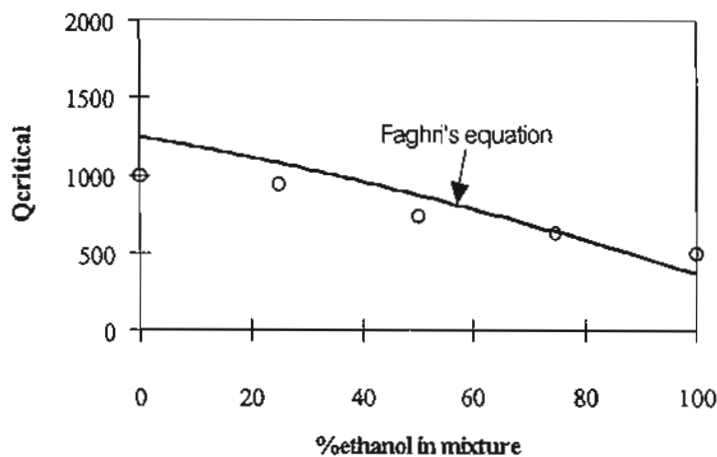


Figure 12. Prediction of the maximum heat transfer rate due to the flooding limit. The binary mixture is ethanol-water.

CONCLUSION

There is a high potential to improve performance of a thermosyphon heat pipe by using binary mixtures. Ethanol-water mixtures give a higher heat transfer rate than water at the low-temperature heat source (less than 80°C). In the case of TEG-water mixtures, the heat transfer rate of the thermosyphon decreases with the content of TEG in the mixture and it is found that a small amount of TEG in the mixture (25%) can enlarge the critical heat flux due to the flooding limit of the small size of the thermosyphon while the heat transfer rate is reduced slightly.

The boiling equation of Rohsenow and the condensation equation of Nusselt can be used to predict the heat transfer coefficients of boiling and condensation inside the thermosyphon. In the case of binary mixtures, the weighted average of the heat transfer coefficient of each component can be used to predict the total heat transfer coefficient. At the critical limit for flooding of the thermosyphon, it is found that Faghri's equation can be applied to calculate the maximum heat transfers of both pure water and ethanol-water mixtures.

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Performance Improvement of Thermosyphon Heat Exchangers by Using Two Kinds of Working Fluids

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ABSTRACT

In this study, the concept of introducing two-fluid thermosyphons is examined. Calculations were performed for both low and high temperature ranges with parallel and counter flow arrangements. For lower temperature application, $125\text{ }^{\circ}\text{C} > T_{hi} > 75\text{ }^{\circ}\text{C}$, use of ammonia in some rows and water in the rest of the thermosyphon can slightly improve the associated heat transfer performance for balanced counter flow arrangement. However, for balanced parallel flow arrangement at both low and high temperature applications, the concept of using two-fluid thermosyphons may not be feasible. The use of two-fluid thermosyphons is especially advantageous for high temperature application. For instance, in the range of $375\text{ }^{\circ}\text{C} > T_{hi} > 350\text{ }^{\circ}\text{C}$, the two-fluid thermosyphons (dowtherm A- water) shows a 15~99% increase of heat transfer performance relative to dowtherm A alone.

Keywords: Thermosyphon, two-fluid, heat transfer augmentation

INTRODUCTION

The thermosyphon is the heat exchanger that carries heat from the higher temperature heat source to the lower temperature heat sink via boiling and condensation of the working fluid inside the tube. Advantages of the thermosyphon heat exchangers

include high thermal conductivity, low cost, and easy construction. The thermosyphon heat exchanger shows potential in energy recovery from industrial waste heat and has become an interesting topic for energy conservation.

The thermosyphon heat pipe shown in Figure 1a can be divided into three parts, namely the evaporator, the adiabatic, and the condenser section. When heat is added to the evaporator section, the working fluid inside the thermosyphon is vaporized. The vapor carries heat from the heat source, flows to the condenser section and rejects heat to the heat sink. The condensate of the working fluid then returns to the evaporator section by gravity.

In practical application, multi-row thermosyphon heat exchangers filled with one kind of working fluid are employed. For applications involved with a medium temperature heat source (40-350 °C), water is known as the best working fluid because of its high latent heat and low investment cost [1]. But it should be noticed that for temperature in the range of 40-100 °C, water may not be the best choice for its comparatively lower heat transfer coefficient than other work fluids (e.g. ammonia). In addition, for other applications having heat source higher than 350 °C, water thermosyphon may be inappropriate due to possible occurrence of critical heat flux. Hence, to enlarge the application scope, we propose to use two kinds of substances in the multi-row thermosyphon heat exchangers application. For low temperature application ($T_{ht} < 100$ °C), combination of ammonia and water heat pipes may be applicable. For application in the higher temperature range ($T_{ht} > 300$ °C), it is likely that the water thermosyphon may fail due to the occurrence of local dry out and

flooding. To avoid this condition, it is possible to apply thermosyphons using downtherm A in several initial rows to reduce the hot-gas temperature and then use water thermosyphons in the rest of tube row to effectively increase the heat transfer performance. It is the main objective of this study to present the feasibility of this two-fluid thermosyphons implementation.

HEAT TRANSFER ANALYSIS

Figure 1b shows the thermal resistances of the thermosyphon heat exchangers. In the present modeling, the pressure drops from the evaporator to the condenser end and the axial conduction along the pipe wall are assumed negligible. Detailed evaluations of the thermal resistance are described as follows.

The external air-side thermal resistance of the evaporator (Z_{eo}) and the condenser (Z_{co}) section can be calculated from

$$Z_{eo} = \frac{1}{h_{eo} A_{eo}} \quad (1)$$

$$Z_{co} = \frac{1}{h_{co} A_{co}} \quad (2)$$

where h_{eo} and h_{co} denote the external air-side heat transfer coefficient in the evaporator and condenser, respectively. For circular finned tube having staggered array, Hewitt et al. [2] proposed the following correlation:

$$\overline{Nu} = 0.242 \text{Re}^{0.658} \left(\frac{f_s}{f_h} \right)^{0.297} \left(\frac{S_t}{S_l} \right)^{-0.091} \text{Pr}^{0.333} \quad (3)$$

where S_t and S_l are the transverse and longitudinal pitch of the staggered array. Related information for calculating the fin surface area A_f , bare tube surface A_b , and fin efficiency η_f are given as:

$$A_f = \frac{nL\pi}{f_s + f_l} (0.5(D_f^2 - D_r^2) + D_f f_l + D_r f_s) \quad (4)$$

$$A_b = \frac{nL\pi}{f_s + f_l} (D_r f_s) \quad (5)$$

$$\eta_f = \frac{\tanh(\sqrt{2h_o / f_l k_f} \times \varphi)}{\sqrt{2h_o / f_l k_f} \times \varphi} \quad (6)$$

$$\varphi = \frac{D_r}{2} \left(\frac{D_f}{D_r} - 1 \right) \left(1 + 0.35 \ln \frac{D_f}{D_r} \right) \quad (7)$$

Note that the fin efficiency η_f is computed from Schmidt's approximation [3].

The wall resistance in the evaporator and condenser can be calculated from Equations (8) and (9), respectively.

$$Z_{ei} = \frac{\ln(r_o / r_i)}{2\pi k_i L_e} \quad (8)$$

$$Z_{ci} = \frac{\ln(r_o / r_i)}{2\pi k_i L_c} \quad (9)$$

The related in-tube resistances in the evaporator (Z_{ei}) and condenser (Z_{ci}) can be calculated from:

$$Z_{ei} = \frac{1}{h_{ei} A_{ei}} \quad (10)$$

$$Z_{ci} = \frac{1}{h_{ci} A_{ci}} \quad (11)$$

Notice that the internal resistance is around 3-5% of the outside resistance. For applications of the thermosyphon, there are many correlations for evaluation of the in-tube heat transfer coefficients for boiling and condensation. Depending on the flow condition, heat transfer mechanism in the evaporator may involve pool boiling or falling film evaporation. Therefore it is necessary to select the suitable correlation in association with heat transfer mechanism. Relevant correlations are described as follows: For lower temperature application ($T_s < 80$ °C), the Rohsenow [4] correlation for pool boiling is used:

$$\left[\frac{Cp_l \Delta T_e}{\lambda} \right] = C_{sf} \left[\frac{q}{\mu_l \lambda} \left(\frac{\sigma}{g(\rho_l - \rho_v)} \right)^{1/2} \right]^{C_1} \left[\frac{Cp_l \mu_l}{k_l} \right]^{C_2+1} \quad (12)$$

where $C_{sf} = 0.013$, $C_1 = 0.33$, and $C_2 = 0$ for water and 0.7 for another fluids.

In addition, in the case of falling film evaporation in the evaporator section for lower temperature, Nusselt's equation [5] can be also used for calculation [6], i.e.,

$$h_{e,f} = 0.943 \left[\frac{g \rho_l (\rho_l - \rho_v) \lambda k_l^3}{\mu_l (T_w - T_s) L_{e,f}} \right]^{1/4} \quad (13)$$

where $h_{e,f}$ is the evaporation heat transfer coefficient of the falling film.

The heat transfer coefficient in the condenser can be calculated from Nusselt's theoretical results [5]:

$$h_{c,i} = 0.943 \left[\frac{g \rho_l (\rho_l - \rho_v) \lambda k_l^3}{\mu_l (T_w - T_s) L_c} \right]^{1/4} \quad (14)$$

For application of water at higher temperature ($T_s \geq 80$ °C), the correlation for boiling and condensation developed by Kiatsiriroat et al. [7] is adopted, i.e.,

Boiling heat transfer coefficient of water:

$$h_e = \frac{18.688}{\Delta T_e X} \left[\left(\frac{\Delta T_e}{Y} \right)^3 \right]^{0.3572}, \quad (15)$$

where $X = \frac{1}{\mu_l \lambda} \sqrt{\frac{g_c \sigma}{g(\rho_l - \rho_v)}}$ and $Y = \frac{\lambda \text{Pr}_l}{Cp_l}$.

Condensation heat transfer coefficient for water;

$$h_{ci} = 0.943 \left[\frac{g \rho_l^2 \lambda k_l^3}{\mu_l (T_w - T_s) L_c} \right]^{0.233} \quad (16)$$

For high temperature application, dowtherm A is regarded as the working medium in this study. ESDU [8,9] proposed several correlations for both evaporator and condenser that is also applicable for downterm A, and is given as below:

Thermal resistance due to pool boiling in the evaporator section:

$$Z_{ei,p} = \frac{1}{\phi_1 g^{0.25} Q^{0.4} (\pi D_i L_e)^{0.6}} \quad (17)$$

$$\phi_1 = 0.32 \frac{\rho_l^{0.65} k_l^{0.3} Cp_l^{0.7}}{\rho_v^{0.25} \lambda^{0.4} \mu_l^{0.1}} \left[\frac{P_v}{P_a} \right]^{0.23} \quad (18)$$

Thermal resistance due to falling film evaporation in the evaporator section:

$$Z_{ei,f} = \frac{0.235 Q^{1/3}}{D_i^{4/3} g^{1/3} L_e \phi_2^{4/3}} \quad (19)$$

$$\phi_2 = \left(\frac{\lambda k_i^3 \rho_i^2}{\mu_i} \right)^{1/4} \quad (20)$$

The total thermal resistance due to boiling in the evaporation can be calculated

as:

$$Z_{ei} = Z_{ei,f}(1 - F) + Z_{ei,p}F \quad (21)$$

where F is filling ratio of the working fluid.

Thermal resistance in the condenser:

$$Z_{ci} = \frac{0.235Q^{1/3}}{D_i^{4/3}g^{1/3}L_c\phi_2^{4/3}} \quad (22)$$

The overall heat transfer resistance of the thermosyphon heat exchanger is then the summation from separate resistance described above,

$$\frac{1}{(UA)_{total}} = \frac{1}{h_{eo}A_{eo,total}} + \frac{\ln(r_o/r_i)}{2\pi k_i L_{e,total}} + \frac{1}{h_{ei}A_{ei,total}} + \frac{1}{h_{co}A_{co,total}} + \frac{\ln(r_o/r_i)}{2\pi k_i L_{c,total}} + \frac{1}{h_{ci}A_{ci,total}} \quad (23)$$

Note that the properties to evaluate the inside heat transfer coefficients are based on the saturated temperature inside the thermosyphon [8], i.e.

$$T_{inside} = T_{hi} - \frac{Z_{ci} + Z_{ci} + Z_{co}}{Z_{total}}(T_{hi} - T_{ci}) \quad (24)$$

Figures 2(a) and 2(b) show the schematic of the arrangement of the thermosyphon heat exchanger considered in the calculations. Both counter and parallel flow arrangement is investigated. The sample heat exchanger consists of seven rows of heat pipe. The overall heat transfer rate of the sample heat exchangers can be calculated as

$$\begin{aligned}
Q_{total} &= \sum_{i=1}^7 Q = Q_1 + Q_2 + Q_3 + Q_4 + Q_5 + Q_6 + Q_7 \\
&= (UA)_{total} (LMTD) \\
&= \dot{m}_h C p_h (T_{hi} - T_{ho}) \\
&= \dot{m}_c C p_c (T_{co} - T_{ci})
\end{aligned} \tag{25}$$

where *LMTD* is log mean temperature difference.

SIMULATION PROGRAM

For better understanding of applicable temperature range of related applications, different simulation algorithms were adopted in the implementation of the thermosyphon heat exchanger. In the first part, performance of the thermosyphon heat exchanger at low temperature range is investigated ($T_{hi} < 100$ °C). In this part, the working fluids in the thermosyphons are ammonia and water. As mentioned in the introduction section, the thermosyphon heat exchanger that uses water in this range may possess a poor heat transfer characteristics. Therefore, it would be beneficial to use the thermosyphons that containing ammonia in some rows of the heat exchanger. This is because of better heat transfer performance of ammonia than water at this temperature range. The simulation program is capable of selecting the suitable working fluid for each row between water and ammonia, and of calculating the overall performance of the heat exchanger. Detailed flow chart for low temperature application can be seen from Figure 3.

As seen in Figure 3, both counter and parallel flow arrangements are implemented. For the counter flow arrangement at the starting stage of iteration, temperatures of the inlet hot and cold streams are prescribed and the inlet and outlet

temperatures of the fluids at each row are assumed by considering all rows to be water. Iteration then starts by setting ammonia to the first row. Comparison is made with original water thermosyphon. Ammonia is selected if the related heat transfer performance outperforms the original water thermosyphon. The selection process continues from the second row to the last row. In case of the parallel flow, similar process has been carried out from the first row to the last row. As a result, appropriate fluid in every row of the thermosyphon heat exchanger is obtained.

In the second part of higher temperature application ($T_{hi} > 300\text{ }^{\circ}\text{C}$), water and dowsin A were used as working fluids. The computer programs were constructed to calculate the performance of the heat exchanger and also to find the suitable working fluid from row to row of the thermosyphon heat exchanger subjected to the constraint of critical heat flux of water thermosyphon. The basic rule of thumb for water is the temperature limit of interior saturation temperature should be less than $300\text{ }^{\circ}\text{C}$ (the highest recommend temperature for water [1]). Beyond this temperature, some critical conditions such as flooding or dry out may occur [1,8,10,11]. These critical conditions not only considerably deteriorate the performance of the heat exchanger but also may damage the thermosyphon as well. In this connection, for higher temperature application, dowsin A may be chosen as an alternative. However, its heat transfer performance is relatively low when comparing with water [8,9]. For optimization the system performance, it is necessary to make a combination of two-fluid thermosyphons. Figure 4 shows the related flow chart for the high temperature application. Relevant conditions for the heat exchanger are tabulated in Table 1.

The procedure in Figure 4 is similar to that in Figure 3 but the selection of the working fluid in each row has been carried out subjected to critical temperature constraint. When the inside temperature is lower than 300°C, water should be used as a working fluid and when the inside temperature is over 300°C, dowtherm A is chosen instead.

RESULTS AND DISCUSSION

Part 1 - Results of Low Temperature

Table 2 shows calculated results from the simulation program in the case of the balanced counter flow arrangement ($\dot{m}_h = \dot{m}_c$) at various working conditions. Calculations were performed at a fixed T_{cl} of 20 °C. Range of T_{hi} is from 75 °C to 125 °C with a mass flow rate of 0.1 ~ 0.4 kg/s. As can be seen from Table 2, one can see that all-water thermosyphons would be the best choice for $T_{hi} = 125$ °C while all-ammonia thermosyphons show the highest performance for $T_{hi} < 75$ °C. For 75 °C $< T_{hi} < 125$ °C, it can be shown that use of both kinds of thermosyphons would be advantageous. However, one can see that only a slight increase of performance is observed. This is because that the heat transfer coefficient for water and ammonia are comparable in this range. It should be pointed out that water thermosyphons should be placed at the hot air inlet consecutively in order to get the best performance. For detailed evaluations the appropriate number of water thermosyphons of various T_{hi} , one can see Figure 5. As seen in the figure, the appropriate number of the number of tube row of water thermosyphons increases with T_{hi} and $\dot{m}$. For an application condition of $\dot{m} < 0.1$ kg/s

and $T_{hi} < 90^{\circ}\text{C}$, performance improvement by use of water thermosyphons is not attainable. Meanwhile, for an application of $\dot{m} \geq 0.4 \text{ kg/s}$ and $T_{hi} > 90^{\circ}\text{C}$, it would be better to use all-water thermosyphons to achieve optimal performance.

In the case of thermosyphon heat exchanger with balanced parallel flow arrangement ($\dot{m}_h = \dot{m}_c$), the calculation indicates that use of ammonia and water is not beneficial throughout the range when comparing to water or ammonia thermosyphon alone. Explanation of this phenomenon can be seen from Figure 6. As depicted from Figure 6, in the case of balanced parallel flow, the temperature profiles of hot and cold stream are quite symmetric and lead to nearly uniform inside temperature of the thermosyphons from the first to the last row. Therefore, the inside resistance for this situation is almost fixed from row to row. Therefore, use of two kinds of thermosyphon is not so beneficial when comparing to those of counter flow arrangement.

Unlike those of balanced parallel flow arrangement, for the case of unbalanced parallel flow thermosyphon heat exchanger ($\dot{m}_h \neq \dot{m}_c$), the temperature profiles of hot and cold stream are not symmetric. The concept of employing the two-fluid thermosyphons is still feasible. Table 3 and Figure 7 shows the calculated results of $\dot{m}_h = 0.2 \text{ kg/s}$ with $\dot{m}_c = 0.1, 0.2$, and 0.4 kg/s , respectively. Analogous to the results of counter-flow arrangements, the calculated results show a detectable influence of the inlet temperature and mass flow. However, it should be pointed out that the applicable temperature range of using two-fluid thermosyphons for unbalanced parallel flow is comparatively small than those encountered in counter flow arrangement. For instance, the applicable range for the un-balanced parallel flow arrangement shown in Figure 7 is

$80\text{ }^{\circ}\text{C} < T_{hi} < 110\text{ }^{\circ}\text{C}$ relative to $75\text{ }^{\circ}\text{C} < T_{hi} < 125\text{ }^{\circ}\text{C}$ for the balanced counter flow arrangement. The results are not surprising due to change of inner temperature of the thermosyphon is more pronounced for counter flow arrangement. The concept of using two-fluid thermosyphon is especially advantageous for large variation of temperatures.

Part 2 – Results of High Temperature

Results of the high temperature application using dowtherm A and water for balanced counter current flow thermosyphon heat exchanger are tabulated in Table 4. The inlet temperature of cold gas is $200\text{ }^{\circ}\text{C}$ and $250\text{ }^{\circ}\text{C}$, respectively. The calculated results indicate that use of two-fluid thermosyphon is much more beneficial than those in low temperature application. Typical increase of heat transfer rate for water-dowtherm A thermosyphons relative to dowtherm A thermosyphon alone is from 15 ~ 99%. However, use of water thermosyphon is subjected to the constraint that the interior saturation temperature of water should be less than $300\text{ }^{\circ}\text{C}$. Therefore, in the case of water- dowtherm A, the heat transfer rate tends to decrease for increase of T_{hi} . In addition, one can experience a further drop of heat transfer performance when T_{ci} is raised to 250°C at $\dot{m} = 0.3$ or 0.4 kg/s . These phenomena come from the effect of the number of tube row that uses water as a working fluid. At higher inlet temperature of hot air, the number of tube row containing water thermosyphons are eventually decreased while the number of tube row for dowtherm A thermosyphons is increased. Figure 8 shows the variation of saturated temperature inside each row of thermosyphon vs. tube row at various conditions. As seen, when the interior saturation temperature

drops below the constraint temperature of 300°C, it would be beneficial to employ the water thermosyphons in these tubes. Because of the considerable improvement of the heat transfer performance of water thermosyphon, the associated heat transfer rate can be increased dramatically. However, it should be noted that for $T_{hi} > 375^\circ\text{C}$, no water thermosyphons can be inserted. In summary, for the counter flow arrangement, the applicable range of the two-fluid thermosyphons concept is limited to $T_{hi} = 325 \sim 375^\circ\text{C}$.

Similar results for the parallel flow thermosyphon heat exchanger using water-dowtherm A as working fluids are tabulated in Table 5. Figures 9 and 10 shows the change of interior saturation temperature vs. the number of tube row for balanced and un-balanced parallel flow arrangement. For balanced parallel flow arrangement shown in Figure. 9, the concept of two-fluid thermosyphon is not feasible. Explanation is identical to those of low temperature application due to the constant temperature situation. Thus for balanced parallel flow arrangement, if the inside saturation temperature is higher than 300 °C, all- dowtherm A thermosyphons may be used. On the other hand, if the inside temperature is lower than 300 °C, all-water thermosyphon would be more suitable. However, in the case of un-balanced parallel flow arrangement, as seen in Figure 10, the temperature profile of hot and cold stream are not symmetric, thus the inside temperature of each row can vary from row to row. Thus the concept of two-fluid thermosyphons is applicable. From the temperature distribution of Figure 10, one can see that it would be very beneficial to insert some rows of water thermosyphons into the heat exchanger (especially in row 5, 6, and 7).

CONCLUSION

In this study, the concept of using two-fluid thermosyphons is implemented for examining the performance improvement of thermosyphon heat exchangers. Calculations were made for both low temperature and high temperature application. Major results are summarized as follows:

- (1) For lower temperature application and $75\text{ }^{\circ}\text{C} < T_{hi} < 125\text{ }^{\circ}\text{C}$, use of ammonia in some rows can slightly improve the associated heat transfer performance for balanced counter flow arrangement.
- (2) For balanced parallel flow arrangement at low and high temperature application, the concept of using two-fluid thermosyphons may not be feasible.
- (3) For un-balanced counter flow arrangement, the concept of using two-fluid thermosyphons is also applicable. However, the associated range of T_{hi} is comparatively smaller than those of counter flow arrangement.
- (4) For un-balanced parallel flow arrangement, the concept of using two-fluid thermosyphons is applicable.
- (5) For higher temperature application, benefits of using two-fluid thermosyphons are much more pronounced than those in lower temperature application. However, the related applicable range of T_{hi} may be smaller than those in high temperature application due to the constraint of critical heat flux of water.

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NOMENCLATURE

A	area, m^2 (ft^2)
C_{1-2}	empirical constants
C_p	specific heat, $J/kg^\circ C$ ($Btu/lb_m^\circ F$)
C_{sf}	liquid-surface constant
D	diameter of tube, m (ft)
f_h	fin height, m (ft)
f_s	fin gap, m (ft)
f_t	fin thickness, m (ft)
F	filling ratio of working fluid
g	gravitational acceleration, $9.81 m/s^2$ (ft/s^2)
g_c	conversion factor, $1 kg m/N s^2$ ($32.17 lbm \cdot ft/lbf sec^2$)
h	heat transfer coefficient, $W/m^2^\circ C$ ($Btu/h ft^2^\circ F$)
k	thermal conductivity, $W/m^\circ C$ ($Btu/h ft^\circ F$)
L	length, m (ft)
$LMTD$	log mean temperature difference, $^\circ C$ ($^\circ F$)
$\dot{m}$	mass flow rate, kg/s (lb_m/s)
n	number of tube
Nu	Nusselt number
P	pressure, Pa (lb_f/in^2)
Pr	Prandtl number
q	heat flux, W/m^2 ($Btu/h ft^2$)
$\dot{Q}$	heat transfer rate, W (Btu/h)
r	radius of tube, m (ft)
Re	Reynolds number
S_d	diagonal pitch of staggered Array, m (ft)
S_l	longitudinal pitch of staggered array, m (ft)
S_t	transverse pitch of staggered Array, m (ft)
T_{ci}	inlet temperature of cold gas, $^\circ C$ ($^\circ F$)
T_{co}	outlet temperature of cold gas, $^\circ C$ ($^\circ F$)
T_{hi}	inlet temperature of hot gas, $^\circ C$ ($^\circ F$)

T_{ho}	outlet temperature of hot gas, °C (°F)
ΔT_c	temperature difference between inside surface and liquid at condenser section, °C (°F)
ΔT_e	temperature difference between inside surface and liquid at evaporator section, °C (°F)
UA	overall heat transfer coefficient-area, W/°C (Btu/h °F)
Z	thermal resistance, °C/W (°F h/Btu)

Greek symbols

ϕ	function
η	efficiency
φ	function
λ	latent heat of vaporization, J/kg (Btu/lb _m)
μ	dynamic viscosity, Pa s (lb _m /ft s)
ρ	density, kg/m ³ (lb _m /ft ³)
σ	surface tension, N/m (lb _f /ft)

Subscripts

a	air, atmospheric
b	bare tube
c	condenser, cold
e	evaporator
f	fin, film
h	hot
i	inlet, inside
l	liquid
o	outlet, outside
r	root
s	saturated
t	tube
w	wall
v	vapor

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Captions of Tables and Figures

Table 1 Testing conditions and related geometrical parameters of the thermosyphon heat exchanger.

Table 2 Results of simulation in the case of balanced counter flow arrangement for low temperature application; working fluid: ammonia (NH₃) – water (H₂O).

Table 3 Results of simulation in the case of unbalanced parallel flow arrangement for low temperature application; working fluid: ammonia (NH₃) – water (H₂O), $\dot{m}_h = 0.2$ kg/s.

Table 4 Results of simulation in the case of balanced counter flow arrangement for high temperature application; working fluid: water- dOWtherm A.

Table 5 Results of simulation in the case of unbalanced parallel flow arrangement for high temperature application; working fluid: water- dOWtherm A.

Figure 1 Schematic of the thermosyphon heat pipe - (a). Operating principle; (b). Thermal resistance circuit.

Figure 2 Flow arrangement of the thermosyphon heat exchanger.

Figure 3 The flow chart of the two-fluid thermosyphons in low temperature application.

Figure 4 The flow chart of the two-fluid thermosyphons in high temperature application.

Figure 5 Relation of the appropriate number of water thermosyphons vs. inlet conditions for balanced counter flow arrangement at $T_{ci} = 20\text{ }^{\circ}\text{C}$.

Figure 6 Temperature profiles of the balanced parallel flow thermosyphon heat exchanger using water and ammonia as working fluids.

Figure 7 Relation of the appropriate number of water thermosyphons vs. inlet conditions for unbalanced parallel flow arrangement at $T_{ci} = 20\text{ }^{\circ}\text{C}$ and $\dot{m}_h = 0.2\text{ kg/s}$.

Figure 8 Interior saturation temperatures of each row of counter flow heat exchanger at various conditions; mass flow rate of air = 0.3 kg/s .

Figure 9 Temperature profile of the balanced parallel flow heat exchanger using water and dowerm A as working fluids.

Figure 10 Temperature profile of the unbalanced parallel flow heat exchanger using water and dowerm A as working fluids.

Table 1 Testing conditions and related geometrical parameters of the thermosyphon heat exchanger.

Items	Conditions
1. Flow arrangement	Parallel and counter flow
2. Temperature of hot air	50-125 °C for low temperature
3. Temperature of cold air	325-400 °C for high temperature
4. Mass flow rate of air	20 °C for low temperature
5. Thermosyphon arrangement	200-250 °C for high temperature
6. Number of tube row	0.05-0.4 kg/s
7. Number of column	Staggered array
8. Diameter of thermosyphon (bare tube)	$S_t = 0.053$ m, $S_d = 0.053$ m, $S_l = 0.046$ m
9. Type of fin	7
10. Size of fin	7
11. Filling ratio of working fluid	0.027 m
12. Material of pipe and fin	Circular fin
	Fin height = 0.1 m and fin pitch = 10
	fins/inch
	50%
	Stainless steel 304

Table 2 Results of simulation in the case of balanced counter flow arrangement for low temperature application; working fluid : ammonia (NH_3) – water (H_2O).

