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# Selecting and Optimizing a Heat Exchanger for Automotive Vehicle Rankine Cycle Waste Heat Recovery Systems

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## Abstract

This paper contributes for the selection and optimization of an adequate compact heat exchanger (HEX) for an automotive vehicle Rankine cycle Waste Heat Recovery Systems (RC-WHRS). Both tube type and plate type HEX were considered. Among the different HEX designs available, a robust cross-flow tube heat exchanger (CFT-HEX), with the working fluid circulating inside the tubes was selected. The selected HEX accomplishes with the specific pressure drops limits on both exhaust gas and working fluid side. The compactness of the CFT-HEX strongly depends on the optimization of the HEX pre-heater section, which mainly depends on the augmentation of the working fluid heat transfer coefficient. The present study data reveals that a transitional flow regime with a working fluid Reynolds number, Re > 3200, allows a significant increase of the Nusselt number (Nu > 20) as compared to laminar flow regime.

*Keywords:* Waste heat recovery, Rankine cycle; heat exchanger; heat transfer performance.

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## 1. Introduction

Escalating fuel prices and future carbon dioxide emission limits are creating a renewed interest in methods to increase the thermal efficiency of vehicles equipped with internal combustion engines (ICE). Over the time ICE

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manufacturers have developed and implemented techniques to increase thermal efficiency. However, around 60-70% of the fuel energy is still lost through the coolant or the exhaust as wasted heat [1,2]. To increase the ICE thermal efficiency and to reduce $CO_2$ emissions, different waste heat recovery (WHR) techniques were recently proposed [1-5]. Among the existing WHR techniques, the most relevant are the electrical turbo-compounding (ETC), the mechanical turbo-compounding (MTC), the thermo-electric generator (TEG) and the Rankine cycle (RC) or organic Rankine cycle (ORC) [1-3].

At the present, Rankine cycle Waste Heat Recovery Systems (RC-WHRS) seem to be the best solution for heavy-duty Diesel engine (HDDE) vehicle applications. The RC-WHRS is based on the steam generation in a secondary circuit that represents an indirect method of WHR. For HDDE, recent studies [4,5] have shown that RC-WHRS allows reducing fuel consumption by up to 6 %. In a RC-WHRS, the waste energy is transferred to the working fluid by a pre-heater plus an evaporator plus a super-heater. Such heat exchanger (HEX) is also known as the evaporator (or boiler), where the working fluid at high pressure changes from liquid to superheated vapor. Subsequently, the working fluid, which has a high enthalpy is expanded in an expander, and output power is generated.

The key components of a RC-WHRS are the heat exchanger (HEX) and the expander, which have a dominant impact on the RC-WHRS efficiency. A RC-WHRS for HDDE vehicle applications requires long life and high performance compact heat exchangers. Although several simulations and theoretical studies about RC-WHRS in ICE have been reported in the literature [1-5], few studies have been developed with regard to the HEX design.

The major outcome of the present work is to contribute for the HEX development. To this end, one starts with the selection of the HEX design suitable for such an application, specifically, reviewing and discussing the heat transfer, pressure drop and technical challenges of both tube and plate type heat exchangers. Next, the fundamental fluid flow and heat transfer phenomena within the channels of the pre-heater section of the HEX are investigated. The obtained experimental data allows to better understand the fundamental transport phenomena involved, so that new guidelines for the HEX optimization will be drawn.

**Nomenclature**

| | |
|---|---|
| *Gr* | Grashof number [-] |
| *Nu* | Nusselt number [-] |
| *Pr* | Prandtl number [-] |
| *Re* | Reynolds number [-] |

## 2. Heat exchanger design

The heat exchanger suitable for a heavy-duty Diesel engine (HDDE) vehicle RC-WHRS should be capable of operating at elevated temperatures of up to 600 ºC, elevated pressures on the working fluid side of up to 40 bar, with long life and in a harsh environment. The thermal hydraulic characteristics of high effectiveness and low pressure drop (to minimize the exhaust back pressure) are required. In addition, light weight and high compactness are required.

