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RESEARCH ARTICLE

THEORICAL INVESTIGATION ON THE EFFECTS OF THE FIN-TUBE CONTACT ZONES ON THE EXCHANGEDCONVECTIVE HEAT FLUX BETWEEN A FINNED-TUBE AIR-COOLED CONDENSER AND ITS SURROUNDINGENVIRONMENT

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Abstract

Adding fins on the tubes of an air-cooled condenser allows to increase its heat transfer area.The purpose of this work is to investigate on the effects of the tube-fin contact zones on thecondenserconvectiveheattransferwiththesurroundingair.Itproposes amethodofdetermining the condenser outer surface and a ratio which compares the losses to the heat fluxgainsduetotheinputoffins.Thisratiocharacterizestheinfluenceofthefin-tubecontactzoneson the exchanged convective heat flux and informs whether or not their effect has to be takeninto account. The developed model allowed to obtain relations which give the condensereffective exchange outer surface which participates to convective heat transfers. The obtainedresults using Matlab software show that the ratio of the losses to the heat flux gains does notdepend on the number of fins. However, it increases with the number of tubes, with their outerdiameterandwiththefinthickness.Itisfoundthat,whenthisratioislesthan0.026,theeffectsof the fin-tube contact zones on the condenser convective heat transfer can be neglected.Otherwise,theseeffectshavetobetakenintoaccountwhendeterminingthecondenserexchanged heat fluxwith its outer environment.

Introduction:-

The performance of a refrigeration system, especially with mechanical vapor compression, ishighly dependent on the efficiency of the condenser which evacuates the heat absorbed by therefrigeranttotheoutsideenvironment[**1;2**].Infact,whentheheattransferisnotsufficient,theworkofthecompressorbecomesimportant,whichcanleadtoanincreaseinenergyconsumption [**3**].

Dependingonthecoolant,adistinctionismadebetweenwater-cooledcondensers,evaporativecondensers and ambient air-cooled condensers [**4**]. The use of air-cooled condensers is morewidespreadduetothefreeairanditsunlimitedavailabilityinlargequantities.However,duetothe very low specific heat and overall heat transfer coefficient, heat removal in this kind ofcondenser technology will require large air volumes and large exchange areas [**5**]. Therefore,fins are added on the tubes of the condenser in order to increase its external heat exchangesurfaceandintensifyheat transfer[**6**].Thefinsareusuallymadeof aluminumandareattachedto the tubes either

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by welding or by soldering or by extrusion, etc. [**6**]. This engenders contactzones between the tubes and the fins which will be inaccessible to the ambient air and whichcan constitute an obstacle for convective heat exchanges between the air and part of the outersurfaces of the tubes and fins. Besides, when the fins are not in good contact with the tubes, itcan dramaticallydecrease the efficiencyon thecondenser heat transfer [**7**].

When the number of tubes and the number of fins become very large, the area occupied by thecontact zones and the total thermal contact resistance can also be important. The input of finstherefore constitutes a gain by increasing the heat exchanger outer surface but also could havesideeffectswhichmanifestthemselvesintheformofheatlosses.Thus,neglectingornotoftheeffect of the tube-fin contacts when determining the outer surface of the condenser and theexchanged convective heat flow must be done accordingto well-defined principles.

Much research has been done on the various factors that influence on the performance of anair-cooled condenser. Most of these studies are concerned with the influence of wind andambient temperature. In fact, it has been well documented that ambient winds can lead to thedistortionofthecondenserinletair flow andtoanincreaseinthe airtemperatureattheinletofthe condenser fans [**8; 9**]. To counter the effect of the wind, proposals have been made such astheinstallationofwindscreensunderthecondenserplatform[**10**],theinstallationofwindwallstoattenuatethedistortionofthei nletairflowandreducethecondenserinletairtemperature
[11] , andthe constructionof walkwaysatthe edge of the condenser platform[**12; 13**].Regarding the ambient temperature, many studies have focused on the pre-cooling of thecondenserinletair flow byspraying water upstream of the condenser[**14; 15; 16**].

However,otherfactorssuchasresistancesrelatedtocondenserfoulinghavebeeninvestigated.In 2017, Howard Cheung et al [**17**], by proposing a method of controlling condenser fouling,showed that it considerably reduces the performance of the system. Lee and al [**18**] madeexperimental and numerical investigations on the efficiency of a condenser for the refrigerantsR-22andR-407Canddeterminedthequantitiesofheatexchangedbytheε-NUTmethod.Whencalculating the global exchange coefficient, they considered the thermal resistances due to fin-tube contacts and the thermal resistances due to pollution and contaminants, but they did notstudy their influence on the efficiency of the condenser. Deng et al [**19**], by carrying out anumerical study on complete condensation and on the effect of freezing on finned-tube air-cooled condensers, showed that this latter decreases the efficiencyof theheat exchanger.

Despite all these aforementioned studies, the influence of the fin-tube contact zones on theconvectiveheatfluxexchangedbetweenthecondenserouterareaanditssurroundingenvironment has not yet been investigated. It seems to be neglected in most studies whendeterminingthecondenserheattransferouterarea.Inthiswork,aratiowhichcharacterizestheeffect of the tube-fin contact zones on the convective heat flux has been introduced. Theobjective is to propose a more precise model for determining the heat transfer outer area of afinned-tubeair-cooledcondenserwhicheffectivelyparticipatestoconvectiveheatfluxexchangeswithitssurroundingenvironment,tostudyan dcomparethelossesandgainsofheatflows due to the input of fins. This ratio will therefore provide information on whether or notthe effect of the tube-fincontact zones can be neglected.

Mathematicalmodellingandmethod

Model assumptions:

1. Thefinsarerectangularand thetubesarecylindrical;
2. TheU-shapedpartsofthetubesarenottakenintoaccountwhendeterminingtheoutersurface;
3. Thetotalcontactareaisconsideredtobeequaltoalltheareasoccupiedbythetubesonthe fins and bythe fins on the tubes;
4. Thetemperatureis uniform overtheentire wall ofthecondenser.

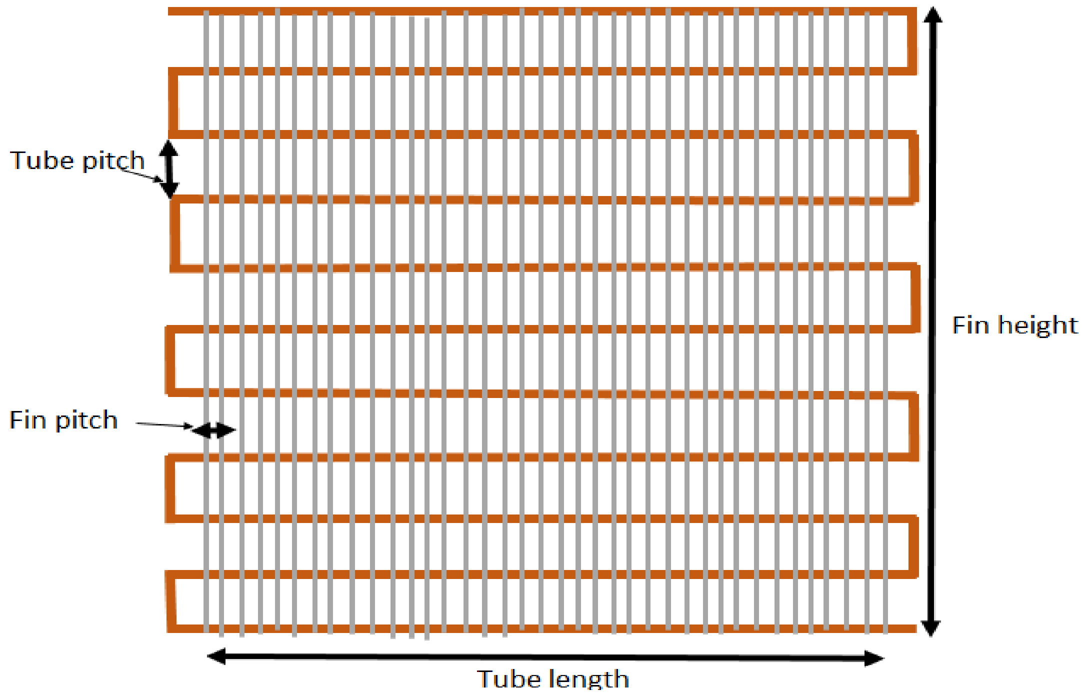


Figure1:- Schematicrepresentationofafinned-tubescondenser.