T_{cl} (°C)	T_{hl} (°C)	$\dot{m}$ (kg/s)	Q_{H_2O} (W)	Q_{NH_3} (W)	$Q_{H_2O-NH_3}$ (W)	No. of tube row		% increase	
						H_2O	NH_3	Relative to H_2O	Relative to NH_3
20	125	0.1	5155.09	4872.82	5155.09	7	0	0.00	5.79
20	125	0.2	7456.00	6779.58	7456.00	7	0	0.00	9.98
20	125	0.3	8882.18	7850.84	8882.18	7	0	0.00	13.14
20	125	0.4	9865.96	8553.55	9865.96	7	0	0.00	15.34
20	100	0.1	3761.03	3778.79	3826.01	3	4	1.73	1.25
20	100	0.2	5467.49	5319.22	5488.06	5	2	0.38	3.17
20	100	0.3	6506.54	6201.61	6506.54	7	0	0.00	4.92
20	100	0.4	7223.34	6787.23	7223.34	7	0	0.00	6.43
20	95	0.1	3491.61	3552.24	3575.62	2	5	2.41	0.66
20	95	0.2	5071.65	5012.1	5113.46	4	3	0.82	2.02
20	95	0.3	6033.43	5851.68	6043.20	6	1	0.16	3.27
20	95	0.4	6696.93	6410.27	6696.93	7	0	0.00	4.47
20	90	0.1	3222.87	3323.42	3330.90	2	5	3.35	0.23
20	90	0.2	4676.74	4700.36	4749.82	3	4	1.56	1.05
20	90	0.3	5561.43	5495.50	5596.36	4	3	0.63	1.84
20	90	0.4	6171.70	6025.85	6182.09	5	2	0.17	2.59
20	85	0.1	2954.92	3092.44	3092.44	0	7	4.65	0.00
20	85	0.2	4283.02	4384.14	4398.77	2	5	2.70	0.33
20	85	0.3	5090.79	5133.17	5170.42	3	4	1.56	0.73
20	85	0.4	5647.95	5634.03	5696.8	4	3	0.86	1.11
20	80	0.1	2687.93	2859.42	2859.42	0	7	6.38	0.00
20	80	0.2	3890.70	4063.53	4063.53	0	7	4.44	0.00
20	80	0.3	4621.82	4764.77	4768.35	1	6	3.17	0.08
20	80	0.4	5126.02	5234.94	5243.41	2	5	2.29	0.16
20	75	0.1	2422.11	2624.39	2624.39	0	7	8.35	0.00
20	75	0.2	3500.14	3738.65	3738.65	0	7	6.81	0.00
20	75	0.3	4154.91	4390.43	4390.43	0	7	5.67	0.00
20	75	0.4	4606.35	4828.61	4828.61	0	7	4.83	0.00

Table 3 Results of simulation in the case of unbalanced parallel flow arrangement for low temperature application; working fluid: ammonia (NH_3) – water (H_2O), $\dot{m}_h = 0.2 \text{ kg/s}$.

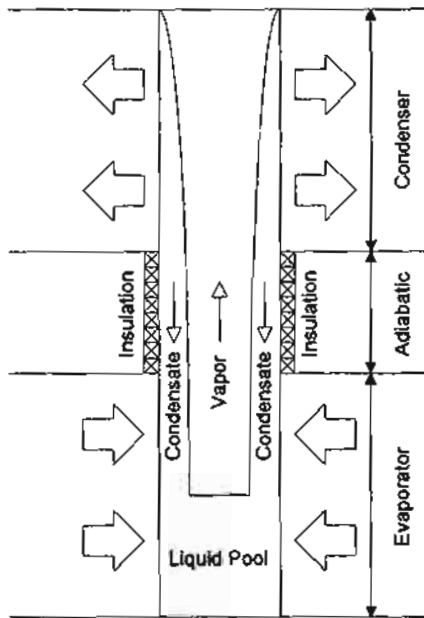
T_c (°C)	T_h (°C)	$\dot{m}_c$ (kg/s)	$Q_{\text{H}_2\text{O}}$ (W)	Q_{NH_3} (W)	$Q_{\text{H}_2\text{O}-\text{NH}_3}$ (W)	No. of tube row		% increase	
						H_2O	NH_3	Relative to H_2O	Relative to NH_3
20	110	0.1	4587.09	4444.08	4587.09	7	0	0.00	3.22
20	110	0.3	6500.33	6219.93	6500.33	7	0	0.00	4.51
20	110	0.4	6859.51	6559.15	6859.51	7	0	0.00	4.58
20	100	0.1	4016.58	3976.97	4033.43	5	2	0.42	1.42
20	100	0.3	5676.76	5582.6	5705.28	5	2	0.50	2.20
20	100	0.4	5988.52	5889.68	6024.55	5	2	0.60	2.29
20	90	0.1	3447.36	3499.02	3508.11	2	5	1.76	0.26
20	90	0.3	4856.82	4927.58	4957.25	3	4	2.07	0.60
20	90	0.4	5121.66	5201.31	5236.26	3	4	2.24	0.67
20	80	0.1	2880.37	3011.31	3011.31	0	7	4.55	0.00
20	80	0.3	4042.06	4255.73	4255.73	0	7	5.29	0.00
20	80	0.4	4260.53	4494.78	4494.78	0	7	5.50	0.00

Table 4 Results of simulation in the case of balanced counter flow arrangement for high temperature application; working fluid: water-dowtherm A.

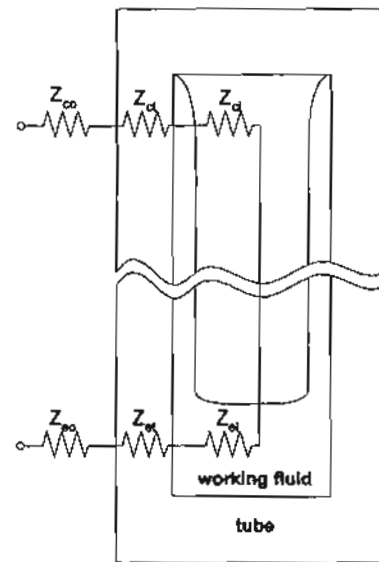
T_{ci} (°C)	T_{hi} (°C)	$\dot{m}$ (kg/s)	$Q_{H_2O-dowthermA}$ (W)	$Q_{dowthermA}$ (W)	No. of tube row		% increase relative to dowtherm A
					H ₂ O	dowtherm A	
200	325	0.2	14,801.24	8,586.18	7	0	72.38
200	325	0.3	19,733.14	9,924.58	7	0	98.83
200	325	0.4	23,910.64	10,794.05	7	0	121.52
200	350	0.2	16,278.12	10,180.31	5	2	59.90
200	350	0.3	21,261.08	11,722.44	5	2	81.37
200	350	0.4	25,367.74	12,717.51	5	2	99.47
200	375	0.2	16,695.81	11,734.56	3	4	42.28
200	375	0.3	21,136.80	13,464.64	3	4	56.98
200	375	0.4	24,643.56	14,574.15	3	4	69.09
250	325	0.2	8,215.36	5,311.22	4	3	54.68
250	325	0.3	10,749.66	6,183.04	4	3	73.86
250	325	0.4	12,843.12	6,756.87	4	3	90.07
250	350	0.2	8,912.57	6,977.37	2	5	27.74
250	350	0.3	11,077.11	8,078.91	2	5	37.11
250	350	0.4	12,735.22	8,796.64	2	5	44.77
250	375	0.2	9,937.70	8,591.39	1	6	15.67
250	375	0.3	9,900.76	9,900.76	0	7	0.00
250	375	0.4	10,746.83	10,746.83	0	7	0.00

Table 5 Results of simulation in the case of unbalanced parallel flow arrangement for high temperature application; working fluid: water-dowtherm A.

T_{ci} (°C)	T_{hi} (°C)	$\dot{m}_h/\dot{m}_c$ (kg/s)	$Q_{H_2O-dowthermA}$ (W)	$Q_{dowthermA}$ (W)	No. of tube row		% increase relative to dowtherm A
					H ₂ O	dowtherm A	
250	375	0.1/0.3	9,581.41	7,255.79	7	0	32.05
250	375	0.15/0.3	12,172.92	8,349.21	6	1	45.80
250	375	0.17/0.3	11,597.18	8,639.07	3	4	34.24
250	375	0.2/0.3	9,070.19	9,070.19	0	7	0.00
250	400	0.05/0.3	6,752.08	6,036.20	7	0	11.86
250	400	0.1/0.3	11,404.50	8,654.23	6	1	31.78
250	400	0.12/0.3	11,769.60	8,309.16	3	4	41.65
250	400	0.2/0.3	10,629.87	10,629.87	0	7	0.00

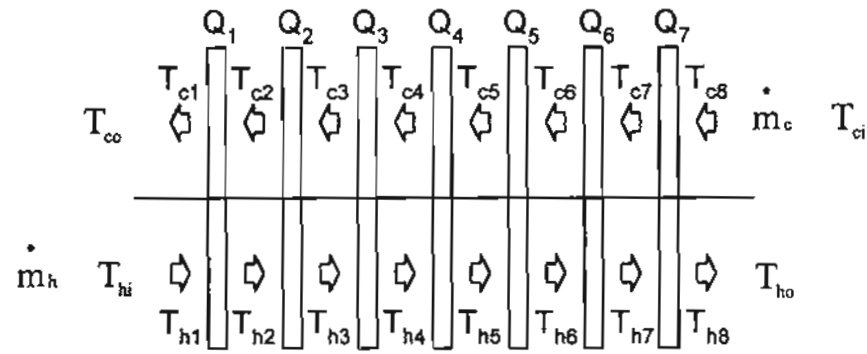


a. Operating principle

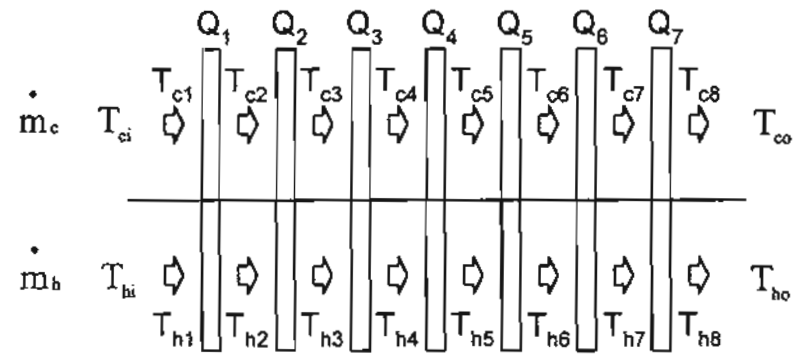


b. Thermal resistance circuit

Figure 1 Schematic of the thermosyphon heat pipe.



a. Counter flow thermosyphon heat exchanger.



b. Parallel flow thermosyphon heat exchanger.

Figure 2 Flow arrangement of the thermosyphon heat exchanger.

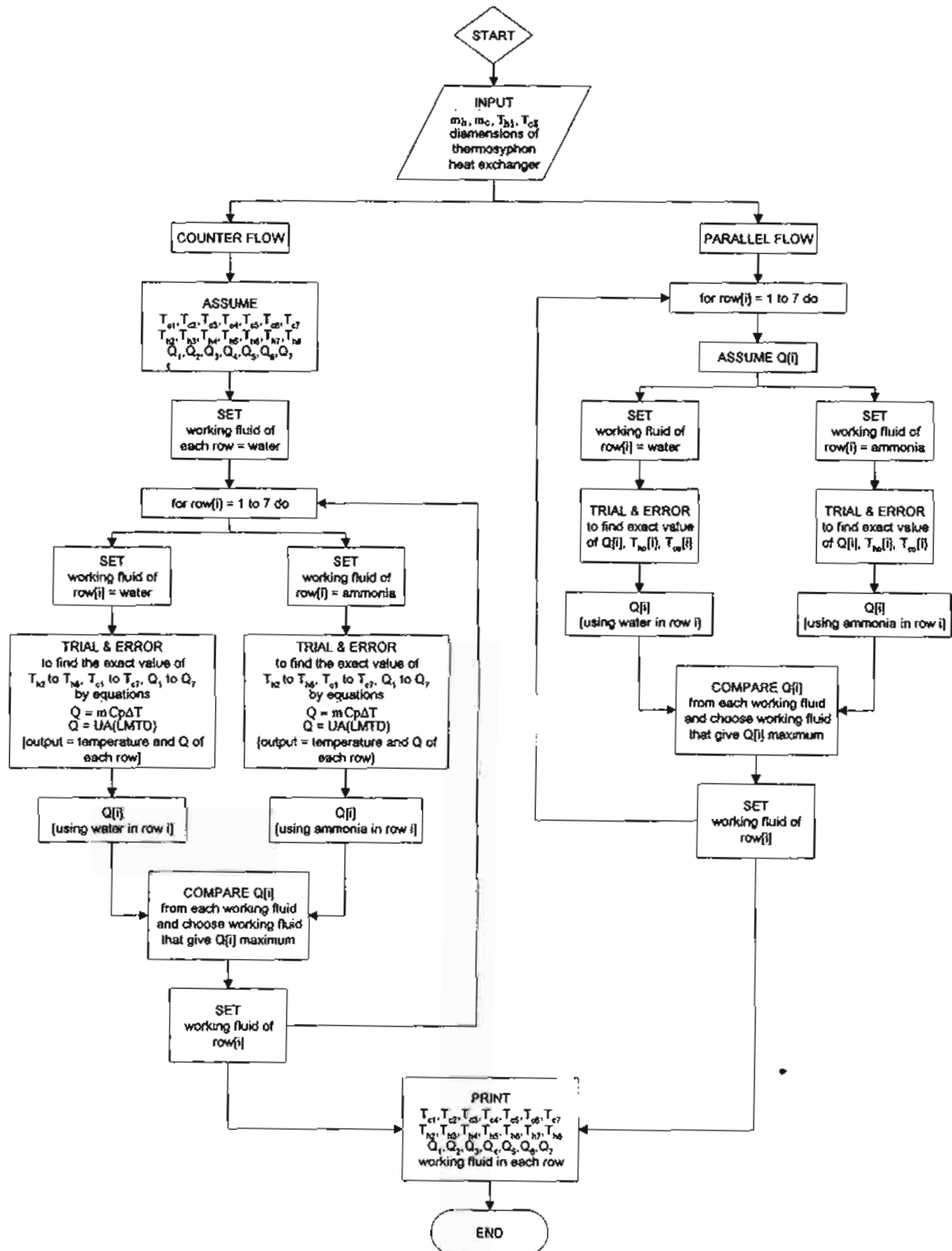


Figure 3 The flow chart of the two-fluid thermosyphons in low temperature application.

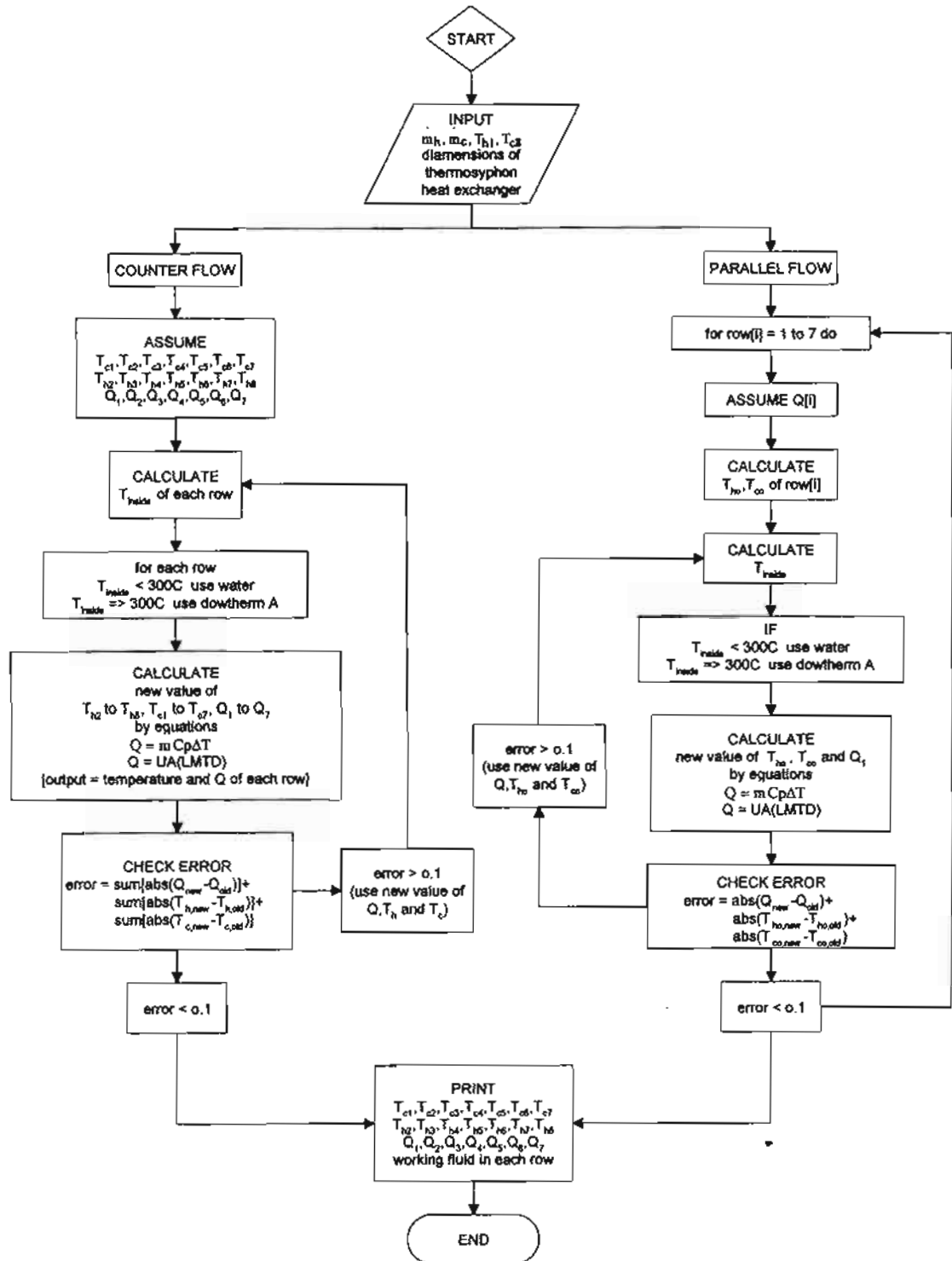


Figure 4 The flow chart of the two-fluid thermosyphons in high temperature application.

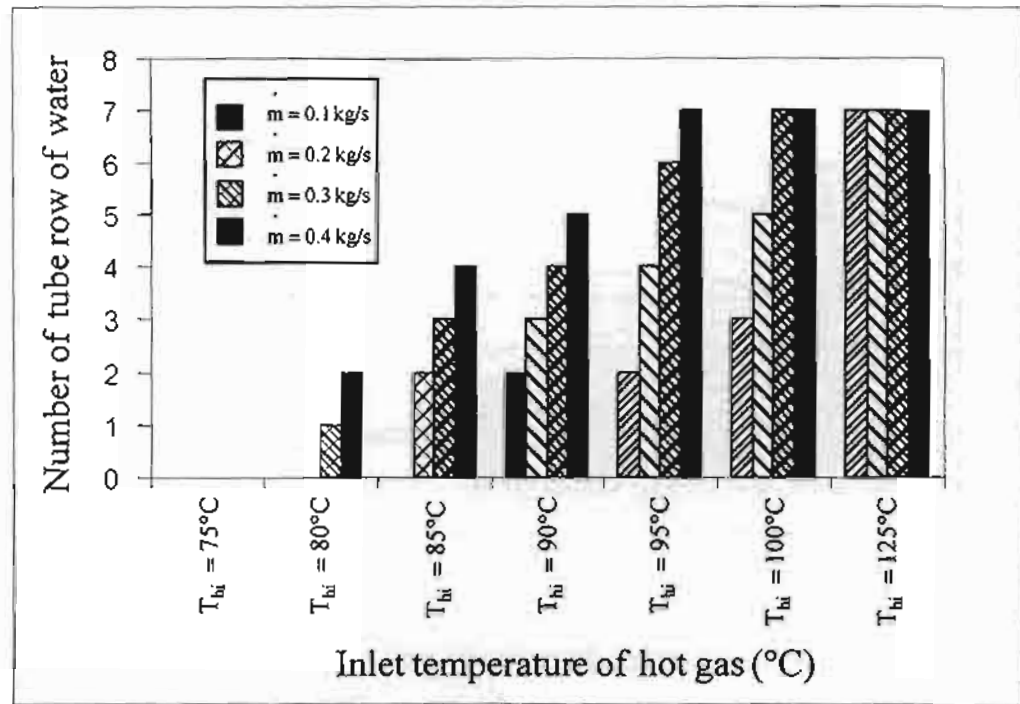


Figure 5 Relation of the appropriate number of water thermosyphons vs. inlet conditions for balanced counter flow arrangement at $T_{ci} = 20^\circ\text{C}$.

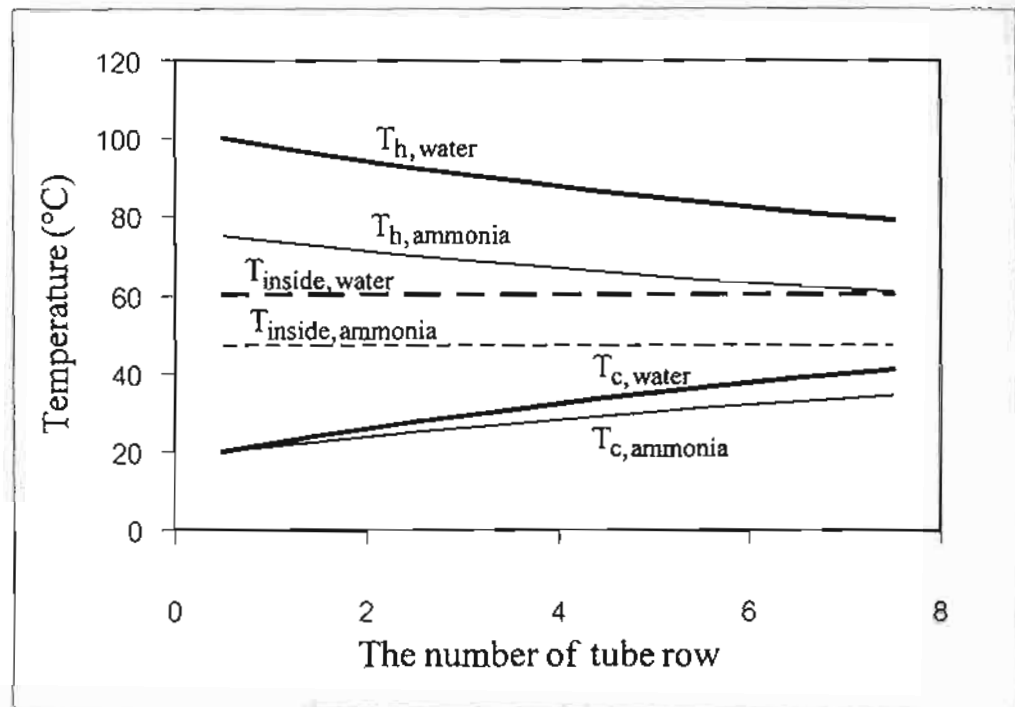


Figure 6 Temperature profiles of the balanced parallel flow thermosyphon heat exchanger using water and ammonia as working fluids.

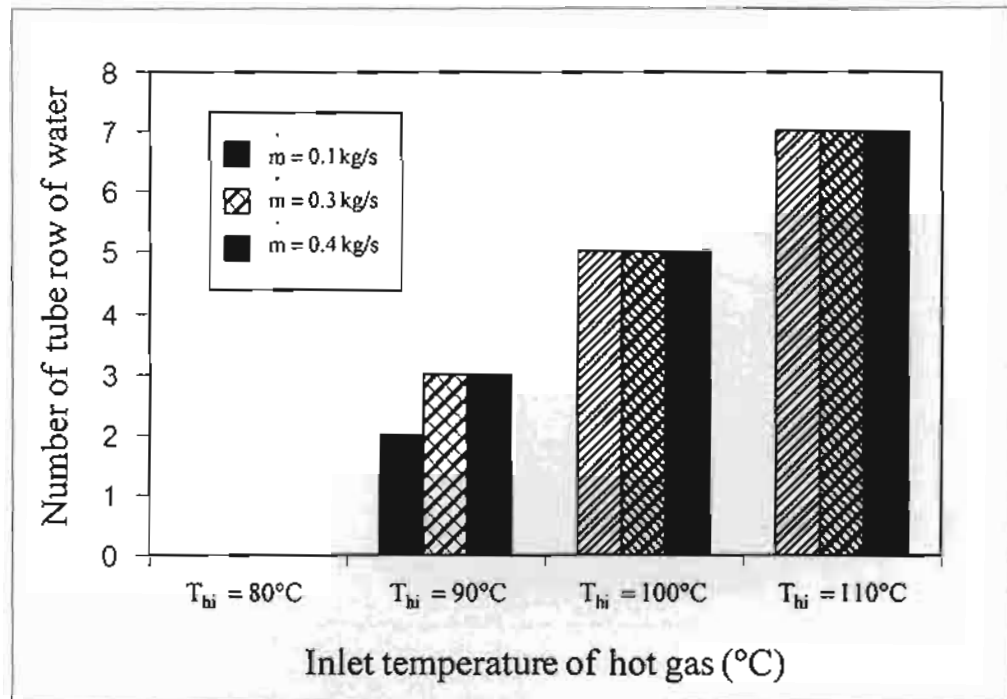


Figure 7 Relation of the appropriate number of water thermosyphons vs. inlet conditions for unbalanced parallel flow arrangement at $T_{ci} = 20^\circ\text{C}$ and $\dot{m}_h = 0.2 \text{ kg/s}$.

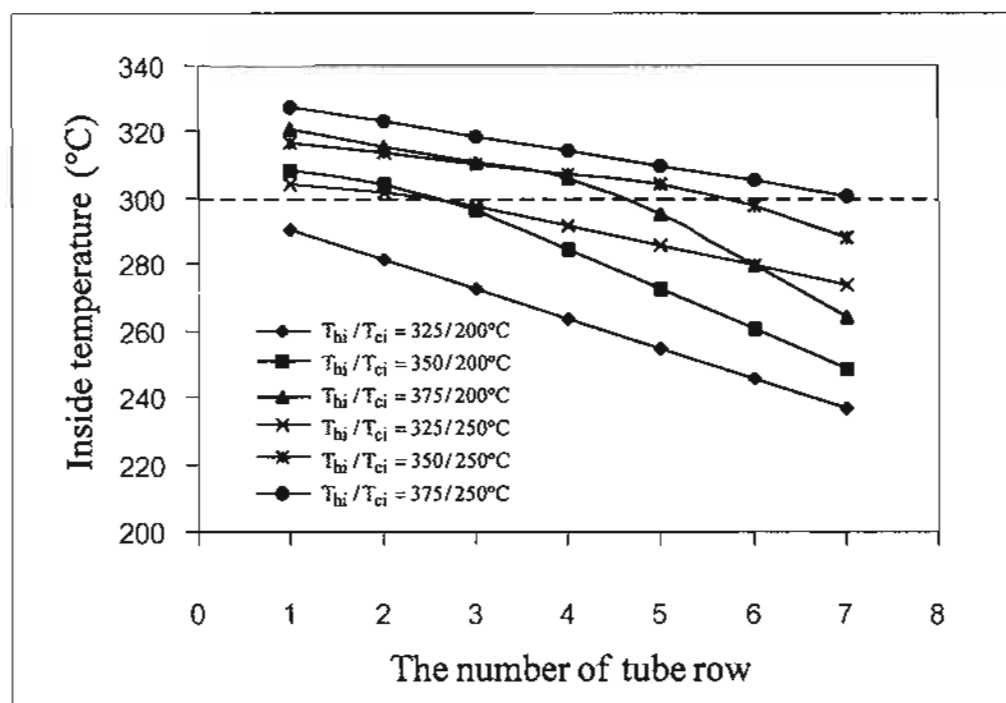


Figure 8 Interior saturation temperatures of each row of counter flow heat exchanger at various conditions; mass flow rate of air = 0.3 kg/s.

