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Highlights

Some hydrocarbons and refrigerants are studied as working fluids in an organic rankine cycle.

The influence of the turbine inlet temperature and the pressure ratio is analyzed.

The viability of implementing this process is demonstrated.

A maximum efficiency of 9% is obtained at an inlet temperature of 110°C using R600a as working fluid.

Comparative study of working fluids for a Rankine cycle operating at low temperature

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Abstract.

The main results of a thermodynamic study on the use of a low temperature heat source (150 ºC as maximum) for power generation through a basic Rankine are reported in this paper. Different working fluids such as water and some hydrocarbons and coolants are studied. The procedure consisted in modifying the input pressure and temperature to the turbine. The efficiency for these fluids is a weak function of temperature, i.e, overheating the inlet fluid to the turbine does not cause a significant change in the efficiency. However, when the pressure ratio in the turbine increases, it is obtained much larger values of efficiency, and also, as the input temperature to the turbine raises, the efficiency increases more sharply. As result, a maximum efficiency 9% was obtained. It is shown the technical viability of implementing this type of process for recovering residual wastes for very low temperature, as well as an energy alternative and/or strengthener of non-conventional energy sources in non-provided zones.

Keywords: Energy efficiency, organic Rankine cycle, power generation, waste heat, renewable energy.

Introduction.

As a result of the power generation, the manufacturing processes, the transport, etc. the global climate has been altered due to emissions of greenhouse gases (GHG) that supposes the realization of such activities [1-2]. The use of fossil fuels has produced a huge release of CO_2, become thus in the greenhouse gas with a greater contribution to global warming [2]. Additionally, the cost of these fossil fuels are increasing every year, so it is essential to use this energy efficiently and therefore, many efforts are aimed at giving a better use to this energy consumed, for example, using waste heat or sources (such as some renewables) [3-5] or with plants where heat and power can be utilized simultaneously. Thus, in the future our energy supply must be renewable and sustainable, efficient and cost-effective, convenient and safe, also contributing to energy independence of the regions, reducing GHG emissions, while at the same time, fostering rural development, technological innovation and trade [6]. The combined heat and power production (CHP) is an efficient and cost-effective means to save energy and reduce pollution. This process can save up to 35% of primary energy [7] i.e., less fuel is required to generate a kWh of electricity [8]. In addition, these facilities allow to get the advantages of decentralized electricity generation, due to which the production is possible in areas with difficulties linked to the network, providing significant savings in infrastructure, in transport fuel to the plant and in the distribution of power generated to end users. CHP plants accounts for just 6% of total electricity production in the European Union; although about 30% of total electricity production in Denmark, the Netherlands, and Finland is cogenerated. In USA about 7% of total electricity generated is cogenerated [7]. The evolution in global policy for environmental protection, designed to incorporate the environmental factors as additional restrictions, may make feasible such type of facilities [9]. Similarly, and as it was mentioned initially, it should join efforts for the development of new technologies that convert renewable energies such solar, biomass, geothermal [3-5], as well as the use of residual and/or low enthalpy heats rejected by the industry which supposes to be a 50% or more of the heat generated in the own installations in electrical or/and mechanical energy [10]. A promising technology for the conversion of these heats is the organic Rankine cycle "ORC" [4,11-13], whose principle of operation is equal to the conventional Rankine cycle, with the difference of using an organic agent as the working fluid. A pump pressurizes the liquid fluid, and it is injected in an evaporator to produce a vapour that is expanded in a turbine connected to a generator; finally, the exit vapour is condensed, starting the new cycle (Fig. 1). However, unlike the conventional Rankine, the change of fluid allows the recovery of energy sources of low enthalpy at work or electricity. Thus, one of the main research lines realized on this issue is the selection of a suitable working fluid due to its great influence in the design of the process [4,11-12]. Depending on the application, the source and the level of the heat to use, the fluid must have optimum thermodynamic properties at the lowest possible temperatures and pressures and also satisfy several criteria such as being economical, nontoxic, nonflammable, environmentally-friendly, allowing a high use of the energy suitability of the heat source, etc. If all these aspects are considered, a few fluids can be used [4,14-15]. In [15] show that these properties of the working fluids are a key point in the cycle performance. Furthermore, the low working temperatures in the ORC cause that the global efficiency be highly sensitive to the inefficiencies in the heat transfer, which depend strongly on the thermodynamic properties of the fluid and on the conditions to which it is operating [14-16]. Hence, there are numerous studies that lead to finding a suitable working fluid to these systems and satisfy as far as possible all these aspects [11-17]. In 1985, it was performed in [11] the study of 68 working fluids, giving the best results only three of them (R11, R113 and R114) which are fluids not recommended today by the global policies of environmental conservation [18]. In [10], it was analyzed the efficiency of the ORC using benzene, ammonia, R134a, R113, R11 and R12, obtaining greater efficiencies for the two last, however, they are also substances of limited use. Other researchers who have analyzed the characteristics and behavior of different fluids for its use in ORC systems are among others [19-21], whose research can be inferred, as good candidates the R245fa and R134a for processes whose source of heat is at low temperature.

Therefore, there are numerous news references on this topic in the literature [22-27]). However, much of them analyze working fluids for heat source >150 ºC, or fluids with a high Ozone Depletion Potential (as R11, R12, R113, R114, R123, R141b, etc.) or a high Global Warming Potential, whose use is, or will be, forbidden, not being longer these

studies interesting for a real application, hence, this work restricts its study on fluids taken into account its effective future use, for environmental reasons as low temperature heat source (<150 ºC).
Other main novelty aspects provided in the present paper are our intend for avoiding all type of “distortions” or “noises” by external parameters on the behaviour of the fluid within the cycle, e.g., for the calculation of the cycle performance, in this paper it is only calculated how much energy is required to heat the inlet flow to the turbine, because it is not the aim to analyze the conditions of the heat source, and, therefore, it is not restricted the study to a given application (waste heat, solar, geothermal, etc.). Other marked difference of this work in comparison with the large literature on this topic is that, in this study, the inlet and output pressure to the turbine are parameterized because because of the difference on the saturation and condensation pressures of each studied fluid at a given temperature, and, therefore, the ratios (P_1/P_2) are maintained equal to 1.5, 2.5 and 3.0, seeking a minimum, an intermediate and a maximum that were common to all fluids.
Finally, in this study was covered and considered the difference in the type of fluid (wet, dry and isentropic). The results obtained allow to determine whether the raise of the inlet temperature to the turbine makes to increase or decrease the cycle performance qualitatively, but, similarly, allows to know in how much increase or decrease this efficiency.