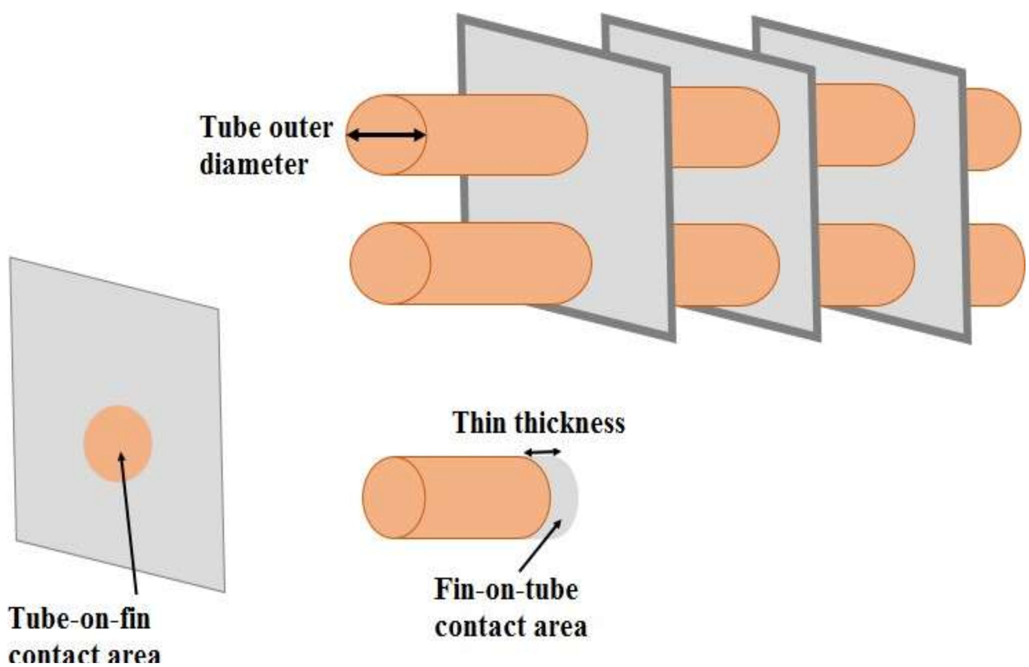


Figure2:- Schematicviewofthetube-fincontact areas.

Determinationofthecondenseroutersurface

The condenser effective outer surface is the difference between its total outer area (withouttaking into account the fin-tube contact zones) and the total area occupied by these contacts.Afterresolution it is given as follows:

$$S_{ef,o,cond} = \pi N_{tot,t} D_o L_{en,t} + N_f H_f \left(E + 2L_{ar,f}\right) - N_{tot,t} N_f \left(\pi D_o E + \frac{\pi D_o^2}{2}\right) \quad \text{Eq.1}$$

The firsttermof this equation(Eq.1) representsthe outer surface of the tubes, thesecondtermtheoutersurfaceofthefins(lateralandfrontal)andthelasttermthetotaltube-fincontactarea.

ThecondenserheatexchangeoutersurfaceS_{exch}isdeterminedconsideringthefinsefficiency η_f.It'sgivenby :

$$S_{exch} = S_{ef,o,t} + \eta_f S_{ef,o,f} \quad \text{Eq.2}$$

With$S_{ef,o,t}$thetubeseffectiveoutersurfaceand $S_{ef,o,f}$thefinseffectiveoutersurface

$$S_{ef,t} = \pi N_{tot,t} D_o L_{en,t} - N_{tot,t} N_f \pi D_o E \quad \text{Eq.3}$$

$$S_{ef,f} = N_f H_f (E + 2L_{ar,f}) - N_{tot,t} N_f \frac{\pi D_o^2}{2} \quad \text{Eq.4}$$

InEq.3, thefirsttermrepresentsthe tubestotaloutersurface($S_{o,t}$)andthesecondtermthetotal contactarea of thefinson thetubes($S_{f/t}$).InEq.4,the firsttermrepresentsthe finstotaloutersurface($S_{o,f}$)andthesecondtermthetotalcontactareaofthetubesonthefins($S_{t/f}$)

With $N_f = \frac{L_{en,t} + p_f/}{E + p_f}$, $N_{v,t} = \frac{H_f + p_{v,t}}{D_o + p_{v,t}}$, $N_{h,t} = \frac{L_{ar,f} + p_{h,t}}{D_o + p_{h,t}}$ et $N_{tot,t} = N_{v,t}.N_{h,t}$

Thefins efficiencyis calculated from the Schmidt's approximation as follows:

$$\eta_f = \frac{\tan h\left(M\left(\frac{D_{in}}{2}\right)\Phi\right)}{M\left(\frac{D_{in}}{2}\right)\Phi} \quad \text{Eq.5}$$

$$\Phi = \left(1.27 \frac{X_M}{\frac{D_{in}}{2}} \left(\frac{X_L}{X_M} - 0{,}3\right)^{\frac{1}{2}}\right)\left(1 + 0.35 \ln\left(1.27 \frac{X_M}{\frac{D_{in}}{2}} \left(\frac{X_L}{X_M} - 0.3\right)^{\frac{1}{2}}\right)\right) \quad \text{Eq.6}$$

With $X_M = \frac{p_{v,t}}{2}$, $X_L = \sqrt{\left(\frac{p_{v,t}}{2}\right)^2 + \frac{p_{v,t}^2}{2}}$ et $M = \sqrt{\frac{2h_{ex}}{E\lambda_{ai}}}$ geometrical parameters.

Thermalbalance

The tubes are considered clean and the thermal resistance due to fouling is neglected. Thecondenser overall heat transfer coefficient is determined by taking into account the tube-fincontact thermal resistances as follows:

$$\frac{1}{U_o S_{ef,o,cond}} = \frac{1}{U_{in} S_{in,t}} = \frac{1}{\eta_o S_{ef,o,cond}\, h_o} + \frac{\ln\left(\frac{D_o}{D_{in}}\right)}{2\pi\lambda_t N_{tot,t} L_{en,t}} + \frac{1}{\eta_{in} h_{in} S_{in,t}} + R_{cont} \quad \text{Eq.7}$$

With η_{in} the efficiency of the condenser internal surface, $\eta_o = 1 - \frac{S_{ef,ai}}{S_{ex,ef}}\left(1 - \eta_f\right)$ the overall efficiency of the condenser outer surface and $R_{cont} = R_{f/t}.S_{f/t}$ the total thermal contact resistance.

