Study and conception of a heat recovery system for hybrid power plants – Development of a thermal design software

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Abstract:

In the wind-diesel compressed air hybrid system (WDCAHS), the wind power surplus is used to compress the air via a compressor and store it in a tank. When air is compressed, its temperature increases. Using a multiple stage compressor allows introducing a cooling system between the stages, which saves energy in the compression process. Thus, in a multi-stage compressor, air can be partially compressed using isentropic compression, and then cooled in a heat exchanger (heat recovery system). This brings the compression process closer to isothermal compression, which is more efficient. There is a distinct efficiency advantage to multiple stages compressor when heat exchangers are used. Indeed, without inter-cooling, the savings would not be realized and the efficiency would be no different from a single stage machine.
The purpose of this work is to design heat recovery system, adaptable to WDCAHS, which cools the compressed air between each compression stage and so store heat recovered in a thermal tank to reheat the air before inject it into diesel engine. The optimum thermal design of a heat exchanger involves the consideration of many interrelated design parameters such as : the inlet and outlet temperatures, the overall heat transfer coefficient, the geometry of the heat exchanger (surface of exchange) and the heat transfer rate between the two fluids (hot and cold). Two most common heat exchanger design problems are those of rating and sizing.

This paper is structured on software developed especially to enable design and rating to be carried out within a total process model integrating a steady state modelling and a detailed analysis of various aspects of heat exchanger design such as heat exchanger configurations, calculation methods, technical constraints, etc. Indeed, there are some software design and rating packages available but we have decided to develop a tool (software) specific to WDCAHS, which enables the designer to study the effects of the many interacting design parameters and achieve an optimum thermal design. This software is supported by extensive component physical property databases, heat exchanger and recovery systems adaptable to WDCAHS, and thermodynamic models. Thus, a design and calculation method is presented which includes key information for data inputs and a verification process of the results from software calculations.

Keywords:

Design of heat exchangers, heat transfer, thermal storage, Hybrid systems, energy efficiency

1. Generalities about thermal energy storage (TES)

Thermal energy storage (TES) is widely recognized as a means to integrate renewable energies into the electricity power systems on the generation side, but its applicability to the demand side is also possible [1, 2]. In recent decades, TES systems have demonstrated a capability to shift electrical loads from high-peak to off-peak hours, so they have the potential to become a powerful instrument in demand-side management programs. Thermal energy storage is a technology that ensures energy security, efficiency and environmental quality. The TES can also be defined as the temporary storage of thermal energy at high or low temperatures. TES systems have the potential of increasing the effective use of thermal energy equipment and of facilitating large-scale switching. They are normally useful for correcting the mismatch between supply and demand energy [3]. Certainly, TES is of particular and significance interest, especially in relation to solar thermal applications such as heating, hot water, cooling, air-conditioning…., because of its intermittent nature. In these applications, a TES system must be able to retain the energy absorbed for at least a few days in order to supply the energy needed on cloudy days when the solar energy is low [4, 5, 6].

The basic principle of a TES system is the same for all the applications: energy is supplied to the TES (charging phase), it is stored (storing phase) and then it is removed from the TES for a later use (discharging phase) [7, 8]. In practical systems, some steps may occur simultaneously, and each step can happen more than once in each storage cycle [4, 5]. There are different criteria that lead to various categories of thermal energy storage technologies. If the criterion is based on the temperature level of stored thermal energy, the thermal storage solutions can be divided into "low temperature thermal energy storage (LTTES)" and "high temperature thermal energy storage (HTTES)" [4, 5]. LTTES operates in a temperature range below 200°C and has been extensively investigated and developed. LTTES applications can be found in building heating and cooling [9], in solar cooking, in solar water boilers and air-heating systems, and in solar greenhouses [10, 11]. HTTES plays a vital role in renewable energy technologies and waste heat recovery. There is a wide range of industrial applications where waste heat can be recovered, as in the manufacturing of construction materials industry, industry and in the metallurgical and mining industry in general [12, 13]. Today, most HTTES usages are however focused upon applications of solar thermal energy [14, 15, 16].

1.1 The wind-diesel compressed air hybrid system (WDCAHS)

In the wind-diesel compressed air hybrid system (WDCAHS), the wind power surplus is used to compress the air at high pressure via a compressor and store it in a tank. The stored compressed air is then used to overfeed the diesel engine and increase its efficiency and decrease its fuel consumption, and consequently its greenhouse gas emissions [7]. This kind of hybridization results only in a small modification of the intake system and an accurate control of the engine's valves.

Nevertheless, the issue of heat transfer remains critical for the CAES. Indeed, when compressed at high pressures, the air sees its temperature increases significantly. Without heat recovery, this part of energy is then lost. In precedent studies carried to quantify fuel savings due to the pneumatic hybridization of diesel generator[7], the compression process and the storage of the air were supposed adiabatic. That means that during compression, the air was not cooled and was then directly stored at its temperature observed at the end of the compression process. Tank's isolation was also considered perfect, so no heat loss was taken into account [18, 19].

Since the compressed air was stored at high temperature (almost 600°C at storage pressure of 300 bars), the volume of the air reservoirs was extremely high. Cooling down the compressed air before storing it may also be a solution to reduce the volume of these tanks; this hypothesis is unrealistic and cannot be assumed to size correctly such system.