1. Theoretical Procedure.

Initially it was started with a pre-selection of fluids, resorting to the results reported by [12], where 20 fluids were studied and the R134a, R152a, R600, R600a and the R290 were found as the more suitable in terms of yields. For comparison, the main physical properties, the security properties based on ASHRAE 34 and the environmental properties as the Ozone Depletion Potential (ODP) and the Global Warming Potential (GWP) of the selected fluids are shown in Table 1.

Basing on the previous results [12] and including water as reference fluid, a thermodynamic analysis of the Rankine cycle was carried out using the process simulator HYSYS® (Hyprotech Co., Canada). The efficiency of this cycle is evaluated as a function of the inlet temperature to the turbine (at a given pressure ratio) as well as the pressure ratio at a fixed inlet temperature to the turbine. The analysis assumes steady state conditions, no pressure drop or heat loss in the evaporator, the condenser or the pipes and the constant isentropic efficiencies of 75% are assumed for the pump as well as for the turbine. The cycle’s total energy efficiency is:

$$\eta = \frac{\dot{W}_{turbine} - \dot{W}_{pump}}{\dot{Q}_{evaporator}} \tag{1}$$

where,

$$\dot{W}_{turbine} = \dot{m} \times \left(h_1 - h_2\right) \tag{2}$$

$$\dot{W}_{pump} = \dot{m} \times \left(h_3 - h_4\right) \tag{3}$$

and

$$\dot{Q}_{evaporator} = \dot{m} \times (h_1 - h_4) \quad (4)$$

An input temperature of the condensation water T_5=15 ºC is considered and a working fluid condensation temperature of T_3=35 ºC. Otherwise, a pinch point of 10 ºC is maintained between T_3 and the output temperature of the condensation water (T_6). In the heating process, the overheating of the inlet fluid to the turbine (T_1) is considered from the condition of saturated steam up to its critical temperature, except for water. The discharge pressure of the turbine (P_2) is equal to the saturation pressure of the fluid in liquid state (P_3) to the temperature T_3 = 35º C, while the inlet pressure to the turbine (P_1) maintains the ratio (P_1/P_2) equal to 1.5, 2.5 and 3.0.

The thermodynamic analysis of the ORC was performed using a process simulator HYSYS®. This simulator is useful for thermodynamic analysis, especially steady state condition, and has the advantage of including fluid properties and ready to use optimization tools. The simulation flow diagram is the same as that presented in Fig. 1 and the method for resolving every one of its components is the following:

- Turbine: the efficiency of a turbine is given as the ratio of the actual power produced ($\dot{W}_{turbine}$) in the expansion process to the power produced for an isentropic expansion ($\dot{W}_{turbine,is}$) :

$$\eta_{turbine} = \dot{W}_{turbine} / \dot{W}_{turbine,is} \times 100\% \quad (5)$$

With the inlet and outlet pressures, the inlet temperature and the efficiency known, the software calculates the expansion rigorously following the isentropic line from the inlet to outlet pressure. Using the enthalpy at that point, as well as the specified efficiency, the software determines the actual outlet enthalpy. From this value and the outlet pressure, the outlet temperature is determined.

- Pump: Calculations are based on the standard pump equation for power, which uses the pressure rise, the liquid flow rate and density:

$$\dot{W}_{pump,is} = (P_4 - P_3) \times \dot{m} / \rho_{liquid} \quad (6)$$

The equation (6) defines the ideal power needed to raise the liquid pressure and the actual power requirement of the pump is defined in terms of the pump efficiency:

$$\eta_{pump} = \dot{W}_{pump,is} / \dot{W}_{pump} \times 100\% \quad (7)$$

When the efficiency is less than 100%, the excess energy goes into raising the temperature of the outlet stream.

Combining (6) and (7) leads to the following expression for the actual power requirement of the pump:

$$\dot{W}_{pump} = \left(P_4 - P_3\right) \times \dot{m} \times 100\% \Big/ \left(\rho_{liquid} \times \eta_{pump}\right) \tag{8}$$

- Evaporator: For the calculation of the cycle performance, it is only necessary to know how much energy is required to heat the process flow directed to the turbine. Because it is not our objective to study the conditions of the utility itself, the available evaporator in the software is used. The inlet stream to the evaporator is heated to the required outlet conditions (the conditions established in the turbine entrance) and according to the Fig. 1, the energy stream provides the enthalpy difference between the two streams.

2. Results and Discussion.

This section presents the results obtained in the simulations where the influence of the inlet temperature to the turbine and the pressure ratio are studied.

2.1 Influence of the inlet temperature to the turbine

According to the assumptions discussed in section 1, the Fig. 2 (a to f) show the results obtained on the cycle efficiency by increasing the inlet temperature to the turbine T_1, with constant P_1/P_2 ratio for R718, R600, R600a, R134a, R152a and R290, respectively.

In order to see how the use of one or other fluid will depend on the heat source that it is intended to recover and on the same thermophysical properties of fluid, it has performed the Fig. 3, in which it is presented the results on the η of the cycle by increasing the T_1 at a constant P_1/P_2 ratio for the 6 fluids under study with the aim to achieve a better visual comparison in a same graph, for a only pressure ratio and for each one of the three ratio analyzed. It is obvious that when the ratio $P_1/P_2 = 1.5$ (which is the lowest of those studied, see Fig. 3.a) the efficiency, η, increases with T_1 for the case of R718, the R290 and R152a; that is “wet fluids”, unlike the “dry fluid” R600 and R600a show reduction in η with the increase in T_1. For the R134a, which is an isentropic fluid, it remains unchanged. For this pressure ratio, the increase of the performance by raising the inlet temperature to the turbine from the saturation condition until its critical temperature was approximately 0.1% for R290 and R152a fluids, while it decreased approximately 0.2% for fluids R600 and R600a. However, it should be noted that η is a weak function of temperature for the case of the fluids studied, i.e, overheating the inlet fluid to the turbine does not cause a significant change in η. Nevertheless, when the ratio P_1/P_2 increases, it is obtained values much higher of η for all fluids as show the Fig. 3.b and 3.c. (The intrinsic properties of water require values P_1/P_2 higher for acquiring an acceptable η) and in addition, as the inlet temperature to the turbine also raises, this effect increases more steeply, i.e. for the greater pressure ratio studied ($P_1/P_2 = 3.0$), the raise of performance by increasing the inlet temperature to the turbine from the saturation condition until its critical temperature was approximately 0.4% for R290 and R152a fluids, while decreased approximately 0.4% for R600 and R600a fluid.