Regarding to the selection of the HEX design, it can be divided into two main categories: tubes and plates. The former group includes all the heat exchangers types and configurations that use tubes to conduct and separate the working fluid from the exhaust gas (hot source). The second group includes all the plate heat exchangers, which are based on configured “sheets”, creating several levels of intermittent cold and hot source passages.

The HEX design and configuration requires taking into account the thermal source (exhaust gases) and working fluid (liquid, two-phase and vapor) operating conditions, temperature difference, mass flow rate and flow direction. The main difficulty would be the specification of a single HEX for constantly changing working conditions on both exhaust and working fluid side. For this, a compromise must be found between performance (effectiveness/pressure drop) and heat exchanger volume for the most relevant engine conditions.

### *2.1. Tube type heat exchanger design*

On the tube type heat exchanger group, the most common design is the simple shell and tube heat exchanger. This design consists of several tubes inside a shell (e.g., cylindrical), having the working fluid circulating generally inside the inner tubes and the hot source flow along and/or across the tubes. Vehicle applications demand a very compact solution, therefore with the highest specific heat transfer surface area possible. Shell and tube design has relatively low specific heat transfer surface area. Therefore, due to compactness requirements, simple shell and tube heat exchanger design cannot be used in RC-WHRS for vehicle applications.

As compared with the simple shell and tube design, the use of a concentric tube heat exchanger (CT-HEX), consists on tube pairs, the inner and outer tube, instead of simple tubes, which allows the increase of heat transfer surface area, for the same volume. The CT-HEX is designed as a tube bundle heat exchanger. The working fluid flows through the annular passage between the inner and the outer tube, which has the advantage that the high fluid pressures are supported by cylindrical inner and outer tubes, so that the tube walls are thinner and thus the use of material is minimal.

The hot exhaust gas flows through the inner surface of the inner tube and the outer surface of the outer tube in counter flow. Hence, the working fluid receives heat from both the inner and the outer tube. Considering that the simple shell and tube design uses a tube diameter equal to the outer tube of the concentric tube heat exchanger, the concentric tube heat exchanger heat transfer area is equal to the shell and tube heat transfer area plus the inner tubes heat transfer area. Furthermore, due to the division of the exhaust gas flow into the inner and the outer flow, a significantly larger flow area is provided, allowing the reduction of the exhaust gas backpressure considerably.

A cross flow tube heat exchanger (CFT-HEX) consists on a tube bank usually arranged in an inline or staggered manner, where exhaust gas flows across the tubes and the working fluid circulates inside the tubes, allowing these heat exchangers to be designed for high pressures. By changing the tube diameter, length and arrangement, great flexibility can be found in a CFT-HEX. Furthermore, the geometry of the working fluids duct can be designed to accomplish with different local fluid densities and velocities, which will allow complying with the specific pressure drop limits on the working fluid side.

With the aim of improving the heat transfer surface area between the two fluids, both the CT-HEX and the CFT-HEX can be improved by making several modifications. In a tubular heat exchanger, the heat transfer area is usually increased by increasing the number of tubes and decreasing the tube diameter and/or by using extended surfaces.

Both tube type heat exchangers (CT-HEX and CFT-HEX) design allow a compact tubular heat exchanger that offer the best geometry to withstand pressure and fluctuating engine exhaust flow on the gas side and with boiling heat transfer on the working fluid side. Furthermore, the usage of corrugated tubes will provide an additional advantage of reducing material stresses, especially under demands of temperature cycle operation due to the elasticity of the tubes.

### *2.2. Plate type heat exchanger design*

Plate type heat exchangers are consistently selected when the compactness of the system is a main requirement. Plate type heat exchanger design is used on the great majority of systems where waste heat recovery is extracted from a low/medium grade heat source. The most common plate type heat exchanger design consists on an arrangement of two metal plates disposed one on top of the other that are completely welded together along the sides.

Plate and fin type heat exchangers have already been used in last generation of EGR coolers for HDDE applications. In an EGR cooler, fins are located at gas side where heat transfer resistance is higher. In a RC-WHRS the HEX (EGR evaporator) replaces the EGR cooler, so that on the exhaust gas side the requirements are similar to an EGR cooler; but the HEX requires high pressure and temperature operation on the working fluid side.