That is why a study about thermal energy storage (TES) for the CAES, applied to the high penetration wind–diesel hybrid systems (HWDS) is necessary. In addition to improve the overall energy efficiency, such system allows cooling the compressed air, thus increasing its density and decreasing significantly the volume of the air reservoir [18, 19].

2. The thermal storage concept in a WDCAHS

Several studies have already been conducted in order to identify the best technology by comparing the different existing storage concepts. Some of these studies are applied on solar power plants [3, 20] and some other ones that analyzed the possible configurations for integrating the TES in the WDCAHS [21, 22]. The figure 1 illustrates the principle schema of WDCAHS with integrated thermal energy storage [6].

The purpose of this article is to perform an ideal design study of heat recovery exchanger system dedicated to a wind-diesel hybrid power plant with compressed air energy storage.

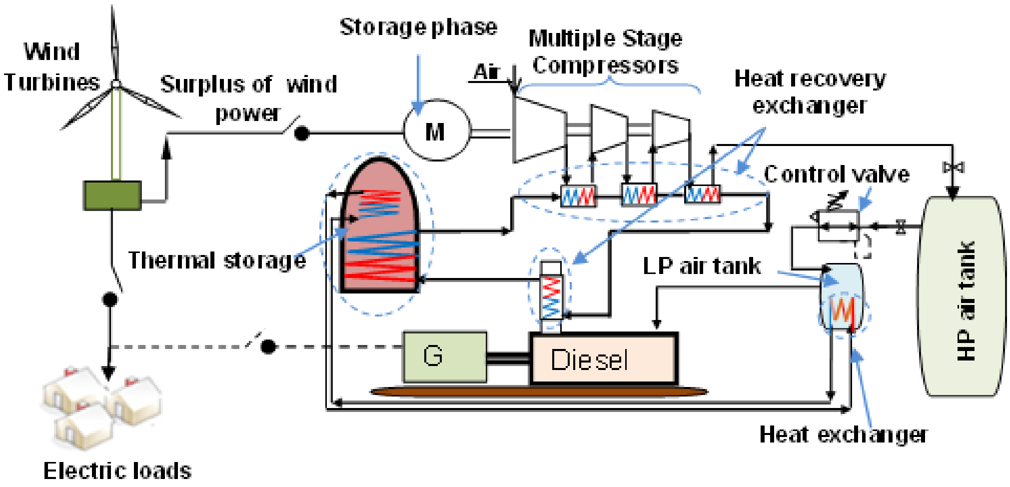


Fig. 1. The heat recovery loop in the wind-diesel compressed air hybrid system WDCAHS

2.1. Compressed air system

3.1.1. Compressors

The compression system chosen for the WDCAHS is shown in Figure 2. This is a reciprocating multistage type compressor. This type of compressors consists of several pistons which are enclosed within the cylinders and equipped with suction and discharge valves. The pistons receive power from main shaft through connecting rod and crankshaft which is driven by electric motor. In multistage air compressor, the fresh air from atmosphere enters in first stage (low pressure) cylinder through suction filter. This air is compressed by piston and then delivered to second stage (middle pressure) cylinder through inter cooler for further compression. In second stage cylinder low pressure, air is compressed and discharged to third stage (high pressure) cylinder through second inter cooler to achieve air pressure up to desired delivery pressure. At high pressure cylinder, the air pressure is increased up to desired discharge range by the piston reciprocating inside the high pressure cylinder. The compressed air is finally discharged into storage tank [18, 19, 23].

The real power of polytropic compression is always higher than the theoretical power reversible and can be calculated for a single-stage compressor, by the following expression [18, 23]:

$$\mathrm{P}_{c_{-1}} = \frac{n_c}{n_c - 1} = \dot{m}_c\, rT_a \left[\left(\frac{P_{ou-c}}{P_a} \right)^{\frac{n_c}{n_c-1}} - 1 \right] \frac{1}{\eta_{p-c}}, \qquad (1)$$

The total power consumed by a multistage compressor (N stages), characterized by the same compression ratio in each stage (in order to have the highest performance) can then be written as follows [18, 23]:

$$\mathrm{P}_c = \frac{n_c N_c}{n_c - 1} = \dot{m}_c\, rT_a \left[\left(\frac{P_{ou-c}}{P_a} \right)^{\frac{n_c-1}{n_c N_c}} \right] \frac{1}{\eta_{p-c}}, \qquad (2)$$

The total ratio of compression (τ_c) process in function of the compression ratio of each stage, (τ_{i-c}), is given by [18, 23]:

$$\tau_c = \frac{P_{ou-c}}{P_a} = (\tau_{i-c})^{N_c}, \qquad (3)$$

$$\tau_{i-c} = \frac{P_1}{P_a} = \frac{P_2}{P_1} = \frac{P_3}{P_2}..... , \qquad (4)$$