It is interesting to indicate that Fig. 3 also shows how the use of one or other fluid will depend on the heat source that is intended to recover and on the thermophysical properties of the fluid, because there are fluids which allow achieving high efficiencies at low temperature ranges when are compared with other fluids that are useful for other different temperature ranges, e.g. with the pressure ratio P_1/P_2 = 2.5 and for the temperature range approximately between 70 and 85 °C, R152a offers the best performance, while for temperatures between 85 and 97 ºC, the best fluid, referred to performance turned out the R290. However, the operation of some of these fluids is limited to a range of temperatures and pressures less, mainly due to restrictions in terms of their chemical stability and security.

2.2. Influence of the pressure ratio.

In Fig. 4 (a to f), it is compiled the results obtained by increasing the P_1/P_2 ratio on the overall cycle efficiency η, and with constant inlet temperatures in the turbine T_1 for R718, R600, R600a, R134a, R152a and R290, respectively.

It can be observed how the efficiency of the system raises with increasing pressure ratio, regardless of the inlet temperature to the turbine and for each of the fluids studied. However, and in line with the comments in section 2.1, for fluids considered as "wet" (R718, R152a and R290), the efficiency of the system reaches the highest values for higher temperatures (Figs. 4.a, 4.e and 4.f, respectively), whereas for “dry fluids” (R600 and R600a), with higher temperatures, lower efficiencies are given (Figs. 4.b and c). With fluid R134a, it is not very perceptible to distinguish some difference.

Therefore, Fig. 5 involves the 6 fluids under study for a single inlet temperature to the turbine T_1 (and for each of the three studied: 75, 85 and 95 ºC in Fig. 5 (a, b and c, respectively) for a better visual comparison. This Fig. 5 shows how the η of the system raises with the increase of the pressure ratio for all the fluids used at a constant T_1. Higher P_1 increases both the net work as the evaporator heat that leads to an improvement in η. However, the increase in the net work is higher than in the heat of the evaporator. The tendency of the organic fluid analyzed is quite similar, detecting how η rises as P_1/P_2 increases, however, when the temperature T_1 and this pressure ratio increase, the slope of the curve tends to decrease (Figs. 5.b and 5.c). This behaviour of the water is hardly appreciable, noting also that the values of η are much lower compared to other fluids.

The influence of fluid flow on the cycle is evident, since the enthalpy differences (between the zone of high pressure and the expanded vapour) of the organic substances are significantly lower than the water. So that higher mass flow rates and power requirements in the evaporator are needed in ORC for the same output power produced by the turbine. At small scale, the use of larger turbines due to higher mass flow of fluids reduces losses when it is compared to steam turbines of the same power, however, increases equipment cost [20].

The realized simulations show among others, that as overheating the steam as increasing the flow of the organic fluid does not significantly affect the efficiency obtained with each, in contrast to the obtained when using water, which increases the cycle efficiency remarkably as temperature and/or flow increases.

In summary, at a given temperature range, it is possible to make a first selection between working fluids. The intended application must take into account the temperature of the heat source in order to discard some fluids. From an environmental point of view, others can be eliminated since they are banned from agreements like the Montreal and Kyoto Protocol. Furthermore, and although the efficiency of the system also increases as system pressure raises, this rise is not always feasible for economic reasons because of the costs.

2.3 Economical analysis

Due to the lack of a real installation to show the cost of this type of machines, a simple economic analysis was carried out to find the maximum investment that can be assumed for a project in which the return on investment was needed within a year. The methodology developed consisted in calculating the kWh that can be produced by ORC machines of 10 kW (for households) and of 100 kW (for the industry), working 8000 hours a year and whose fuel cost is zero, thus obtaining the saving that not buying this quantity of energy would suppose. Thus, the maximum reasonable investment is assumed to be 90% of this saving, as 10% is reserved for fixed operation costs. Table 2 shows the cost of kWh in 2008 for both industry and households in the target countries of the present study (Spain, the U.S.A. and Colombia).

Figure 6 presents a semi-log graph to capture the results of this economic analysis. Evidently, due to the greater cost of the kWh for Spain, a higher cost of the project can be assumed in both sectors. For the case of The U.S.A. and Colombia, the cost of the project at residential level is practically the same, whereas for the industrial sector, a notable difference is detected. The thermophysical properties of the work fluid also influences the cost of the heat exchanger through the transmission heat coefficient; a fluid with low viscosity and high conductivity will have a high coefficient of heat transmission, so its heat exchanger will be much less expensive. The nature of the available source thermal energy must also be considered when deciding whether to use a subcritical or supercritical cycle.

Conclusions.

Based on the simulations carried out, it can mention that the system’s efficiency proposed is a weak function of temperature, because overheating the inlet fluid to the turbine does not cause a significant change in the overall efficiency of the cycle. However, when the pressure ratio in the turbine increases (obviously limited by the temperature of the heat source), it is obtained much larger values of efficiency and also, as the inlet temperature to the turbine raises, the efficiency increases more sharply.

Based on the results, it can conclude that using organic working fluids in a Rankine cycle, good efficiencies are achieved for the recovery of low enthalpy resources, however, there is not a fluid that fits all characteristics (in terms of efficiency, toxicity, environmental, economic, etc.) taken into account in a real cycle ORC.

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TABLES.

Table 1. Physical, security and environmental properties of the selected fluids.

Table 2. Cost of the kWh in the industry and in the households in the selected countries.

Table 1.

Substance	Physical data					Security*	Enviromental*		
	Type[1]	Weight (kg/kmol)	T_b[2] (º C)	T_c[3] (º C)	P_c[4] (bar)	Group	Lifetime (years)	ODP	GWP
R718 (Water)	w	18.0	99.9	373.9	220.6	A1	_	0	<1
R600 (Butane)	d	58.1	-0.5	152.0	38.0	A3	0.018	0	~ 20
R600a (isobutane)	d	58.1	-11.7	134.7	36.3	A3	0.019	0	~ 20
R134a (1,1,1,2-Tetrafluoroethane)	i	102.0	-26.1	101.1	40.6	A1	14	0	1430
R152a (1,1-Difluoroethane)	w	66.0	-24.0	113.3	45.2	A2	1.4	0	124
R290 (Propane)	w	44.1	-42.1	96.7	42.5	A3	0.041	0	~ 20

[1]i=isentropic, w=wet, d=dry. [2]T_b= Normal boiling temperature. [3]T_c= Critical temperature. [4]P_c= Critical pressure.
[*] Source [12]

Table 2.