From the equation Eq. 7, the overall heat exchange coefficient can be expressed as a functionof the condenser inner or outer surface. Expressing it as a function of the condenser outersurface and considering that η_{in} = 1 since there are no fins on the condenser inner surface, wehave:

$$U_o = \frac{1}{\frac{1}{\eta_o h_o} + \frac{S_{ef,o,cond}\ \ln\left(\frac{D_o}{D_{in}}\right)}{2\pi\lambda_t N_{tot,t} L_{en,t}} + \frac{S_{ef,o,cond}}{h_{in} S_{in,t}} + S_{ef,o,cond}\ R_{cont}} \quad \text{Eq.8}$$

TheairsideexternalexchangecoefficientcanbeobtainedusingtheColburnfactorjdeterminedexperimentallyby[**20**]:

$$h_o = \frac{\lambda_a j R_{e,D_o} P_{r,a}^{\frac{1}{3}}}{D_o} \quad \text{Eq.9}$$

With $j = 0.394 R_{e,D_o}^{-0.392} \left(\frac{E}{D_o}\right)^{-0.0449} N_{tot,t}^{-0.0897} \left(\frac{p_f}{D_o}\right)^{-0.212}$

Therefrigerant sideinternal exchangecoefficientisgiven asfollows:

$$h_{in} = \frac{N_{u,refr}\ \lambda_{refr}}{D_{in}} \quad \text{Eq.10}$$

Inalaminar flowfor$R_{e,D_{in}}$<2000,Lévêque'sexperimentalcorrelationsareapplicableasafunction of a parameterA such that:

$$A = \frac{1}{R_{e,D_{in}} P_{r,refr}} \frac{L_{on,t}}{D_{in}} \quad \text{Eq.11}$$

Inthat case:

$$N_{u,refr}=3.66 \text{ for } A>0.05 \quad \text{Eq.12}$$

$$N_{u,refr}=1.06A^{-0.4} \text{ for } A<0.05 \quad \text{Eq.13}$$

TheColburn'sformula for$N_{u,refr}$in aturbulentflowisgiven by:

$$N_{u,refr} = 0.023P_{r,ref}^{\frac{1}{3}} R_{e,D_{in}}^{0.8} \quad \text{Eq.14}$$

Thisformulaisvalidfor$10^4<R_{e,D}<1.2.10^5$,$0.7<P_{r,refr}<100$andforaperfectlyestablished flow regime in the tube ($\frac{L_{en,t}}{D_{in}}>60$).

For$\frac{L_{en,t}}{D_{in}}<60$, the Colburn correlation must be corrected to take into account that the velocity profileof thefluid in the tube is notyet fullyestablished.In thatcase:

$$N_{u,refr} = 0.023P_{r,refr}^{\frac{1}{3}} R_{e,D_{in}}^{0.8} \left(1+\left(\frac{L_{en,t}}{D_{in}}\right)^{0.7}\right) \quad \text{Eq.15}$$

Theamountofheateffectivelyexchangedbetweentherefrigerantandtheoutsideenvironmentis given by:

$$Q_{cond}=U_O S_{ef,o,cond}(T_{ref}-T_a) \quad \text{Eq.16}$$

The amount of heat effectively exchanged by convection between the condenser outer surfaceand its surrounding air is expressed as follows:

$$Q_{conv}=h_O S_{ef,o,cond}(T_w-T_a) \quad \text{Eq.17}$$

When the fin-tube contact zones are not taken into account during the calculation, the quantityofheatexchangedbetweentherefrigerantandtheambientairandthatexchangedbyconvectionbetween thecondenser outersurface and the surroundingair are givenrespectivelyby:

$$Q_{cond}'= U_o'S_{o,cond}(T_{ref}-T_a) \quad \text{Eq.18}$$

$$Q_{conv}'=h_o S_{o,cond}(T_w-T_a) \quad \text{Eq.19}$$

With $U_o' = \frac{1}{\frac{1}{\eta_o' h_o}+\frac{S_{o,cond}\ \ln\left(\frac{D_o}{D_{in}}\right)}{2\pi\lambda_t N_{tot,t} L_{en,t}}+\frac{S_{o,cond}}{h_{in} S_{in,t}}}$ et $\eta_o' = 1-\frac{S_{o,f}}{S_{o,cond}}\left(1-\eta_f\right)$

The condenser convective heat flux exchanged by the bare tubes without adding fins is givenby:

$$Q_{conv,t}=h_o S_{o,t}(T_w-T_a) \quad \text{Eq.20}$$

Sincetheheatflowdischargedfromtherefrigeranttotheambientairisequaltotheheatflowtransmittedtothecondenserouter wallbytherefrigerant,thecondenserouterwalltemperature

$T_w$can be obtained from thetemperatureof therefrigerantT_{ref}as follows:

$$T_w = T_{ref} - Q_{cond}\left(\frac{ln\left(\frac{D_o}{D_{in}}\right)}{2\pi\lambda_t N_{tot,t} L_{en,t}} + \frac{1}{h_{in} S_{in,t}}\right) \quad \text{Eq.21}$$

Inthisstudy,thetemperatureistakenasconstantovertheentirewallofthecondenser.Thetemperatureoftherefrigerantinthecondenseristhereforeequaltoitscondensing temperatureT_k = T_{ref}(without de-superheating and sub-cooling). By choosing a condensertemperaturedifferenceΔT_{cond} = 15 Kbetweentheambientairandtherefrigerant,thecondensingtemperature can be obtained from the condenserair inlet temperatureby:

$$T_k = T_{ref} = T_a + 15 \quad \text{Eq.22}$$

Theheat fluxgains dueto theinput of fins are given by:

$$Q_{conv,gain} = Q_{conv} - Q_{conv,t} \quad \text{Eq.23}$$

Theheat fluxlosses duetothecontact zones is expressed asfollows:

$$Q_{conv,lost} = Q_{conv}' - Q_{conv} \quad \text{Eq.24}$$

DETERMINATIONOF THERATIOOF THELOSSESTOTHEHEATFLUXGAINS

The ratio of the losses to the heat flux gains characterizes the effects of the fin-tube contactzones on the condenser exchangedconvectiveheat flux.It is determined asfollows:

$$r = \frac{Q_{conv,lost}}{Q_{conv,gain}} \quad \text{Eq.25}$$

Method:-

ThepropertiesoftheR134aasachosenrefrigerantareobtainedfromtheSOLKANE8softwareat its condensing temperature in its state of saturated vapor. The air flow at the inlet of thecondenseris taken at its stateofambient temperatureat thespeed rate of2 $m.s^{-1}$ (see**Table1**).In this study, the thermal conductivity of the inlet air flow is taken as 0.026 $W.m^{-1}.K^{-1}$. Thecondensing temperature ofthe R134a refrigerant is determined from the temperature of thecondenser inlet air flow by setting a condenser temperature difference of ΔT_{cond} = 15 K. Thevertical pitch of the tubes is taken as equal to their horizontal one. The calculation code isdevelopedonMatlabsoftwareinordertovisualizetheevolutionsofthe exchangedconvectiveheat flows and the ratio of the losses to the heat flux gains (see **figure 3**). The results areobtained by varying the condenser parameters such as, the total number of tubes, their outerdiameter,thetotalnumberoffinsandtheirthickness.Beforevaryingthenumberoftubes(fins),first we determine their maximum number in the condenser, then we maintain the number offins (tubes) at its maximum value before varying the number of tubes (fins) from 1 to theirmaximumnumber.Everytimeweputonetube(fin),wedeterminethefin-tubecontactsurfaces,the exchanged heat flows and the ratio of the losses to the heat flux gains. Throughout thisstudy, the pitches of the tubes and fins are constant when varying the other condenser above-mentioned parameters. The tube thickness is also kept constant, therefore any change in thetube outer diameter will implya change in the tube innerdiameter.

Table1:-Inputparameters forthemodel.