After each compression phase, the air is cooled between stages through the integrated heat recovery. The geometry of the compressor and the design of the gas passages between stages will facilitate the dissipation of the heat generated by compression into the surrounding structure [18, 23]. The temperature evolution during compression can be expressed as follows:

$$T_{ou} = T_a.(\tau)^{\frac{n_c-1}{n_c}} \qquad (5)$$

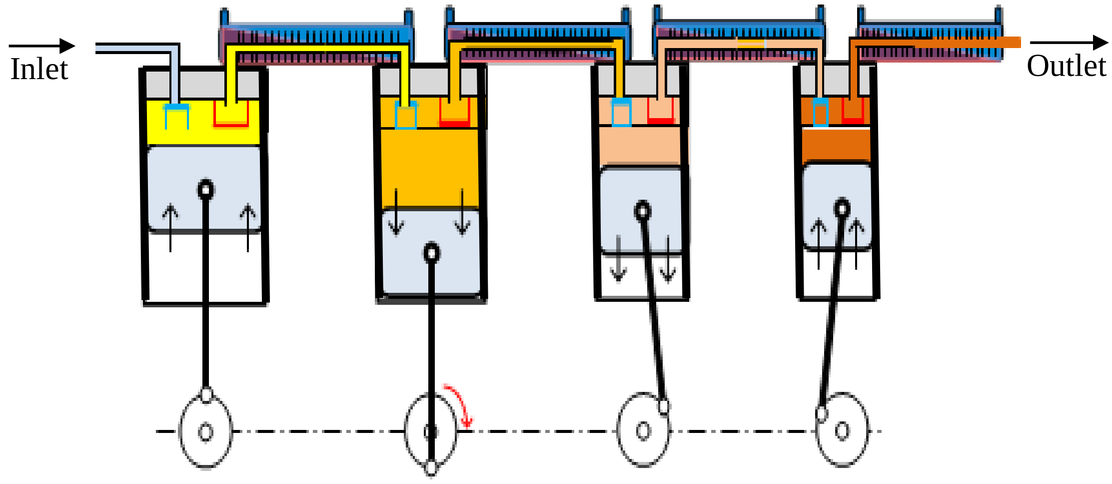


Fig. 2. Multi-stage air compressor with inter-cooler

3. Heat Exchangers

Heat exchangers are devices used to transfer heat between two or more fluid streams at different temperatures. Heat exchangers are widespread used in power generation, chemical processing, electronics cooling, air-conditioning, refrigeration, and in automotive applications. In the following sections, we will examine the basic theory of heat exchangers. In addition, we will examine various aspects of heat exchanger design and analysis [16, 24].

3.1. Heat exchanger design methods

The goal of heat exchanger design is to relate the inlet and outlet temperatures, the overall heat transfer coefficient, and the geometry of the heat exchanger, to the rate of heat transfer between the two fluids. The two most common heat exchanger design problems are those of rating and sizing. We will limit ourselves to the design of the heat recuperators. That is, the design of a two fluid heat exchanger used for the purposes of recovering heat released by the air compression [16].

We will begin first, by discussing the basic principles of heat transfer for a heat exchanger. We may write the enthalpy balance on either fluid stream to give:

$$Q_c = \dot{m}_c \left(h_{c2} - h_{c1}\right) \qquad (6)$$

and

$$Q_h = \dot{m}_h \left(h_{h1} - h_{h2}\right) \qquad (7)$$

For constant specific heats with no change of phase, we may also write:

$$Q_c = (\dot{m}_{cp})_c.(T_{c2} - T_{c1}) \qquad (8)$$

and

$$Q_h = (\dot{m}_{cp})_h.(T_{h1} - T_{h2}) \qquad (9)$$

From energy conservation we know that $Q_c = Q_h = Q$, and that we may relate the heat transfer rate Q and the overall heat transfer coefficient U, to the same mean temperature difference ΔT_m and to the total surface area for heat exchange A, that U is based upon. Later we shall show that [23].

$$\Delta T_m = f\left(T_{h1}, T_{h2}, T_{c1}, T_{c2}\right) \qquad (10)$$

It is now clear that the problem of heat exchanger design comes down to obtaining an expression for the mean temperature difference. Expressions for many flow configurations, i.e. parallel flow, counter flow, and cross flow, have been obtained in the heat transfer field. We will examine these basic expressions later. Two approaches to heat exchanger design that will be discussed are the LMTD method and the effectiveness NTU method. Each of these methods has particular advantages depending upon the nature of the problem specification [16].

3.2. Overall Heat Transfer Coefficient

A heat exchanger analysis always begins with the determination of the overall heat transfer coefficient. The overall heat transfer coefficient may be defined in terms of heat exchangers. Individual thermal resistance of the system combining each of these resistances in series gives:

$$\frac{1}{U.A} = \frac{1}{(\eta_0.h.A)_i} + \frac{1}{S_w.k_w} + \frac{1}{(\eta_0.h.A)_0} \qquad (11)$$

The surface efficiency accounts for the effects of any extended surface which is present on either side of the parting wall. It is related to the fin efficiency of an extended surface in the following manner:

$$\eta_0 = \left(1 - (1 - \eta_{fins})\frac{A_{fins}}{A}\right) \qquad (12)$$

The thermal resistances include: the inner and outer film resistances, inner and outer extended surface efficiencies, and conduction through a dividing wall which keeps the two fluid streams from mixing. The shape factor for a cylindrical wall configuration is given below [16].

Equation (11) is for clean or unfolded heat exchanger surfaces.

$$U\ A = U_0 A_0 = U_i A_i \qquad (13)$$

however,

$$U_0 \neq U_i \qquad (14)$$

Finally, the order of magnitude of the thermal resistances in the definition of the overall heat transfer coefficient can have a significant influence on the calculation of the overall heat transfer coefficient. Depending upon the nature of the fluids, one or more resistances may dominate making additional resistances.