EGR coolers material for a high pressure EGR circuit is typically stainless steel. Feasible fins in this material are basically of two types: offset and wavy fins. Dimensional characteristics for the fin are: transversal pitch, height and longitudinal pitch. Quantification of additional surface provided by the fin is given by the transversal pitch and height. The lower these values are the higher heat transfer area is achieved. However, it must be balanced with the flow restriction, given the fact that a high density fin arrangement leads to high pressure drop of the gas flow.

Plate and fin type design allows high flexibility in the geometrical design of the fluid ducts. Therefore, such type of heat exchanger can be optimally designed to meet the high efficiency and low pressure drop requirements. Nevertheless, this design requires a higher material usage. When compared to a simple shell and tube heat exchanger, a plate and fin type heat exchanger design presented an improvement by 65% in volume reduction, counter balanced by an increase in weight by 20.6%, Mavridou et al. [6].

Plate type heat exchanger design allows high compactness. However, a heat exchanger for vehicle exhaust (or EGR) heat recovery operates at high pressures (on the working fluid side) and temperatures (on both gas and working fluid side), which is an important issue for plate heat exchangers. The exhaust gas flow of an ICE varies extremely during operation and is further subjected to temperature fluctuations. Regarding to this, a problem arises for plate type heat exchangers. For example, a plate pair can be heated much more greatly than an adjacent plate pair. As a result, plates of the more greatly heated plate pair expand much more. Consequently, shear stress increases and can damage the solder joint between plates. Hence, the heat exchanger plates must be capable of securely managing such fluctuations in flow rate and temperature and securely ensuring the desired evaporation of the working fluid.

Evaporator heat exchangers face the risk of instable operation due to phase change of the working fluid, e.g.: aperiodic and periodic instabilities, pressure drop instability induced by large steam, bubbles (slug flow), and oscillatory instability of parallel evaporations ducts. The evaporator heat exchanger needs to withstand high thermal and mechanical loads over a long period of time. The cost and the fouling problem are also important issues for plate heat exchangers. Following this review of the different design strategies, their advantages and disadvantages, the present study considers the cross flow tube heat exchanger (CFT-HEX) as an adequate design solution for automotive vehicle RC-WHRS applications.

## 3. Heat exchanger operating conditions

In a heavy-duty Diesel engine with high pressure EGR line, the EGR heat exchanger is provided with exhaust gas on a high temperature level as compared to the exhaust gas heat exchanger. As a reference, it can be considered that the EGR gas temperatures lie between 350 ºC and 600 °C; and in exhaust line after the exhaust gas aftertreatment system between 250 ºC and 400 °C. The gas mass flow rate in the exhaust line is about 3 times higher than the EGR line.

The working fluid of the RC-WHRS directly impacts the safety, size, performance and cost-effectiveness of the system. For HDDE vehicles RC-WHRS working fluids under investigation include: water, alcohols (such as ethanol), organic fluids (e.g. R245fa), among others [7]. Water has relative good efficiency, but has freezing problems and risk of droplet formation on the expansion process, which is an issue for the use of a turbine expander. As compared with R245fa, ethanol offers better results in terms of cycle efficiency and power recuperation. So that, at the present ethanol is the selected working fluid.

On the working fluid side, the HEX has three zones: the pre-heater (liquid), the evaporator (two-phase) and the super-heater (vapour). The temperature of the working fluid at the HEX inlet ranges from 60 ºC to 90 ºC [4,5]. The working fluid evaporation pressure will be within 10 to 40 bar. As an example, at 20 bar ethanol boils at 182 ºC. The maximum working fluid temperature for ethanol is limited by the risk of degradation [5, 7]; hence for safety reasons the present study considers a limit of 230 ºC for the superheated ethanol at the HEX outlet.

Taking into account the exhaust gas and the ethanol working fluid operating conditions referred above, model predictions had revealed that the pre-heater section corresponds to the largest part of the HEX. Furthermore, the power density for both evaporator and super-heater are similar and very high. On the other hand, the power density on the pre-heater section of the HEX is very low. Accordingly, a more compact HEX for a HDDE vehicle RC-WHRS strongly depends on the increase of the power density on the pre-heater section.