Parameters	Values
Thermalconductivity oftheR134arefrigerant,λ_{refr}($W.m^{-1}.K^{-1}$)	$15.34.10^{-3}$
Thermalcapacity ofR134a,$C_{p,ref}$($J.kg^{-1}.K^{-1}$)	1118
ViscosityofR134a,μ_{ref}(Pa.s)	$12.8980.10^{-6}$
Condensertemperaturedifference,ΔT_{cond}(K)	15
Tubelength,$L_{en,t}$(m)	0.44
Tubeouterdiameter,D_o(m)	0.016

Tube inner diameter, D_{in}(m)	0.013
Thermal conductivityofthetubes(copper), λ_t((W.m^{-1}. K^{-1}))	360
Tubeverticalpitch, $p_{v,t}$	0.045
Tubehorizontalpitch, $p_{h,t}$	0.045
Widthofthefins, $L_{ar,f}$(m)	0.1
Finthickness, E(m)	0.0003
Fin pitch, p_f	0.001
Finheight, H_f(m)	0.416
Finthermal conductivity(aluminium), λ_f((W. m^{-1}.K^{-1}))	203
Airmaximalspeedrate, $V_{a,max}$(m.s^{-1})	2
Condenserairinlettemperature, T_a(K)	298
Airviscosity, μ_a(Pa.s)	$1.7894.10^{-6}$
Airthermalconductivity, λ_a((W.m^{-1}.K^{-1}))	0.026
Airthermalcapacity, $C_{p,a}$((J.kg^{-1}.K^{-1}))	1008
Air massvolume, ρ_a(Kg.m^{-3})	1.225
Fin-on-tube thermalcontactresistance, $R_{f/t}$(m^2.K.W^{-1})	0.00205

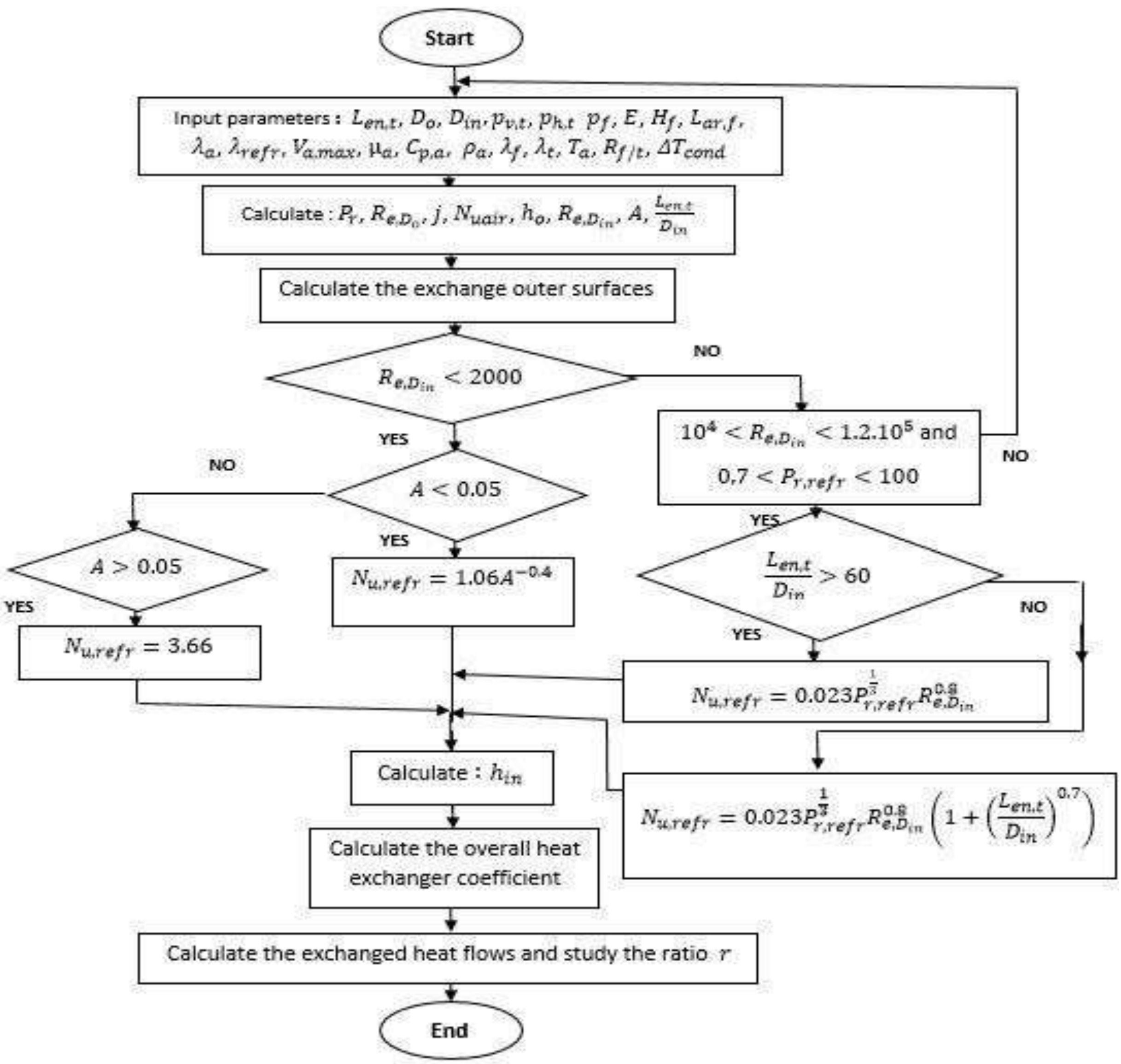
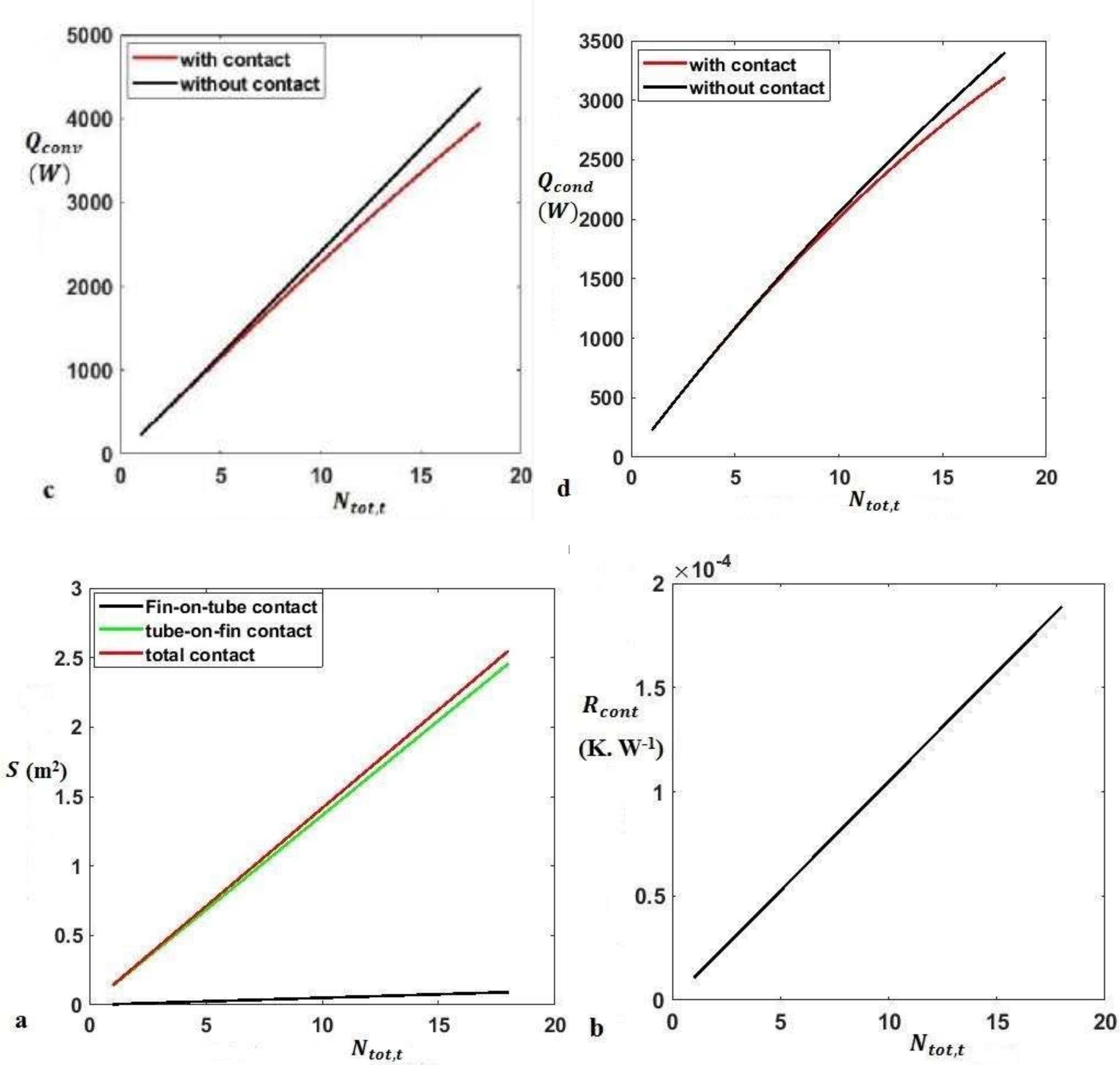


Figure3:- CalculationcodeonMatlabsoftware.