3.3. LMTD Method

The log mean temperature difference (LMTD) is derived in all basic heat transfer texts. It may be written for a parallel flow or counter flow arrangement as follows [16]:

$$\Delta T_{LMTD} = \frac{\Delta T_2 - \Delta T_1}{\ln\left(\frac{\Delta T_2}{\Delta T_1}\right)} \qquad (16)$$

The LMTD expression assumes that the overall heat transfer coefficient is constant along the entire flow length of the heat exchanger. If it is not, then an incremental analysis of the heat exchanger is required. The LMTD method is also applicable to cross flow arrangements when used with the cross flow correction factor. The heat transfer rate for a cross flow heat exchanger may be written as:

$$Q = FUA\Delta T_{LMTD} \quad (17)$$

The LMTD method assumes that both inlet and outlet temperatures are known. When this is not the case, the solution to a heat exchanger problem becomes some-what tedious. An alternate method based upon heat exchanger effectiveness is more appropriate for this type of analysis. If $\Delta T = \Delta T_1 = \Delta T_2$, then the expression for the LMTD reduces simply to ΔT [16].

3.4. NTU Method

The effectiveness / number of transfer units (NTU) method were developed to simplify a number of heat exchanger design problems. The heat exchanger effectiveness is defined as the ratio of the actual heat transfer rate to the maximum possible heat transfer rate if there were infinite surface area. Heat exchanger effectiveness depends upon whether the hot fluid or cold fluid is a minimum fluid. That is the fluid which has the smaller capacity coefficient $C = \dot{m}\, C_p$. If the cold fluid is the minimum fluid then the effectiveness is defined as [16]:

$$\in = \frac{C_{\max}(T_{h,in} - T_{h,ou})}{C_{\min}(T_{h,in} - T_{c,in})} \quad (18)$$

Otherwise, if the hot fluid is the minimum fluid, then the effectiveness is defined as:

$$\in = \frac{C_{\max}(T_{c,ou} - T_{c,in})}{C_{\min}(T_{h,in} - T_{c,in})} \quad (19)$$

We may now define the heat transfer rate as:

$$Q = \in C_{\min}(T_{h,in} - T_{c,in}) \quad (20)$$

It is now possible to develop expressions which relate the heat exchanger effectiveness to another parameter referred to as the number of transfer units (NTU). The value of NTU is defined as [16]:

$$\text{NTU} = \frac{U.A}{C_{\min}} \quad (21)$$

It is now a simple matter to solve a heat exchanger problem when:

$$\in = f(NTU, C_r) \quad (22)$$

$$\text{Cr} = \frac{\text{Cmin}}{\text{Cmax}} \quad (23)$$

Numerous expressions have been obtained which relate the heat exchanger effectiveness to the number of transfer units. The handout summarizes a number of these solutions and the special cases which may be derived from them. For convenience the $\in -NTU$ relationships are given for a simple double pipe heat exchanger for parallel flow and counter flow [16]:

3.4.1. Parallel Flow

$$\in = \frac{1 - \exp\left[- \text{NTU}(1 + C_r)\right]}{1 + C_r} \quad (24)$$

or

$$NTU = \frac{-\ln\left[1 - \in (1 + C_r)\right]}{1 + C_r} \quad (25)$$

3.4.2. Counter Flow

$$\in = \frac{1-\exp[-\mathrm{NTU}(1+C_r)]}{1+C_r\exp[-\mathrm{NTU}(1+C_r)]}, C_r < 1 \quad (26)$$

and

$$\in = \frac{\mathrm{NTU}}{1+\mathrm{NTU}}, C_r = 1 \quad (27)$$

or

$$\mathrm{NTU} = \frac{1}{C_r - 1}, \ln\left(\frac{\in - 1}{\in C_r - 1}\right), C_r < 1 \quad (28)$$