Country	€/kWh		Reference
	Households	Industry	
Spain	0.1124	0.0915	[28]
USA	0.0750	0.0455	[29]
Colombia	0.0766	0.0613	[30]

FIGURE CAPTIONS

Figure. 1. **Schematics diagram of the Rankine process.**

Figure 2. **Influence of the inlet temperature to the turbine T_1 on the overall efficiency of the cycle with constant P_1/P_2 ratio for R718 (a), R600 (b), R600a (c), R134a (d), R152a (e), R290 (f), respectively.**

Figure 3. **Influence of the input temperature to the turbine T_1 on the total efficiency of the cycle with a constant ratio P_1/P_2 =1.5 (a), 2.5 (b) and 3.0 (c), respectively.**

Figure 4. **Influence of the P_1/P_2 ratio on the overall efficiency of the cycle, with constant inlet temperature to the turbine T_1 for R718 (a), R600 (b), R600a (c), R134a (d), R152a (e), R290 (f), respectively.**

Figure 5. **Influence of the ratio P_1/P_2 on the total efficiency of the cycle with a constant input temperature to the turbine T_1 = 75 ºC (a), 85 ºC (b) and 95 ºC (c), respectively.**

Figure 6. **Maximum investment (in the logarithm scale) at industrial and household level for the three countries under study.**

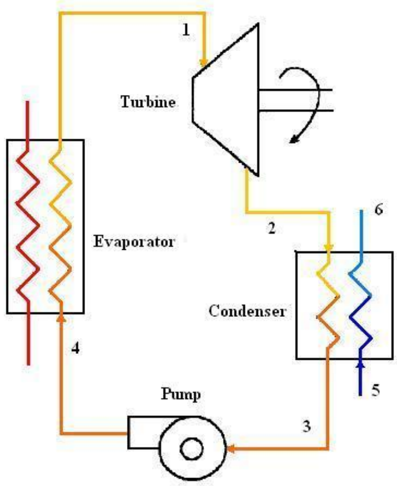
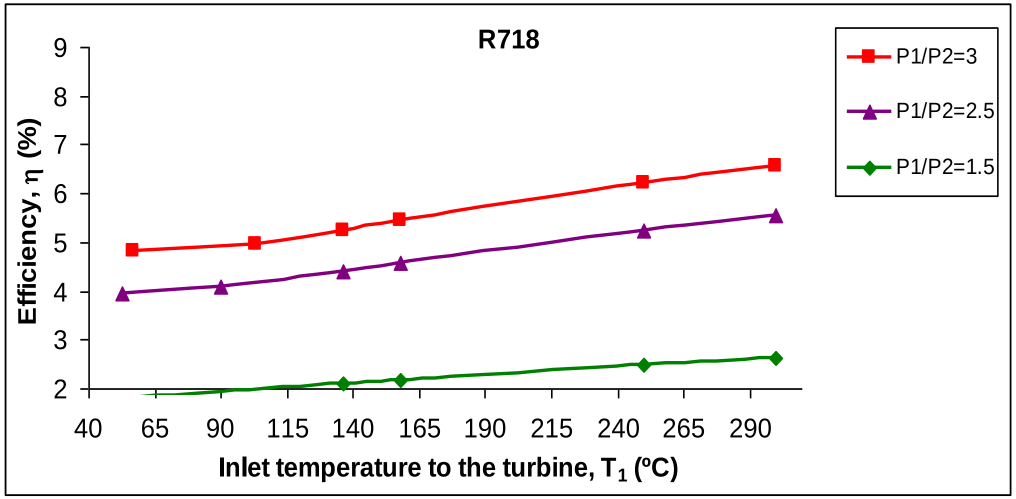
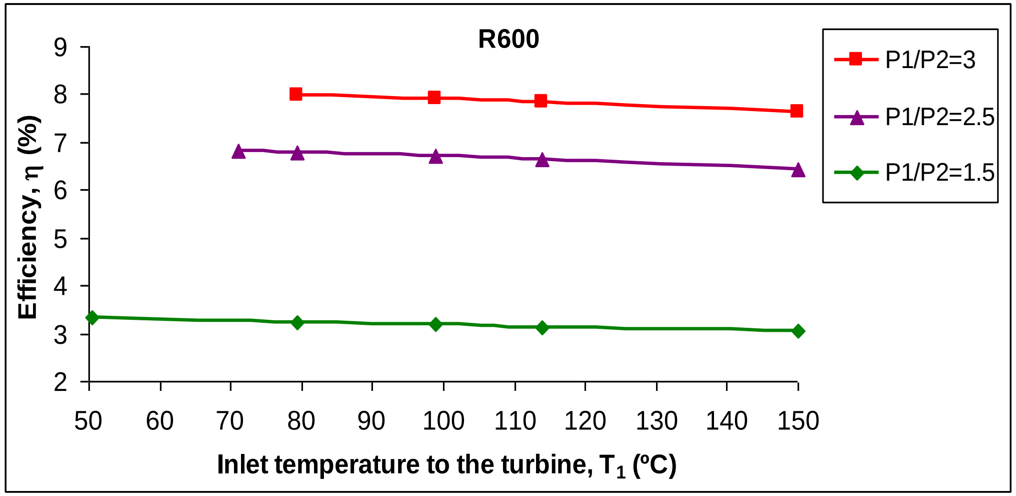


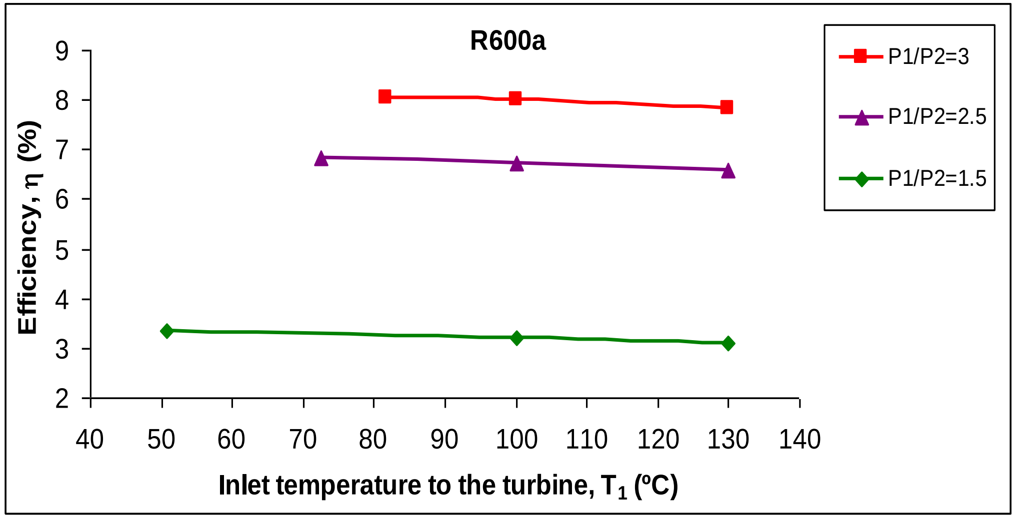
Figure 1.