Results and discussions3-1Influenceoftubes



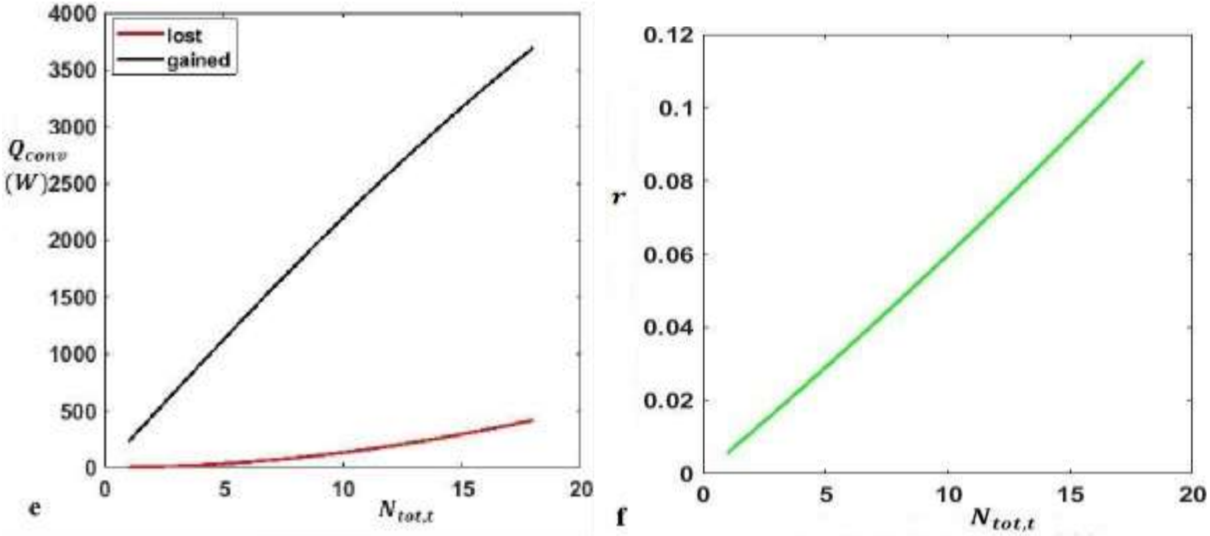
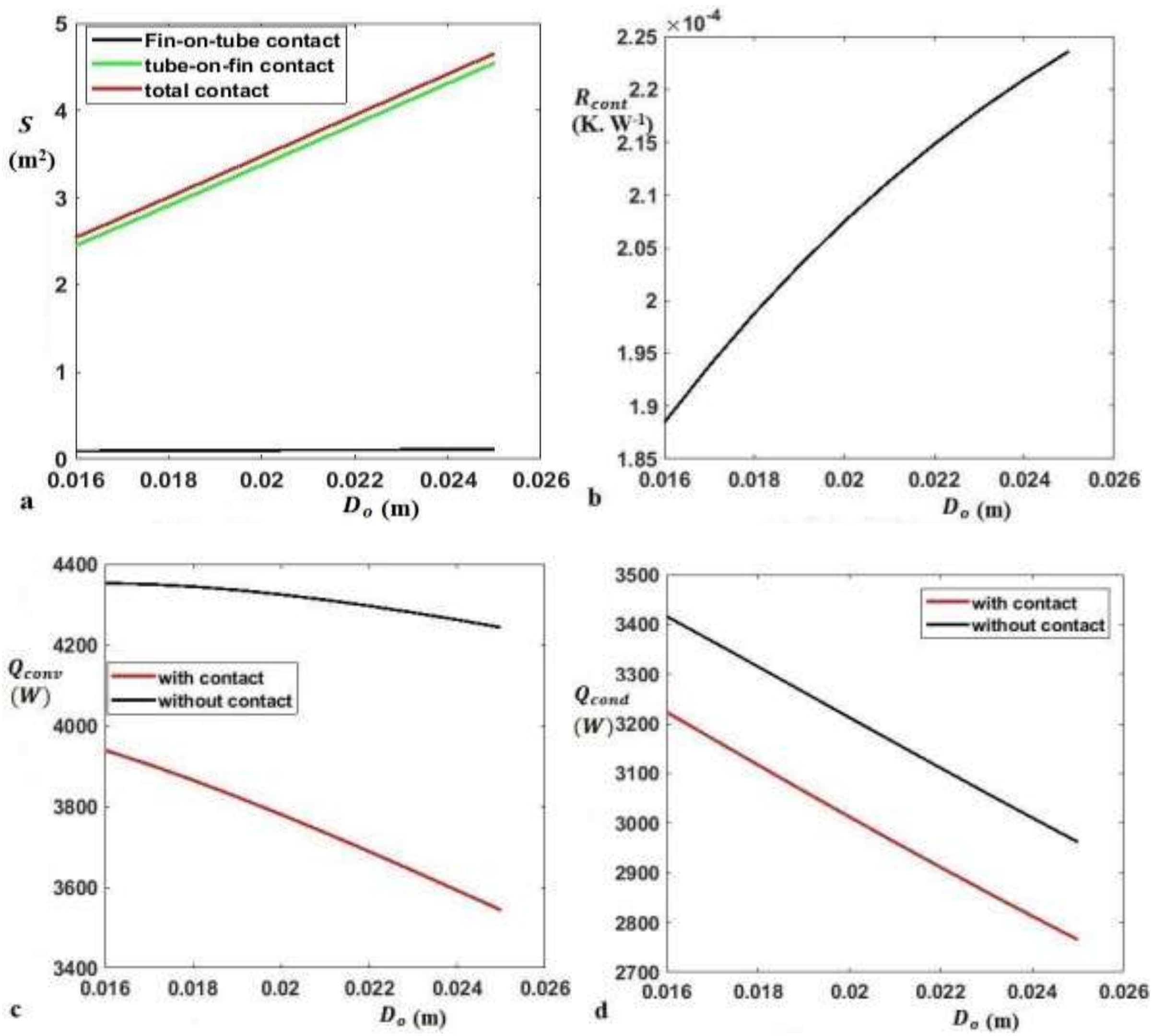


Figure 4:- The fin-tube contact surface (a), the total thermal contact resistance (b), the convective heatflows (c), the condenser exchanged heat flows (d), the heat flux gains and losses (e) and the ratio oflossesto theheatflux gains(f)asafunctionofthe condenser total numberof tubes.