$$\mathrm{NTU} = \frac{\in}{1-\in}, C_r = 1 \quad (29)$$

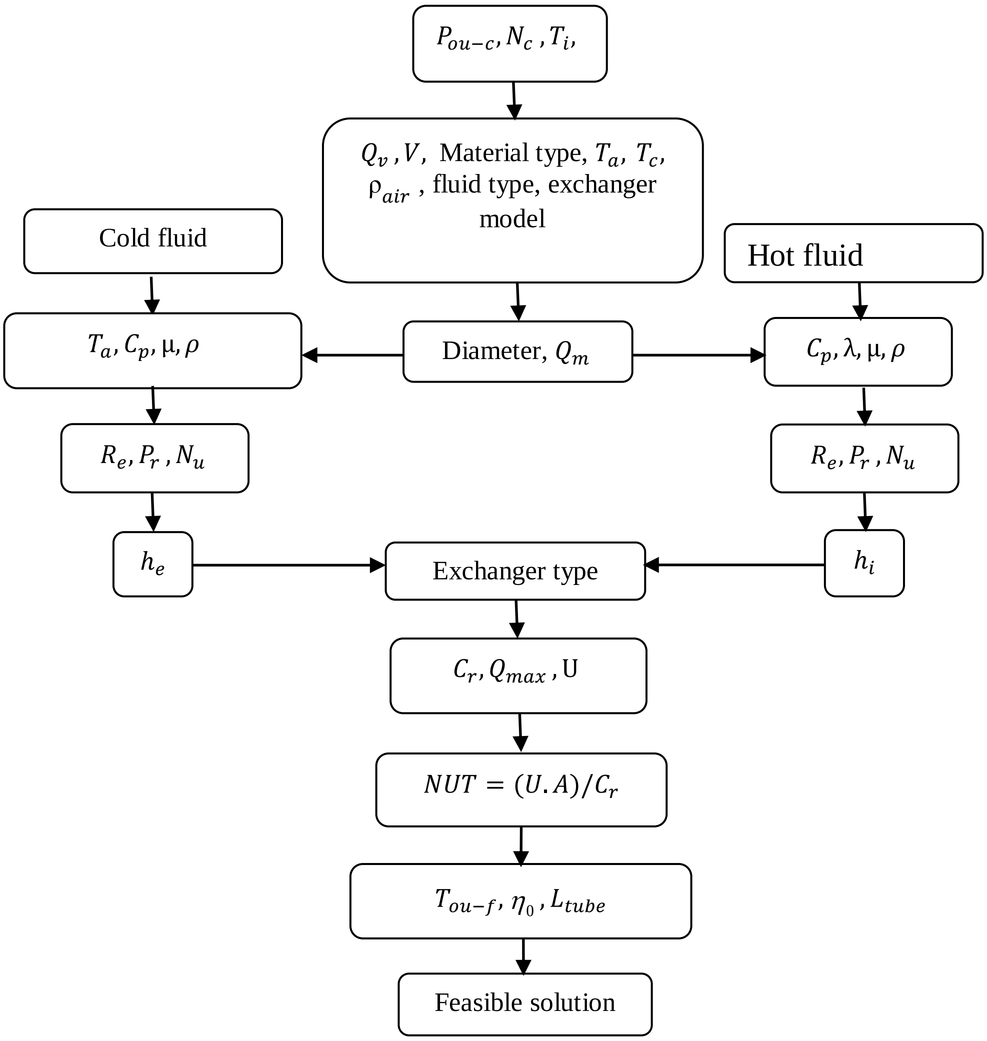


Fig. 3. The processing methods used to design heat exchanger design

3.5. Analysis of Extended Surfaces

Extended surfaces also known as fins are widely used as a means of decreasing the thermal resistance of a system. Indeed, the addition of fins as a means of increasing the overall heat transfer rate is widely employed in compact heat exchanger and heat recuperator design. The aim of this part is to develop and present the theory used to calculate of extended surfaces. Two choices solutions for types of fins will be presented in detail [16]. The annular fins and straight fins, Figures 4 show the methodology used to determine the extended surface and the different results obtained by using the developed software for two case study of heat exchanger: parallel flow and counter flow.

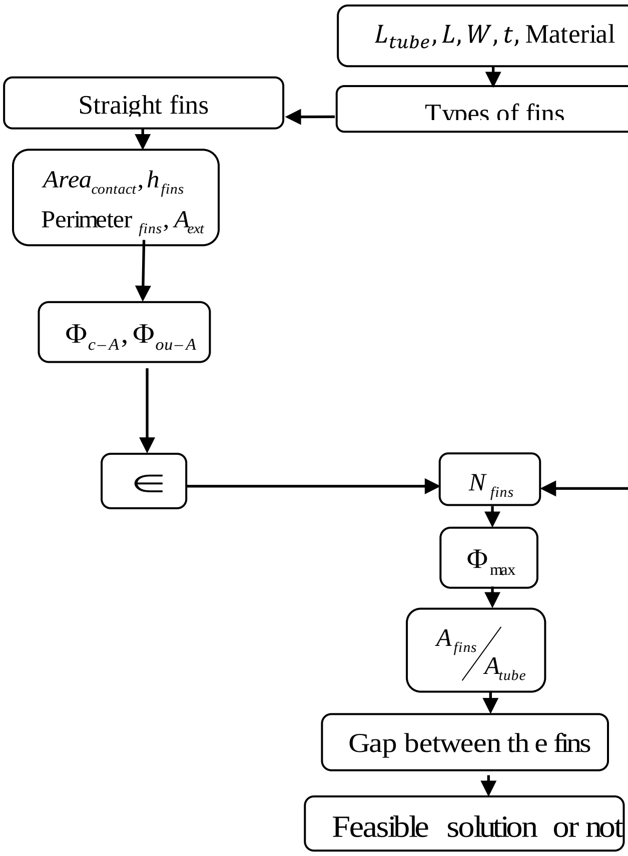
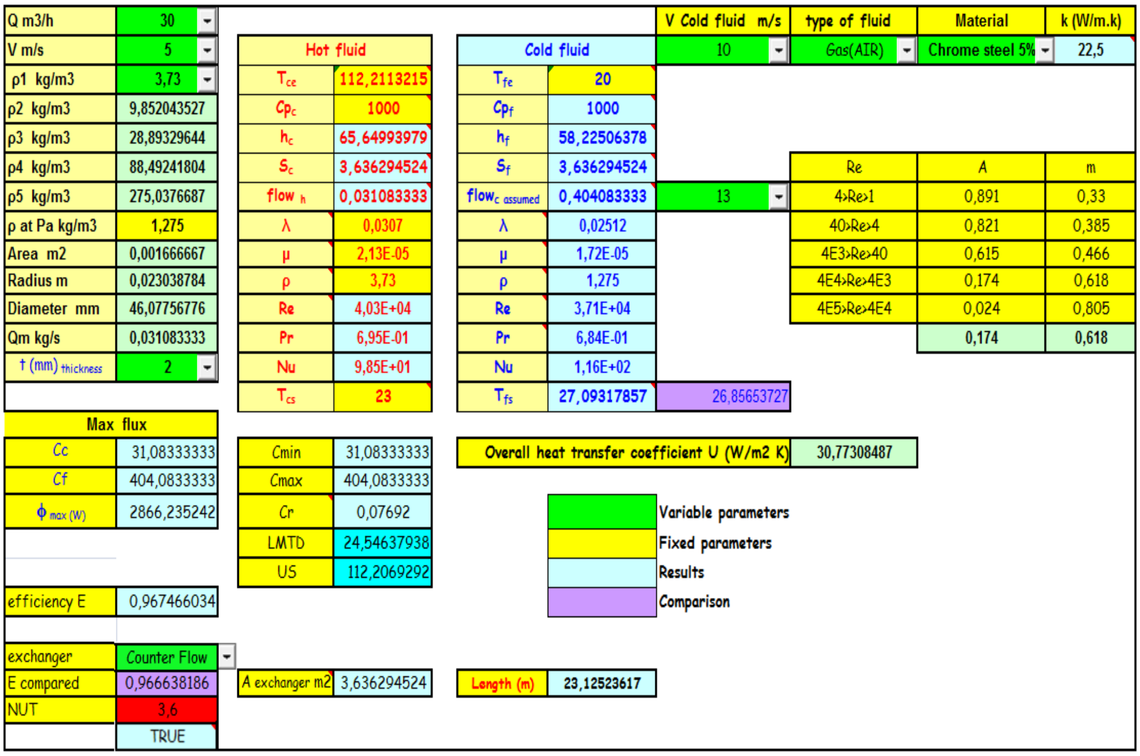