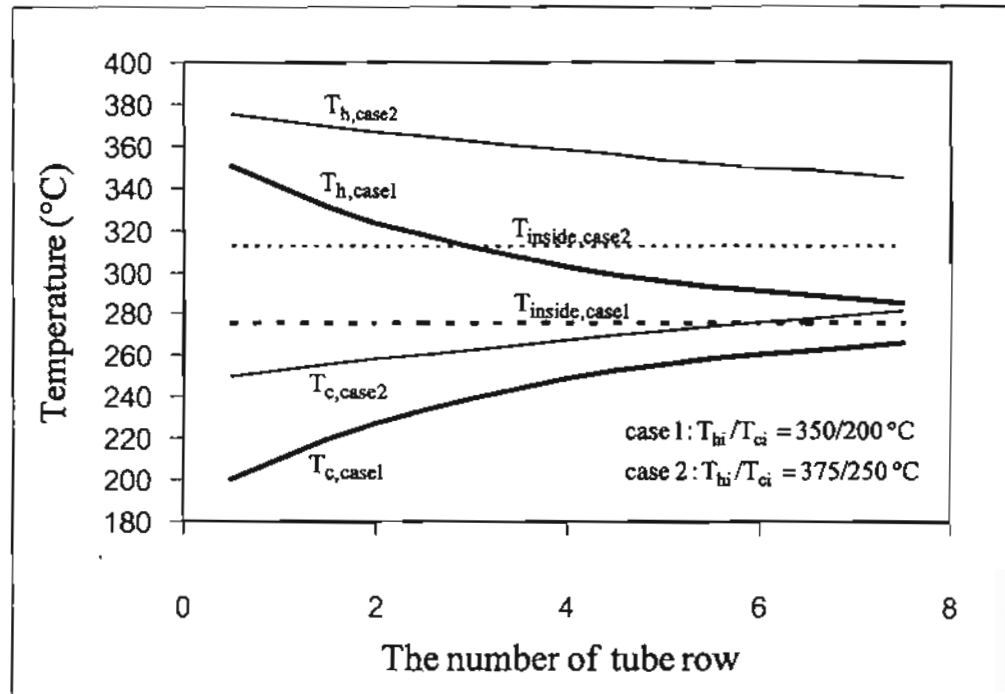


Figure 9 Temperature profile of the balanced parallel flow heat exchanger using water and dowtherm A as working fluids.

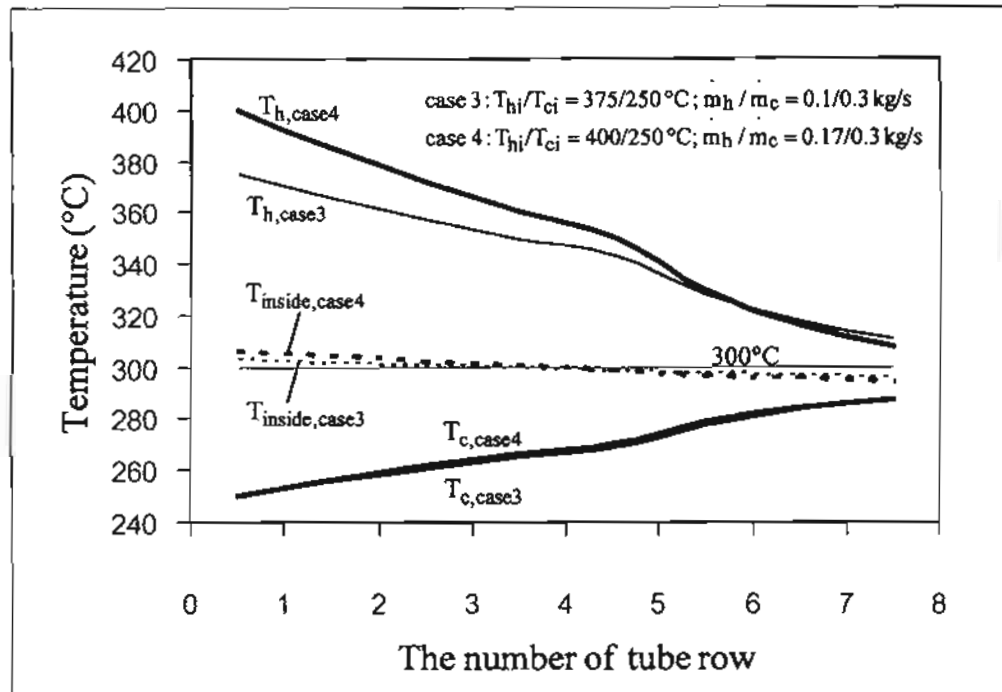


Figure 10 Temperature profile of the unbalanced parallel flow heat exchanger using water and dowtherm A as working fluids.

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Enhancement of Heat Transport in Thermosyphon Air Preheater at High Temperature with Binary Working Fluid: A Case Study of TEG-water

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Abstract - In this research, the critical heat flux due to flooding limit of thermosyphon heat pipe using TEG-water mixture has been investigated. From the experiment it is found that, use of TEG-water mixture can extend the heat transport limitation compared with pure water and higher heat transfer is obtained compared with pure TEG at high temperature applications. Moreover it is found that ESDU equation can be used to predict the critical heat flux of the thermosyphon in case of TEG-water mixture.

For thermosyphon air preheater at high temperature applications, it is found that with selected mixture content of TEG-water in each row of the thermosyphon the performance of the system could be increased approximately 30-80% compared with pure TEG for parallel flow and 60-115% for counter flow configurations. The performances also increase approximately 80-160% for parallel flow and 140-220% for counter flow compared with those of pure dowtherm A which is the common working fluid at high temperature applications.

Keywords: thermosyphon air preheater, critical heat flux

1. INTRODUCTION

The thermosyphon air preheater, at present, has become an important equipment for energy recovery from industrial waste heat because of its low investment cost and high thermal conductivity.

The thermosyphon air preheater consists of many tubes of thermosyphon heat pipes. The thermosyphon heat pipe shown in Fig. 1a can be divided into three parts, evaporator, adiabatic and condenser. When heat is added to the evaporator section, the working fluid inside the tube boils, vaporises and carries heat from the high temperature heat source to the low temperature heat sink where the working fluid condenses. The condensate from the condenser returns to the evaporator by gravitational force.

Water, normally, has been used as a working fluid inside the thermosyphon because of its high latent heat of vaporization and wide working temperature range, (approximately 50-300°C) [1-3]. However, in some conditions of waste heat such as the flue gas from ceramic furnace, the temperature of hot gas is higher than 300°C and the critical heat flux (CHF) due to flooding limit may occur when using pure water as the working fluid. Dowtherm A is recommended for this high temperature range [4] however the heat transfer rate when using dowtherm A is very low compared to the water [5].

From a previous study [6], it was found that using Triethylene glycol (TEG; $C_6H_{14}O_4$) – water mixture as a working fluid could reduce the flooding phenomenon of the thermosyphon while the heat transfer rate slightly decreases. However, the information of the TEG concentration effect on the heat transport is very limited. In this

research work the critical limit of the thermosyphon heat pipe using TEG-water has been investigated. Selection of the suitable fraction of TEG in the working fluid mixture in each row of a thermosyphon air preheater operated at high temperature has also been carried out.

2. FLOODING LIMIT OF THE THERMOSYPHON

2.1. Critical heat flux equation

Flooding phenomenon of working fluid inside the thermosyphon always occurs at high operating temperature. ESDU [7] proposes a the correlation to calculate this critical heat flux due to flooding limit and is in the form of

$$Q_{max} = f_1 f_2 f_3 \lambda A_{cs} (g\sigma(\rho_l - \rho_v))^{0.25} \rho_v^{0.5}, \quad (1)$$

where A_{cs} is cross-sectional area of the thermosyphon, D_i is inside pipe diameter. The factor f_1 is the function of Bond number (Bo) and it can be evaluated from

$$f_1 = -0.0331Bo^2 + 0.8161Bo + 3.2134, \quad (2)$$

where

$$Bo = D_i \left(\frac{g(\rho_l - \rho_v)}{\sigma} \right)^{0.5}. \quad (3)$$

The factor f_2 is a function of the dimensionless pressure parameter (K_p) which is defined as

$$K_p = \frac{P_v}{(g(\rho_l - \rho_v)\sigma)^{0.5}} \quad (4)$$

and

$$f_2 = K_p^{-0.17} \quad \text{if } K_p \leq 40000 \quad (5)$$

$$f_2 = 0.165 \quad \text{if } K_p > 40000. \quad (6)$$

The function f_3 is unity if the pipe is in vertical direction.

2.2. Experimental Set-up

The CHF due to flooding limit of a TEG-water thermosyphon has been investigated and Fig. 2 shows a schematic diagram of the experimental apparatus. A 9.5 mm diameter copper tube with 1 mm thickness has been used as a thermosyphon. The length of evaporator, adiabatic and condenser sections are 40, 20 and 40 cm respectively. Four K-type thermocouples are attached at the outer surface of the tube along the length of the evaporator section and the others four are also attached at the condenser section for measuring the surface temperature. The pressure gage is mounted on the top of the tube for measuring inside pressure. Hot paraffin oil bath with an electric heater and a temperature control box is used as the heat source of the evaporator section. The air bubble is also injected to the paraffin oil to make a uniform temperature. The condenser section is inserted into the cooling jacket. The mass flow rate of cooling water is controlled by a constant head tank and the water flows through the jacket to absorb heat from the condenser section. The inlet and the outlet temperatures of cooling water are measured by a set of thermocouples and a flow meter is used to measure the mass flow rate. These values are used to calculate the heat transfer rate of the thermosyphon.

In this study, the temperature of hot paraffin oil is controlled from 90-200°C and the temperature of the cooling water is kept constant at 30°C and the mass flow rate is 0.0054 kg/s. The working fluid inside the thermosyphon is TEG-water mixture at 0, 25, 50, 75 and 100% by volume of TEG. The filling ratio is 50% of the evaporator volume. The experiments have been carried out under steady-state conditions.

2.3. Critical heat flux of TEG-water mixture

Fig. 3 shows the critical heat flux of the thermosyphon heat pipe using various contents of TEG-water mixture. It is found that using TEG-water can extend CHF due to flooding limit and this limit is proportional to the content of TEG in the mixture. The heat transfer rate of the thermosyphon using pure TEG is the lowest compared to those of pure water and the binary mixture because of very low latent heat of pure TEG and its high normal boiling point (278°C at normal pressure for TEG).

Fig. 4 shows the comparison of the experimental results critical heat flux with those calculated from the ESDU correlation. Both results agree quite well. Note that the properties of the binary mixture can be evaluated from the methods in references [8-13].

3. THERMOSYPHON AIR PREHEATER USING TEG-WATER – A CASE STUDY

3.1. Thermal resistances of the thermosyphon air preheater

Fig. 1b shows the thermal resistance circuit of the thermosyphon. In this research work, the pressure drops from the evaporator to the condenser end and the axial

conduction along the pipe wall are assumed to be negligible. Evaluation methods of the thermal resistance are described as follows:

The external air-side thermal resistances of the evaporator (Z_{eo}) and the condenser (Z_{co}) section can be calculated from

$$Z_{eo} = \frac{1}{h_{eo} A_{eo}}, \quad (7)$$

$$Z_{co} = \frac{1}{h_{co} A_{co}}. \quad (8)$$

h_{eo} and h_{co} are the external air-side heat transfer coefficient at the evaporator and the condenser, respectively. For plain plate finned tube having staggered array, Webb [14] proposed a following correlation as

$$j = 0.14 Re^{-0.328} \left(\frac{S_t}{S_l} \right)^{-0.502} \left(\frac{f_s}{D_o} \right)^{0.031}, \quad (9)$$

where j is Colburn factor which is defined as:

$$j = \frac{h_o}{\rho V_{max} C p_f} Pr^{2/3}. \quad (10)$$

Related information for calculating the fin surface area A_f , the bare tube surface A_b , and the fin efficiency η_f are given as:

$$A_f = 2n_f \left((n_c + 0.5) S_t S_l - 0.25 \pi n_c D_o^2 \right), \quad (11)$$

$$A_b = \left(\frac{\pi n_c L_e}{f_s + f_t} \right) D_o f_s, \quad (12)$$

$$\eta_f = \frac{\tanh \left(\sqrt{2h_o / f_t k_f} \times \varphi \right)}{\sqrt{2h_o / f_t k_f} \times \varphi}, \quad (13)$$

$$\varphi = (\phi - 1)(1 + 0.35 \ln \phi), \quad (14)$$

$$\phi = 12.7 \frac{X_M}{r} \left(\frac{X_L}{X_M} - 0.3 \right)^{1/2}, \quad (15)$$

$$X_L = 0.5 \sqrt{\left(S_i / 2 \right)^2 + S_i^2}, \quad (16)$$

$$X_M = 0.5 S_i. \quad (17)$$

Note that the fin efficiency η_f is computed from Schmidt's approximation [15,16].

The wall resistances in the evaporator and the condenser sections can be calculated from

$$Z_{et} = \frac{\ln(r_o / r_i)}{2\pi k_t L_e}, \quad (18)$$

$$Z_{ct} = \frac{\ln(r_o / r_i)}{2\pi k_t L_c}. \quad (19)$$

The boiling and condensation resistances in the evaporator (Z_{el}) and condenser section (Z_{cl}) can be calculated from

$$Z_{el} = \frac{1}{h_{el} A_{el}} \quad (20)$$

$$Z_{cl} = \frac{1}{h_{cl} A_{cl}} \quad (21)$$

Kiatsiriroat et al. [6] has proposed correlations for calculating the boiling and condensation heat transfer coefficients inside the thermosyphon as

Boiling:

$$h_e = \frac{C_1}{\Delta T_e X} \left[\left(\frac{\Delta T_e}{Y} \right)^3 \right]^{C_2}, \quad (22)$$

where $X = \frac{1}{\mu_l \lambda} \sqrt{\frac{g_c \sigma}{g(\rho_l - \rho_v)}}$ and $= \frac{\lambda Pr_l}{Cp_l}$.

Condensation:

$$h_d = C_3 \left[\frac{g \rho_l^2 \lambda k_l^3}{\mu_l (T_w - T_s) L_c} \right]^{C_4} \quad (23)$$

The correlation constants C_1 - C_4 for water and TEG are shown in Table 1.

In case of binary working fluid, Kiatsiriroat et al. [6] also proposed a correlation for predicting the heat transfer coefficient as

$$h_{mixture} = h_1 x_1 + h_2 (1 - x_1), \quad (24)$$

where x_1 means the mole fraction of component 1 in the mixture.

For high temperature application, dowtherm A, is normally used as the working fluid. ESDU [7] proposed correlations for calculating the thermal resistances due to boiling and condensation inside the thermosyphon.

Thermal resistance due to pool boiling in the evaporator section:

$$Z_{ei,p} = \frac{1}{\phi_1 g^{0.25} Q^{0.4} (\pi D_e L_e)^{0.6}}, \quad (25)$$

where $\phi_1 = 0.32 \frac{\rho_l^{0.65} k_l^{0.3} Cp_l^{0.7}}{\rho_v^{0.25} \lambda^{0.4} \mu_l^{0.1}} \left[\frac{P_v}{P_o} \right]^{0.23}$. (26)

Thermal resistance due to falling film evaporation in the evaporator section:

$$Z_{ei,f} = \frac{0.235 Q^{1/3}}{D_e^{4/3} g^{1/3} L_e \phi_2^{4/3}}, \quad (27)$$

where $\phi_2 = \left(\frac{\lambda k_l^3 \rho_l^2}{\mu_l} \right)^{1/4}$. (28)

The total thermal resistance due to boiling in the evaporation can be calculated

as:

$$Z_{ei} = Z_{ei,f}(1 - F) + Z_{ei,p}F \quad (29)$$

where F is filling ratio of the working fluid.

Thermal resistance in the condenser:

$$Z_{ci} = \frac{0.235Q^{1/3}}{D_i^{4/3}g^{1/3}L_c\phi_2^{4/3}} \quad (30)$$

The overall thermal resistance of the thermosyphon air preheater can be calculated from

$$\frac{1}{(UA)_{total}} = Z_{eo} + Z_{ei} + Z_{ei} + Z_{ci} + Z_{ci} + Z_{co} \quad (31)$$

where $(UA)_{total}$ is the overall thermal resistance of the thermosyphon air preheater.

The overall heat transfer rate of the thermosyphon air preheater from Fig. 5 for parallel flow and counter flow arrangements can be evaluated as

$$\begin{aligned} Q_{total} &= (UA)_{total} \Delta T_{LMTD} \\ &= \dot{m}_h C p_h (T_{hi} - T_{ho}) \\ &= \dot{m}_c C p_c (T_{co} - T_{ci}) \end{aligned} \quad (32)$$

ΔT_{LMTD} could be calculated from

for parallel flow

$$\Delta T_{LMTD} = \frac{(T_{hi} - T_{ci}) - (T_{ho} - T_{co})}{\ln \left[\frac{T_{hi} - T_{ci}}{T_{ho} - T_{co}} \right]}, \quad (33)$$

for counter flow

$$\Delta T_{lmtd} = \frac{(T_{hi} - T_{co}) - (T_{ho} - T_{ci})}{\ln \left[\frac{T_{hi} - T_{co}}{T_{ho} - T_{ci}} \right]}. \quad (34)$$

3.2. Simulation program

This part is the case study of using TEG-water as a working fluid in the thermosyphon air preheater. Table 2 shows testing conditions and related geometrical parameters of the thermosyphon air preheater using in the simulation program. The concept of the program is to find out the suitable mixture content of TEG-water in each row of the air preheater. Note that the contents of TEG in the binary mixture are 0, 25, 50, 75 and 100% by volume.

Fig. 6 shows the flow chart of the calculation method. It can be divided into two parts, counter flow and parallel flow. In the counter flow part, firstly input all of working conditions and also dimensions of the air preheater then assume values of inlet and outlet temperatures of the hot and the cold streams and also heat transfer rate in each row. Next, set the working fluid of each row as water and using trial and error technique for finding out the real value of the temperatures and the heat transfer rate of each row. Then check the critical heat flux of each row and select the suitable working fluid which gives the highest heat transfer rate. Re-calculate until all the computational values are constant. The calculation continues to the last row of the heat exchanger.

For the parallel flow part, starting with row 1 by assume the value of Q and using water as working fluid. Then, by trial and error technique, the real value of the outlet temperature of the hot and the cold air and also the heat transfer rate are obtained.

Next check the critical heat flux of this row and compute the heat transfer rate then select the suitable mixture content and recalculate again until every value is steady. Next starting with row 2 by the same method until the last row is carried out.

3.3. Results and discussion

Tables 3 and 4 show the results of the heat exchanger from the simulation program for the parallel flow and the counter flow respectively. It is found that the heat transfer rate increases significantly in some conditions compared to pure TEG when using suitable mixture content in each row of the air preheater. Normally water cannot be used as a working fluid in this temperature range (300-400°C) because of the flooding limit. However, TEG-water can extend this condition and the performance increases approximately 30-80% for parallel flow and 60-115% for counter flow compared to pure TEG. When compared with a common working fluid, dowtherm A, it is also found that using TEG-water can also increase the heat transfer rate approximately 80-160% for parallel flow and 140-220% for counter flow configurations.

In case of parallel flow, it is found that the content of TEG should be high in the first rows and water is a good working fluid in the last rows. These results can be explained by Fig. 7 which shows the temperature profiles of the parallel flow air preheater. Normally the gap between the hot and the cold streams is large at the initial rows and narrow at the last rows. Consequently the heat transfer rate of the initial row is higher than that of last row and it has a high opportunity to reach the critical limit and

therefore high content of TEG or pure TEG are the suitable working fluids in this range. On the other hand, water or lower content of TEG in the mixture is suitable for the last rows. In Fig. 7 it also shows the comparison between using TEG-water and pure TEG. This result comes from the simulation program shown in Table 3 (No.7). It is found that using 75% and 25% of TEG in rows 3 and 4 and pure water in row 5 and 6, the temperature difference between the hot and the cold air streams is smaller significantly compared with pure TEG, therefore, higher heat exchange is obtained.

In case of counter flow, from Table 4, it is found that in case of balance counter flow ($\dot{m}_h = \dot{m}_c$) the mixture content in each row is nearly the same. This phenomenon can be explained by Fig. 8 which shows the comparison of temperature profiles of balance counter flow between using TEG-water mixture and pure TEG. Actually in this case the temperature gap of the hot and the cold streams is nearly constant in this case. Consequently the heat transfer rate of each row is also nearly constant and this value is suitable for only one mixture content. However, in case of unbalance counter flow ($\dot{m}_h \neq \dot{m}_c$), the temperature difference is not constant including the heat transfer rate, therefore, the mixture content in each row will affect the performance. Fig. 9 shows the result obtained from Table 4 (No.18). Use of TEG content of 75% in row 1-3 and 100% in rows 4-6 shows better heat exchange compared with pure TEG in all rows.

It could be seen that, the system performance when using TEG-mixture is also higher than using dowtherm A which is a common working fluid of high temperatures. From Tables 3 and 4, the performance increases about 80-160% for parallel flow and 140-220% for counter flow.

4. CONCLUSIONS

In this research, the concept of using TEG-water in the thermosyphon heat pipe has been studied. Major results are summarized as follow:

1. Using of TEG-water can extend the critical limit due to flooding inside the thermosyphon and the limit is directly proportional to the content of TEG in the mixture.
2. Suitable mixture content of TEG-water in each row of the thermosyphon air preheater can increase the performance of the system approximately 30-80% for parallel flow and 60-115% for counter flow air preheater compared with pure TEG.
3. The heat exchanger with suitable content of TEG-water also shows better performance than that with dowtherm A.

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NOMENCLATURE

A	area (m^2)
A_{cs}	cross sectional area (m^2)
Bo	Bond number
C_{1-4}	empirical constants
C_p	specific heat (J/kgK)
D	diameter (m)
f_{1-3}	parameter
f_s	fin gap (m)
f_t	fin thickness (m)
g	gravitational acceleration (9.81 m/s^2)
g_c	conversion factor (1 kgm/Ns^2)
h	heat transfer coefficient ($\text{W/m}^2\text{K}$)
j	Colburn factor
k	thermal conductivity (W/mK)
K_p	parameter
L	length (m)
$\dot{m}$	mass flow rate (kg/s)
n_c	number of column of tube bank
n_f	total number of fin
Q	heat transfer rate (W)

r	radius (m)
S_l	longitudinal pitch of tube bank (m)
S_t	transverse pitch of tube bank (m)
T	temperature ($^{\circ}\text{C}$)
ΔT_e	temperature difference between inside surface of evaporator tube and working fluid ($^{\circ}\text{C}$)
ΔT_c	temperature difference between inside surface of condenser tube and working fluid ($^{\circ}\text{C}$)
ΔT_{lmtd}	log mean temperature difference ($^{\circ}\text{C}$)
x	mole fraction
X	factor
X_L	factor
X_M	factor
Y	factor
Z	thermal resistance (K/W)

Greek symbols

ϕ	factor
η	efficiency
φ	factor
λ	latent heat (J/kg)
μ	dynamic viscosity (Pas)
ρ	density (kg/m^3)
σ	surface tension (N/m)

Subscripts

b	bare tube
c	condenser section, cold air
e	evaporator section
f	fin, film evaporation
h	hot air
i	inside, inlet
l	liquid
o	outside, outlet
p	pool boiling
s	surface
v	vapor
w	working fluid

CAPTIONS OF FIGURES AND TABLES

Fig. 1. Schematic of the thermosyphon heat pipe. a. Operating principle. b. Thermal resistance circuit.

Fig. 2. The experimental set-up.

Fig. 3. Critical heat flux of thermosyphon heat pipe using TEG-water mixtures.

Fig. 4. Comparison of critical heat flux from the experiment and the ESDU model at various mixtures content of TEG.

Fig. 5. Thermosyphon air preheater. a. Parallel flow arrangement. b. Counter flow arrangement.

Fig. 6. Flow chart for calculating the suitable mixture content of TEG-water in each row of the thermosyphon air preheater.

Fig. 7. Comparison of temperature profile of parallel flow thermosyphon air preheater between using TEG-water mixtures and pure TEG (no. 7 in Table 3).

Fig. 8. Comparison of temperature profile of balance counter flow thermosyphon air preheater between using TEG-water mixtures and pure TEG (no. 3 in Table 4).

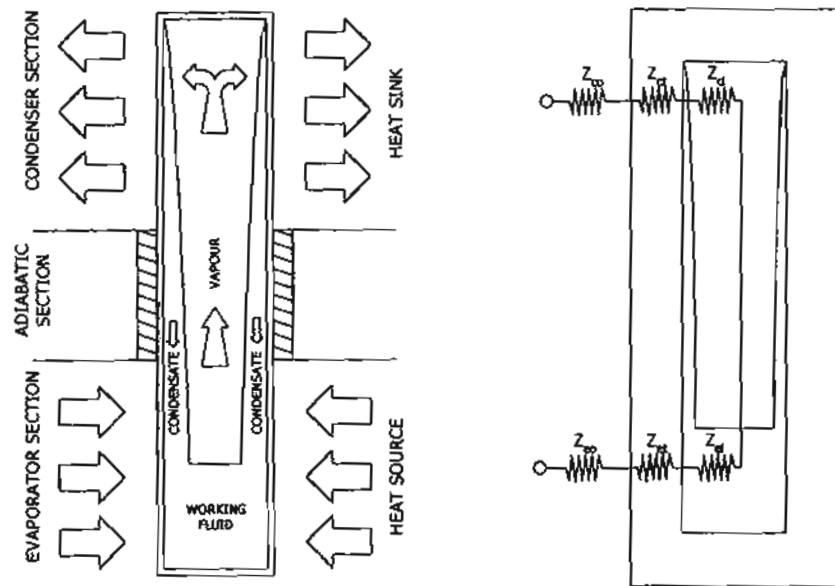
Fig. 9. Comparison of temperature profile of unbalance counter flow thermosyphon air preheater between using TEG-water mixtures and pure TEG (no.18 in Table 4).

Table 1. Empirical constants of Kiatsiriroat et al. Correlation.

Table 2. Testing conditions and related geometrical parameters of the thermosyphon air preheater.

Table 3. Results of simulation in case of parallel flow arrangement; inlet temperature of cold air = 30°C.

Table 4. Results of simulation in case of counter flow arrangement; inlet temperature of cold air = 30°C.



a. Operating principle

b. Thermal resistance circuit

Fig. 1. Schematic of the thermosyphon heat pipe.

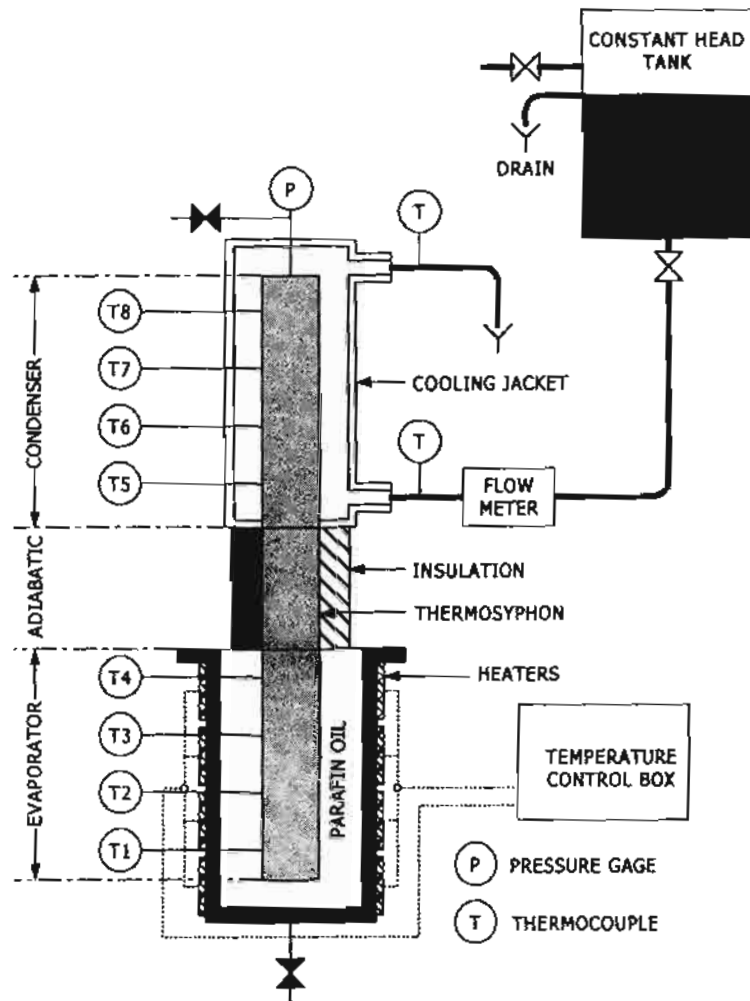


Fig. 2. The experimental set-up

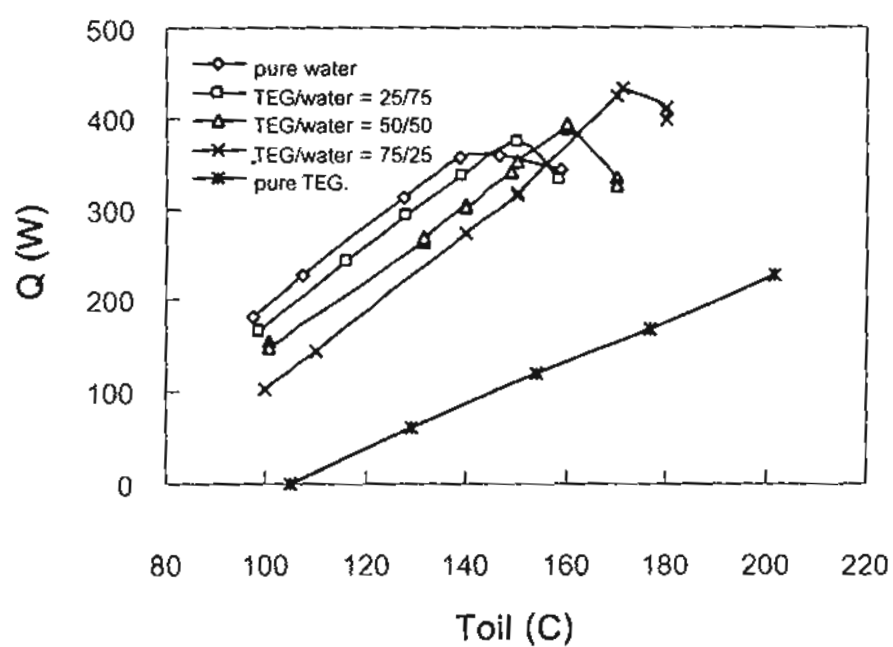


Fig. 3. Critical heat flux of thermosyphon heat pipe using TEG-water mixtures.

