Modelling, sizing and testing a scroll expander for a waste heat recovery application on a gasoline engine

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Abstract. Waste heat recovery technologies in a mobile application emerge every time energy becomes a valuable resource. It has been the case in the 70s with oil crisis and it is starting to regain some interests now due to the continuously rising price of oil and due to the restrictive standards imposed by the different governments. This paper deals with the recovery on the exhaust gases of an internal combustion engine by using a Rankine system. The study focuses on the expander, which is one of the most important components of the system. The use of a scroll expander operating with steam is currently investigated through simulation and experimentation. This paper presents the modelling of a scroll expander. The model is a detailed model including various losses such as leakage, friction or under or over expansion. This model has been used to design and size a tailor-made scroll expander. This was necessary due to the small amount of expanders on the market and also to have a machine that fits our application. After designing the machine, a prototype has been built. It has also been tested on our prototype bench of waste heat recovery on a gasoline engine, by means of a Rankine cycle. Measured performance will be presented, analysed and compared to predictions by the model. The first results will be presented here and discussed in order to give recommendations for the design of next prototypes.

1. Introduction

After various restrictive laws on pollutants in the exhaust gases of passenger car engines, future legislations will be focused on the dioxide carbon emissions. Figure 1 shows that Europe is currently the leader on the average CO₂ emissions and its target for 2020 is 95 g of CO₂ per km. Similar goals are set by other governments, such as USA or China, within the same period of time. However, the trend is to reduce the carbon dioxide emissions.
In order to reduce the carbon dioxide emissions, the engine efficiency has to be improved and its fuel consumption decreased. With an efficiency order of magnitude of about 35%, passenger car gasoline engines show various sources of losses. The major losses in an engine are thermal losses and are present in the engine coolant and in the exhaust gases. Those thermal losses represent roughly 60% of the energy released during the combustion process and this percentage can vary widely when the engine is solicited at part load [2, 3].

The coolant and the exhaust gases are two losses that can be exploited. However, this paper focusses on the waste heat available in the exhaust gases only. That choice has been justified by the fact that the exhaust gases have a higher exergy content than the coolant [4].

Lots of technologies can be used to recover waste heat from the gases. Currently, almost none of them are mature and then are still in research phase. Legros et al. [1] have conducted an extensive review of those technologies and provide a comparison between them. The outcome of that comparison shows some advantages of the Rankine cycle heat engine.

The present paper will present the design steps of the expander of the Rankine cycle system used in a waste heat recovery application and introduce some preliminary experimental results.

2. Sizing the expander
Available expanders are not numerous on the market and most of them are not suitable for a mobile application such as waste heat recovery on passenger cars. Therefore, a new expander has to be designed to match more closely the project requirements.

Volumetric expanders are preferred to turbine expanders. Indeed, the latter work at higher speeds and are less suited for small power applications (ranging from 0 to 10kW). Moreover, the volumetric expanders tends to show better isentropic efficiencies [5]. Volumetric expanders are characterized by a displacement and, for most of them, a volume ratio.

Among the different technologies of volumetric expanders, the most common are scroll, screw and piston expanders [5]. Isentropic efficiency is a key parameter of the machine because the energy that has to be released to the condenser will be lower with a higher efficiency for a given amount of energy transfer to the evaporator. Scroll expanders usually show higher efficiencies than piston expanders [6, 7] and screw expander are usually for larger power range.
Scroll expander technology has been selected for the present waste heat recovery application and the working fluid is water. The design of the expander is divided in two steps. The first one aims at defining the characteristics of the expander and the second one will define the geometry that fits those characteristics. The following subsections will introduce to the driving cycles and the methodology and reasoning used to size the scroll expander.

2.1. Driving cycles

Passenger car fuel consumption are evaluated on normalized driving cycle. The current cycle in Europe is called NEDC, or New European Driving Cycle. This cycle is rather old and will soon be changed to have a cycle more representative of the actual behavior of car drivers. The future cycle is called WLTC and stands for World harmonized Light vehicles Test Cycle.