Fig. 4. Organisation char of design extended surface.



Q m3/h	30
V m/s	5
ρ1 kg/m3	3,73
ρ2 kg/m3	9,852043527
ρ3 kg/m3	28,89329644
ρ4 kg/m3	88,49241804
ρ5 kg/m3	275,0376687
ρ at Pa kg/m3	1,275
Area m2	0,001666667
Radius m	0,023038784
Diameter mm	46,07756776
Qm kg/s	0,031083333
t (mm) thickness	2

Max flux	
Cc	31,08333333
Cf	404,0833333
ϕ max (W)	2866,235242
efficiency E	0,967466034
exchanger	Counter Flow
E compared	0,966638186
NUT	3,6
	TRUE

Hot fluid	
Tce	112,2113215
Cpc	1000
hc	65,64993979
Sc	3,636294524
flow h	0,031083333
λ	0,0307
μ	2,13E-05
ρ	3,73
Re	4,03E+04
Pr	6,95E-01
Nu	9,85E+01
Tcs	23
Cmin	31,08333333
Cmax	404,0833333
Cr	0,07692
LMTD	24,54637938
US	112,2069292
A exchanger m2	3,636294524

Cold fluid		
Tfe	20	
Cpf	1000	
hf	58,22506378	
Sf	3,636294524	
flowc assumed	0,404083333	13
λ	0,02512	
μ	1,72E-05	
ρ	1,275	
Re	3,71E+04	
Pr	6,84E-01	
Nu	1,16E+02	
Tfs	27,09317857	26,85653727
Overall heat transfer coefficient U (W/m2 K)	30,77308487	
Length (m)	23,12523617	

V Cold fluid m/s	type of fluid	Material	k (W/m.k)
10	Gas(AIR)	Chrome steel 5%	22,5

Re	A	m
4>Re>1	0,891	0,33
40>Re>4	0,821	0,385
4E3>Re>40	0,615	0,466
4E4>Re>4E3	0,174	0,618
4E5>Re>4E4	0,024	0,805
	0,174	0,618