R718
Efficiency, η (%)
Inlet temperature to the turbine, T1 (ºC)
P1/P2=3
P1/P2=2.5
P1/P2=1.5
9
8
7
6
5
4
3
2
40
65
90
115
140
165
190
215
240
265
290

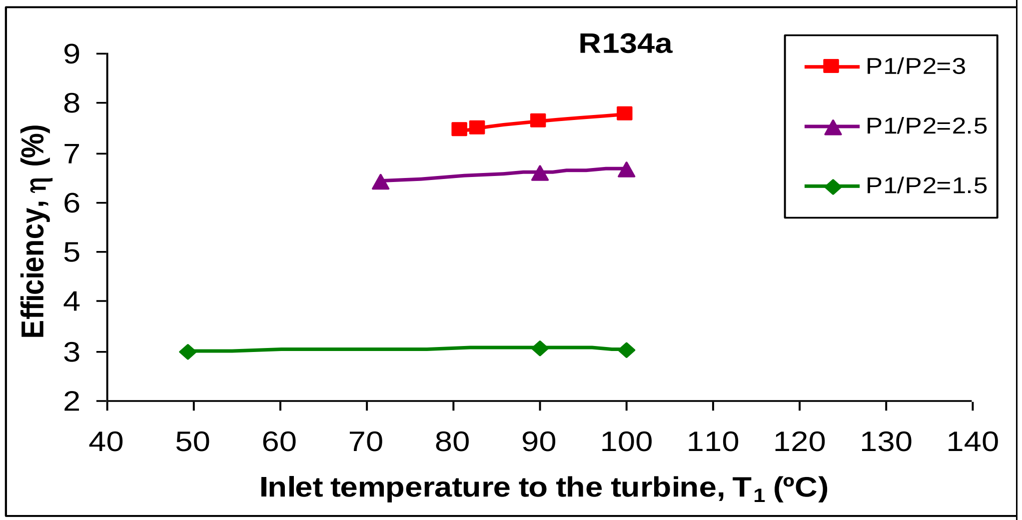
a.


R600
Efficiency, η (%)
Inlet temperature to the turbine, T1 (ºC)
P1/P2=3
P1/P2=2.5
P1/P2=1.5
9
8
7
6
5
4
3
2
50
60
70
80
90
100
110
120
130
140
150

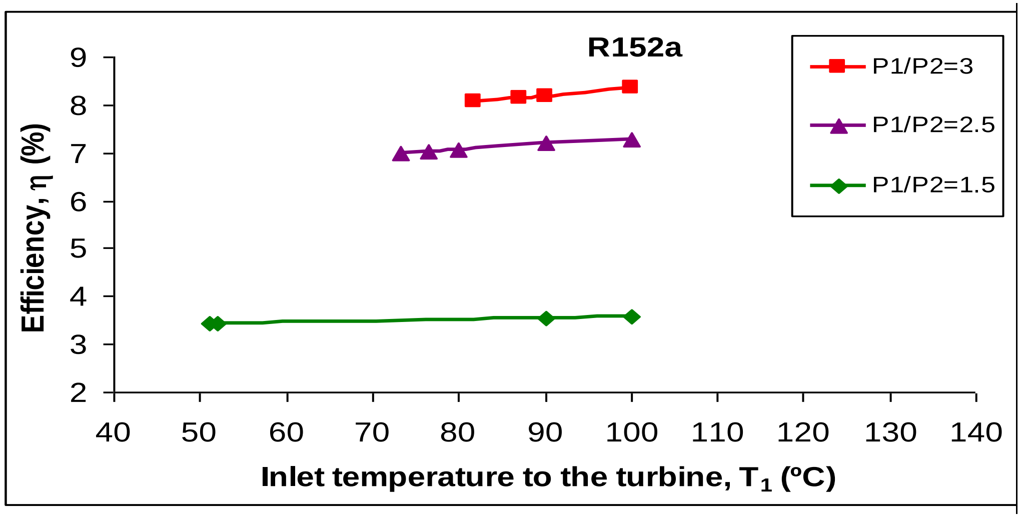
b.


R600a
Efficiency, η (%)
Inlet temperature to the turbine, T1 (ºC)
P1/P2=3
P1/P2=2.5
P1/P2=1.5
9
8
7
6
5
4
3
2
40
50
60
70
80
90
100
110
120
130
140

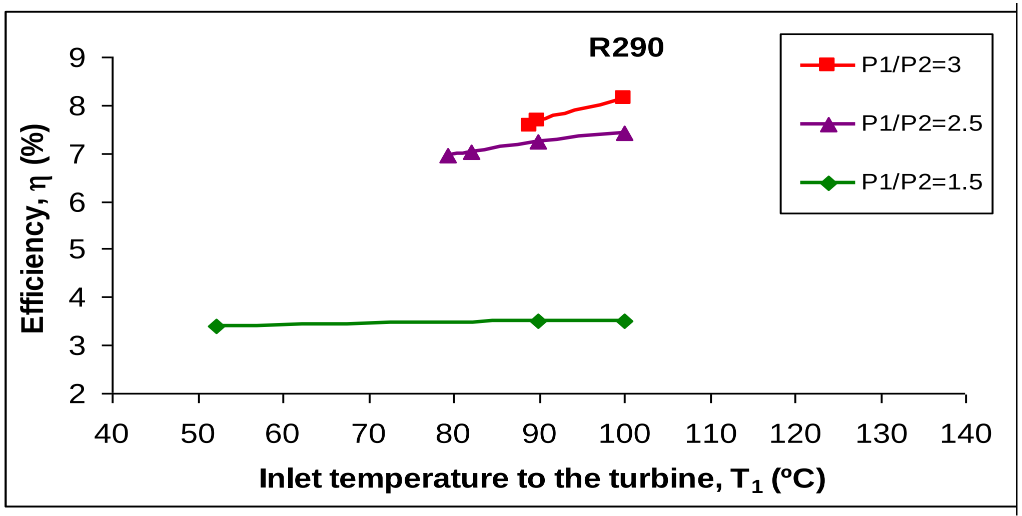
c.


R134a
Efficiency, η (%)
Inlet temperature to the turbine, T1 (ºC)
P1/P2=3
P1/P2=2.5
P1/P2=1.5
9
8
7
6
5
4
3
2
40
50
60
70
80
90
100
110
120
130
140

d.