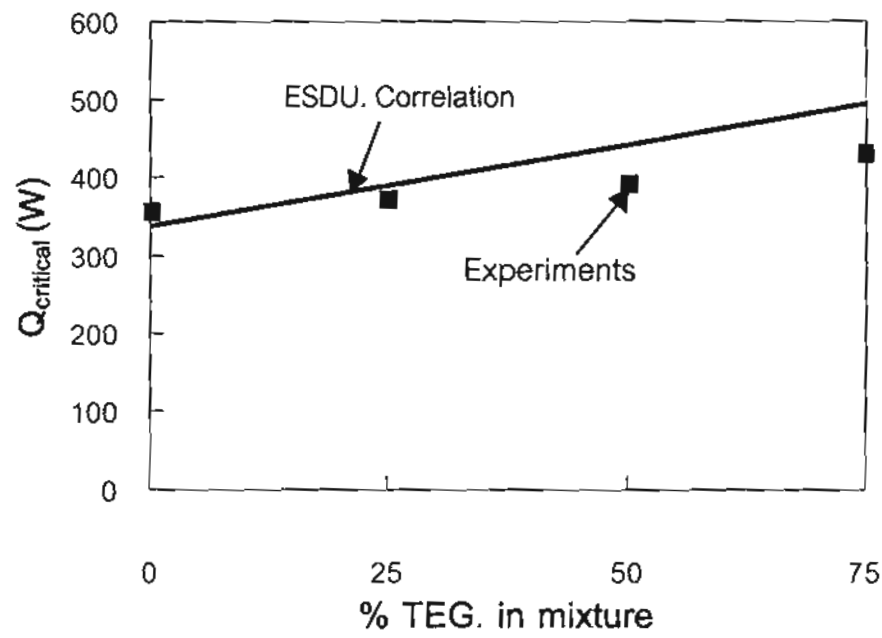


Fig. 4. Comparison of critical heat flux from the experiment and the ESDU model at various mixtures content of TEG.

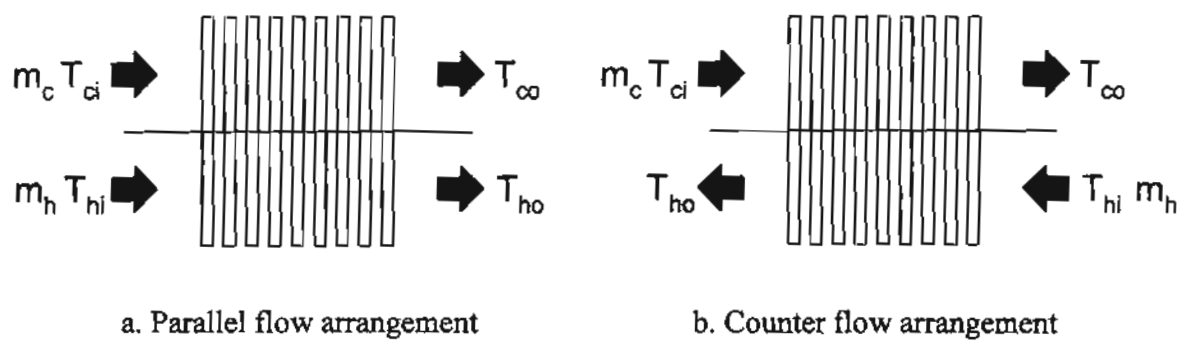


Fig. 5. Thermosyphon air preheater.

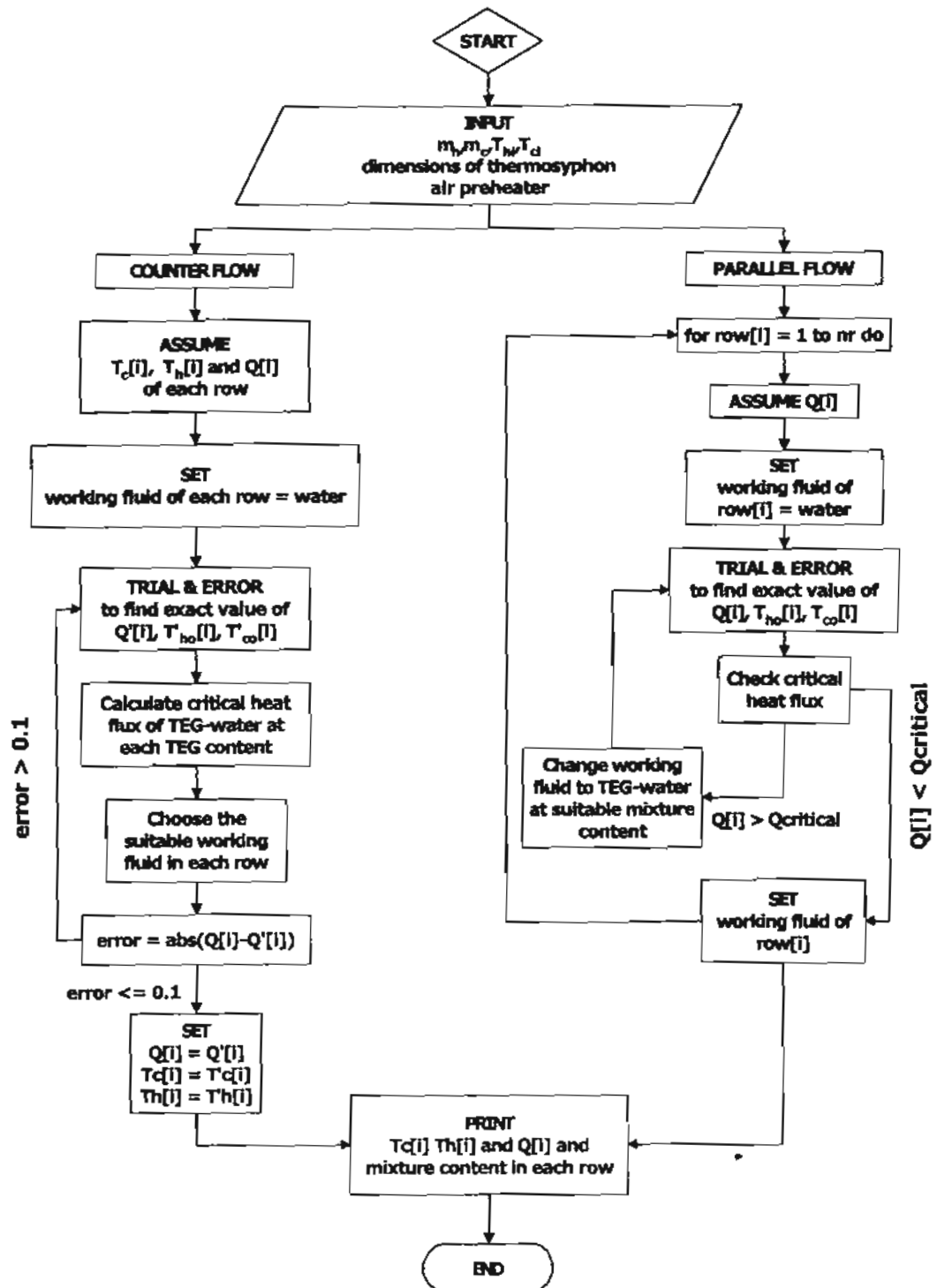


Fig. 6. Flow chart for calculating the suitable mixture content of TEG-water in each row of the thermosyphon air preheater.

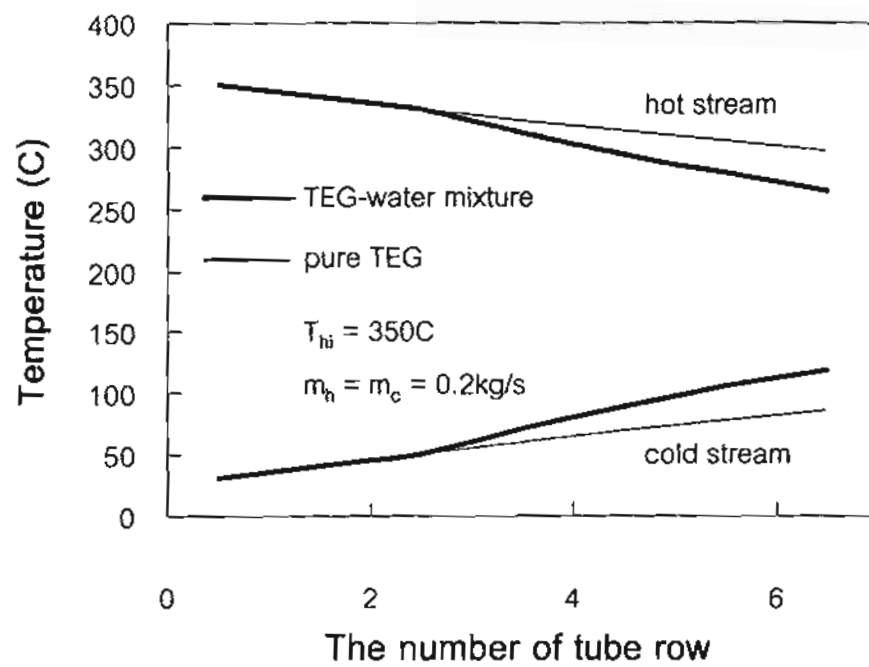


Fig. 7. Comparison of temperature profile of parallel flow thermosyphon air preheater between using TEG-water mixtures and pure TEG (no. 7 in Table 3).

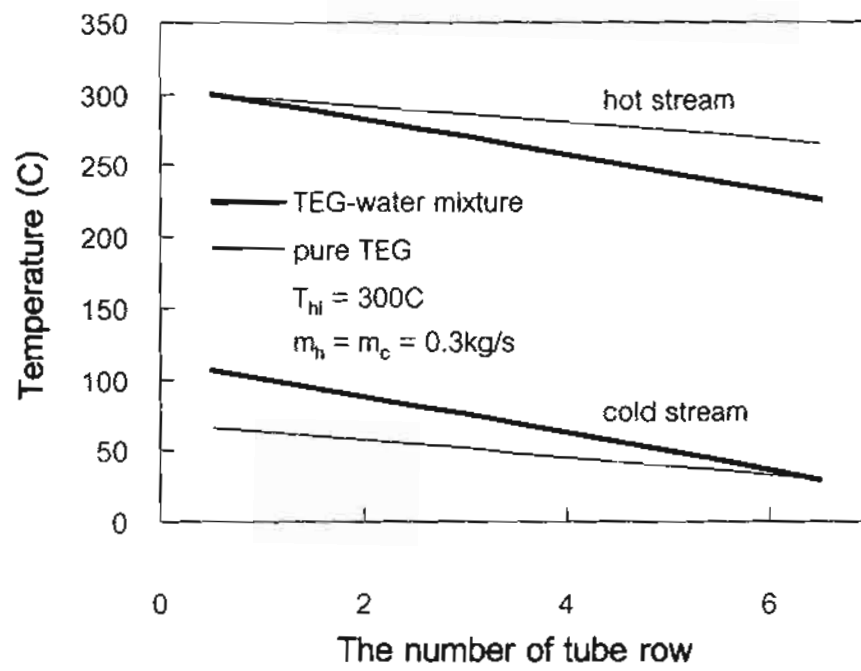


Fig. 8. Comparison of temperature profile of balance counter flow thermosyphon air preheater between using TEG-water mixtures and pure TEG (no. 3 in Table 4).

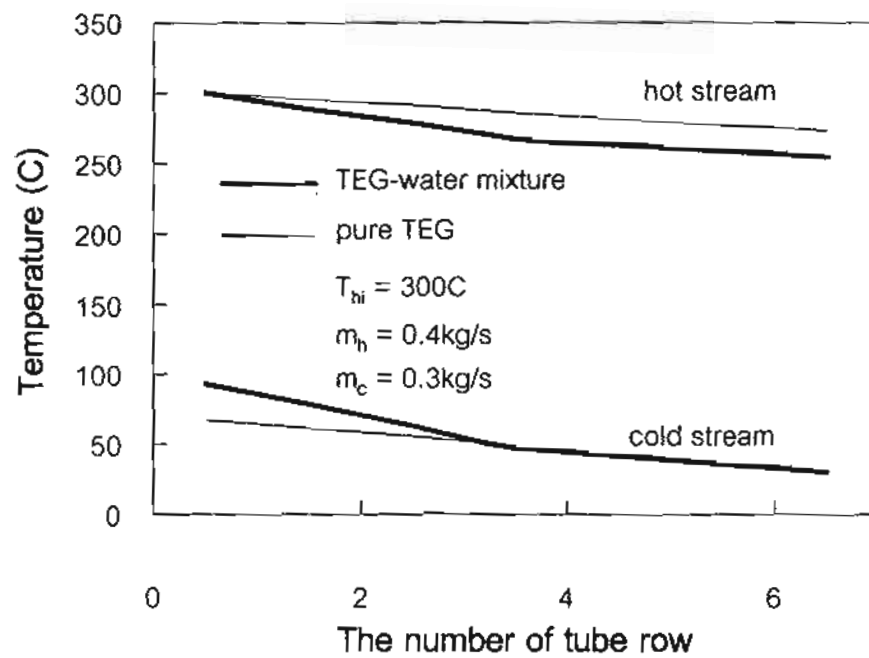


Fig. 9. Comparison of temperature profile of unbalance counter flow thermosyphon air preheater between using TEG-water mixtures and pure TEG (no.18 in Table 4).

Table 1. Empirical constants of Kiatsiroat et al. correlation.

Working fluid	C_1	C_2	C_3	C_4
Water	18.688	0.3572	0.943	0.233
TEG	20.565	0.3662	0.943	0.180

Table 2. Testing conditions and related geometrical parameters of the thermosyphon air preheater.

Items	Conditions
1. Flow arrangement	Parallel and counter flow
2. Temperature of hot air	300-400 °C
3. Temperature of cold air	30 °C
4. Mass flow rate of air	0.1-0.5 kg/s
5. Thermosyphon arrangement	Staggered array $S_t = 0.02$ m, $S_d = 0.02$ m, $S_l = 0.0173$ m
6. Number of tube row	6
7. Number of column	10
8. Diameter of thermosyphon (bare tube)	0.0095 m
9. Type of fin	Plain plate fin
10. Size of fin	fin pitch = 6 fins/inch fin thickness = 0.0005 m
11. Filling ratio of working fluid	50%
12. Material of pipe and fin	Copper for pipe and Aluminum for fin

Table 3. Results of simulation in case of parallel flow arrangement; inlet temperature of cold air = 30°C.

No	$\dot{m}_h$ (kg/s)	$\dot{m}_c$ (kg/s)	T_{hi} (°C)	% TEG in each row						$Q_{TEG-water}$ (W)	Q_{TEG} (W)	$Q_{dowtherm A}$ (W)	% increase	
				1	2	3	4	5	6				Relative to TEG	Relative to dowtherm A
1	0.1	0.1	300	0	0	0	0	0	0	10055	6934	5377	45.00	87.01
2	0.2	0.2	300	50	25	0	0	0	0	17301	9446	6656	83.16	159.92
3	0.3	0.3	300	100	100	100	100	75	25	15340	10827	7263	41.68	111.21
4	0.4	0.4	300	100	100	100	100	100	100	11724	11724	7626	0.00	53.74
5	0.5	0.5	300	100	100	100	100	100	100	12361	12361	7871	0.00	57.04
6	0.1	0.1	350	0	0	0	0	0	0	12065	8236	6371	46.49	89.38
7	0.2	0.2	350	100	100	75	25	0	0	17926	11168	7846	60.52	128.48
8	0.3	0.3	350	100	100	100	100	100	100	12768	12768	8539	0.00	49.53
9	0.4	0.4	350	100	100	100	100	100	100	13801	13801	8952	0.00	54.16
10	0.5	0.5	350	100	100	100	100	100	100	14533	14533	9231	0.00	57.44
11	0.1	0.1	400	25	0	0	0	0	0	14046	9533	7340	47.34	91.36
12	0.2	0.2	400	100	100	100	100	75	25	16820	12865	8990	30.74	87.10
13	0.3	0.3	400	100	100	100	100	100	100	14668	14668	9758	0.00	50.31
14	0.4	0.4	400	100	100	100	100	100	100	15827	15827	10215	0.00	54.94
15	0.5	0.5	400	100	100	100	100	100	100	16646	16646	10521	0.00	58.21
16	0.1	0.3	300	0	0	0	0	0	0	13681	8335	6102	64.15	124.20
17	0.2	0.3	300	100	75	25	0	0	0	18311	10038	6917	82.41	164.71
18	0.4	0.3	300	100	100	100	100	100	100	11294	11294	7459	0.00	51.42
19	0.5	0.3	300	100	100	100	100	100	100	11605	11605	7586	0.00	52.98
20	0.1	0.3	350	50	0	0	0	0	0	16341	9898	7227	65.10	126.11
21	0.2	0.3	350	100	100	100	100	75	25	16038	11866	8151	35.16	96.75
22	0.4	0.3	350	100	100	100	100	100	100	13297	13297	8757	0.00	51.84
23	0.5	0.3	350	100	100	100	100	100	100	13650	13650	8899	0.00	53.39
24	0.1	0.3	400	100	75	25	0	0	0	17967	11456	8324	56.83	115.85
25	0.2	0.3	400	100	100	100	100	100	100	13666	13666	9338	0.00	46.35
26	0.4	0.3	400	100	100	100	100	100	100	15253	15253	9994	0.00	52.62
27	0.5	0.3	400	100	100	100	100	100	100	15640	15640	10146	0.00	54.15

Table 4. Results of simulation in the case of counter flow arrangement; inlet temperature of cold air = 30°C.

No	m_h (kg/s)	m_c (kg/s)	T_{hi} (°C)	% TEG in each row						$Q_{TEG-water}$ (W)	Q_{TEG} (W)	$Q_{dowtherm A}$ (W)	% increase	
				1	2	3	4	5	6				Relative to TEG	Relative to dowtherm A
1	0.1	0.1	300	0	0	0	0	0	0	10884	7120	5448	52.86	99.78
2	0.2	0.2	300	0	0	0	0	0	0	18465	9543	6684	93.49	176.24
3	0.3	0.3	300	50	50	50	50	50	50	23340	10888	7278	114.37	220.70
4	0.4	0.4	300	100	100	100	100	100	100	11765	11765	7635	0.00	54.09
5	0.5	0.5	300	100	100	100	100	100	100	12391	12391	7878	0.00	57.29
6	0.1	0.1	350	0	0	0	0	0	0	13081	8453	6452	54.75	102.76
7	0.2	0.2	350	25	25	25	25	25	25	21033	11280	7877	86.47	167.03
8	0.3	0.3	350	100	100	100	100	100	100	12836	12836	8555	0.00	50.03
9	0.4	0.4	350	100	100	100	100	100	100	13847	13847	8962	0.00	54.50
10	0.5	0.5	350	100	100	100	100	100	100	14566	14566	9237	0.00	57.68
11	0.1	0.1	400	0	0	0	0	0	0	15330	9778	7429	56.77	106.36
12	0.2	0.2	400	50	50	50	50	50	50	24019	12989	9023	84.92	166.20
13	0.3	0.3	400	100	100	100	100	100	100	14743	14743	9776	0.00	50.81
14	0.4	0.4	400	100	100	100	100	100	100	15877	15877	10225	0.00	55.27
15	0.5	0.5	400	100	100	100	100	100	100	16682	16682	10529	0.00	58.45
16	0.1	0.1	300	0	0	0	0	0	0	14296	8430	6133	69.59	133.11
17	0.2	0.2	300	25	25	25	25	25	25	20290	10113	6937	100.63	192.47
18	0.3	0.3	300	75	75	75	75	75	75	18987	11344	7470	67.38	154.16
19	0.4	0.4	300	100	100	100	100	100	100	14619	11648	7596	25.51	92.46
20	0.5	0.5	300	100	100	100	100	100	100	17065	10009	7262	70.50	135.00
21	0.1	0.1	350	25	25	25	25	25	25	23517	11952	8174	96.76	187.72
22	0.2	0.2	350	75	75	75	75	75	75	13354	13354	8770	0.00	52.27
23	0.3	0.3	350	100	100	100	100	100	100	13698	13698	8910	0.00	53.74
24	0.4	0.4	350	100	100	100	100	100	100	19808	11582	8362	71.02	136.89
25	0.5	0.5	350	100	100	100	100	100	100	13762	13762	9362	0.00	47.00
26	0.1	0.1	400	50	50	50	50	50	50	15314	15314	10008	0.00	53.03
27	0.2	0.2	400	100	100	100	100	100	100	15692	15692	10157	0.00	54.49

· Kiatsiriroat T., Nuntaphan A. and Tiansuwan J., Parameters Affecting Exergy Loss of Thermosyphon Air Preheater, **The 6th International Heat Pipe Symposium**, 5-9 November 2000, Chiang Mai, Thailand.

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Parameters Affecting Exergy Loss of Thermosyphon Air Preheater

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ABSTRACT

The parameters affecting the exergy loss of the thermosyphon air preheater was studied in this research work. The effect of the ratio of mass capacity flow rate, the inlet temperature ratio, the pipe arrangement and the flow direction had been considered as the parameters and the simulation program was used to calculate the exergy loss of the system. The dimensionless of exergy loss per dimensionless heat transfer rate formulated from the simulation program was used as the criteria for various conditions.

From the simulation program, it was found that the parallel flow and aligned array give more exergy loss than the counter current flow and staggered array respectively. Moreover it was found that the exergy loss per unit heat transfer vary in reverse direction with the ratio of mass capacity flow rate but directional to the inlet temperature ratio.

Key Words : Exergy analysis, Thermosyphon air preheater.

INTRODUCTION

At present, the second law efficiency of thermodynamic that used for evaluate the performance of the mechanical system has become interesting topic of a new research work. The entropy generation or the exergy loss is used as a scale to determine quantitatively the quality of thermal energy transformation.

Gool [1] used the exergy concept for analysis the industrial processes to find the exergy loss and the second law efficiency. Nikolaidis and Probert [2] studied the behavior of two-stage compound compression cycle, with flash intercooling, using R-22 by exergy method. Kotas and Jassim [3] and McGovern [4] analyzed the exergetic cost with the optimization

technique to find the suitable condition and size of the thermal system.

In the case of heat exchanger, normally the effectiveness is used as a parameter to determine the performance. However this quality gives very limited information on the quality of energy transformation in the system [5]. Consequently, in this research, the concept of second law efficiency of thermodynamic is applied to evaluate the performance of the thermosyphon air preheater. The thermosyphon air preheater is an economic type of heat exchanger because of its many advantages such as low investment cost, high thermal conductivity, no energy consumption and moving part. Normally this equipment is used to recover heat from the exhaust gas to produce hot air. So it can improve the performance of overall system.

THEORETICAL ANALYSIS

The method for analysis of exergy loss of Teke et al. [6] had been used in this research work and the details are as follow.

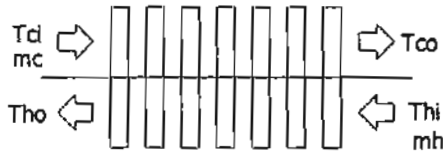


Figure 1 Flow direction of the thermosyphon air preheater.

Figure 1 shows the flow pattern of the thermosyphon air preheater and the exergy loss from the heat transfer is

$$E = T_0 \left[m_h C_{p_h} \ln \left(\frac{T_{ho}}{T_{hi}} \right) + m_c C_{p_c} \ln \left(\frac{T_{co}}{T_{ci}} \right) \right] \quad [1]$$

where E is the exergy loss of the air preheater. The effectiveness of the air preheater when the heat capacity flow rate of hot fluid is minimum, can be evaluated as

$$\varepsilon = \frac{T_{hi} - T_{ho}}{T_{hi} - T_{ci}} \quad [2]$$

The inlet temperature ratio and the ratio of heat capacity flow rate is defined as

$$Tr = \frac{T_{hi}}{T_{ci}} \quad [3]$$

$$Cr = \frac{m_h C_{p_h}}{m_c C_{p_c}} \quad [4]$$

where Tr is inlet temperature ratio and Cr is heat capacity flow rate ratio. A dimensionless exergy loss and a dimensionless heat transfer rate are defined as

$$e = \frac{E}{m_h C_{p_h} T_0} \quad [5]$$

$$q = \frac{Q}{m_h C_{p_h} T_{ci}} \quad [6]$$

where e is a dimensionless exergy loss and q is a dimensionless heat transfer rate.

From equations 1-6 the dimensionless of exergy loss per dimensionless heat transfer rate can be evaluated as

$$\frac{e}{q} = \frac{\ln[1 - \varepsilon(1 - (1/Tr))] + (1/Cr) \ln[1 + \varepsilon Cr(Tr - 1)]}{\varepsilon(Tr - 1)} \quad [7]$$

where e/q is the dimensionless of exergy loss per dimensionless heat transfer rate. Higher e/q is mean that the irreversibility of system increases.

The simulation program had been constructed by the method of Nuntaphan et al. [7] to calculate the e/q at various conditions such as the inlet temperature, the flow pattern of the air preheater, the pipe arrangement and the mass flow rate of hot and cold streams.

EXPERIMENTAL SETUP

In this research work, the counter and parallel flow heat pipe air preheaters with staggered and aligned arrangement had been used to calculate the exergy loss per dimensionless heat transfer rate. Importance conditions of these heat exchangers are shown in Table 1.

RESULTS AND DISCUSSION

Figure 2 shows the relation between the dimensionless of exergy loss per dimensionless heat transfer rate (e/q) and the inlet temperature ratio (Tr) at various conditions of heat capacity flow rate ratio (Cr), pipe arrange and flow pattern.

From figure 2, it was found that the factor e/q vary with the inlet temperature ratio. It means that higher difference of inlet

temperature of hot and cold stream gives higher e/q . The increase of factor e/q indicates that the irreversibility of the system increase. In the case of the heat exchanger, the second low efficiency equals to unity in the reversible process. It means that no exergy loss. From equation 1, no exergy loss occurs when the inlet temperature of hot stream equals to the outlet temperature of cold stream and on the other hand the outlet temperature of hot stream equals to the inlet temperature of cold stream. Moreover the heat capacity flow rate ratio must equals to unity. These mean that there is no gap between the temperature profile of hot and cold stream. So, it can be concluded that the exergy loss can be reduced by reduce the gap of the temperature profile of both stream.

The effect of the inlet temperature ratio plays an important role on the factor e/q and the increase of this factor creates the enlargement of the temperature gap between hot and cold stream. Consequently, high e/q is obtained.

The same affect as the inlet temperature, the heat capacity flow rate ratio equals to unity give lowest factor e/q . Because the rate of flow can decreases the temperature gap of hot and cold stream. In the case of flow pattern, counter flow has lower factor e/q than parallel flow because of it nature of the flow pattern. Counter flow can reduces the temperature gap more efficiently than parallel flow. It means that the area between the two stream of counter flow is lower than that of parallel flow. However at low ratio of heat capacity flow rate, factor e/q of parallel flow is slightly higher than counter flow because the unbalance of the flow pattern.

The pipe arrangements have also affect to the factor e/q . Staggered array gives lower factor e/q than aligned array. Normally staggered array has high heat transfer rate than aligned array. Therefore, the temperature change of both streams is greater than that of aligned array and it means that the gap of temperature or both

streams is reduced and the irreversibility of system is decreased.

CONCLUSION

The exergy loss of the thermosyphon air preheater can be decreased by reduce the temperature gap of hot and cold stream. The increase in the inlet temperature different and decrease the heat capacity flow rate bring to enlarge this temperature gap and high exergy loss is obtained.

The counter flow air preheater has lower exergy loss than that of parallel flow because of it's more efficiently transfer of heat and the area of gap between two streams is lower than that of parallel flow. The staggered array arrangement of pipe in the thermosyphon air preheater give lower exergy loss than that of aligned arrangement because of its high efficiency of heat.