In the pre-heater section, the working fluid heat transfer coefficient is very low, which limits the overall heat transfer coefficient. This is mainly because the working fluid flow in the channels of the CFT-HEX pre-heater section is in the laminar flow regime. Furthermore, during the design process it was identified that the best compromise between high heat transfer coefficients and relatively low pressure drop is usually in the transitional flow regime, for which theoretical and empirical relations are still scarce. According to a recent review paper by

Meyer [8], flow in the transitional flow regime has mainly been investigated by Professor Ghajar and his co-workers that were the first to investigate the heat transfer and pressure drop in the transitional flow regime [9-13].

In the present practical application the fluid flows in the laminar and transitional flow regimes under both hydraulic and thermal developing flow conditions. The literature studies considered either fully developed flow [11], or average measurements of developing flow across a tube length [8]. Therefore, the heat transfer and pressure drop characteristics of developing flow in the transitional flow regime have not yet received the required attention. This paper presents a modular test section that permits the investigation of fundamental transport phenomena in the laminar and transitional flow regime, which occurs within the channels of the pre-heater section of the CFT-HEX.

## 4. Experiments

In a cross flow tube heat exchanger (CFT-HEX) the exhaust gas flows across the tube bank and the working fluid flows inside the core tubes, perpendicularly to the exhaust gas [14]. Furthermore, the working fluid presents a multipass flow arrangement through the HEX, with return chambers at the top and bottom of each row of tubes. It is important to point out that the HEX must have a large quantity (generally, more than 200) of short length tubes (100 to 400 mm) with internal diameter from 4 to 7 mm that are connected with return chambers. In this context, previous work [14] has been carried out to design and implement a RC-WHRS. The developed test bench includes the vehicle and the RC-WHRS that uses water as working fluid.

In the present work, the fundamental fluid flow and heat transfer mechanisms are investigated for a single smooth horizontal pipe with $L/D_h \approx 72$, which corresponds to the practical operating conditions of the HEX integrated in the RC-WHRS. Those conditions were replicated in the experimental set-up that uses water as the working fluid. Atmospheric pressure was considered at this early stage of the work, which minimizes design complexity.

Fig. 1 depicts a schematic representation of the experimental set-up, which includes: 1) a reservoir; 2) a fine adjustment valve; 3) a volumetric fluid flow meter; 4) a non-return valve; 5) a preheating section; 6) a test section; 7) a visualization section; 8) a condenser; 9) a pump; 10) a T connecting valve.

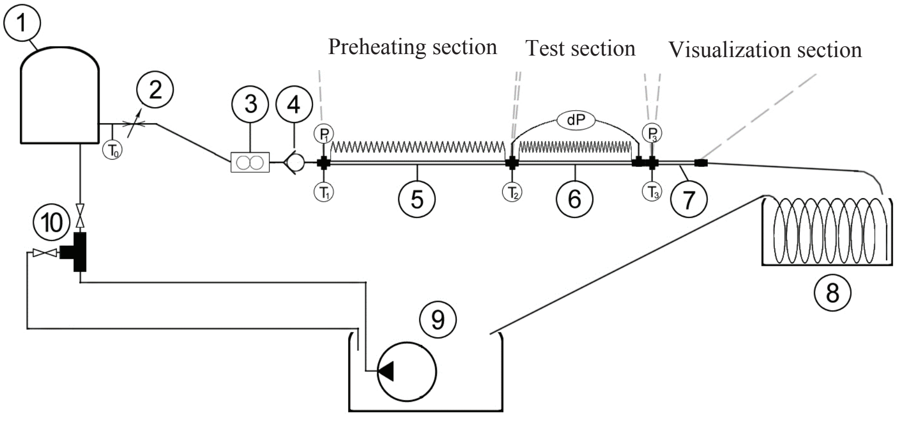

Fig. 1. Schematic representation of the experimental set-up.