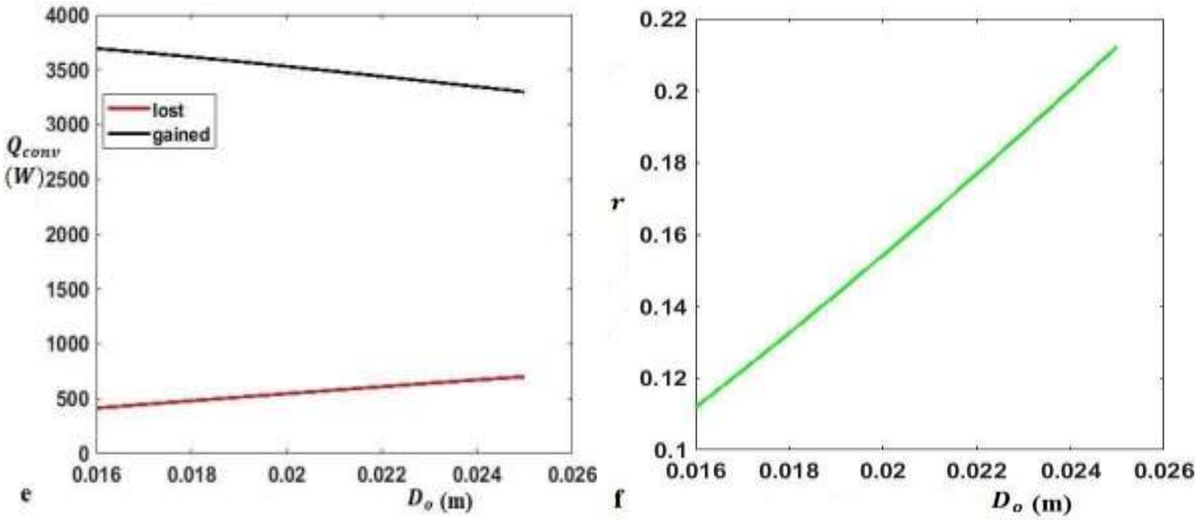
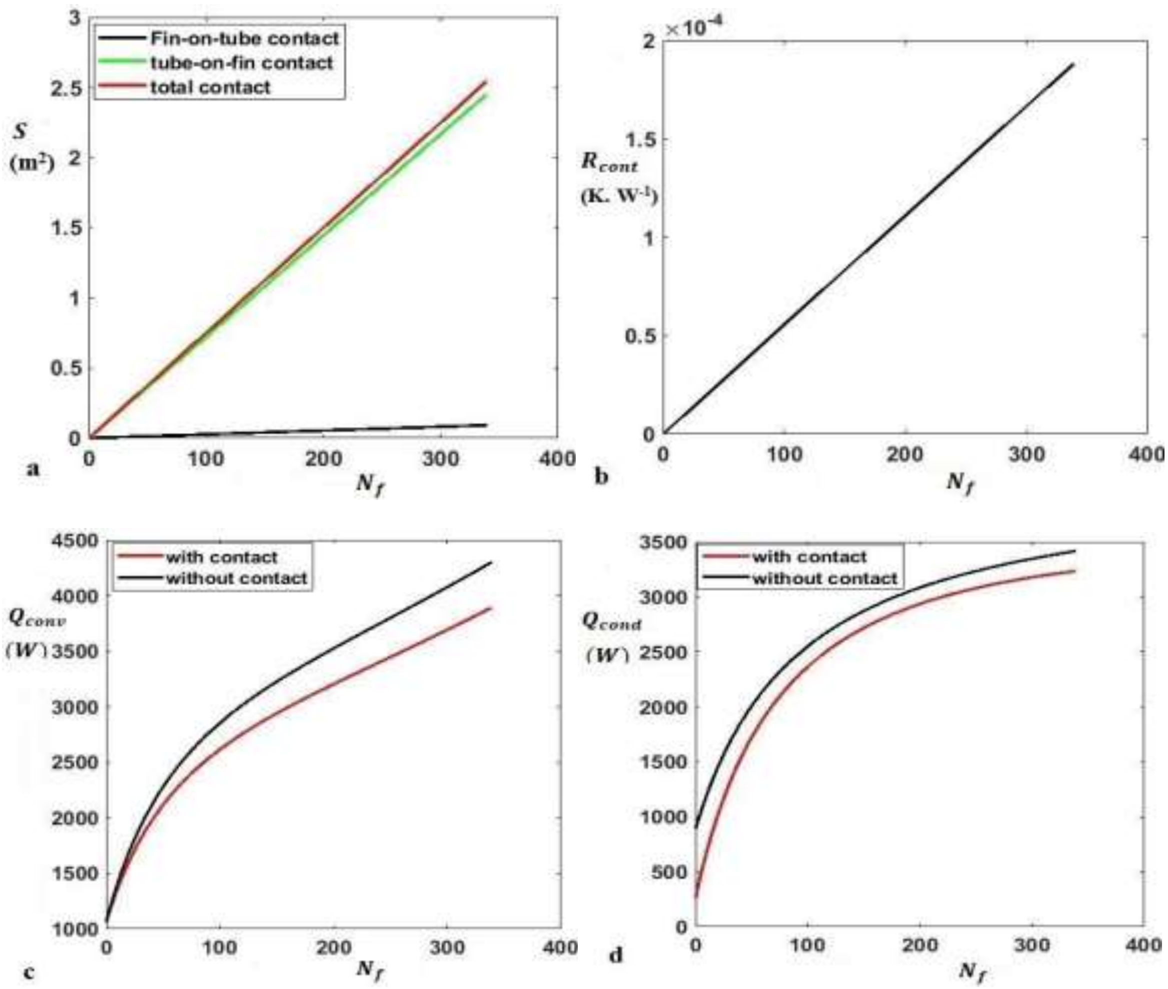


Figure 5:- The fin-tube contact surface (a), the total thermal contact resistance (b), the convective heatflows (c), the condenser exchanged heat flows (d), the heat flux gains and losses (e) and the ratio oflossesto the heatflux gains(f)asa function ofthetubeouter diameter.

For the same number of fins in the condenser, the ratio of the losses to the heat flux gains duetotheinputoffinsincreasesasafunctionofthenumberoftubesandasafunctionoftheirouterdiameter, as it's illustrated respectively in **figures 4f** and **5f**. This is due to the increase in thefin-tube contact surfaces as noted in **figures 4a** and **5a** which show that the contact areaoccupied by the tubes on the fins is very large whatever the number of tubes and their outerdiameter.Thesechangescanbejustifiedbythefactthateachtubeoccupiesthetwolateralfacesof each fin. Consequently, the more the number of tubes or their outer diameter is increased,the more the outer surface of the fins decreases and the more the contact area of the tubes onthe fins increases. These two figures also show that the contact surface of the fins on the tubeshardly increases with the number of tubes (see **fig. 4a**) and with their outer diameter (**see fig.5a**). It is also observed from these figures that the total fin-tube contact area is greater with thetube outer diameter thanwith their number.