An analysis of those cycles has been conducted. A frequential distribution of the available power in the exhaust gases reveals that the power range is very widespread but it is mainly located in the first class as shown on Figure 2. The frequential distribution may slightly varies depending on the engine or the driving cycle but the trend is always the same. On a mid-size passenger car gasoline engine, more than 99% of the time the power available in the exhaust gases is less than 50kW with a peak power of 84kW.

That result coupled to the highly transient behavior of an engine do not allow a design of a nominal point. Instead, an optimization has been conducted. By reducing the power range to the three first classes (0 to 50kW), an optimization of the produced power by the expander can be done.

2.2. Characteristics of the expander

The characteristics of the expander are its displacement and its volume ratio. Those characteristics were determined based on a Rankine cycle steady-state model coupled to the exhaust gases on a driving cycle. The model is similar to the one developed by Quoilin et al. [8].

2.2.1. Volume ratio. The volume ratio is defined by the ratio of the volume of the chamber right before the opening to the discharge chamber and the displacement. This ratio is responsible of over- or under-expansion losses. Those losses appear because the pressure at the end of the expansion is different from the exhaust pressure. In the case of over-expansion, the pressure after the expansion is under the exhaust pressure. The under-expansion leads to a pressure superior to the exhaust pressure after the expansion and if the volume ratio would have been higher, the produced power could have been increased.
Looking at existing scroll machines, their volume ratio varies roughly between 2.4 to 5 [9, 10]. It must also be noted that the higher the volume ratio, the more contact points there are in the expander. Those contact points defines the different chambers and that number of pairs of contact points should be minimised in order to reduce the internal leakage [5].

The sensitivity of the volume ratio on the produced power is also very interesting. An expression has been derived and is given in equation (1). Dividing that expression by the isentropic power gives a ratio of the residual power after a reduction of the volume ratio. Plotting this expression reveals that a significant decrease of the volume ratio can be made without reducing too much the mechanical power.

$$w_{red} = P_{su}v_{su} \left( \frac{1-\gamma}{\gamma} \left( \frac{r_{v,red} - 1}{\gamma - 1} \right) + r_{v,red} \left( r_{p,red} - \frac{1}{r_{p}} \right) \right)$$

As a conclusion, a small built-in volume ratio is preferable for two reasons. First, a smaller volume ratio will lead to a more compact machine and secondly, working with a smaller volume ratio will reduce the number of contact pairs and therefore, internal leakage.

2.2.2. Displacement. The displacement of the machine is the parameter that has been optimized to maximise the power produced by the expander. The displacement is function of several parameters such as the working fluid nature, the evaporating pressure and the mass flow rate.

In order to maximise the mechanical power, the Rankine cycle simulation code is run with inputs set as the mean value of the three power classes defined in the previous subsection. Three simulation results are then obtained and by varying the displacement and the evaporating pressure, three cartography maps can be drawn, displaying the rotational speed in function of the displacement and the evaporating pressure. One of this cartography is presented in the Figure 3 and is for the second class.

![Figure 3: Cartography map of the rotational speed in function of the displacement and the evaporating pressure](image)

To understand Figure 3, bounds on the variables must be first set. The boundary of the rotational speed has been estimated based on the observation of what is achievable technically in scroll machines. For the purpose of presentation of the map, rotational speed that would have overcome the upper bound is set to that bound.
Therefore, if the displacement of the expander is too small, the machine cannot physically run sufficiently fast to be able to recover the available power in the exhaust gases. Using evaporating pressure that is too low leads to the same conclusion.

On another hand, if we look at the lowest available power class, the highest displacement/evaporating pressure would like to ridiculously low rotational speed. Low rotational speed leads to high internal leakage and therefore, reduced isentropic efficiency.

Based on that reasoning, an optimal displacement should be encountered on the overall cycle. Running the simulation code on the overall cycle for several displacement provides the Figure 4. A maximum of mechanical power appears for a given displacement. The overspeed frequency is also displayed and it can clearly be seen that lowest displacements results in high overspeed frequency. In a case of overspeed, the cycle controller would have to reduce the amount of energy injected into the cycle, decreasing then the mechanical power that can be produced.