Variable parameters
Fixed parameters
Results
Comparison

Fig. 5. Sheet of thermal analysis

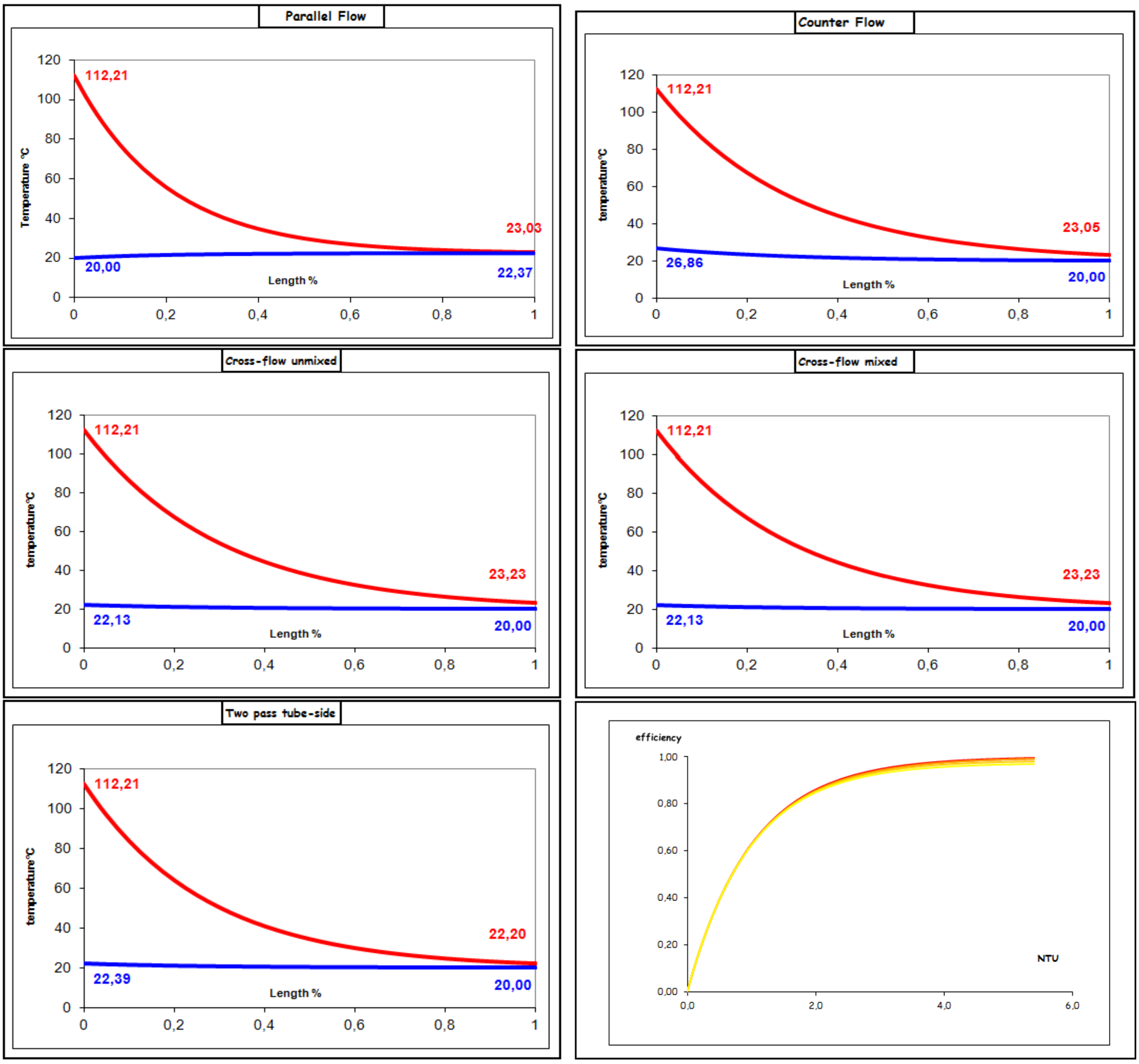


Fig. 6. Efficiency and the temperature profile in heat transfer for different exchangers.