ACKNOWLEDGEMENT

The Authors gratefully acknowledge the support provided by the Thailand Research Fund for carrying out this study.

NOMENCLATURE

C_p	= specific heat (J/kgK)
Cr	= heat capacity flow rate ratio
e	= dimensionless exergy loss
E	= exergy loss (W)
T	= temperature (K)
T_0	= ambient temperature (K)
Tr	= inlet temperature ratio
q	= dimensionless heat transfer rate
Q	= heat transfer rate (W)

Subscripts

c	= cold stream
h	= hot stream
m	= mass flow rate (kg/s)

Greek symbol

ϵ	= effectiveness
i	= inlet
o	= outlet

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Table 1 Testing condition of heat exchangers.

Items	Conditions
1. Flow arrangement	Parallel and counter flow
2. Thermosyphon arrangement	
Staggered array	$S_t = 0.053$ m, $S_d = 0.053$ m, $S_l = 0.046$ m
Aligned array	$S_t = 0.053$ m, $S_l = 0.053$ m
3. Number of tube row	7
4. Number of column	7
5. Diameter of thermosyphon (bare tube)	0.027 m
6. Length	
Evaporator section	0.4 m
Condenser section	0.4 m
Adiabatic section	0.2 m
7. Type of fin	Circular fin
8. Size of fin	Fin height = 0.1 m fin pitch = 10 fins/inch
9. Filling ratio of working fluid	50%
10. Material of pipe and fin	Stainless steel 304

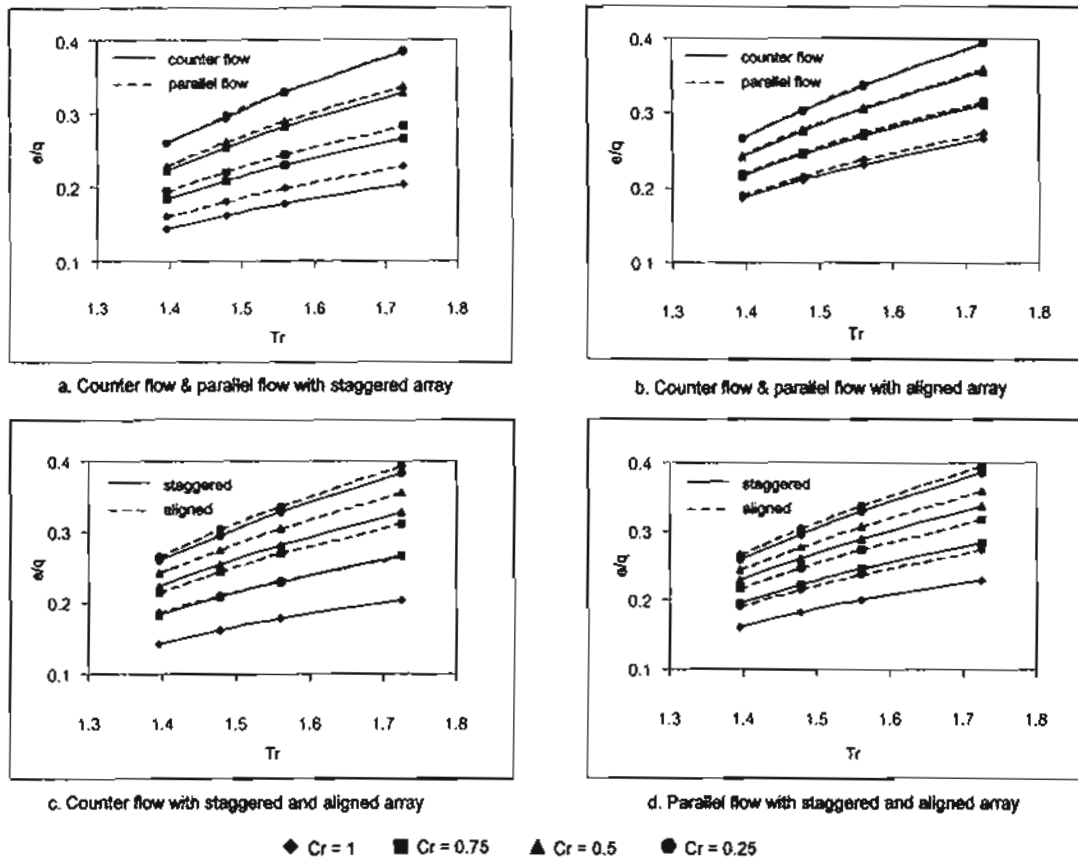


Figure 2 The dimensionless of exergy loss per dimensionless heat transfer rate at various working conditions of the thermosyphon air preheater.

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SECOND LAW ANALYSIS OF A THERMOSYPHON AIR PREHEATER WITH BINARY MIXTURES

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1. INTRODUCTION

A longan dryer is used to removed moisture from longan, an important fruit of Thailand. Normally consumes a lot of thermal energy. To save such energy, a heat exchanger is designed to recover waste heat from the dryer by using the exhaust flue gas from the dryer is the input hot stream which ranges from 60-80°C. The cold stream gas is the ambient air ranging 25-40°C.

A thermosyphon heat exchanger is used to recover the waste heat. The simple working fluid inside is water. However, there is an idea to improve the heat exchanger by using binary mixture and the fluid pairs of ethanol-water and TEG.-water have been selected. In this paper, comparison of the heat exchanger performance of the binary mixtures with the pure water have been carried out and the second law efficiency is the tool to estimate the system.

2. THERMOSYPHON HEAT EXCHANGER

2.1 The Longan Dryer

Longan is an important fruit of Thailand. In order to increase the life of the fruit, and decrease the weight and volume of the fruit for storage and handling they are often dried. A direct dryer is usually used to dry the whole fruit, outer skin and seed included. Ambient air is usually combusted with LPG and the resulting hot gas passed directly over the longans.

In order to conserve energy the flue gas ranges of about 60-80°C from the dryer may be used to pre-heat ambient air of about 25-40°C before it is combusted with LPG.

ABSTRACT Performance of different binary working fluids in a thermosyphon heat exchanger has been considered by evaluating the 2nd law efficiency. The binary working fluids used are ethanol-water and TEG.-water and the heat exchanger is designed as an air preheater of a longan dryer; ambient air is considered as the cold stream and the flue gas from the dryer as the hot stream. The performances have been compares with that of pure water.

It was found that the ethanol-water working fluid resulted in the highest second law efficiency. An efficiency of 0.34 was found at 57 rows for the 19.05 mm. thermosyphon size, and at 90 rows for the 12.7 mm. thermosyphon size at a flue gas temperature of 80°C.

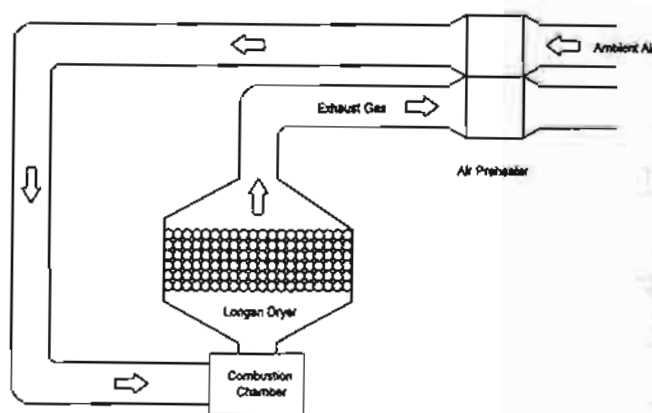


Fig. 1 The longan drying process and air preheater

2.2. Principals of Thermosyphon

A thermosyphon is a heat transfer device that utilizes the high heat transfer ability of boiling and condensation process. A small quantity of working fluid is placed in the bottom of a tube from which air has been evacuated. The tube is then sealed. Heat is applied to the lower end of the tube, causing vapor to form and move up to the cold end where it is condensed. The condensed liquid returns via gravity to the evaporator section. Because the latent heat of vaporization is large considerable quantities of heat may be transported with a small temperature difference. A thermosyphon has high effective thermal conductance.

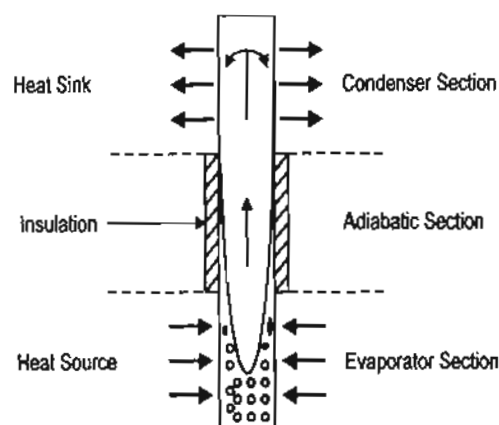


Fig. 2 Thermosyphon

2.3. Internal Resistance in Thermosyphon

In order to determine the amount of heat transferred through a heat pipe, the resistance in the heat pipe must be determined.

The internal resistance of the heat exchanger incorporates the boiling and condensing heat transfer resistance, an axial resistance, vapor/liquid interface resistance and pressure drop resistance and the wall resistance in the evaporator and condenser sections of the thermosyphon.

Experimental data tests were carried out on a single thermosyphon. The thermosyphon had cooling water flowing over the condenser section and the evaporator section was heated by a hot oil bath. The temperature of the oil bath, the evaporator wall, the inlet and outlet cooling water temperatures and the temperature of the condenser wall were measured. These were used to determine the internal resistance of the thermosyphon.

An energy balance over the condenser section enables Q , the heat transferred to be determined.

$$Q = \dot{m}_w C_{pw} \Delta T_w \quad (1)$$

This could then be used to determine internal thermal resistance of heat pipe from

$$R_{in} = \frac{(T_{ev} - T_{cd})}{Q} \quad (2)$$

Where T_{ev} and T_{cd} are surface temperature of heat pipe at evaporator and condenser section respectively.

The relations between internal resistance and log mean temperature difference for ethanol-water, TEG-water and water are shown in equation 3 and table 1.

$$R_{in} = a(\text{LMTD}) + b \quad (3)$$

Table 1 The values of a and b in equation 3

Substant	a	b
water ($d = 12.7$ mm.)	-0.0160	1.5798
ethanol-water ($d = 12.7$ mm.)	-0.0010	0.2195
TEG.-water ($d = 12.7$ mm.)	-0.0100	1.2453
water ($d = 19.05$ mm.)	-0.0047	0.4989
ethanol-water ($d = 19.05$ mm.)	-0.0013	0.1774
TEG.-water ($d = 19.05$ mm.)	-0.0060	0.6353

2.4 Heat Exchanger Configuration

The heat exchanger was assumed to be of a staggered, counter-current configuration. The spacing between thermosyphons was set at 0.023m, in the direction of flow and perpendicular to the direction of flow.

The benefits of a thermosyphon heat exchanger over a the standard heat exchangers is that there is no opportunity for cross-contamination, there are no moving parts required, the design is compact, there is no need for an external energy supply and they have low maintenance and operating costs.

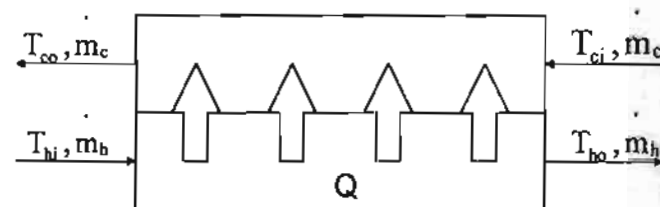


Fig 3 Counter-current Thermosyphon Heat Exchanger Configuration

2.5 Second Law Efficiency

The first law of thermodynamics states that energy must always be conserved. The concept of first law efficiency relates to minimizing energy losses from a process. In the case of a heat pipe the energy losses are minimal and the first law efficiency can be taken as 100%.

The second law of thermodynamics states the entropy of an isolated system cannot decrease. Thus, although energy must always be conserved, a given amount of energy will not always have the same potential for use. Potential for use or availability of a system can be destroyed by irreversibility. Irreversibility of a system may be due to a finite temperature difference, pressure differences and friction. In the case of a heat exchanger the main factor is that of minimizing the temperature difference.

The 2nd law efficiency uses the concept of availability in assessing the effectiveness of energy resource utilization. By comparing the second law efficiency obtained for various designs the most thermodynamically effective use of energy can be determined.

In a counter-current heat exchanger as considered here and assuming the pressure drop over the length of a

heat pipe is negligible, the second law efficiency can be evaluated as

$$\varepsilon = \frac{\dot{m}_c \left(C_{pc} (T_{co} - T_{ci}) - T_{amb} C_{pc} \ln \left(\frac{T_{co}}{T_{ci}} \right) \right)}{\dot{m}_h \left(C_{ph} (T_{hi} - T_{ho}) - T_{amb} C_{ph} \ln \left(\frac{T_{hi}}{T_{ho}} \right) \right)} \quad (4)$$

2.6 External Resistance

The external resistance over the evaporator and condenser are evaluated according to convective heat transfer relations for the flow over a bank of tubes.

$$R_{ev,o} = 1 / (h_{ev,o} A_{ev,o}) \quad (5)$$

$$R_{cd,o} = 1 / (h_{cd,o} A_{cd,o}) \quad (6)$$

The average heat transfer coefficient for the entire tube bundle can be found using the correlation

$$Nu = C_1 Re_m^m \quad (7)$$

Where C_1 and m are empirical constants. This equation is valid for flow over 10 or more rows of tubes.

2.7 Simulation Program

Determination of the 2nd law efficiency was conducted using the Turbo Pascal computer program. A flow chart of input data and calculations is shown in the appendix

3. RESULT AND DISCUSSION

From the simulation program it is found that, as the number of rows was increased the resulting change in outlet temperatures was found to reduce until the addition of another row did not produce a significant change in temperature. At this point the second law efficiency plateaus, and the maximum obtainable second law efficiency is found.

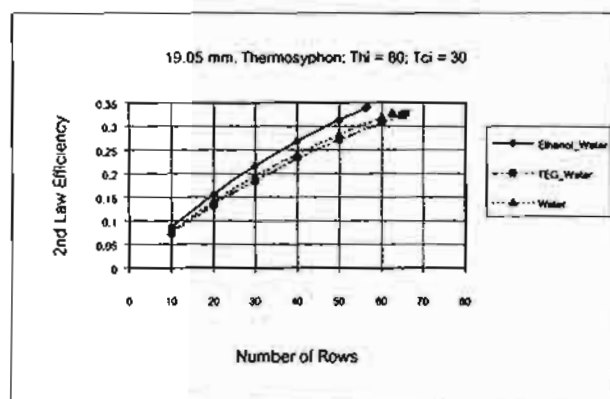
Ethanol-water was found to be the better working fluid in the thermosyphons considered (Refer to Fig. 4). It produced the highest efficiency for the lowest number of rows in both the 19.05 and 12.7 mm. thermosyphon heat exchangers. The efficiency was found to plateau at a value of 0.34 in both cases. The number of rows required was 57 for the 19.05mm. and 89 for the 12.7 mm. at a flue gas temperature of 80° C, and ambient air temperature of 30° C.

The difference in performance between TEG-water and pure water was found to be minimal (Refer to Fig. 4). In addition to this TEG has a high boiling point and under the conditions of operation this component was probably not vaporizing. Hence the TEG-water mix was seen to behave in a similar way to the water.

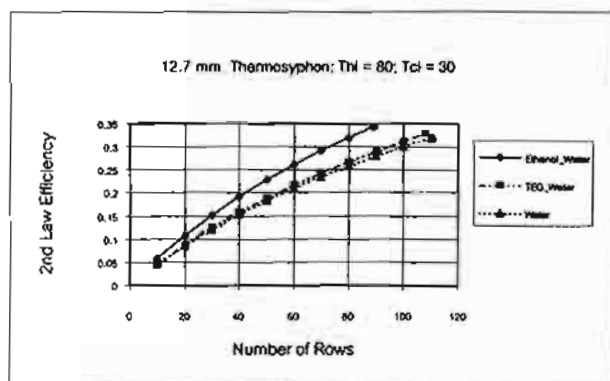
A plot of the temperature profile for ethanol-water heat exchanger and pure water heat exchanger are shown in Fig. 5. From this it can be seen using the ethanol-water

binary mixture reduces the temperature difference between the hot and cold streams. This in turn reduces the irreversibility of the system and increases the second law efficiency.

Because Ethanol has a lower boiling point than water the ethanol water mixture boils at a lower temperature. As the liquid is vaporized the pressure in the tube increases. In the case of ethanol-water more vapor is formed and therefore the pressure in the tube is greater than the pure water thermosyphon. As a result of this the ethanol-water vapor is able to condense at a lower temperature. The effect of this is that the temperature difference are lower for the ethanol-water mixture and hence the 2nd Law efficiency greater.



a



b

Fig. 4 Effect of number of rows on 2nd law efficiency

The two different thermosyphon sizes resulted in large differences in the number of rows required (Refer to Fig. 6). This difference is expected as the heat transfer surface area of the 19.05 mm. thermosyphon is 0.09 m² compared with 0.06 m² for the 12.7 mm. thermosyphon. Therefore a 33% increase in the number of rows can be expected. The choice of a 19.05 mm. thermosyphon is preferable, as capital investment cost would be lower for this sized thermosyphon.

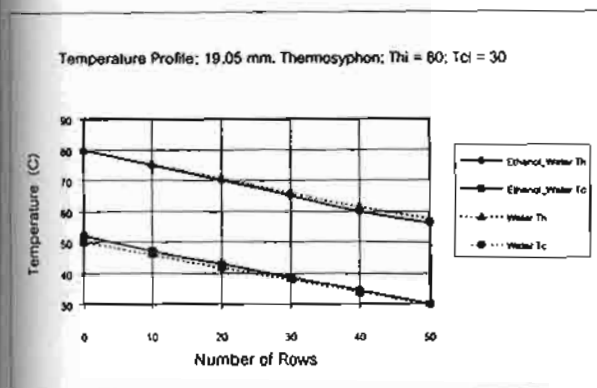


Fig. 5 Temperature profile along the heat exchanger

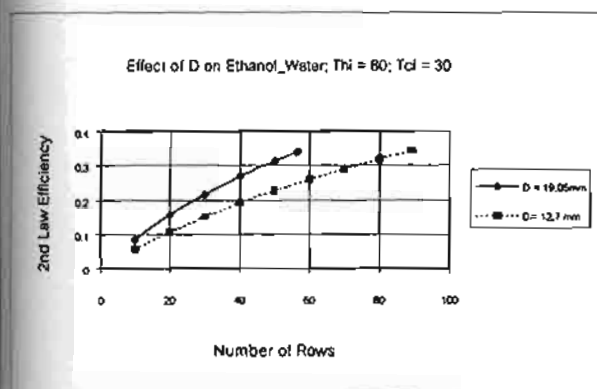


Fig. 6 Effect of diameter on the the 2nd law efficiency

4. CONCLUSION

It was found that the ethanol-water working fluid resulted in the highest second law efficiency. The use of ethanol-water reduced the temperature difference throughout the heat exchanger compared to the other working fluids; a reduction in the temperature difference corresponds to a decrease in the irreversibility and an increase in the 2nd law efficiency. An efficiency of 0.34 was found at 57 rows for the 19.05 mm. thermosyphon size, at a flue gas temperature of 80°C and ambient air temperature of 30°C.

5. ACKNOWLEDGMENT

The Authors gratefully acknowledge the partial support provided by the Thailand Research Fund for carrying out this study.

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7. NOMENCLATURES

A	=	Area (m ²)
C _p	=	Specific heat (J/kg-K)
LMTD	=	Log mean temperature difference
m'	=	Mass flow rate (kg/s)
Q	=	Heat transfer rate (W)
R	=	Thermal resistant (K/W)
Re	=	Reynold's Number
T	=	Temperature (°C)
Subscript		
amb	=	Ambient
c	=	Cold
cd	=	Condenser
ev	=	Evaporator
h	=	Hot
i	=	Inlet
in	=	Inside
o	=	Outlet
w	=	Water
Greek Symbol		
ε	=	2 nd Law Efficiency

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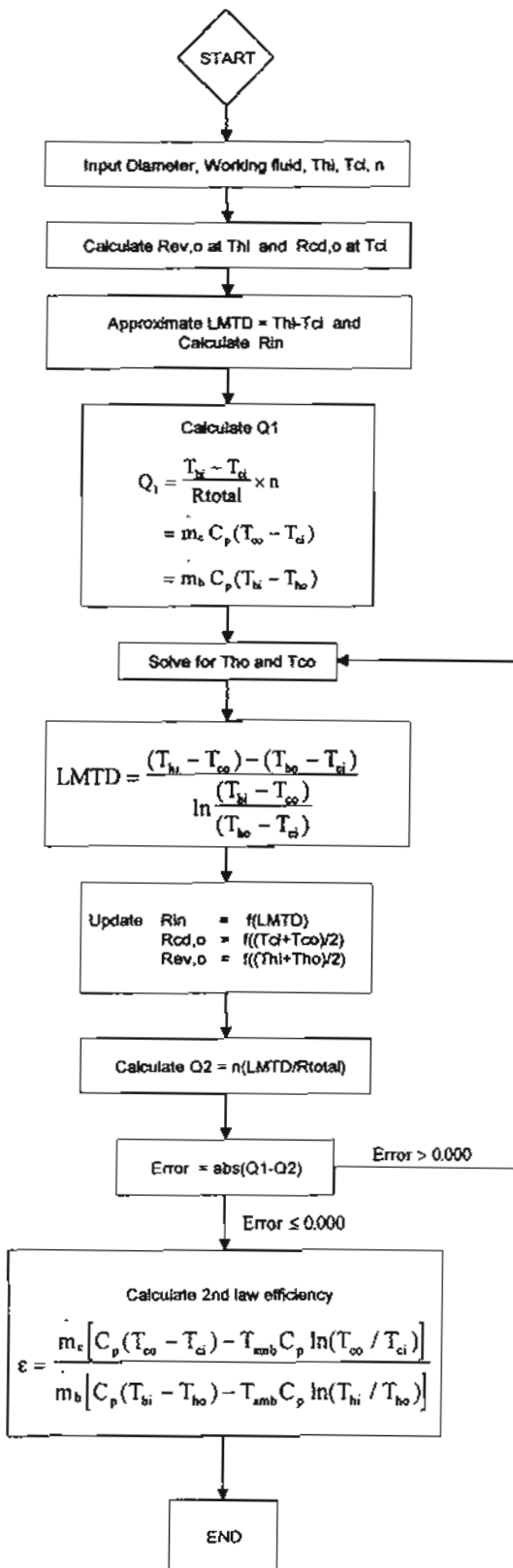
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APPENDIX



Simulation flowchart

Nuntaphan A., Tiansuwan J. and Kiatsiriroat T., Heat Transfer Coefficients of Thermosyphon Heat Pipe at Medium Operating Temperature, **The 1st Regional Conference on Energy Technology Toward a Clean Environment**, 1-2 December 2000, Chiang Mai, Thailand.

Heat Transfer Coefficients of Thermosyphon Heat Pipe at Medium Operating Temperature

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Abstract

This research work studied the heat transfer coefficient of thermosyphon heat pipe at medium operating temperature. The pipe is 100 cm stainless steel tube with 40 cm evaporator and condenser sections and 20 cm for adiabatic section. Hot oil bath and cooling water are used as heat source and heat sink respectively. Nine points of thermocouples are inserted into the pipe for measuring inside temperatures along the thermosyphon and the other nine points also for measuring the surface temperature at the same positions. Water, ethanol, methanol and acetone are used as working fluids with 50% filling ratio. The hot oil temperature is varied from 40-120°C and the temperature of cooling water is kept at 30°C at constant flow rate. The measured inlet and outlet temperatures of cooling water are used to calculate the heat transfer rate of the thermosyphon.

From the experiments it is found that when the hot oil temperature is less than 60°C, the heat transfer rate of water is lower than those of other working fluids. However at higher temperature heat source, the heat transfer rate of water is the best. Boiling Equation of Rohsenow is modified to predict the boiling heat transfer coefficient of the thermosyphon at various types of working fluids. For the condenser section, it is found that modified Nusselt correlation of condensation agrees well with the experiments.

Introduction

The thermosyphon heat pipe, one type of heat exchanger, at present, plays an important role in the energy conservation program because of its good advantages such as high thermal conductivity, low operating cost and easy to construct. Figure 1 shows the schematic picture of the operating thermosyphon. When heat is added to the evaporator section, the working fluid inside the thermosyphon is boiled, vaporized and then the vapor flows to the condenser section where the condensation process occurs and rejects heat to the outside of the thermosyphon.

Because the heat transfer of the thermosyphon deals with boiling and condensation inside a tube, it is important to know the heat transfer coefficient of both processes. Many researches have been done to find the correlations to predict these phenomena. ESDU [1] collected many data and correlations of both boiling and condensation inside the thermosyphon and proposed these new correlations in term of the thermal resistance as follows:

Boiling

$$Z_b = Z_{pool}F + Z_{film}(1 - F), \quad [1]$$

where

$$Z_{film} = \frac{0.235Q^{1/3}}{D^{4/3}g^{1/3}L_e\phi_1^{4/3}}, \quad Z_{pool} = \frac{1}{\phi_2g^{0.2}Q^{0.4}(\pi DL_e)^{0.6}},$$

$$\phi_1 = \left(\frac{\lambda k_f \rho_f^2}{\mu_f} \right)^{0.25}, \quad \phi_2 = 0.32 \frac{\rho_f^{0.65} k_f^{0.3} C_{pf}^{0.7}}{\rho_g^{0.25} \lambda^{0.4} \mu_f^{0.1}} \left[\frac{P_v}{P_a} \right]^{0.23},$$

Condensation

$$Z_{cd} = \frac{0.235 Q^{1/3}}{D^{4/3} g^{1/3} L_{cd} \phi_1^{4/3}} \quad [2].$$

Although the correlation of ESDU can be used with in high operating range and various type of working fluid, however in some conditions, especially at low operating temperature, the use of ESDU correlations may give some errors. Shiraishi et al [2] found that at low operating temperature (32-60°C) equations 1 and 2 can predict the heat transfer coefficient of the thermosyphon within 70-150% of the experimental values. Hahne and Gross [3] found that at a working temperature between 37-76°C, the correlations of ESDU give a lower prediction of the heat transfer coefficient (31-76% for boiling and 46-86% for condensation).

At present there are attempt to use the thermosyphon to recover waste heat from agricultural industries in Thailand. The heat source is normally lower than 80°C. Therefore using correlations of ESDU may be give some error for predicting the heat transfer coefficient. In this research, correlations of boiling and condensation heat transfer coefficients have been reconstructed to predict the heat transfer rate of the thermosyphon heat exchanger at medium operating temperature with some common working fluids such as water, ethanol, methanol and acetone.

Materials and Methods

1. Experimental Setup

Figure 2 shows the schematic diagram of the experimental setup. Stainless 304 with 27 mm diameter, 2 mm thickness and 1 m length has been used as the thermosyphon heat pipe. This tube can be divided into 3 parts, 40 cm for evaporator, 20 cm for adiabatic section and 40 cm for condenser section. Nine K-type thermocouples are inserted into the thermosyphon for measuring inside temperatures at the length 5, 15, 25, 35, 50, 65, 75, 85 and 95 cm from the evaporator end. The other nine thermocouples are attached at the outside surface of the thermosyphon at the same positions of the inside measurement. The hot paraffin oil bath controlled by the electric heater is used as the heat source of the thermosyphon and the injection of bubble air into the hot oil is also used to make well mix of hot oil temperature. At the condenser section, there is a stream of cooling water to remove heat out of the thermosyphon. The flow rate of cooling water is kept constant at 0.0057 kg/s by the constant head tank at normal temperature (approximately 30°C). The inlet and outlet temperatures of cooling water are measured by the thermocouples to calculate the heat transfer rate of the thermosyphon.

Four types of common working fluids, water, ethanol, methanol and acetone, have been used with 50% filling ratio of the evaporator section.

2. Analysis

The heat transfer rate of the thermosyphon can be calculated as

$$Q = \dot{m}_w C_{pw} (T_{wo} - T_{wi}) \quad [3].$$

The boiling and condensation heat transfer coefficient can be calculated as

$$h_b = \frac{Q}{A_{ei} \Delta T_e} \quad [4]$$

$$h_{cd} = \frac{Q}{A_{cd} \Delta T_{cd}} \quad [5]$$

where h_b and h_{cd} are the heat transfer coefficient of the evaporator and condenser respectively.

Rohsenow equation [4] is modified to predict the boiling heat transfer coefficient of the thermosyphon by find the new empirical constant as follow

$$Nu = C_1 Re^{C_2} Pr^{C_3} \quad , \quad [6]$$

where

$$Nu = \frac{h_b}{k_l} \left[\frac{\sigma}{g(\rho_l - \rho_g)} \right]^{1/2}, \quad Re = \frac{q}{\lambda \rho_l} \left[\frac{\sigma}{g(\rho_l - \rho_g)} \right]^{1/2} \frac{\rho_l}{\mu_l},$$

$$Pr = \frac{C_p \mu_l}{k_l}.$$

In this work factor C_3 was set as 0.40 [5].