The pre-heating section consists of an AISI 304 stainless steel channel with 2000 mm length and 6 mm internal diameter. Heat flux is imposed at the wall by means of a heating wire with 35.5 Ω. Temperature and pressure were measured at the inlet of the pre-heating section using a K-type thermocouple and an absolute pressure transducer (MPX4250A). The test section also consists of an AISI 304 stainless steel channel with 412 mm length and 5.85 mm internal diameter. The heating wire has 170.4 Ω and is coiled around 200 mm of the channel's length. Temperature was measured at the inlet and outlet of the test section by two K-type thermocouples. Pressure was measured at the outlet using an absolute pressure transducer (MPX4250A). Pressure drop along the channel was measured using a differential pressure transducer (Honeywell 26PC series).

## 5. Results and discussion

As aforementioned, in the present practical application the working fluid flows in the pre-heater section of the HEX in the laminar or transitional flow regimes, under both hydraulic and thermal developing flow conditions. Fig. 2 compares the experimental and predicted Nusselt number as a function of the Reynolds number. The results depicted in the Fig. 2 were obtained by a series of 130 experiments with an imposed heat flux of 92 kW/m$^2$. The correlations used for comparison are the Gnielinski (2010) for the laminar regime [15] and the Gnielinski (1976) for the turbulent regime [16]. Both correlations include entry region effects.

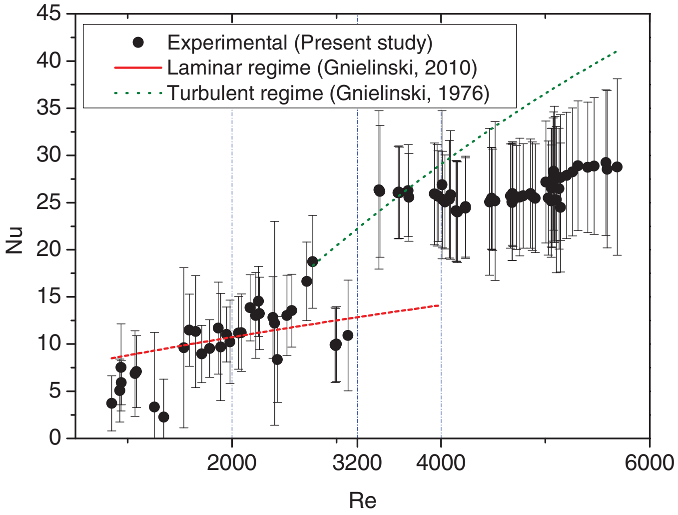

Fig. 2. Experimental Nusselt number as a function of the Reynolds number (800 < Re < 6000).

As expected, for laminar flow, the Nusselt number is larger than the analytically obtained value (Nu = 4.364). This is mainly due to the: i) developing thermal boundary layer along the whole channel; and ii) secondary flow, which increase the amount of mixing in the tube. These observations are consistent with those of Meyer [8], who refers that the Nusselt number for laminar flow regime is strongly influenced by secondary flow effects and increased with increasing heat flux.

Fig. 2 reveals that the Nusselt number increases considerably for Re $\approx$ 3200 which is a good indicator of transition. It is important to point out that the Nusselt number in the turbulent flow regime is independent of the inlet geometry and secondary flow [8]. Furthermore, it is well known that in the turbulent regime the Nusselt number increases with the Reynolds number, as it can be observed in the Fig. 2 for Re > 4500. The correlation derived by Gnielinski (1976) is for the turbulent regime [15]. In fact, turbulent conditions are not reached in the test section and the flow is characterized by fluctuations between laminar and turbulent regimes (transitional regime). Hence, the Nusselt number in the transition region is over predicted by the Gnielinski (1976) correlation [16], which was also observed by Meyer [8]. This author refers that correlations for turbulent regime usually over predict the Nusselt number for Reynolds number between the critical value (onset of transition) and about 10000.

Mixed convection combines forced convection and natural convection. In the HEX under investigation there are operating conditions where the natural convection and forced convection mechanisms act together. To identify the relevance of the different heat transfer mechanisms involved, Ghajar and Tam [17] presented a flow regime map where a new boundary for forced-to-mixed convection transition was defined. In addition, Ghajar and Tam [17] also defined an analytical solution for the determination of the laminar-to-turbulent transition upper and lower bound for re-entrant configuration tubes.