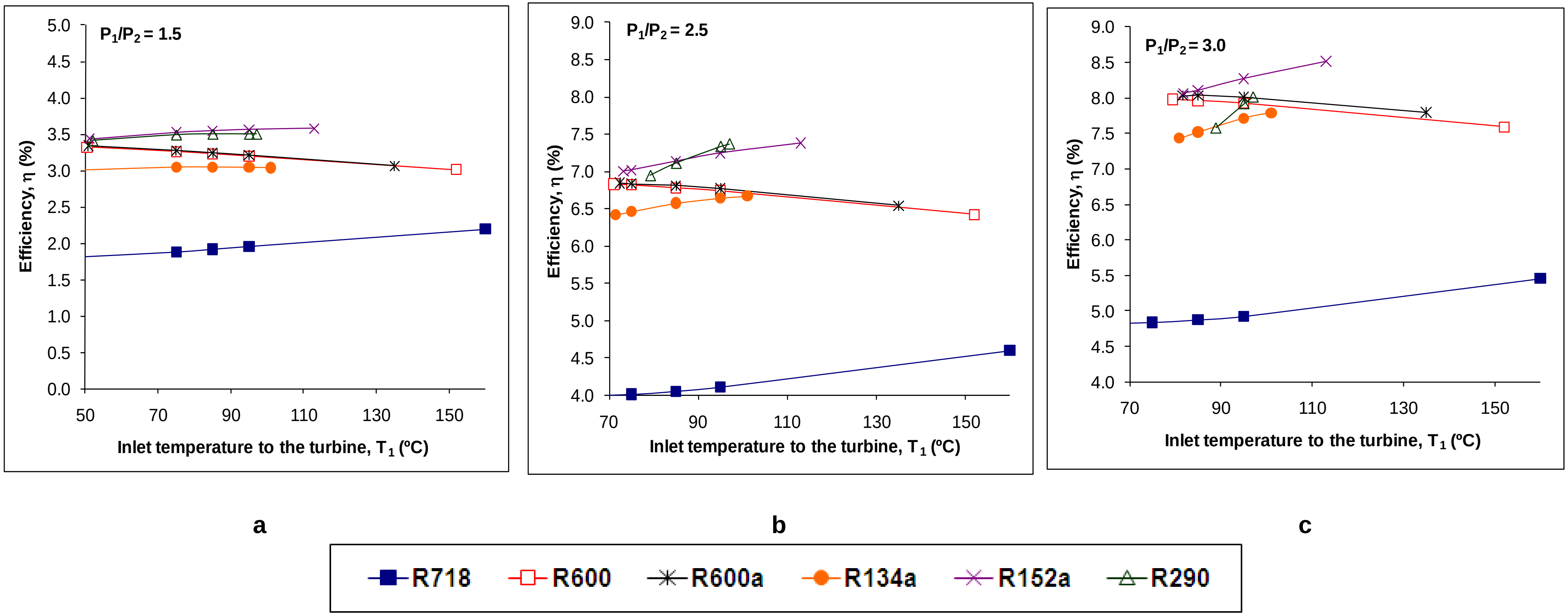

R152a
P1/P2=3
P1/P2=2.5
P1/P2=1.5
Efficiency, η (%)
9
8
7
6
5
4
3
2
40
50
60
70
80
90
100
110
120
130
140
Inlet temperature to the turbine, T1 (ºC)

e.


R290
P1/P2=3
P1/P2=2.5
P1/P2=1.5
Efficiency, η (%)
9
8
7
6
5
4
3
2
40
50
60
70
80
90
100
110
120
130
140
Inlet temperature to the turbine, T1 (ºC)

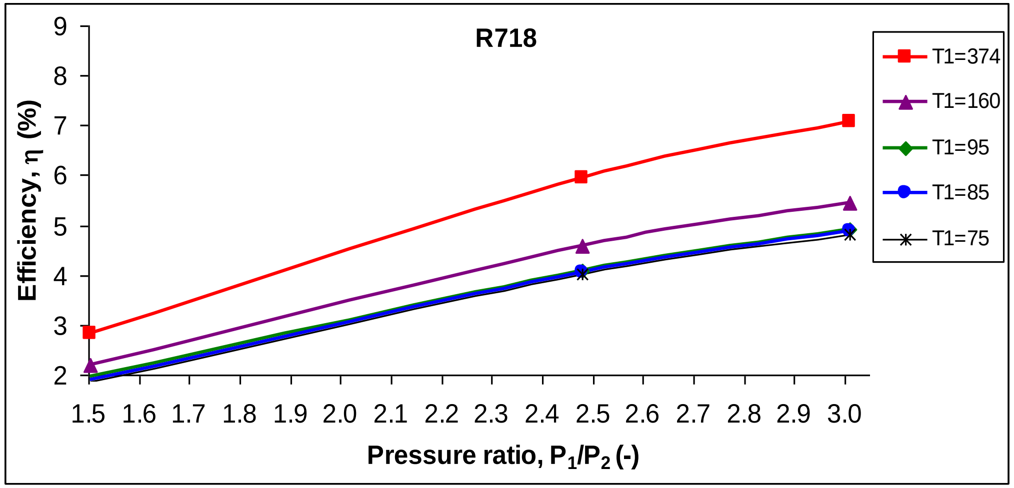
f.

Figure 2.

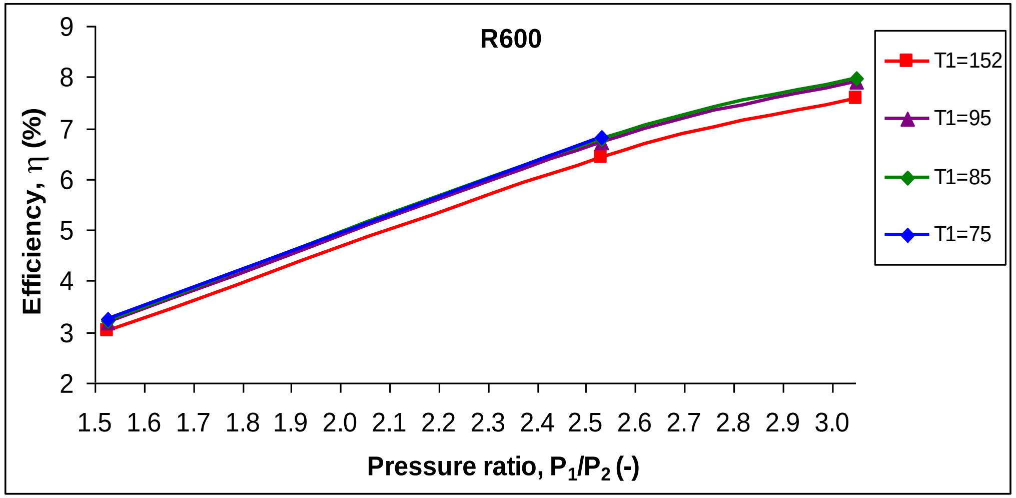


P1/P2 = 1.5
Efficiency, η (%)
Inlet temperature to the turbine, T1 (ºC)
P1/P2 = 2.5
Efficiency, η (%)
Inlet temperature to the turbine, T1 (ºC)
P1/P2 = 3.0
Efficiency, η (%)
Inlet temperature to the turbine, T1 (ºC)
a
b
c
R718
R600
R600a
R134a
R152a
R290