For the condenser section, the modified heat transfer coefficient of Nusselt [4] has been used to calculate the heat transfer coefficient as

$$h_{cd} = C_4 \left(\frac{\rho_l (\rho_l - \rho_g) g k^3 \lambda}{\mu_l L_{cd} \Delta T_{cd}} \right)^{C_5} \quad [7]$$

ΔT_{cd} is the temperature difference between inside surface and inside temperature of the condenser section.

Results and Discussion

Figure 3 shows the heat transfer rate of the thermosyphon heat pipe, using water, ethanol, methanol and acetone at various temperatures of hot oil. From this Figure it is found that when the temperature is lower than 60°C, the heat transfer rate of water is lower than that of ethanol, methanol and acetone and the heat transfer rate of ethanol, methanol and acetone is slightly difference. These phenomena come from the higher boiling point of water and its difficulty to boil at this condition. However when the temperature is higher than this point, water can boil and the heat transfer rate of water higher than those of other working fluids because of high latent heat of the vaporization of water. The heat transfer rate of methanol and ethanol are the same at this range since the physical properties of these working fluids are slightly difference. However the heat transfer rate of acetone is lower than the other working fluids because of its very low latent heat of vaporization.

Figure 4 shows the comparison between the experimental and calculated data of the Nusselt number of boiling modified from the Rohsenow equation. It was found that the new correlation, with the correlation coefficient shown in Table 1 can predict 96.3% of the experimental value with in $\pm 20\%$. Figures 5a and b show the comparison of Nusselt number of boiling of water and ethanol at various temperature of working fluid with the previous work [6]. It is found that, for water, when the temperature of working fluid lower than 50°C, the Nusselt number drop drastically with the reduction of working fluid temperature. However when the temperature more than 50°C the Nusselt number is comparatively constancy since the nucleate boiling of water has occurred. In case of boiling of ethanol inside the thermosyphon, it is found that the result from the present correlation agree well with Kiatsiriroat et al. correlation [6] because at this temperature range the ethanol can be boiled easily.

Table 1 The correlation coefficient C_1 , C_2 , C_4 and C_5

Working Fluids	C_1	C_2	C_4	C_5
Water	41.27	0.37	0.94	0.245
Ethanol	14.64	0.15	0.94	0.275
Methanol	24.09	0.20	0.96	0.270
Acetone	27.44	0.22	0.95	0.280

For the condensation heat transfer coefficient it is found that modified Nusselt equation of condensation can predicts 73.6% of the experimental data within $\pm 30\%$ and Figure 6 also shows this results. Moreover it is found that the correlation constants C_4 and C_5 shown in Table 1 are slightly difference from the former Nusselt correlation, which equal to 0.943 and 0.25 respectively.

Conclusions

From the experiment it can be concluded that;

1. When the temperature of hot oil is lower than 60°C the heat transfer rate of water is lower than those of the other working fluids because of higher boiling point of water.
2. As the temperature of hot oil is over 60°C the heat transfer rate of water is the best because of its very high latent heat of vaporization.
3. Modified boiling equation of Rohsenow can predict 96.3% of the Nusselt number of boiling inside the thermosyphon with in $\pm 20\%$.
4. Modified Nusselt equation of condensation can predict 73.68% of the condensation heat transfer coefficient inside the thermosyphon with in $\pm 30\%$.

Acknowledgement

The authors gratefully acknowledge the support provided by Thailand Research Fund for carrying out this study.

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Nomenclatures

A	area, m^2
C_{1-5}	empirical constants
C_p	specific heat, $\text{J/kg}^\circ\text{C}$
D	diameter of tube, m
F	filling ratio of working fluid
g	gravitational acceleration, 9.81 m/s^2
h	heat transfer coefficient, $\text{W/m}^2^\circ\text{C}$
k	thermal conductivity, $\text{W/m}^\circ\text{C}$
L	length, m
$\dot{m}$	mass flow rate, kg/s
Nu	Nusselt number
P	pressure, Pa
Pr	Prandtl number
q	heat flux, W/m^2
Q	heat transfer rate, W
Re	Reynolds number
ΔT_{cd}	temperature difference between inside surface and liquid at condenser section, $^\circ\text{C}$
ΔT_e	temperature difference between inside surface and liquid at evaporator section, $^\circ\text{C}$

T	temperature ($^\circ\text{C}$)
Z	thermal resistance, $^\circ\text{C/W}$
Greek symbols	
ϕ	function
λ	latent heat of vaporization, J/kg
μ	dynamic viscosity, Pa s
ρ	density, kg/m^3
σ	surface tension, N/m
Subscripts	
a	ambient
b	boiling
cd	condenser, condensation
e	evaporator
$film$	film boiling
g	gas
i	inlet, inside
l	liquid
o	outlet, outside
$pool$	pool boiling
s	surface
v	vapour
w	water

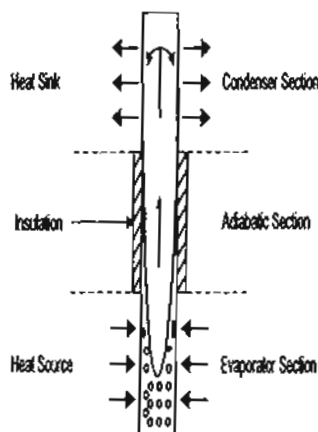


Figure 1 The thermosyphon heat pipe.

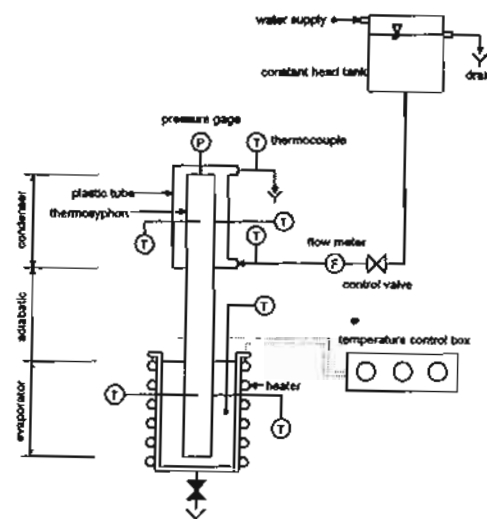


Figure 2 The experimental setup.

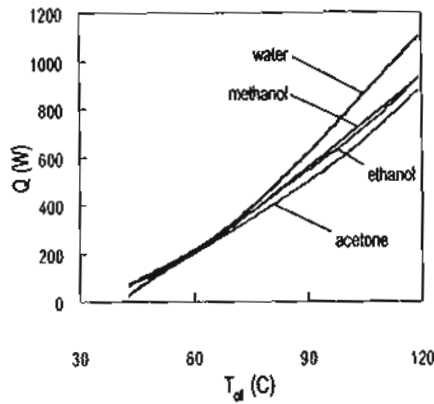


Figure 3 Variation of heat transfer rate of various kinds of working fluids with hot oil temperature.

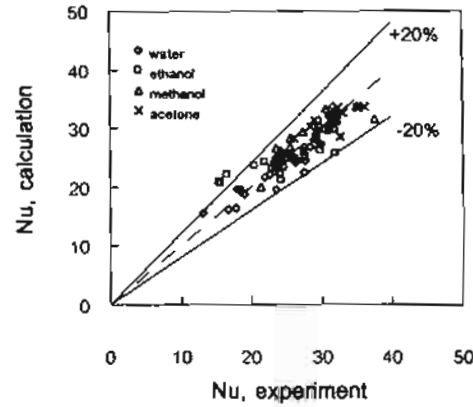
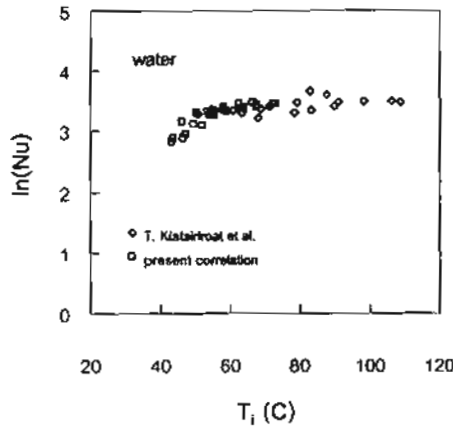
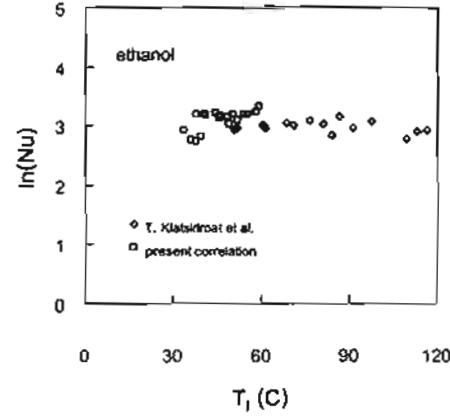


Figure 4 Comparison of Nusselt number of boiling inside the thermosyphon between the experimental and calculation data.



a. water



b. ethanol

Figure 5 Comparison of Nusselt number of boiling of water and ethanol between present correlation and Kiatsiriroat et al. correlation.

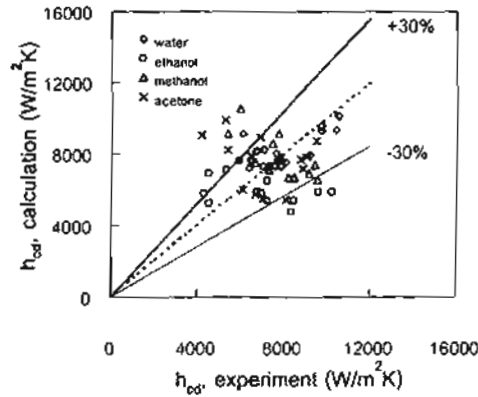


Figure 6 Comparison of condensation heat transfer coefficients inside the thermosyphon between the experimental and calculation data.

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การวิเคราะห์พฤติกรรมทางความร้อนของท่อความร้อนแบบเทอร์โมไซฟอน ที่ใช้สารทำงานคู่ผสม

Thermal Behavior Analysis of Thermosyphon Heat Pipe Using Binary Working Fluids

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บทคัดย่อ

งานวิจัยนี้ ศึกษาพฤติกรรมทางความร้อนของท่อความร้อนแบบเทอร์โมไซฟอนที่ใช้สารทำงานคู่ผสม เอทานอล-น้ำ และไตรเอทิลีนไกลคอล-น้ำ โดยทำการศึกษาลักษณะของพารามิเตอร์ต่างๆ ที่เกี่ยวข้อง กล่าวคือ อัตราส่วนผสมของสารทำงาน อัตราส่วน (อัตราส่วนความยาวของส่วนระเหยต่อขนาดเส้นผ่านศูนย์กลางของท่อความร้อน : L/d) ของท่อความร้อน และช่วงอุณหภูมิการทำงาน

จากการทดลองพบว่า อัตราการถ่ายเทความร้อนของท่อความร้อนที่ใช้ เอทานอล-น้ำเป็นสารทำงาน ในกรณีที่อุณหภูมิแตกต่างเชิงลักระหว่างน้ำมันร้อนและน้ำที่ถ่ายเทความร้อนมีค่าต่ำ (ไม่เกิน 100°C) อัตราการถ่ายเทความร้อนที่ได้จะมีค่าสูงกว่าท่อความร้อนที่ใช้ น้ำเป็นสารทำงาน และมีค่าใกล้เคียงกับอัตราการถ่ายเทความร้อนของท่อความร้อนที่ใช้เอทานอลเป็นสารทำงาน และในกรณีที่อุณหภูมิแตกต่างเชิงลักระหว่าง (สูงกว่า 100°C) อัตราการถ่ายเทความร้อนของท่อความร้อนที่ใช้ น้ำเป็นสารทำงานจะมีค่าสูงกว่าในกรณีที่ใช้สารทำงานคู่ผสม เอทานอล-น้ำ

ในกรณีของท่อความร้อนที่มีค่าอัตราส่วนสั้นเกินไป และช่วงอุณหภูมิการทำงานสูง พบว่ามักจะเกิดการท่วม (flooding) ของสารทำงานขึ้นภายในท่อความร้อน การผสมไตรเอทิลีนไกลคอลในน้ำประมาณ 25% จะช่วยลดภาวะวิกฤติดังกล่าวได้ในขณะที่อัตราการถ่ายเทความร้อนมีค่าลดลงเล็กน้อย

Abstract

Thermal behavior of a thermosyphon heat pipe using binary working fluids, ethanol-water and triethyleneglycol(TEG)-water,

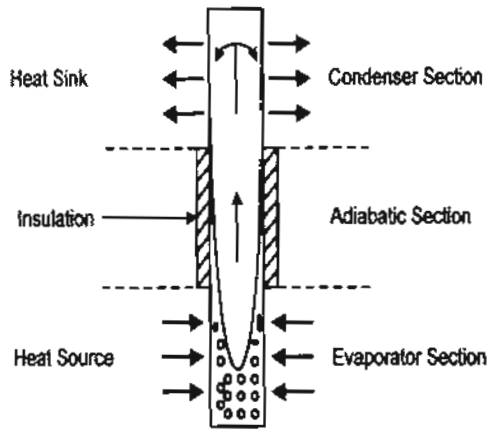
and the affecting parameters such as the mixtures content, the aspect ratio of pipe and the working temperature had been studied in this research work.

From the experimental, at low log mean temperature difference (lower than 100°C) of the hot oil and cold fluid stream, it is found that the thermosyphon using ethanol-water as a working fluid has higher heat transfer rate than that of pure water and close to pure ethanol. But at high log mean temperature difference (more than 100°C), the heat transfer rate of water is the best.

In case of high aspect ratio of the pipe and high working temperature, the working fluid inside the pipe is always flooded. Low concentration of TEG in water (around 25%) can decrease this critical phenomenon, while the heat transfer rate is slightly reduced.

1. บทนำ

ท่อความร้อนแบบเทอร์โมไซฟอน (Thermosyphon Heat Pipe) เป็นอุปกรณ์ในการส่งถ่ายความร้อนจากแหล่งความร้อนสูงไปยังแหล่งความร้อนต่ำ โดยอาศัยหลักการเดือดและการควบแน่นของสารทำงานในท่อความร้อน หลักการทำงานของท่อความร้อนแบบเทอร์โมไซฟอน แสดงดังรูปที่ 1



รูปที่ 1 หลักการทำงานของท่อความร้อนแบบเทอร์โมไซฟอน

ท่อความร้อนแบบเทอร์โมไซฟอนสามารถแบ่งออกเป็น 3 ส่วนคือ ส่วนระเหย (Evaporator Section) ส่วนที่ไม่มีการถ่ายเทความร้อน (Adiabatic Section) และส่วนควบแน่น (Condenser Section) โดยที่แต่ละส่วนมีหน้าที่ดังต่อไปนี้

ส่วนระเหย จะมีหน้าที่ดึงเอาความร้อนออกจากแหล่งความร้อนสูง (Heat Source) โดยสารทำงานที่อยู่ภายใน เมื่อได้รับความร้อนจะเกิดการเดือดเป็นไอและไหลขึ้นด้านบนไปยังส่วนควบแน่น โดยส่วนระเหยนี้ในท่อความร้อนแบบเทอร์โมไซฟอน จะเป็นส่วนที่อยู่ต่ำที่สุด

ส่วนที่ไม่มีการถ่ายเทความร้อน ส่วนนี้จะมีหน้าที่ในการส่งผ่านของไหลระหว่างส่วนระเหยและส่วนควบแน่น โดยทั่วไปแล้วส่วนนี้จะมีการหุ้มฉนวน เพื่อป้องกันความร้อนถ่ายเทออกนอกท่อความร้อน ในบางครั้งส่วนนี้อาจจะไม่มียกได้ ส่วนมากแล้วส่วนนี้จะมีในท่อความร้อนที่ส่งผ่านความร้อนระยะทางไกลๆ

ส่วนควบแน่น จะอยู่ปลายด้านบนสุดในท่อความร้อนแบบเทอร์โมไซฟอน โดยมีหน้าที่ถ่ายเทความร้อนออกจากท่อความร้อนให้กับของไหลที่มารับความร้อน (Heat Sink) โดยการควบแน่น และสารทำงานที่ควบแน่นแล้วจะไหลลงสู่ด้านล่างเพื่อรับเอาความร้อนจากส่วนระเหย และเกิดการไหลเวียนเป็นวัฏจักรต่อไป

ท่อความร้อนแบบเทอร์โมไซฟอน โดยทั่วไปแล้วจะใช้สารทำงานเดี่ยว เช่น น้ำ, เอทานอล, สารทำความเย็นต่างๆ เป็นต้น ซึ่งในการเลือกใช้สารดังกล่าวต้องให้เหมาะสมกับช่วงอุณหภูมิใช้งานของท่อความร้อนนั้นๆ ท่อความร้อนที่ใช้ในการดึงความร้อนทั้งกลับมาใช้ประโยชน์ จากกระบวนการทางอุตสาหกรรมต่างๆ นิยมใช้น้ำเป็นสารทำงาน เนื่องจากน้ำมีค่าความร้อนแฝงของการกลายเป็นไอสูงและ สามารถใช้งานได้ในช่วงอุณหภูมิของแหล่งความร้อนที่กว้าง (60-250°C) แต่อย่างไรก็ตามถ้าหากใช้สารคู่ผสม (Binary Mixture) มาใช้ในท่อความร้อนเช่น เอทานอล-น้ำ เมื่อให้ความร้อนแก่ส่วนระเหยของท่อความร้อน พบว่าสารระเหยง่ายซึ่งในที่นี้คือเอทานอล จะเกิดการเดือดกลายเป็นไอก่อนน้ำ ส่วนน้ำจะเดือดตามมาเมื่ออุณหภูมิเพิ่มสูงขึ้นจนถึงจุดเดือดของมัน ทำให้ช่วงอุณหภูมิการทำงานต่างๆ นั้นอัตราการถ่ายเทความร้อนของท่อความร้อนที่ใช้สารทำงานคู่ผสมเอทานอล-น้ำ มีค่าสูงกว่าในกรณีที่ใช้น้ำเป็นสารเดี่ยวในการทำงาน และในช่วงอุณหภูมิสูงที่สุดส่วนผสมบางค่า

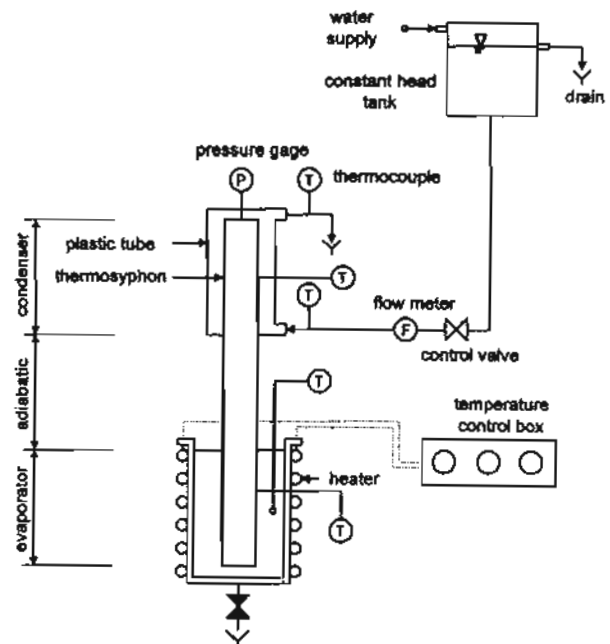
สมรรถนะในการถ่ายเทความร้อนไม่แตกต่างกันมากนักเมื่อเทียบกับการใช้น้ำเพียงอย่างเดียว

จากแนวคิดดังกล่าวข้างต้นเป็นที่มาของงานวิจัยนี้ ซึ่งการใช้สารคู่ผสมในท่อความร้อนแบบเทอร์โมไซฟอน จะช่วยทำให้สมรรถนะของท่อความร้อนดังกล่าวมีค่าสูงขึ้น เนื่องจากเหตุผลดังที่กล่าวมาข้างต้น ในโครงการวิจัยนี้ จะศึกษาถึงพฤติกรรมทางความร้อนของท่อความร้อนแบบเทอร์โมไซฟอนที่ใช้สารคู่ผสมโดยจะศึกษาตัวอย่างของสารคู่ผสม คือเอทานอล-น้ำ ซึ่งเอทานอลมีจุดเดือดต่ำกว่าน้ำ (ประมาณ 20°C) และ ไตรเอธิลีนไกลคอล-น้ำ โดย ไตรเอธิลีนไกลคอลมีจุดเดือดสูงกว่าน้ำมาก (ประมาณ 170°C)

2. อุปกรณ์และวิธีการดำเนินการวิจัย

2.1 อุปกรณ์ในการวิจัย

งานวิจัยนี้จะทำการศึกษาการถ่ายเทความร้อนของท่อความร้อนแบบเทอร์โมไซฟอน ที่ใช้สารทำงานผสม โดยถ่ายเทความร้อนจากแหล่งอุณหภูมิสูง ซึ่งในที่นี้คือ อ่างน้ำมันร้อน ไปยังแหล่งอุณหภูมิต่ำ ซึ่งก็คือ น้ำหล่อเย็น ลักษณะของอุปกรณ์ในการวิจัยแสดงดังรูปที่ 2 ซึ่งมีส่วนประกอบที่สำคัญประกอบด้วย ท่อความร้อน, อ่างน้ำมันร้อน, ระบบน้ำหล่อเย็น และ อุปกรณ์การวัดและควบคุมอื่นๆ



รูปที่ 2 อุปกรณ์ในการวิจัย

ท่อความร้อนแบบเทอร์โมไซฟอน ทำจากท่อสแตนเลส 304 ขนาดเส้นผ่านศูนย์กลาง 25.4 และ 12.7 mm. หนา 2.0 mm. ความยาวรวม 100 cm. โดยส่วนระเหยยาว 40 cm. ส่วนควบแน่นยาว 40 cm. ส่วนอะเดียบัติกยาว 20 cm.

อ่างน้ำมันร้อน ประกอบด้วยขดลวดความร้อนขนาด 3000 W ควบคุมด้วยชุดปรับอุณหภูมิทางไฟฟ้า ติดตั้งเข้ากับภาชนะทรงกระบอกขนาดเส้นผ่านศูนย์กลาง 7 cm. ยาว 50 cm. และบรรจุด้วยน้ำ

มันพาราฟิน ซึ่งทำหน้าที่เป็นตัวกลางในการถ่ายเทความร้อนระหว่างขดลวดความร้อนและส่วนระเหยของท่อความร้อน อุณหภูมิของน้ำมันร้อนสามารถปรับค่าได้อยู่ระหว่าง 50-225 °C

ระบบน้ำหล่อเย็นของส่วนควบแน่น จะประกอบด้วยท่อโลหะทรงกระบอกขนาดเส้นผ่านศูนย์กลาง 5.08 cm. ยาว 40 cm. สวมทับส่วนควบแน่นของท่อความร้อน และหุ้มฉนวนภายนอกเพื่อป้องกันความร้อนถ่ายเทสู่สิ่งแวดล้อม น้ำหล่อเย็นจะถูกควบคุมอัตราการไหลโดยการใช้ถังน้ำที่มีความสูงคงที่

อุปกรณ์การวัดอุณหภูมิ จะประกอบด้วยเทอร์โมคัปเปิลชนิด K โดยทำการวัดอุณหภูมิที่ตำแหน่งต่างๆ คือ วัดอุณหภูมิผิวท่อความร้อนที่ระยะ 5,20,35,50,65,80 และ 95 cm. จากปลายของส่วนระเหย วัดอุณหภูมิน้ำมันพาราฟิน วัดอุณหภูมิน้ำหล่อเย็นขาเข้าและขาออกจากส่วนควบแน่นของท่อความร้อน

อุปกรณ์การวัดความดัน จะใช้เกจความดัน วัดความดันภายในท่อความร้อนที่ด้านปลายของส่วนควบแน่น

2.2 วิธีการวิจัย

ทดสอบการถ่ายเทความร้อนของท่อความร้อนเมื่อใช้สารทำงานได้แก่ น้ำ, เอทานอล และ ไตรเอธิลีนไกลคอล ที่อัตราส่วนการเติม 50% ของส่วนระเหย และสารผสมได้แก่ เอทานอล-น้ำ และ ไตรเอธิลีนไกลคอล-น้ำ ที่อัตราส่วนผสมของ เอทานอล และไตรเอธิลีนไกลคอล 25, 50, 75% โดยปริมาตร ที่อัตราส่วนการเติม 50% โดยปริมาตรของส่วนระเหยโดยค่าอุณหภูมิของน้ำมันพาราฟินจะอยู่ ระหว่าง 50-225°C และควบคุมอัตราการไหลของน้ำหล่อเย็นที่อุณหภูมิห้องให้คงที่ที่ 8.5 g/s

ค่าความดันและอุณหภูมิที่จุดต่างๆ จะบันทึกที่สภาวะคงตัว เพื่อทำการศึกษาพฤติกรรมทางความร้อนของท่อความร้อน

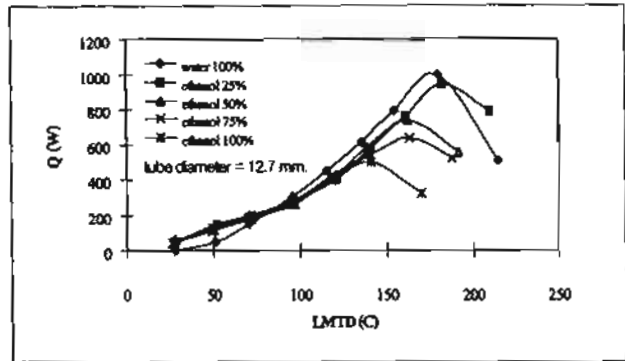
3. ผลการทดสอบและการวิเคราะห์

3.1 อัตราการถ่ายเทความร้อนของท่อความร้อน

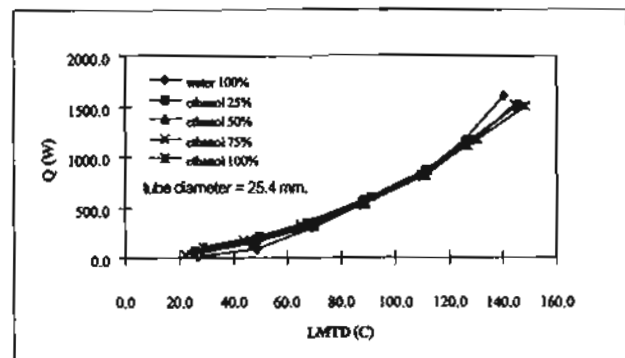
ความสัมพันธ์ระหว่าง อัตราส่วนการถ่ายเทความร้อนของท่อความร้อน กับอุณหภูมิแตกต่างเชิงลอการิทึมระหว่างอุณหภูมิน้ำมันและน้ำที่มารับความร้อน ของท่อความร้อนแบบเทอร์โมไซฟอนที่ใช้ น้ำ เอทานอล ไตรเอธิลีนไกลคอล เอทานอล-น้ำ และ ไตรเอธิลีนไกลคอล-น้ำ เป็นสารทำงานแสดงดังรูปที่ 3 ก,ข,ค,ง โดยที่นิยามของค่าอุณหภูมิแตกต่างเชิงลอการิทึม ในงานวิจัยนี้ ได้นิยามไว้ดังต่อไปนี้

$$\Delta T_{\text{lmtd}} = \frac{(T_{\text{wo}} - T_{\text{wi}})}{\ln \left(\frac{T_{\text{oil}} - T_{\text{wi}}}{T_{\text{oil}} - T_{\text{wo}}} \right)} \quad (1)$$

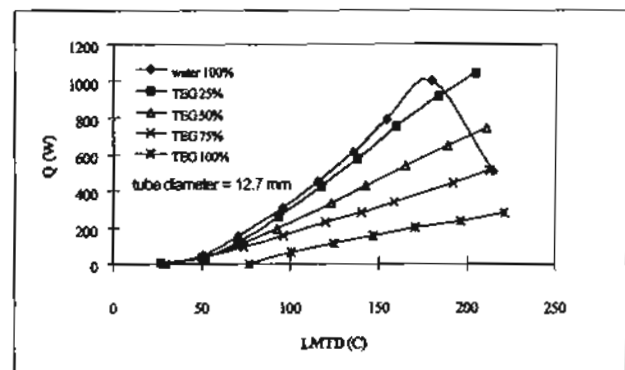
T_{wi} และ T_{wo} คืออุณหภูมิของน้ำหล่อเย็นเข้าและออกที่มารับความร้อนที่คอนเดนเซอร์ และ T_{oil} คืออุณหภูมิน้ำมันที่ส่วนระเหยของท่อความร้อน



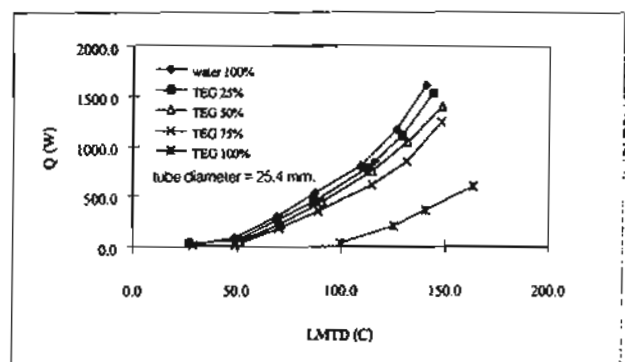
3ก. คู่มือสารทำงาน เอทานอล-น้ำ ที่สัดส่วนต่างๆ ขนาดท่อ 12.7 mm.