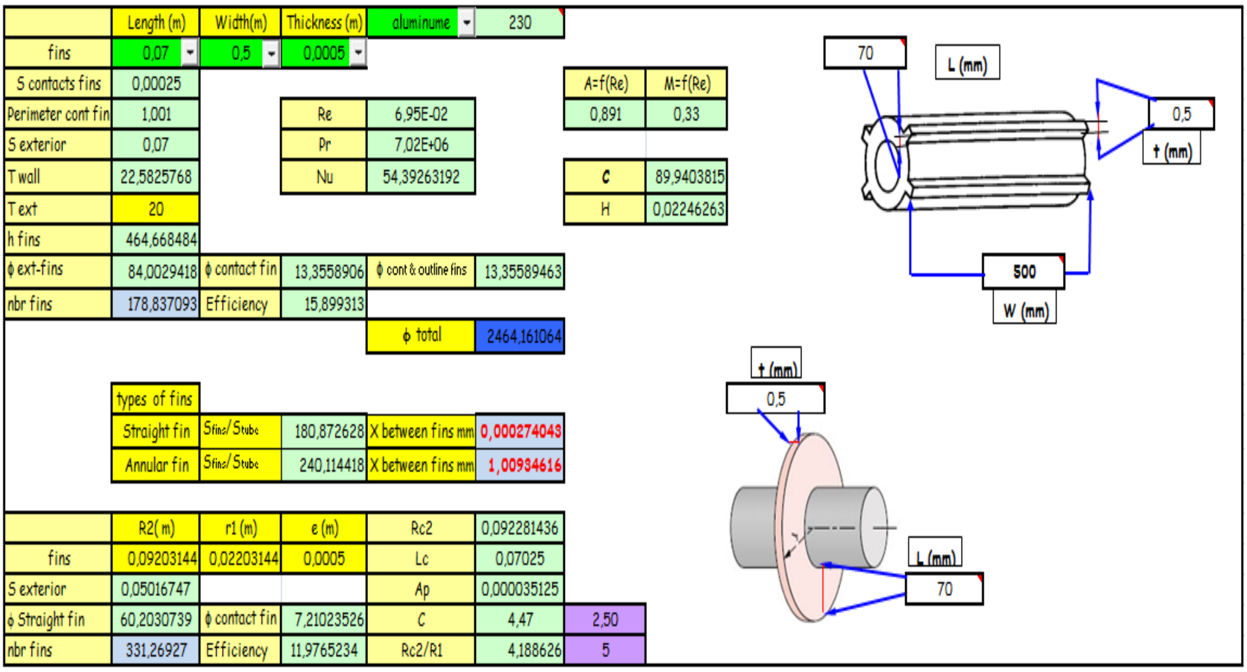


Fig. 7. Spreadsheet of extended surfaces

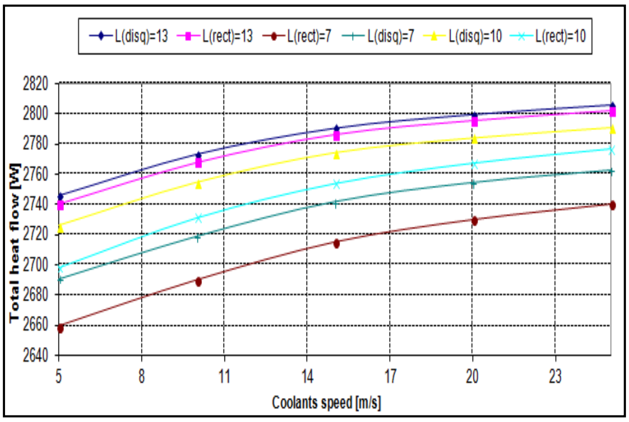


Fig. 8. Total heat flux according to the Coolants Velocity

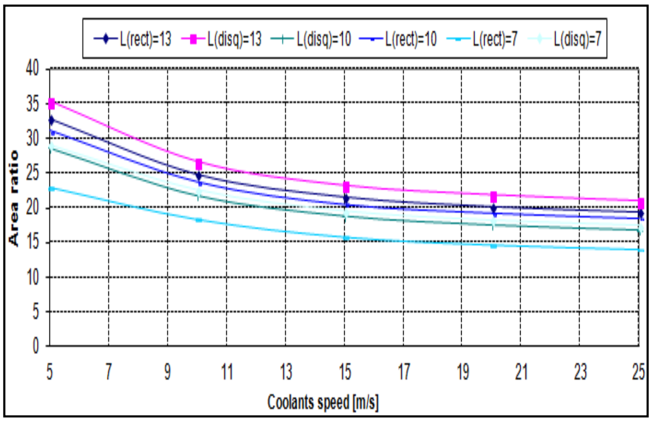


Fig. 9. Area ratio according to the coolants velocity

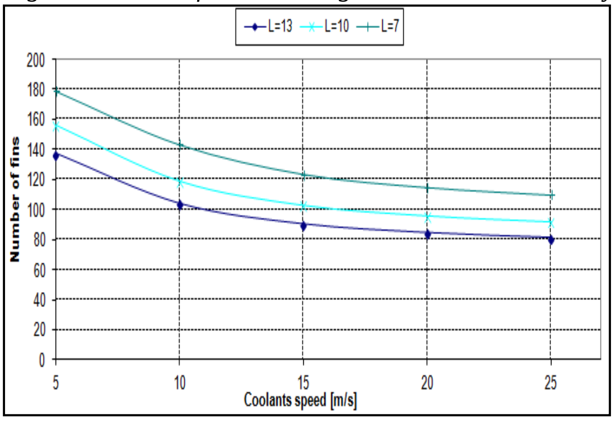


Fig. 10. Straight fins number according to the coolants velocity

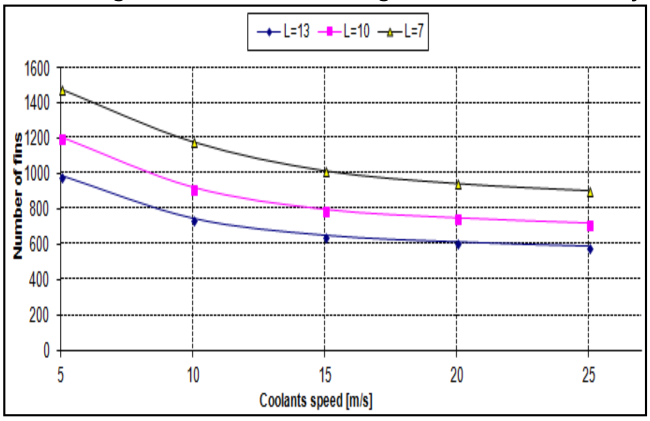


Fig. 11. Annular fins number according to the coolants velocity

CONCLUSION

This software has been developed for heat exchangers design for wind-diesel compressed air hybrid system (WDCAHS). The development is based on the LMTD Method and NTU Method. The software allows the user to input parameters and select the exchanger configurations. By allowing the user to experiment and correlate the solutions to different design requirements, the software could assist the user to better understand the temperature profile and efficiency in heat transfer for different exchangers. Furthermore, the program lets see the value the total heat flux and the exchanger surfaces and make the choice of fins. Future work would involve further development of the software to include other fluids with phase change and costing of the heat exchanger.

NOMENCLATURE

P Power compression, w

N Number of stage

n polytropic compression index

η Compression polytropic efficiency

P Pressure, pa

T Temperature, k

Q Heat flow, w

V Flow velocity, m/s

C_p specific heat, j/(kg k)

h Heat transfer coefficient w/(m^2 k)

LMTD Logarithmic mean temperature difference, k

NUT Number of transfer units

L Length, m
W Width, mm
t Thickness, mm
Re Reynolds number,
Pr Prandtl number,
Nu Nusselt Number,
$\dot{m}$ Mass flow, kg/s
h Enthalpy cold fluid and enthalpy hot fluid, kj/kg
ΔT Temperature difference, k
A Area of heat exchange, m^2
S Shape factor
k Thermal conductivity w/(m k)

Greek symbols

τ Compression ratio
η Efficiency
μ Fluid viscosity, Pa.s;
ρ Fluid density, kg/m3
Δ Difference

Subscripts and superscripts

f Fins
$-c$ Compressor
a Atmospheric
p Polytropic
c Cold
h Hot
w Wall
i Within
e Outside
ou Output
in Input
m Mean
1,2 Each end of the heat exchanger, K
0 Inner and outer surfaces

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