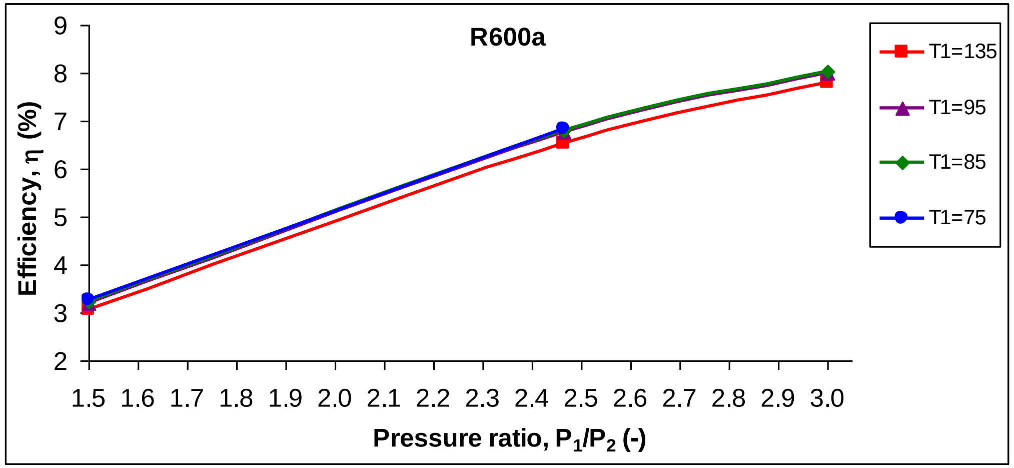
Figure 3.


R718
Efficiency, η (%)
9
8
7
6
5
4
3
2
1.5 1.6 1.7 1.8 1.9 2.0 2.1 2.2 2.3 2.4 2.5 2.6 2.7 2.8 2.9 3.0
Pressure ratio, P1/P2 (-)
T1=374
T1=160
T1=95
T1=85
T1=75

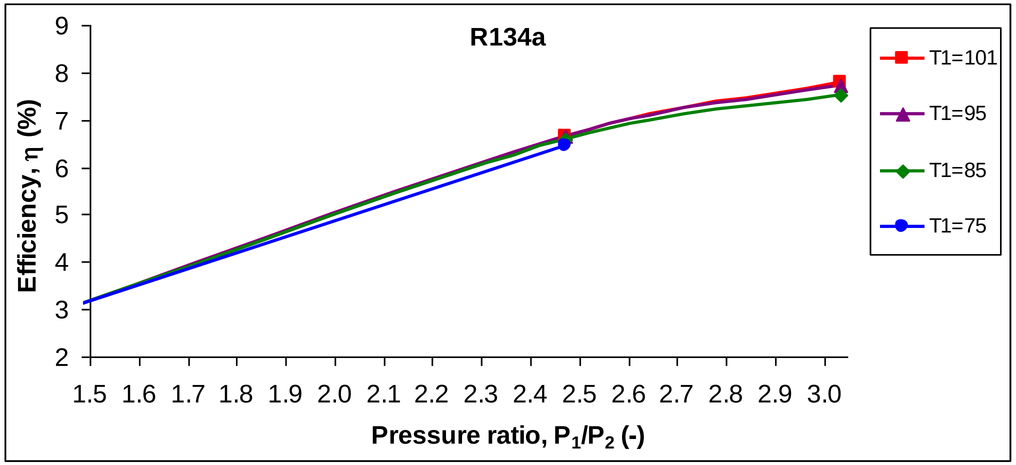
a.


R600
Efficiency, η (%)
9
8
7
6
5
4
3
2
1.5 1.6 1.7 1.8 1.9 2.0 2.1 2.2 2.3 2.4 2.5 2.6 2.7 2.8 2.9 3.0
Pressure ratio, P1/P2 (-)
T1=152
T1=95
T1=85
T1=75

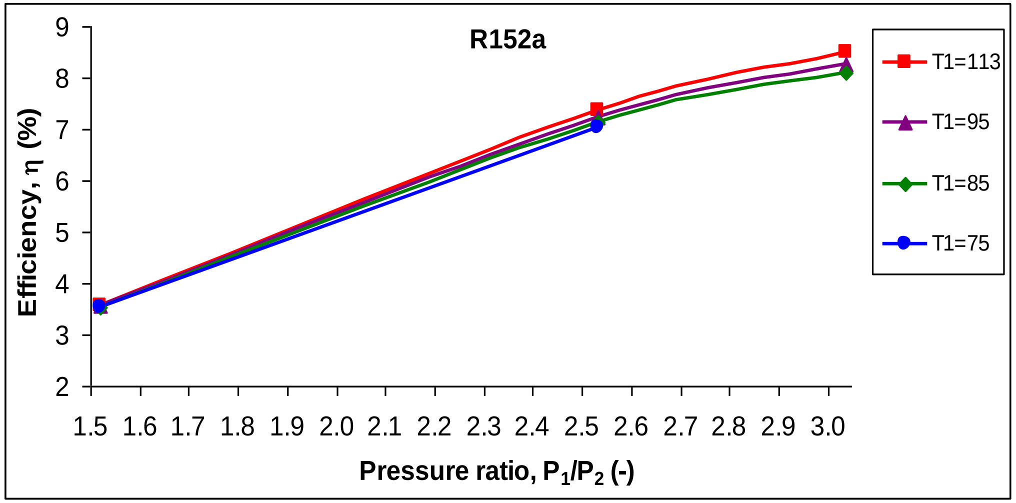
b.