3ข. คู่มือสารทำงาน เอทานอล-น้ำ ที่สัดส่วนต่างๆ ขนาดท่อ 25.4 mm.



3ค. คู่มือสารทำงาน TEG-น้ำ ที่สัดส่วนต่างๆ ขนาดท่อ 12.7 mm.



3ง. คู่มือสารทำงาน TEG-น้ำ ที่สัดส่วนต่างๆ ขนาดท่อ 25.4 mm.

รูปที่ 3 ความสัมพันธ์ของค่าอัตราการถ่ายเทความร้อน และค่าความแตกต่างอุณหภูมิเชิงลอการิทึมต่างๆ ของท่อความร้อน

จากการทดสอบพบว่า ในกรณีของท่อความร้อนที่ใช้สารทำงานคู่ผสม เอทานอล-น้ำ ในช่วงอุณหภูมิการทำงานต่ำ อัตราการถ่ายเทความร้อนที่ได้จะมีค่าสูงกว่า ท่อความร้อนที่ใช้เป็นสารทำงาน และมีค่าใกล้เคียงกับท่อความร้อนที่ใช้เอทานอลเป็นสารทำงาน ในขณะที่ในช่วงอุณหภูมิการทำงานสูง อัตราการถ่ายเทความร้อนของท่อความร้อนที่ใช้เป็นสารทำงานจะมีค่าสูงกว่ากรณีที่ใช้สารทำงานคู่ผสม ดังรูปที่ 3 ก และ ข ปรากฏการณ์ดังกล่าวเกิดขึ้นเนื่องจากการที่อุณหภูมิจุดเดือดของเอทานอลมีค่าต่ำกว่าน้ำทำให้ในช่วงอุณหภูมิการทำงานต่ำ เอทานอลในสารทำงานคู่ผสมสามารถเดือด และส่งผ่านความร้อนได้ ในขณะที่น้ำยังไม่เริ่มเดือด ในกรณีช่วงอุณหภูมิการทำงานสูง น้ำสามารถเดือดและส่งผ่านความร้อนได้ดีกว่ากรณีใช้สารทำงานคู่ผสม หรือเอทานอล ทั้งนี้เนื่องจากค่าความร้อนแฝงของการกลายเป็นไอของน้ำมีค่าสูงกว่า เอทานอล

นอกจากนี้ยังพบว่า ในกรณีท่อความร้อนขนาดเส้นผ่านศูนย์กลาง 12.7 mm. ที่ช่วงอุณหภูมิการทำงานสูงจะเกิดภาวะวิกฤติเนื่องจากการท่วม (Flooding) ขึ้นภายในท่อความร้อน โดยอัตราการถ่ายเทความร้อนสูงสุดจะแปรผกผันกับปริมาณเอทานอลในสารทำงานคู่ผสม ภาวะวิกฤติดังกล่าวเกิดขึ้นเนื่องจากอัตราส่วนสนทรรณ์ (อัตราส่วนความยาวของส่วนระเหยต่อขนาดเส้นผ่านศูนย์กลางของท่อความร้อน : L/d) ไม่เหมาะสม

ในการใช้สารทำงานคู่ผสม ไตรเอธิลีนไกลคอล-น้ำ ในท่อความร้อน จากรูปที่ 3 ค และ ง พบว่าการส่งผ่านความร้อนของท่อความร้อนจะแปรผกผันกับปริมาณของไตรเอธิลีนไกลคอล และยังพบว่า ภาวะวิกฤติเนื่องจากการท่วมภายในท่อความร้อนขนาดเส้นผ่านศูนย์กลาง 12.7 mm. จะไม่เกิดขึ้น

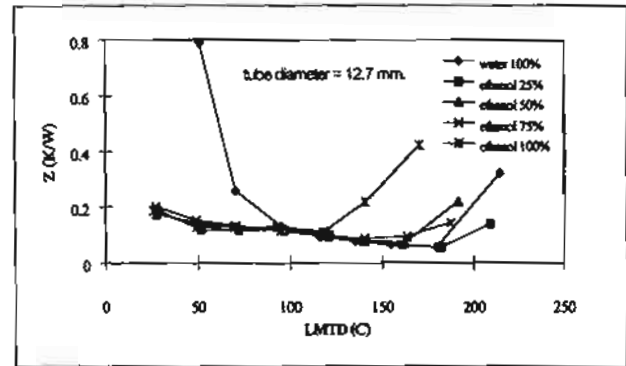
3.2 ความต้านทานการถ่ายเทความร้อนภายในท่อความร้อน

ความสัมพันธ์ระหว่างความต้านทานการถ่ายเทความร้อนภายในท่อความร้อน และค่าอุณหภูมิแตกต่างเชิงล็อก แสดงดังรูปที่ 4 ก, ข, ค และ ง โดยความต้านทานการถ่ายเทความร้อนภายในท่อความร้อน นิยามดังสมการดังนี้

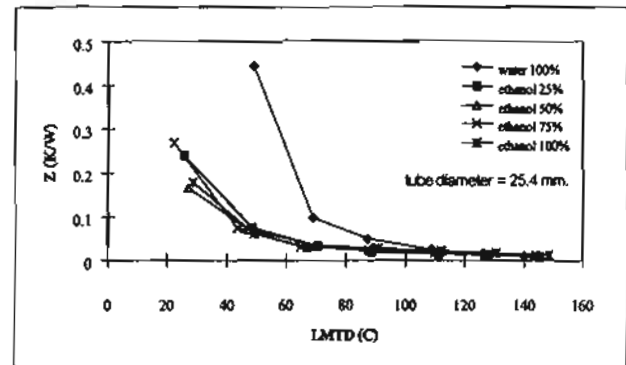
$$Z = \frac{T_{c,ave} - T_{h,ave}}{Q}$$

(2)

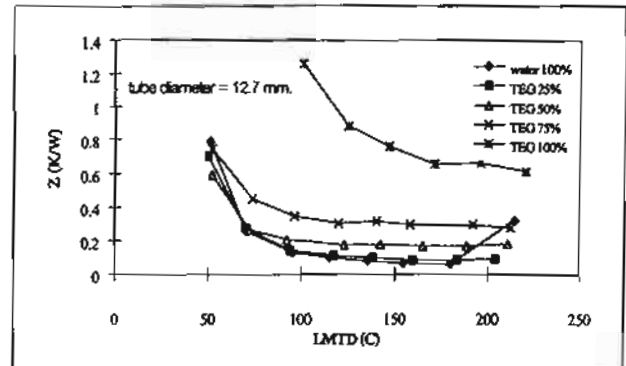
โดยที่สมการที่ 2 จะตั้งสมมุติฐานว่า การถ่ายเทความร้อนตามแนวแกนของท่อความร้อนมีค่าน้อยมากเมื่อเทียบกับการถ่ายเทความร้อนภายในท่อความร้อน



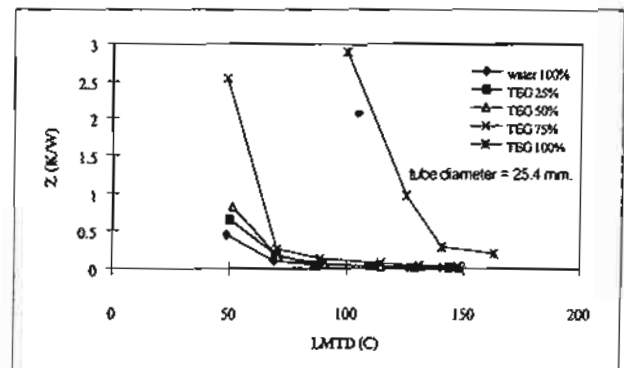
4ก. คู่สารทำงานเอทานอล-น้ำ ที่สัดส่วนต่างๆ ขนาดท่อ 12.7 mm.



4ข. คู่สารทำงาน เอทานอล-น้ำ ที่สัดส่วนต่างๆ ขนาดท่อ 25.4 mm.



4ค. คู่สารทำงาน TEG-น้ำ ที่สัดส่วนต่างๆ ขนาดท่อ 12.7 mm.



4ง. คู่สารทำงาน TEG-น้ำ ที่สัดส่วนต่างๆ ขนาดท่อ 25.4 mm.

รูปที่ 4 ความสัมพันธ์ระหว่างความต้านทานการถ่ายเทความร้อนภายในท่อความร้อนและอุณหภูมิแตกต่างเชิงล็อก

จากการทดลองพบว่า ความสัมพันธ์ของความต้านทานการถ่ายเทความร้อนและอุณหภูมิแตกต่างเชิงล็อกสอดคล้องกับกรณีของความสัมพันธ์ระหว่างอัตราการถ่ายเทความร้อนกับอุณหภูมิแตกต่างเชิงล็อก และมีแนวโน้มที่จะเข้าสู่ค่าคงที่เมื่ออุณหภูมิแตกต่างเชิงล็อกมีค่าสูงขึ้น และกลับมีค่าสูงขึ้นอีกเมื่อเกิดการท่วมขึ้นภายในท่อความร้อน

จากรูปที่ 4 ก และ ข จะพบว่าในกรณีช่วงอุณหภูมิแตกต่างเชิงล็อกต่ำ ค่าความต้านทานการถ่ายเทความร้อนของน้ำจะมีค่าสูงกว่าของสารทำงานกลุ่มผสม เอทานอล-น้ำ และเอทานอลบริสุทธิ์ อย่างเห็นได้ชัด ทั้งนี้เนื่องมาจากช่วงอุณหภูมิดังกล่าวน้ำยังไม่เริ่มเดือด ในขณะที่เอทานอลสามารถเดือดและส่งผ่านความร้อนได้แล้ว ในกรณีที่อุณหภูมิแตกต่างเชิงล็อกมีค่าสูงขึ้น พบว่าค่าความต้านทานของการถ่ายเทความร้อนของท่อความร้อนขนาดเส้นผ่านศูนย์กลาง 25.4 mm. มีค่าใกล้เคียงกัน ในขณะที่ค่าความต้านทานของการถ่ายเทความร้อนของท่อความร้อนขนาด 12.7 mm. มีค่าเพิ่มขึ้นอีกครั้งหนึ่ง ทั้งนี้เนื่องจากเกิดการท่วมขึ้นภายในท่อความร้อน

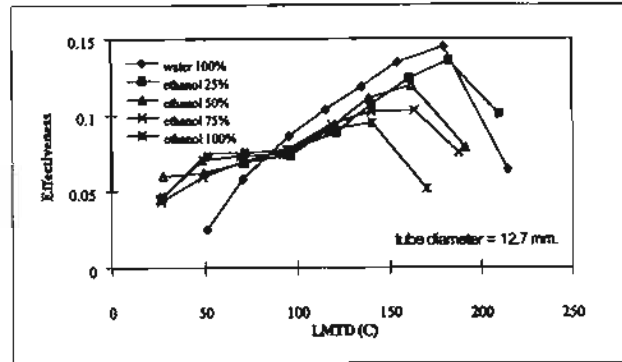
รูปที่ 4 ค และ ง เป็นกรณีของความต้านทานการถ่ายเทความร้อนของท่อความร้อนที่ใช้สารทำงานกลุ่มผสม ไตรเอทิลีนไกลคอล-น้ำ จากรูปดังกล่าวพบว่า ความต้านทานการถ่ายเทความร้อนจะแปรผกผันกับปริมาณไตรเอทิลีนไกลคอล ในสารทำงานกลุ่มผสม และยังพบว่าภาวะวิกฤติเนื่องจากการท่วมของสารทำงานภายในท่อความร้อนของท่อความร้อนขนาดเส้นผ่านศูนย์กลาง 12.7 mm. จะไม่เกิดขึ้น แต่กรณีของน้ำเพียงอย่างเดียวค่าความต้านทานจะเพิ่มขึ้นอีก เมื่อเกิดการท่วม

3.3 ค่าประสิทธิภาพของการถ่ายเทความร้อน

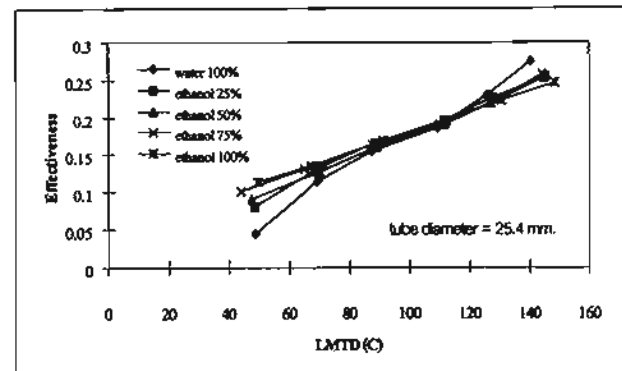
ค่าประสิทธิภาพของการถ่ายเทความร้อนของท่อความร้อนสามารถนิยามได้ดังสมการต่อไปนี้

$$\text{Effectiveness} = \frac{T_{wo} - T_{wi}}{T_{oil} - T_{wi}} \quad (3)$$

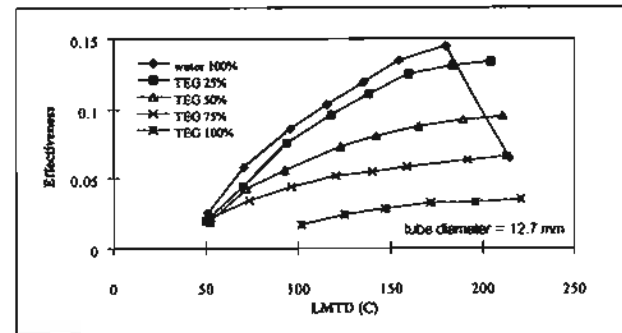
ความสัมพันธ์ระหว่างค่าประสิทธิภาพและอุณหภูมิแตกต่างเชิงล็อกแสดงดังรูปที่ 5 ก,ข,ค และ ง ซึ่งความสัมพันธ์ดังกล่าว สอดคล้องกับความสัมพันธ์ระหว่างอัตราการถ่ายเทความร้อนกับอุณหภูมิแตกต่างเชิงล็อก (รูปที่ 3) และความความสัมพันธ์ระหว่างความต้านทานการถ่ายเทความร้อน กับอุณหภูมิแตกต่างเชิงล็อก (รูปที่ 4)



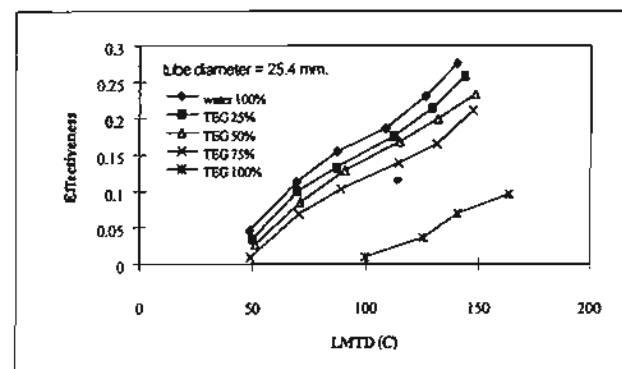
4ก. คู่อสารทำงานเอทานอล-น้ำ ที่สัดส่วนต่างๆ ขนาดท่อ 12.7 mm.



4ข. คู่อสารทำงาน เอทานอล-น้ำ ที่สัดส่วนต่างๆ ขนาดท่อ 25.4 mm.



4ค. คู่อสารทำงาน TEG-น้ำ ที่สัดส่วนต่างๆ ขนาดท่อ 12.7 mm.



4ง. คู่อสารทำงาน TEG-น้ำ ที่สัดส่วนต่างๆ ขนาดท่อ 25.4 mm.

รูปที่ 5 ความสัมพันธ์ของค่าประสิทธิภาพของการถ่ายเทความร้อนและค่าอุณหภูมิแตกต่างเชิงล็อก

3.4 การประยุกต์ต่อความร้อนแบบเทอร์โมไซฟอนที่ใช้สารทำงานคู่ผสมกับเครื่องแลกเปลี่ยนความร้อนแบบท่อความร้อน

1. กรณีใช้สารทำงานคู่ผสมเอธานอล-น้ำ

ในกรณีเครื่องแลกเปลี่ยนความร้อนที่ใช้งานในช่วงอุณหภูมิไม่สูงมากนัก (อุณหภูมิแตกต่างเชิงลึกไม่เกิน 100°C) การใช้สารทำงานคู่ผสมเอธานอล-น้ำจะทำให้สมรรถนะของเครื่องแลกเปลี่ยนความร้อนสูงกว่าในกรณีที่ใช้น้ำเป็นสารทำงานเพียงอย่างเดียว ทั้งนี้เนื่องจากในช่วงที่อุณหภูมิของกระแสร้อนมีค่าลดลง (ต่ำกว่า 100°C) อัตราการถ่ายเทความร้อนของท่อความร้อนที่ใช้สารทำงานคู่ผสมจะมีค่าสูงกว่าท่อความร้อนที่ใช้ใช้น้ำเป็นสารทำงาน ทำให้สมรรถนะในการถ่ายเทความร้อนรวมของเครื่องแลกเปลี่ยนความร้อนมีค่าสูงขึ้น

2. กรณีใช้สารทำงานคู่ผสมไตรเอธิลีนไกลคอล-น้ำ

ในบางกรณีท่อความร้อนที่ต้องการใช้มีขนาดยาวมาก ๆ เช่นท่อความร้อนที่ใช้ดึงความร้อนจากใต้ดิน ขนาดของอัตราส่วน L/d จะไม่เหมาะสม ในกรณีที่อุณหภูมิการทำงานมีค่าสูง มักจะทำให้เกิดภาวะวิกฤติเนื่องจากการท่วมได้ การผสม ไตรเอธิลีนไกลคอลลงไปในน้ำเพียงเล็กน้อย (ประมาณ 25%) จะช่วยลดภาวะวิกฤติเนื่องจากการท่วมลงได้ ในขณะที่อัตราการถ่ายเทความร้อนมีค่าลดลงเพียงเล็กน้อยเท่านั้น

4. บทสรุป

ท่อความร้อนแบบเทอร์โมไซฟอนที่ใช้สารทำงานคู่ผสมมีศักยภาพในการพัฒนาขึ้นมาใช้ประโยชน์ ทั้งนี้เนื่องจากว่าในการเลือกอัตราส่วนและสารคู่ผสมที่เหมาะสม ในสภาวะการทำงานบางอย่าง ท่อความร้อนดังกล่าวสามารถให้สมรรถนะที่สูงกว่าในกรณีของท่อความร้อนที่ใช้สารเดี่ยวเป็นสารทำงาน ทั้งนี้ขึ้นอยู่กับสภาวะการทำงานของท่อความร้อนดังกล่าวด้วย ดังเช่นในการวิจัยนี้พบว่าในกรณีช่วงอุณหภูมิการทำงานต่ำท่อความร้อนที่ใช้สารทำงานคู่ผสมเอธานอล-น้ำ จะมีอัตราการถ่ายเทความร้อนที่สูงกว่าท่อความร้อนที่ใช้ใช้น้ำเป็นสารทำงาน และนอกจากนี้ยังพบว่าการเติมไตรเอธิลีนไกลคอลลงไปในน้ำเพียงเล็กน้อย (ประมาณ 25%) จะช่วยลดภาวะวิกฤติเนื่องจากการท่วมขึ้นภายในท่อความร้อนที่มีอัตราส่วน L/d ไม่เหมาะสมลงได้

กิตติกรรมประกาศ

คณะผู้วิจัยขอขอบพระคุณสำนักงานกองทุนสนับสนุนการวิจัยที่ให้ทุนสนับสนุนในการวิจัยครั้งนี้

รายการสัญลักษณ์

- d = เส้นผ่านศูนย์กลางท่อความร้อน (m)
- k = ค่าการนำความร้อนของท่อ (W/mK)
- L = ความยาว (m)
- Q = อัตราการถ่ายเทความร้อนของท่อความร้อน (W)
- r = รัศมี (m^2)
- T = อุณหภูมิ (K)
- ΔT_{limd} = อุณหภูมิแตกต่างเชิงลึก ($^{\circ}\text{C}$)

Z = ความต้านทานทางความร้อน (KW)

Subscripts

- ave = ค่าเฉลี่ย
- c = ส่วนควบแน่น
- e = ส่วนระเหย
- i = ขาเข้า, ด้านใน
- l = ของเหลว
- o = ขาออก, ด้านนอก
- v = ไอ
- w = ผนัง, น้ำ

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อากาศแบบเทอร์โมไซฟอนโดยที่ค่าสัมประสิทธิ์การถ่ายเทความร้อนไม่คงที่, การประชุมวิชาการเครือข่าย
วิศวกรรมเครื่องกลแห่งประเทศไทยครั้งที่ 14, 2-3 พฤศจิกายน 2543, เชียงใหม่.

การคำนวณสมรรถนะของเครื่องอุ่นอากาศแบบเทอร์โมไซฟอนโดยที่ ค่าสัมประสิทธิ์การถ่ายเทความร้อนไม่คงที่ Performance Analysis of Thermosyphon Air Preheater With Variable Heat Transfer Coefficient

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บทคัดย่อ

งานวิจัยนี้ ได้เสนอวิธีการในการคำนวณสมรรถนะของเครื่องอุ่นอากาศแบบเทอร์โมไซฟอน ซึ่งโดยทั่วไปจะตั้งสมมุติฐานว่าอัตราการถ่ายเทความร้อนของท่อความร้อนแต่ละท่อในเครื่องอุ่นอากาศมีค่าคงที่ โดยในงานวิจัยนี้ได้ทำการเปรียบเทียบวิธีการคำนวณดังกล่าวกับวิธีการที่กำหนดให้อัตราการถ่ายเทความร้อนของท่อความร้อนในเครื่องอุ่นอากาศแปรเปลี่ยนไปตามสภาวะของอุณหภูมิอากาศที่มาแลกเปลี่ยนความร้อนกัน โดยทำการสร้างโปรแกรมจำลองสถานการณ์เพื่อคำนวณสมรรถนะของเครื่องอุ่นอากาศ โดยอาศัยหลักการทั้งสองแบบเปรียบเทียบกัน จากโปรแกรมจำลองสถานการณ์พบว่า การตั้งสมมุติฐานว่าอัตราการถ่ายเทความร้อนของท่อความร้อนมีค่าคงที่ จะใช้ได้ดีกับกรณีของเครื่องอุ่นอากาศแบบไหลสวนทางกันที่มีอัตราการไหลของอากาศร้อนและเย็นเท่ากัน ส่วนในกรณีของอัตราการไหลของอากาศร้อนและเย็นไม่เท่ากันและกรณีของเครื่องอุ่นอากาศแบบไหลตามกัน พบว่าสมมุติฐานดังกล่าว ให้การคำนวณสมรรถนะของเครื่องอุ่นอากาศคลาดเคลื่อน

Abstract

This research work presents an approach to calculate performance of the thermosyphon air preheater. Normally, the heat transfer rate of the thermosyphon in the air preheater is assumed constant. In this work, the heat transfer rate depends on the working conditions of the air in the thermosyphon heat exchanger. The simulated results had been compared with those of the conventional approach. It is found that both approaches had the same result for counter current flow with equal mass flow rates of hot and cold streams. For unbalance counter current flow and parallel flow, the conventional are showed oversized value.

1. บทนำ

เครื่องอุ่นอากาศแบบเทอร์โมไซฟอน เป็นอุปกรณ์แลกเปลี่ยนความร้อนชนิดหนึ่งที่มีใช้ในการดึงความร้อนทิ้งจากไอเสียที่เกิดจากการเผาไหม้ของหม้อไอน้ำ มาใช้ในการอุ่นอากาศก่อนเข้าห้องสันดาป ซึ่งเป็นการเพิ่มสมรรถนะของหม้อไอน้ำให้สูงขึ้น และนอกจากนี้เครื่องอุ่นอากาศแบบเทอร์โมไซฟอน ยังสามารถนำมาประยุกต์ใช้ในระบบการทางอุตสาหกรรมต่างๆ ได้มากมาย ทั้งนี้เนื่องจากอุปกรณ์ดังกล่าวมีข้อดีหลายประการ อาทิเช่น ให้อัตราการถ่ายเทความร้อนสูง ความดันตกคล้อยน้อย ราคาถูก และสามารถติดตั้งได้ง่าย

ในการออกแบบเครื่องอุ่นอากาศแบบเทอร์โมไซฟอน โดยปกติแล้วจะตั้งสมมุติฐานให้อัตราการถ่ายเทความร้อนของท่อความร้อนแต่ละท่อมามีค่าคงที่เท่ากัน [1] แต่ในทางปฏิบัติแล้ว อัตราการถ่ายเทความร้อนของท่อความร้อนในแต่ละตำแหน่งในเครื่องอุ่นอากาศจะมีค่าไม่เท่ากัน ขึ้นอยู่กับอุณหภูมิของอากาศที่มาแลกเปลี่ยนความร้อนกัน ดังนั้นการให้สมมุติฐานข้างต้นอาจจะไม่ถูกต้องนัก ทำให้การคำนวณออกแบบขนาดของเครื่องอุ่นอากาศเกิดความผิดพลาด ดังนั้น ในงานวิจัยนี้ จะทำการทดสอบสมมุติฐานดังกล่าวว่าสามารถใช้ได้ในลักษณะใดบ้าง โดยจะทำการออกแบบโปรแกรมคอมพิวเตอร์เพื่อใช้ในการคำนวณสมรรถนะของเครื่องอุ่นอากาศ ทั้งในกรณีของการใช้สมมุติฐานอัตราการถ่ายเทความร้อนของท่อความร้อนมีค่าคงที่ และในกรณีให้อัตราการถ่ายเทความร้อนของท่อความร้อนแปรเปลี่ยนไปตามสภาวะของอุณหภูมิอากาศที่มาแลกเปลี่ยนความร้อนกัน เพื่อศึกษาถึงความเหมาะสมในการนำเอาสมมุติฐานอัตราการถ่ายเทความร้อนคงที่ มาใช้ในการออกแบบเครื่องอุ่นอากาศแบบเทอร์โมไซฟอน

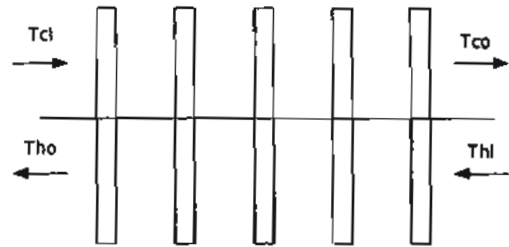
2 ทฤษฎีที่เกี่ยวข้องกับงานวิจัย

การคำนวณการถ่ายเทความร้อนของเครื่องอุ่นอากาศแบบเทอร์โมไซฟอน สามารถพิจารณาได้ดังรูปที่ 1 ซึ่งมีรายละเอียดดังต่อไปนี้

อัตราการถ่ายเทความร้อนรวมของเครื่องอุ่นอากาศแบบเทอร์โมไซฟอนสามารถคำนวณได้จากสมการดังนี้

$$Q = (UA)_t \Delta T_{lmt} \quad [1]$$

โดยที่ Q คืออัตราการถ่ายเทความร้อนรวม $(UA)_t$ คือค่าสัมประสิทธิ์การถ่ายเทความร้อนรวมพื้นที่ ของเครื่องอุ่นอากาศ ΔT_{lmt} คือ อุณหภูมิแตกต่างเชิงล็อก



รูปที่ 1 การถ่ายเทความร้อนของเครื่องอุ่นอากาศแบบเทอร์โมไซฟอน

ค่าสัมประสิทธิ์การถ่ายเทความร้อนรวมพื้นที่ ของเครื่องอุ่นอากาศแบบเทอร์โมไซฟอนสามารถคำนวณได้ดังนี้

$$\begin{aligned} \frac{1}{(UA)_t} = & \frac{1}{h_{ev,o} A_{ev,ot}} + \frac{1}{h_{cd,o} A_{cd,ot}} \\ & + \frac{1}{h_{ev,i} A_{ev,it}} + \frac{1}{h_{cd,i} A_{cd,it}} \\ & + \frac{\ln(r_o / r_i)}{2\pi k_m L_{ev,t}} + \frac{\ln(r_o / r_i)}{2\pi k_m L_{cd,t}} \end{aligned} \quad [2]$$