To identify the relevance of the natural and forced convection mechanisms in the pre-heater section of the HEX under practical operating conditions, the HEX was tested on the developed RC-WHRS (see, Santos et al., [14]) at three different ICE operating conditions: Diab.Test 1, Diab.Test 2 and Diab.Test 3, corresponding to three different exhaust gas thermal powers: 0.75, 3.5 and 9 kW, respectively. The experimental data cover a wider range of flow conditions, namely in terms of Reynolds number (331 < Re < 2935). The experimental points of Diab.Test 1, Diab.Test 2 and Diab.Test 3 were plotted into the Ghajar and Tam [17] flow regime map depicted in Fig 3.

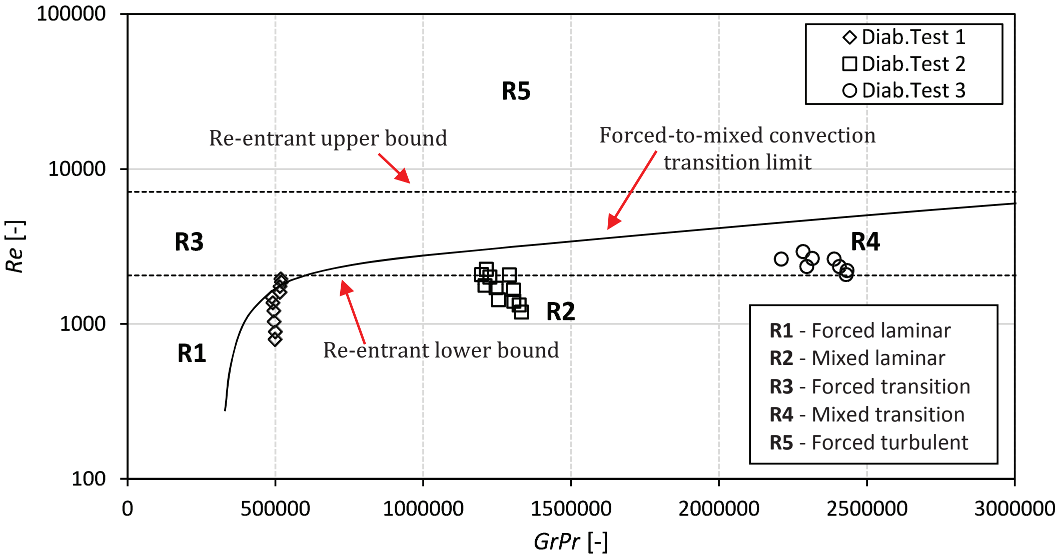

Fig. 3. Reynolds number plot as function of *GrPr* in the Ghajar and Tam [17] flow regime map.

The Ghajar and Tam [17] flow regime map identifies 5 different regions as depicted in the caption of Fig. 3. This figure also reveals that almost all of the experimental points are below the forced-to-mixed convection transition line, indicating a dominant mixed convection mechanism. In addition, Diab.Test 3 is within the flow transition regime, while Diab.Test 1 and Diab.Test 2 are almost exclusively in a laminar regime. Hence, the convection mechanisms present in these experimental conditions are mixed-laminar for most of Diab.Test 1 and 2 and Mixed transition for Diab.Test 3.

## 6. Conclusion

Among the different HEX designs available, a robust cross-flow tube heat exchanger (CFT-HEX), with the working fluid circulating inside the tubes was selected.

The working fluid flow in the pre-heater section of the HEX occurs in the laminar or transitional flow regime under both hydraulic and thermal developing flow conditions. The experimental data revealed that in the laminar regime, Nusselt numbers were higher than those predicted for fully developed flow, which is due to the mixed convection mechanism and the developing thermal boundary layer. On the other hand, in the transitional regime Nusselt number results were always over-predicted by the correlations from the literature that were developed for turbulent conditions, not reached in the present study.

Overall the results show that the correlations found in literature can be used with strong restrictions for the particular flow conditions studied here. Taking into account the optimization of the HEX pre-heater section, the present study data reveals that Re > 3200 allows a significant increase of the Nusselt number (Nu > 20) as compared with laminar flow regime.

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