R600a
Efficiency, η (%)
9
8
7
6
5
4
3
2
1.5 1.6 1.7 1.8 1.9 2.0 2.1 2.2 2.3 2.4 2.5 2.6 2.7 2.8 2.9 3.0
Pressure ratio, P1/P2 (-)
T1=135
T1=95
T1=85
T1=75

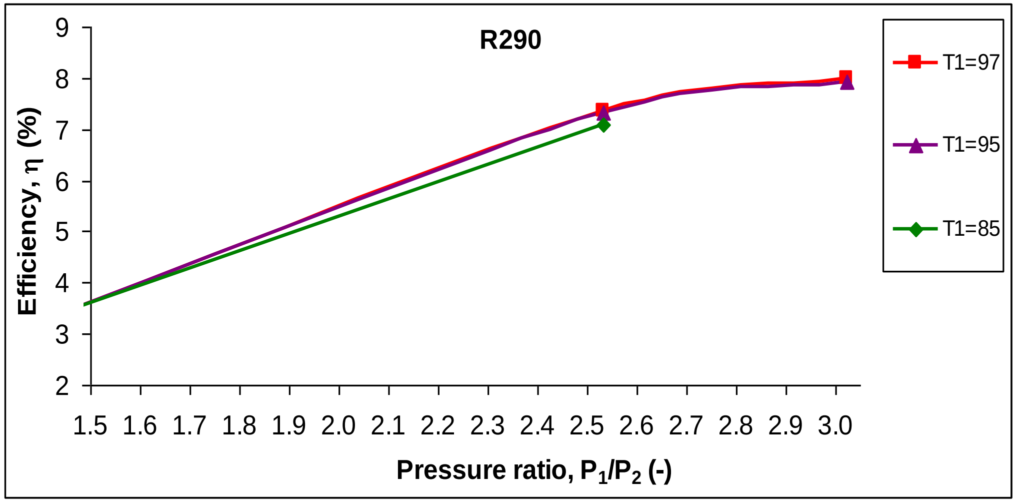
c.


R134a
Efficiency, η (%)
9
8
7
6
5
4
3
2
1.5 1.6 1.7 1.8 1.9 2.0 2.1 2.2 2.3 2.4 2.5 2.6 2.7 2.8 2.9 3.0
Pressure ratio, P1/P2 (-)
T1=101
T1=95
T1=85
T1=75

d.

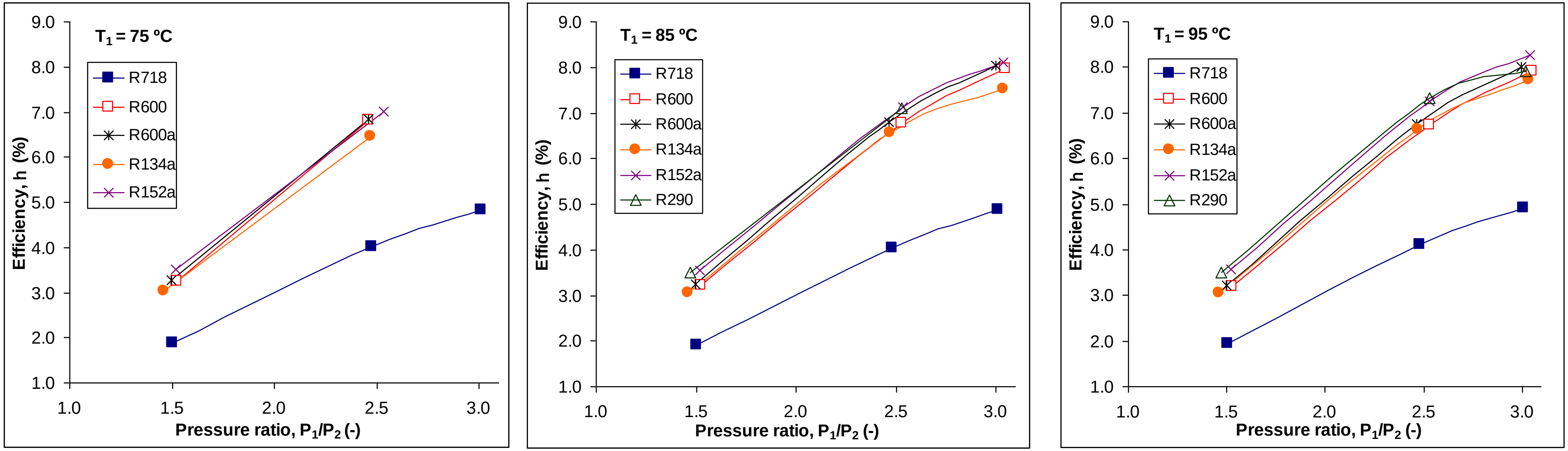

R152a
Efficiency, η (%)
Pressure ratio, P1/P2 (-)
T1=113
T1=95
T1=85
T1=75
9
8
7
6
5
4
3
2
1.5 1.6 1.7 1.8 1.9 2.0 2.1 2.2 2.3 2.4 2.5 2.6 2.7 2.8 2.9 3.0

e.


R290
Efficiency, η (%)
Pressure ratio, P1/P2 (-)
T1=97
T1=95
T1=85
9
8
7
6
5
4
3
2
1.5 1.6 1.7 1.8 1.9 2.0 2.1 2.2 2.3 2.4 2.5 2.6 2.7 2.8 2.9 3.0

f.

Figure 4.


T1 = 75 ºC
T1 = 85 ºC
T1 = 95 ºC
R718
R600
R600a
R134a
R152a
R290
Efficiency, h (%)
Pressure ratio, P1/P2 (-)
1.0
2.0
3.0
4.0
5.0
6.0
7.0
8.0
9.0
1.5
2.5

a. b. c.

Figure 5.

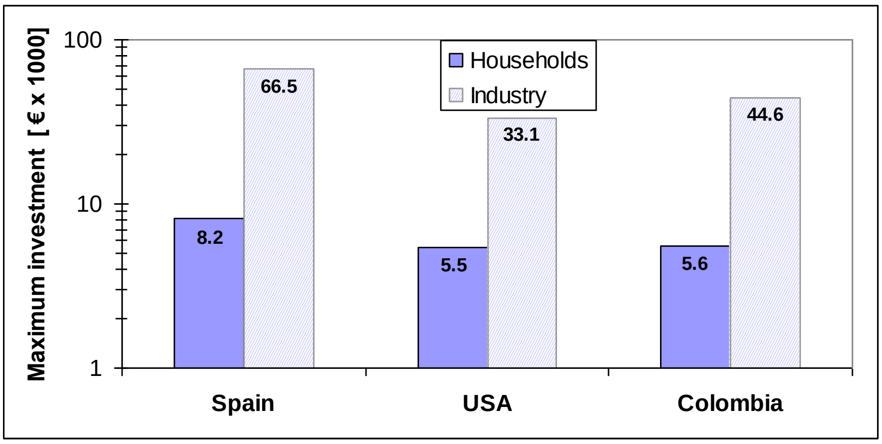

Maximum investment [€ x 1000]
100
10
1
Households
Industry
66.5
33.1
44.6
8.2
5.5
5.6
Spain
USA
Colombia

Figure 6.