In **figures 4b** and **5b**, there is an increase in the fin-tube total thermal contact resistance,respectively, as a function of the number of tubes and as a function of their outer diameter,which is dueto theincreasein thefin-tube total contact surfacein **figures 4a** and **5a**.In **figure4e**,theheatfluxgainsandtheheatfluxlossesduetotheinputoffinsincreasewiththenumberof tubes. Thetrend ofthe heat flux gains as afunction of thenumberof tubes can be explainedby the fact that increasing the number of tubes leads to an increase in the condenser outersurface. However, the increase in the number of tubes also implies an increase in the contactsurface of the tubes on the fins (see **fig. 4a** and **fig. 5a**), having the effect of reducing the outersurface of the fins which causes heat flux losses. We notice that there is almost no loss of heatflowforanumberoftubesrangingfrom0to5whichcorrespondstoaratiolessthan0.026($r<$ 0.026).As for the changes in the heat flux as noted in **figure 5e**, they clearly reveal thatthe heat flux gains drop very quickly when the tube outer diameter increases, unlike the heatfluxlosseswhichincreaseslowly.Thiscouldbeduetothefactthat,ontheonehand,increasingthe tube outer diameter is accompanied by a decrease in their number since their pitches(horizontal and vertical) are kept constant, which reduces the heat gains. On the other hand,increasing the tube outer diameter increases the total fin-tube contact area and the fin-tubethermal contact resistance (see **fig. 5a** and **fig. 5b**, respectively), which increases heat losses.**Figures4c**showstheconvectiveheatflowsexchangedbetweenthecondenserandthesurrounding air, respectively, with and without taking into account the effects of the fin-tubecontact zones. It can be seen that when the number of tubes varies from 0 to 5 approximately,thesetwoheatfluxesarealmostthesame.Ontheotherhand,whenthenumberoftubesexceeds5($r>$ 0.026),thereisacleardifferencebetweenthesetwoheatflowswhichincreaseprogressively with the number of tubes. The same trend is observed in **figure 4d** for the heatflows exchanged between the refrigerant and the surrounding air with and without taking intoaccounttheeffectsofthefin-tubecontactzones;theonlydifferenceisthatthesetwoheatflowsare distinguished when the number of tubes reaches approximately 7. It is noticed that theexchanged heat flow without taking into account the effects of the contact zones is alwaysgreater than that with takinginto account theseeffects regardless of the numberof tubes As for the exchanged convective heat flows as a function of the tube outer diameter with andwithout taking into account the effects of the fin-tube contact zones (see **fig. 5c**), a very rapiddecrease in the two heat fluxes can be seen and during which, a gap which becomes more andmore important as the tube outer diameter increases is noticed. The same trend is observed in**figure 5d** for the heat flows exchanged between the refrigerant and the ambient air with andwithout taking into account the effects of the contact zones; the only difference is that the gapis a constant regardless of the tube outer diameter. These trends are explained by the fact thatincreasing the tube outer diameter reduces the heat flux gains while promoting the heat fluxlosses (see **fig. 5f**).

Influenceof fins



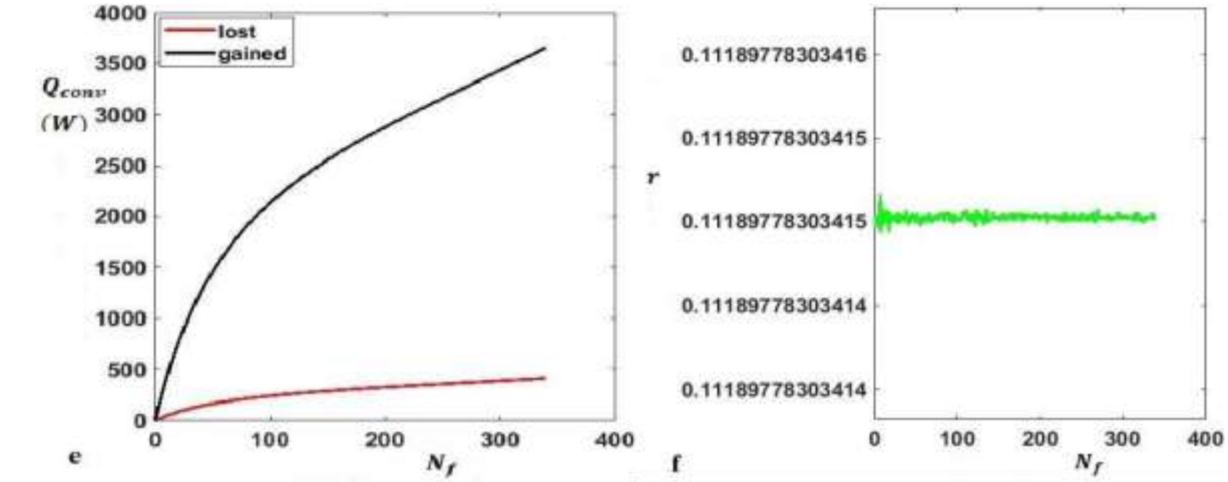
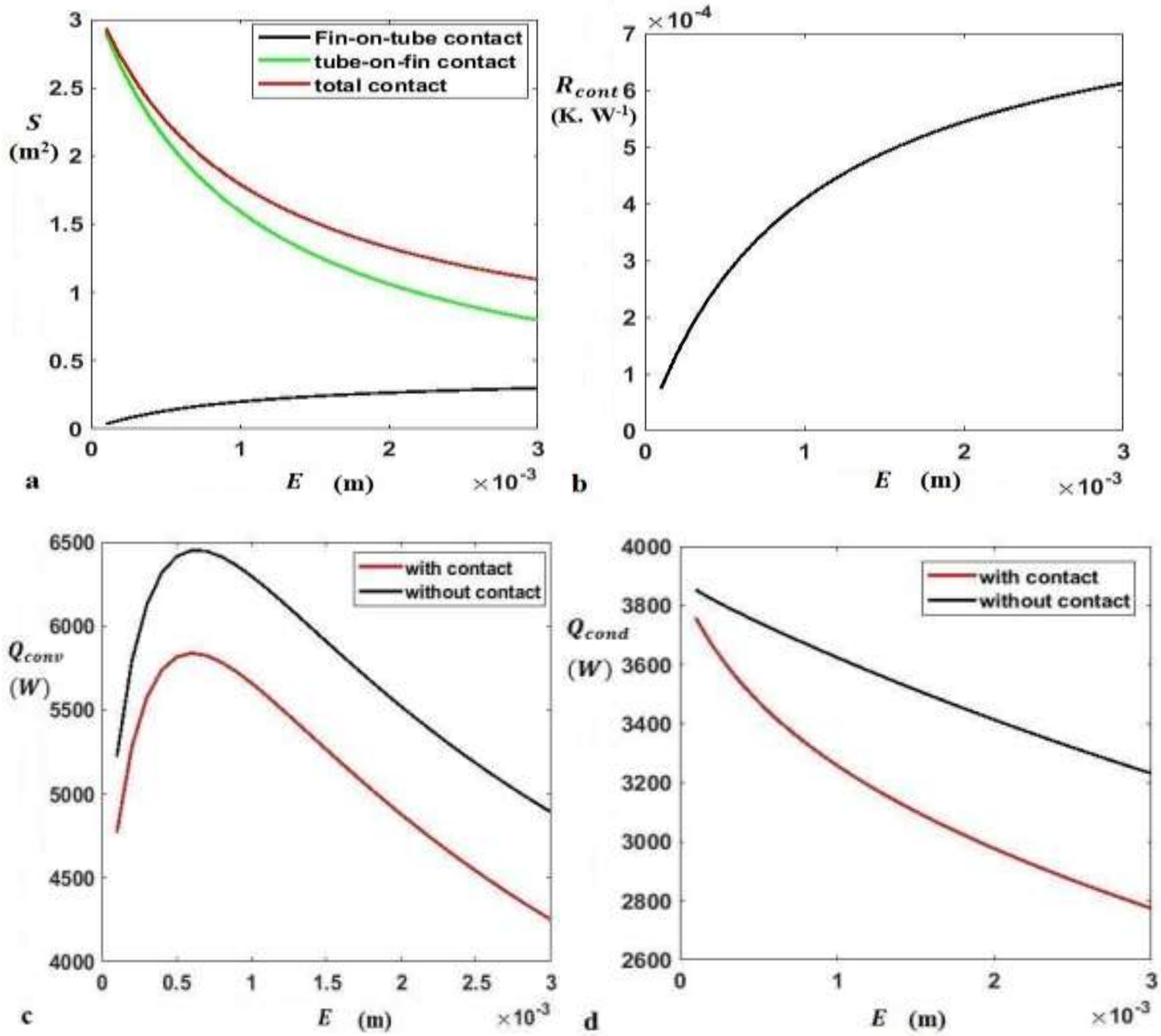


Figure 6: The fin-tube contact surface (a), the total thermal contact resistance (b), the convective heatflows (c), the condenser exchanged heat flows (d), the heat flux gains and losses (e) and the ratio oflossesto the heatflux gains(f)asa function ofthe condenser total numberof fins