Klatsiriroot et.al. [2] ได้ทำการศึกษาถึงค่าสัมประสิทธิ์การถ่ายเทความร้อนของการเดือดในท่อเทอร์โมไซฟอน ที่ใช้น้ำเป็นสารทำงาน และได้สร้างสมการคำนวณค่าสัมประสิทธิ์การถ่ายเทความร้อนของการเดือด โดยดัดแปลงสมการของ Roshenow ดังนี้

$$\begin{aligned} h_{ev,i} = & \frac{18.688}{\Delta T_{ev} C_1} \left[\left(\frac{\Delta T_{ev}}{C_2} \right)^3 \right]^{0.3572} \\ C_1 = & \frac{1}{\mu_{wl} \lambda_w} \sqrt{\frac{g_c \sigma_w}{g(\rho_{wl} - \rho_{wv})}} \\ C_2 = & \frac{\lambda_w Pr_{wl}}{Cp_{wl}} \end{aligned} \quad [3]$$

โดยที่ ΔT_{av} คืออุณหภูมิแตกต่างระหว่างผิวและสารทำงานที่ส่วนระเหยของท่อความร้อน

นอกจากนี้ยังได้ทำการสร้างสมการเอมไพริคัลเพื่อคำนวณค่าสัมประสิทธิ์การถ่ายเทความร้อนของการควบแน่นภายในท่อความร้อนแบบเทอร์โมไซฟอนที่ใช้ น้ำเป็นสารทำงาน โดยดัดแปลงมาจากสมการของ Nusselt ดังนี้

$$h_{cd,i} = 0.943 \left(\frac{\rho_{wl}^2 g \lambda_w k_{wl}^3}{\mu_{wl} L_{cd} \Delta T_{cd}} \right)^{0.233} \quad [4]$$

โดยที่ ΔT_{cd} คืออุณหภูมิแตกต่างระหว่างผิวและสารทำงานที่ส่วนควบแน่นของท่อความร้อน

ESDU [3] ได้เสนอสมการในการคำนวณค่าสัมประสิทธิ์การถ่ายเทความร้อนของอากาศร้อนที่ไหลผ่านกลุ่มท่อติดครีบบนไหลตามขวาง (cross flow) ดังนี้

$$h = \frac{\overline{Nu} k_a}{D_o} \quad [5]$$

$$\overline{Nu} = 0.242 Re^{0.658} \left(\frac{f_s}{f_h} \right)^{0.297} \left(\frac{S_t}{S_l} \right)^{-0.091} Pr^{0.333}$$

3. โปรแกรมการจำลองสถานการณ์

โปรแกรมการจำลองสถานการณ์ที่ใช้ในการคำนวณสมรรถนะของเครื่องอุ่นอากาศแบบเทอร์โมไซฟอนในงานวิจัยนี้ สามารถแบ่งออกเป็น 2 ส่วนคือ โปรแกรมที่สร้างขึ้นจากสมมติฐานอัตราการถ่ายเทความร้อนของท่อความร้อนคงที่ และไม่คงที่ (ขึ้นกับสภาวะของอากาศที่ไหลแลกเปลี่ยนความร้อน) โดยลักษณะของเครื่องอุ่นอากาศที่ใช้ในการคำนวณมีรายละเอียดดังต่อไปนี้

จำนวนท่อความร้อน	49	ท่อ
จำนวนแถวของท่อความร้อน	7	แถว
การจัดเรียง	แบบเหลื่อมกัน	
S_1	0.053	m
S_2	0.046	m
S_3	0.053	m
ขนาดท่อความร้อน	0.027	m
ความยาวของส่วนระเหย	0.4	m
ความยาวของส่วนควบแน่น	0.4	m

สารทำงาน	น้ำ
ลักษณะของครีป	แบบวงกลม
ความหนาแน่น	10 ครีป/inch
ความสูงครีป	0.01 m
วัสดุ (ท่อและครีป)	stainless 304
อุณหภูมิขาเข้าของอากาศร้อน	200 °C
อุณหภูมิขาเข้าของอากาศเย็น	30 °C

Flow-charts ของโปรแกรมในการคำนวณสมรรถนะของเครื่องอุ่นอากาศแบบเทอร์โมไซฟอน ในกรณีของการใช้สมมติฐานอัตราการถ่ายเทความร้อนเท่ากัน และสมมติฐานที่อัตราการถ่ายเทความร้อนไม่เท่ากัน ในกรณีของการไหลสวนทางและไหลตามกัน แสดงในรูปที่ A1-A3 ในภาคผนวกตามลำดับ โดยในงานวิจัยนี้ จะใช้โปรแกรม Turbo Pascal for Windows 1.5 ในการคำนวณสมรรถนะของระบบ

รูป ก1 ซึ่งแสดง Flow-chart ในการคำนวณสมรรถนะของเครื่องอุ่นอากาศแบบไหลสวนทางหรือแบบไหลตามกัน โดยกำหนดให้ค่าอัตราการถ่ายเทความร้อนมีค่าเท่ากันทุกท่อ ซึ่งเป็นการคำนวณแบบวนลูป เพื่อหาค่าอัตราการถ่ายเทความร้อนรวมและอุณหภูมิขาออกของอากาศร้อนและเย็น

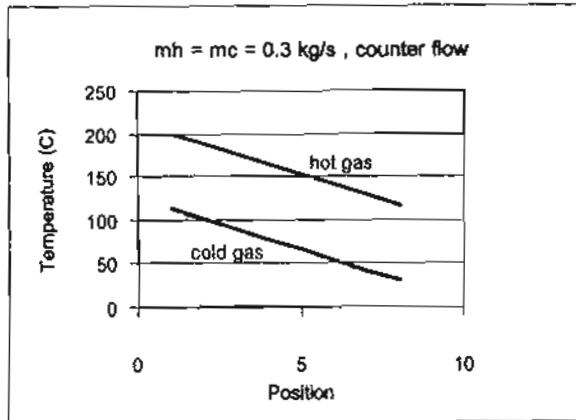
รูป ก2-3 เป็นการคำนวณอัตราการถ่ายเทความร้อนของท่อความร้อนในเครื่องอุ่นอากาศที่ละแถว เพื่อหาค่าอุณหภูมิของอากาศด้านเข้าและออกจากท่อความร้อนตลอดจนอัตราการถ่ายเทความร้อนในแต่ละแถว

4. ผลการจำลองสถานการณ์และการวิเคราะห์

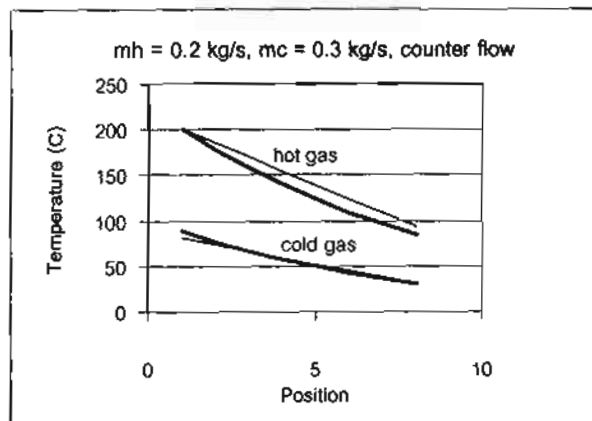
รูปที่ 2 ก-ง แสดงการเปรียบเทียบอุณหภูมิของอากาศร้อนและเย็นที่ได้จากผลการจำลองสถานการณ์ในกรณีของการใช้สมมติฐานอัตราการถ่ายเทความร้อนคงที่และไม่คงที่ ในกรณีของเครื่องอุ่นอากาศแบบไหลสวนทางกัน ที่มีอัตราการไหลของอากาศร้อนและเย็นเท่ากัน จากรูปจะพบว่าอุณหภูมิของอากาศร้อนและเย็นที่ได้จากการคำนวณทั้งสองแบบมีค่าเท่ากัน ซึ่งหมายถึงในกรณีนี้การใช้สมมติฐานแบบเดิมให้การคำนวณที่ถูกต้องแม่นยำ

รูปที่ 2 ข-ง เป็นกรณีของอุณหภูมิของอากาศร้อนและเย็น ในกรณีของเครื่องอุ่นอากาศแบบไหลสวนทางกันโดยอัตราการไหลของอากาศร้อนและเย็นไม่เท่ากัน และในกรณีของเครื่องอุ่นอากาศแบบไหลตามกัน ทั้งในกรณีของอัตราการไหลของอากาศแบบเท่ากันและไม่เท่ากันตามลำดับ ในกรณีดังกล่าวนี้จะพบว่าการใช้สมมติฐานที่ว่าอัตราการถ่ายเท

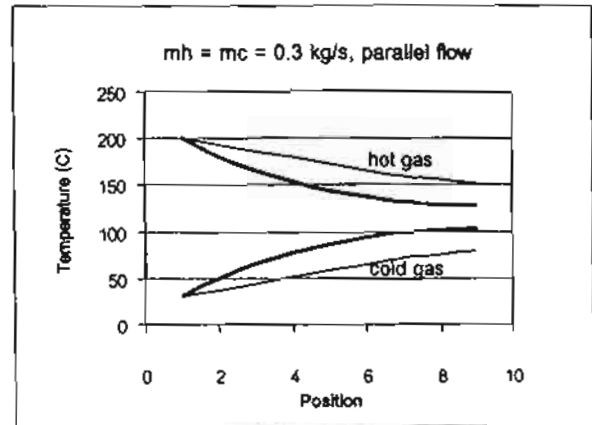
ความร้อนคงที่จะให้ความคลาดเคลื่อนในการคำนวณอุณหภูมิของอากาศสูง โดยเฉพาะอย่างยิ่งในกรณีของเครื่องอุ่นอากาศแบบไหลสวนทางกัน ซึ่งส่งผลให้การคำนวณอุณหภูมิขาออกของอากาศร้อนและเย็นที่ได้สูงและต่ำกว่าความเป็นจริง ซึ่งถ้านำไปใช้ในการออกแบบเครื่องอุ่นอากาศจะทำให้ได้เครื่องอุ่นอากาศที่มีขนาดใหญ่เกินความจำเป็น



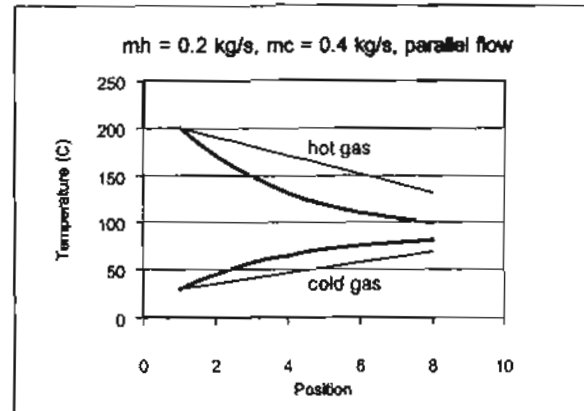
- ก. อุณหภูมิของอากาศร้อนและเย็นในกรณีของเครื่องอุ่นอากาศแบบไหลสวนทางกัน โดยมีอัตราการไหลของอากาศร้อนและเย็นเท่ากัน



- ข. อุณหภูมิของอากาศร้อนและเย็นในกรณีของเครื่องอุ่นอากาศแบบไหลสวนทางกัน โดยมีอัตราการไหลของอากาศร้อนและเย็นไม่เท่ากัน



- ค. อุณหภูมิของอากาศร้อนและเย็นในกรณีของเครื่องอุ่นอากาศแบบไหลตามกัน โดยมีอัตราการไหลของอากาศร้อนและเย็นเท่ากัน



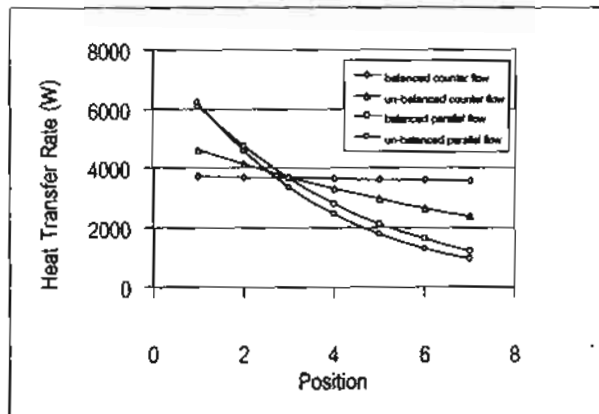
- ง. อุณหภูมิของอากาศร้อนและเย็นในกรณีของเครื่องอุ่นอากาศแบบไหลตามกัน โดยมีอัตราการไหลของอากาศร้อนและเย็นไม่เท่ากัน

รูปที่ 2 อุณหภูมิของอากาศร้อนและเย็นของเครื่องอุ่นอากาศแบบต่างๆ ที่ได้จากการคำนวณโดยใช้สมมุติฐานทั้งสองแบบ

note : ——— อัตราการถ่ายเทความร้อนคงที่
———— อัตราการถ่ายเทความร้อนไม่คงที่

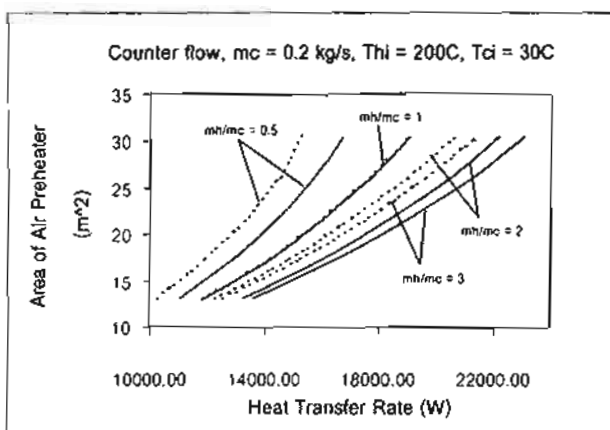
รูปที่ 3 แสดงอัตราการถ่ายเทความร้อนของเครื่องอุ่นอากาศแบบไหลสวนทางและไหลตามกันทั้งในกรณีอัตราการไหลของอากาศเท่ากันและไม่เท่ากัน จากรูปพบว่า ในกรณีของเครื่องอุ่นอากาศแบบไหลสวนทางกันที่มีอัตราการไหลของอากาศร้อนและเย็นเท่ากัน จะมีค่าอัตราการถ่ายเทความร้อนในแต่ละแถวของท่อความร้อนคงที่ ซึ่งส่งผลให้ผลต่าง

ของอุณหภูมิอากาศร้อนและเย็นที่ตำแหน่งต่างๆ ในเครื่องอุ่นอากาศประเภทนี้มีค่าคงที่ สำหรับในกรณีอื่นๆ พบว่าอัตราการถ่ายเทความร้อนมีค่าไม่คงที่ ดังนั้นการใช้สมมติฐานอัตราการถ่ายเทความร้อนคงที่สำหรับในกรณีนี้เป็นสิ่งที่ไม่ถูกต้อง



รูปที่ 3 อัตราการถ่ายเทความร้อนของท่อความร้อนในแต่ละแถวของเครื่องอุ่นอากาศแบบต่างๆ

รูปที่ 4 แสดงพื้นที่ในการแลกเปลี่ยนความร้อนของเครื่องอุ่นอากาศแบบเทอร์โมไฮฟอนแบบไหลสวนทางกัน ที่ได้จากการคำนวณทั้งสองแบบ จากรูปพบว่า ในกรณีของอัตราการไหลของอากาศร้อนและเย็นไม่เท่ากัน การใช้สมมติฐานอัตราการถ่ายเทความร้อนคงที่ จะทำให้พื้นที่ ที่ได้จากการคำนวณมีค่าสูงเกินความจำเป็น สำหรับในกรณีอัตราการไหลเท่ากัน พบว่าพื้นที่ที่ได้จากการคำนวณทั้งสองแบบมีค่าเท่ากัน



รูปที่ 4 พื้นที่ผิวของเครื่องอุ่นอากาศที่ได้จากการคำนวณทั้งสองแบบ

----- Q constant _____ Q un-constant

5. สรุปผลการวิจัย

การออกแบบเครื่องอุ่นอากาศแบบไหลสวนทางกันโดยมีอัตราการไหลของอากาศร้อนและเย็นคงที่ สามารถใช้สมมติฐานอัตราการถ่ายเทความร้อนคงที่ได้ แต่ในกรณีของอัตราการไหลของอากาศร้อนและเย็นไม่เท่ากัน หรือในกรณีของเครื่องอุ่นอากาศแบบไหลสวนทางกัน การใช้สมมติฐานดังกล่าวจะมีความคลาดเคลื่อนในการคำนวณสูง

6. กิตติกรรมประกาศ

คณะผู้จัดทำขอขอบพระคุณสำนักงานกองทุนสนับสนุนการวิจัยที่ให้ทุนอุดหนุนในการทำวิจัยครั้งนี้

เอกสารอ้างอิง

1. Engineering Science Data Unit, "Heat Pipe Performance of Two Phase Closed Thermosyphons", Item No. 81038, UK., 1981
2. Kiatsiriroat T., Nuntaphan A. and Tiansuwan J., "Thermal Performance Enhancement of Thermosyphon Heat Pipe with Binary Working Fluids" Experimental Heat Transfer, Vol 13, No 2, 2000
3. Hewitt G.F., Shires G.L. and Bott T.R., "Process Heat Transfer", CRC Press., 1994

รายการสัญลักษณ์

- A = Area (m^2)
 C_p = Heat capacity (J/kgK)
 D = Diameter of pipe (m)
 f_h = Fin height (m)
 f_s = Gap between fins (m)
 g = Gravitational acceleration ($9.81 m/s^2$)
 g_c = Constant ($1kgm/Ns^2$)
 h = heat transfer coefficient (W/m^2K)
 k = Thermal conductivity (W/mK)
 L = Length (m)
 m = mass flow rate (kg/s)
 Nu = Nusselt number
 Pr = Prandtl number
 Q = Heat transfer rate (W)

Re = Renold number

S_d = Pitch of tubes on the diagonal plane (m)

S_f = Pitch of tubes in direction of flow (m)

S_t = Pitch of tubes in plane perpendicular to flow (m)

T = Temperature ($^{\circ}\text{C}$)

ΔT_{LMTD} = Log mean temperature difference ($^{\circ}\text{C}$)

Greek Letters

μ = Viscosity (kg/ms)

ρ = Density (kg/m^3)

λ = Latent heat of vaporization (J/kg)

σ = Surface Tension (N/m)

subscript

a = air

c = cold air

cd = condenser

ev = evaporator

h = hot air

i = inlet, inside

l = liquid phase

m = metal

o = outlet, outside

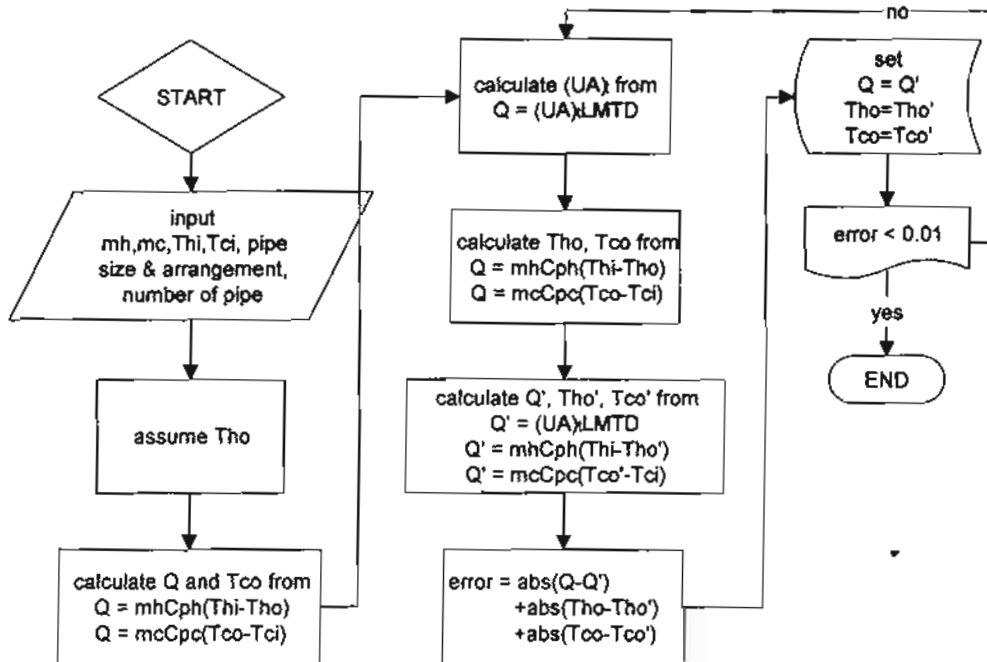
s = surface

t = total

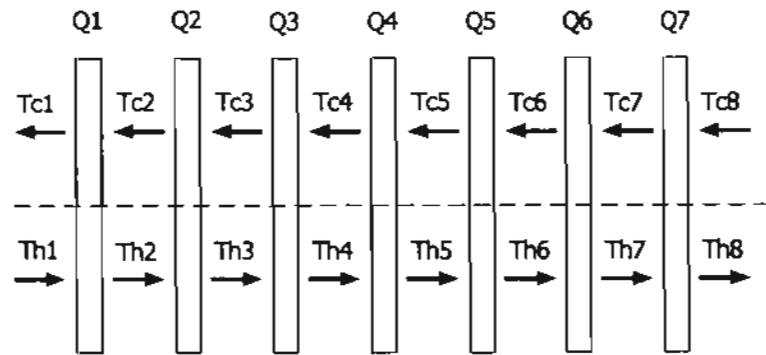
v = vapor phase

w = water

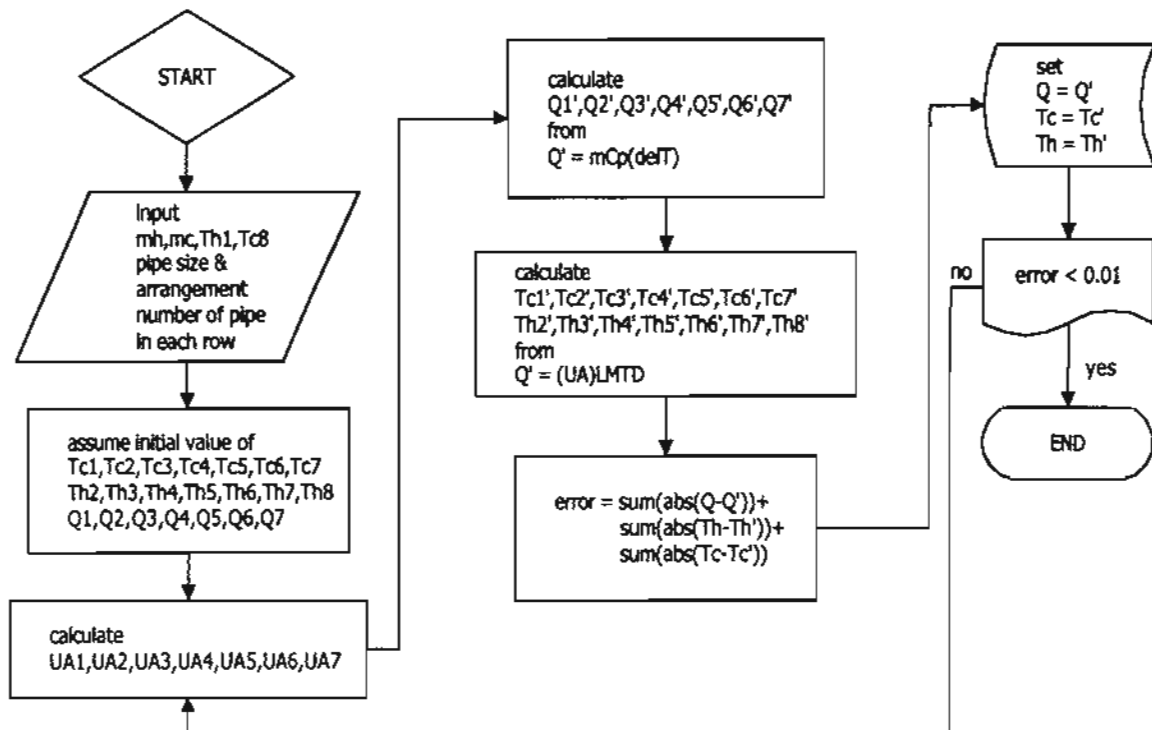
ภาคผนวก



รูปที่ ก1 แผนภาพในการคำนวณสมรรถนะของเครื่องปรับอากาศแบบเทอร์โมอิเล็กทริก
ในกรณีของอัตราการถ่ายเทความร้อนคงที่

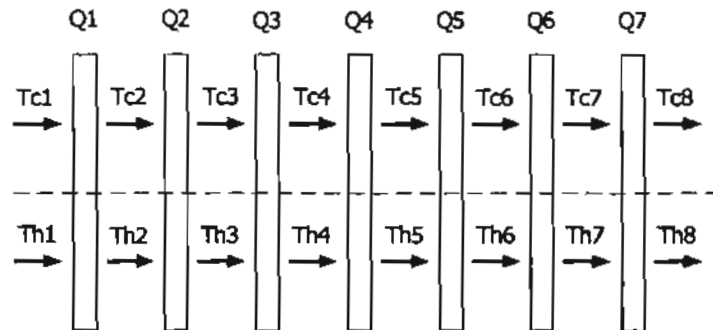


ก. ทิศทางการไหลของกระแสน้ำและเย็นในเครื่องอุ่นอากาศแบบไหลสวนทางกัน

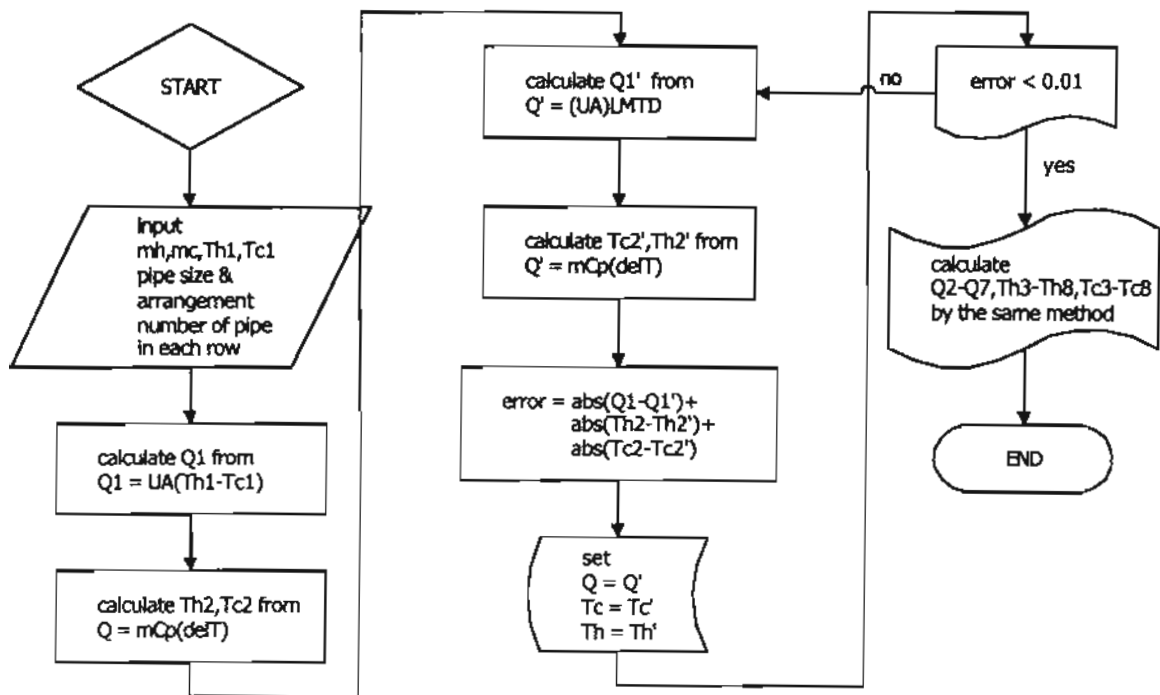


ข. แผนภาพในการคำนวณสมรรถนะของเครื่องอุ่นอากาศแบบไหลสวนทางกัน

รูปที่ ก2 แผนภาพในการคำนวณสมรรถนะของเครื่องอุ่นอากาศแบบเทอร์โมไซฟอนแบบไหลสวนทางกัน
ในกรณีของอัตราการถ่ายเทความร้อนไม่คงที่คงที่



ก. ทิศทางการไหลของกระแสน้ำร้อนและเย็นในเครื่องอุ่นอากาศแบบไหลตามกัน



ข. แผนภาพในการคำนวณสมรรถนะของเครื่องอุ่นอากาศแบบไหลตามกัน

รูปที่ ก3 แผนภาพในการคำนวณสมรรถนะของเครื่องอุ่นอากาศแบบเทอร์โมไซฟอนแบบไหลตามกัน
ในกรณีของอัตราการถ่ายเทความร้อนไม่คงที่คงที่