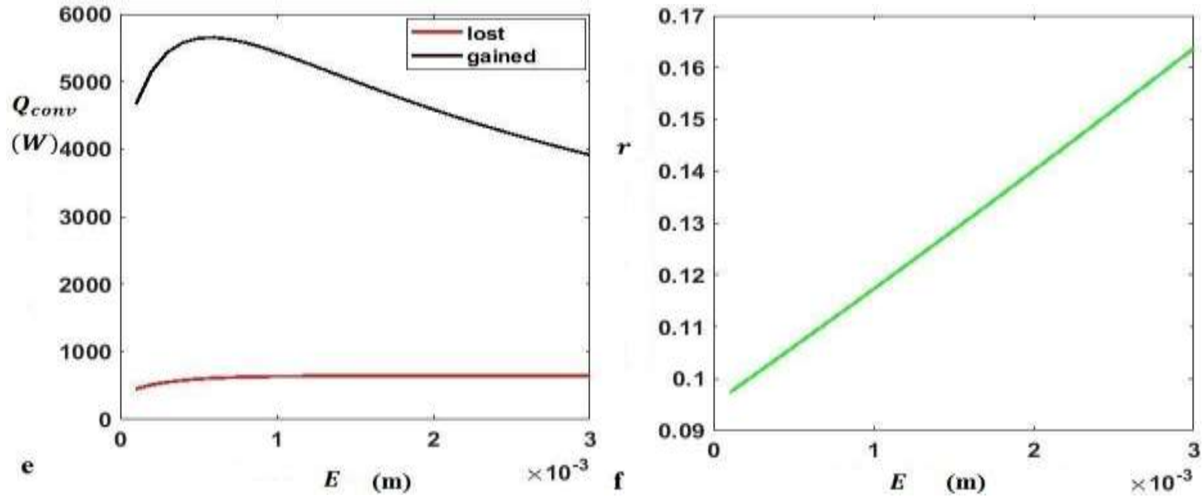


Figure 7:- The fin-tube contact surface (a), the total thermal contact resistance (b), the convective heatflows (c), the condenser exchanged heat flows (d), the heat flux gains and losses (e) and the ratio oflossesto the heatflux gains(f)asa function ofthefinthickness

Figure 6f shows that, for the same number of tubes, the variation in the number of fins doesnotaffecttheratioofthelossestotheheatfluxgainsduetotheinputoffins.Ontheotherhand,when their thickness increases, as shown in **figure 7f**, this ratio grows swiftly, faster than thatasafunctionofthenumberoftubesandslowerthanthatasafunctionofthetubeouterdiameter(**fig.4f** and **fig.5f**, respectively). In **figure 6a**, it is noticed that the contact area occupied by thefinsonthetubesisverysmallregardlessofthenumberoffinsandisverynegligiblecomparedto that occupied by the tubes on the fins. From **figure 7a**, it can be seen a somewhat rapidincrease in the contact surface of the fins on the tubes and a gradual decrease in the contactsurface of the tubes on the fins and the total contact surface as a function of the fin thickness.The changes in the contact surfaces are justified by the fact that increasing the number of finsimplies an increase in the area occupied by the tubes on the fins and since the fin thickness isverysmall comparedto thetubeouterdiameter,thentheareaoccupiedbythefinsonthe tubesis negligible compared to that occupied by the tubes on the fins (**see fig. 6a**). However, whenthefinthicknessincreases,sincetheirpitchiskeptconstant,thenumberoffinsinthecondensermust necessarily decrease while the area occupied by the fins on the tubes increases, whichexplains the appearanceof thecontact surfaces asnoted in**figure 7a**.

The evolution of the heat flux gains due to the input of fins noted in **figure 6e** shows that itincreasesveryquicklyasafunctionofthenumberoffins,whichisduetothefactthatincreasingthenumberoffinsmakesitpossibletoincreasethecondenseroutersurface.Ontheotherhand,as it is illustrated in **figure 7e**, when the fin thickness increases, the heat flux gains increaserapidly until reaching a maximum for a critical thickness value of about 0.5 mm and begins todropgraduallywithincreasingthefinthickness.Thefirsttrendisduetothefactthatincreasingthe fin thickness allows the condenser front surface to increase, which promotes a largerexchange surface with the surrounding air. However, when this thickness reaches a criticalvalue (0.5 mm), the number of fins in the condenser begins to decrease with any additionalincreaseinthefinthickness.Thisdecreasewillleadtoareductioninthecondenseroutersurfacewhich explains the drop in the heat flux gains. As for the evolution of the heat flux losses in**figure 6e**, it increases slowly with the number of fins while keeping almost a constant rate in**figure 7a** with the increase in the fin thickness. Changes in the heat flux losses are due to theshapeof the contact surfaces noted respectivelyin**figures 6a** and **7a**.

In **figures6c**and**7c** areillustratedthe evolutiontheconvectiveheat flows exchanged betweenthe condenser and the surrounding air, respectively, as a function of the number of fins and asa function of the fin thickness with and without taking into account the effects of the fin-tubecontact zones. It can be seen from **figures 6c** that these heat flows increase with the number offins. From this figure, it is observed that the exchanged heat flux without taking into accounttheeffectsofthecontactzonesisgreaterthanthatwithtakingintoaccounttheseeffectsexceptfor small numbers of fins (between 1 and 15) where the two heat flows are almost equal.However, as shown in **figure 6d**, there is a constant gap between the heat flows exchangedbetweentherefrigerantandthesurroundingairwithandwithouttakingintoaccounttheeffectsofthecontactzones.Theincreaseintheheatfluxesin**figures6c**and**6d**isduetotheevolutionof theheat fluxgains in**figure 6e**.

In**figure7c**,thetwoexchangedconvectiveheatflowsbetweentheouterwallofthecondenserand the surrounding air with and without taking into account the effects of the contact zonesfirst keep a small constant gap until the fin thickness reaches 0.5 mm, then, any additionalincrease in the fin thickness causes the heat flows to drop while keeping a larger constant gap.Thesedifferentpacesareduetotheevolutionoftheheatfluxgainsin**figure7e**.Asforthegapbetweentheexchangedheatflowsbetweentherefrigerantandtheambientairwithandwithouttaking into account the effects of the contact zones, as shown in **figure 7d**, it increases rapidlydepending on the fin thickness while the heat flows fall. Changes in the fin-tube total thermalcontact resistance as a function of the number of fins (see **fig. 6b**) and as a function of theirthickness (see **fig. 7b**) are due to the shape of the fin-tube contact surfaces respectively in**figures 6a** and **7a**. It can be seen that, before the fin thickness reaches its critical value (0.5mm),thisresistanceincreasesmorerapidlyas afunctionofthe finthicknessthanas afunctionof their number. This can be explained by the fact that the contact surface of the fins on thetubes is greater as a function of the fin thickness than as a function of their number. However,when the fin thickness reaches its critical value, the thermal contact resistance 'growth beginsto slow down due to the fact that after this critical value, any increase in the value of the finthickness causesa decrease in the numberof fins.

Conclusion:-

In this work, the effect of the tube-fin contact zones on convective heat transfers between thecondenseroutersurfaceanditssurroundingairhasbeeninvestigated.Amethodofdeterminingthe condenser outer area has been proposed and a ratio which compares the losses to the heatflux gains due to the input of fins has been defined and studied. It is concluded that the effectsofthefin-tubecontactzonescanbeneglectedwhentheratioofthelossestotheheatfluxgainsis less than 0.026. Otherwise, when this ratio is greater than 0.026, these effects have to beconsidered when determining the condenser exchanged convective heat flux with the ambientair. It is also drawn from this study that the condenser exchanged heat flux without taking intoaccount the effects of the fin-tube contact zones is always greater than that with consideringthese effects regardless of the number of tubes and their outer diameter and the number of finsand their thickness.

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