![Figure 4: Mechanical power and overspeed frequency in function of the displacement for a WLTC](image)

2.3. Definition of the geometry

The definition of the geometry of the scroll expander requires more than a simplified model such as the one used in the previous analysis. A detailed model of scroll expander has therefore been built but the explanation of such a model is out of the scope of this article. More information can be found in the literature [11-13].

Based on the characteristics defined in the previous subsection, a geometry can be defined. Several geometry can exist and return the same characteristics. An optimisation could have been run to define the best geometry. However, the simulation model was not calibrated and losses could not be predicted correctly. This imprecision in the model could result in different results of optimisation and would make inefficient the use of an optimisation.

The shape of the suction port has also been studied to reduce the pressure drop losses. In scroll machine, the moving involute can pass over the admission port, leading to a reduction of admission area. This decrease leads in a pressure drop. Since the clearance volume in scroll expander is not a drawback, its minimisation was not mandatory and a non-negligible clearance volume has been implemented in order to maximize the suction port cross sectional area. Parametric study show that such an increase of the area of the suction port allows the reduction of more than 50% of the pressure drop at the highest rotational speed [11].

A 2D view of the expander geometry is provided in Figure 5. The shape of the suction port can be seen.
2.4. High temperature investigations

The investigated Rankine cycle and expander uses water as the working fluid. In order to achieve higher cycle efficiency, evaporating pressure needs to be increased and therefore, the temperature of the working fluid entering the expander is also high. Literature reports few scroll expanders but all of them showed a maximal operating temperature of 200°C. In order to increase that temperature limit, investigations on the tribology of the expander have been conducted. Those investigations have been also mandatory due to the weak miscibility of water and oil. The mix of water and lubricating oil makes the design very complex by adding a lot of constraints to the system (complexity of the separation, additional pump…). Therefore, the design of the expander has been focussed on an oil-free expander. The study of the material in contact in the scroll expander has revealed autolubricating materials that work with high temperature and provide low friction coefficient, reducing at the same time the friction losses. The involutes are made of coated aluminium and the tip seal are made of an autolubricating material. The details for the study fall out of the scope of this article and more details can be found in [14].

3. Experimental results

After the design of the expander, the manufacturing of the machine has been achieved. The scroll expander has been tested in a Rankine cycle test bench. The Rankine cycle is coupled to a gasoline engine to provide exhaust gases into the evaporator. The scroll expander is coupled to an alternator, controlled by a variable resistance. The alternator consists in a synchronous electric machine with an efficiency higher than 90%.

Pressure and temperature measurements are available at the supply and exhaust of the expander. The mass flow rate of working fluid and the electrical power produced by the generator are also measured. Temperature next to the bearings are also measured. The first test campaign is presented in this article. The measurements were made by varying the rotational speed of the alternator for various mass flow rate. A diagram of the test bench and the sensors position are displayed in Figure 6. For simplicity of control, the cycle is an open loop.
Isentropic efficiency and filling factor are presented in Figure 7. It can directly be noted that the maximal isentropic efficiency reached is about 26%. That efficiency is relatively low in comparison of what could be observed in literature. Isentropic efficiency is defined by the equation (2).

\[
\varepsilon_s = \frac{\dot{W}_{el}}{M(h_{su} - h_{ex,s})}
\]  

(2)

In order to understand why the efficiency is so low, the filling factor can be evaluated with the equation (3). The filling factor is presented in Figure 7. For very low rotational speeds, the filling factor in very high and leads to very poor efficiency. The increase of rotational speed tends to reduce the filling factor and a minimal value of around 2 could be reached. Although the decrease of filling
factor is impressive, a value of 2 is still important and an analysis of the expander revealed that flank leakages are important due to a pretty high leakage clearance.

\[
\varphi = \frac{M}{\rho_s V_s N_{rot}}
\]

Looking at the temperature of the bearings, a trend with the rotational speed is shown in the Figure 8. That dependence with the rotational speed provides evidence of mechanical losses in the bearings.

![Figure 8: Bearing temperature in function of the rotational speed](image)

4. Conclusion
A design methodology has been defined and applied to size a new oil-free scroll expander for a waste heat recovery steam Rankine cycle. This has been mandatory since the scroll expanders available on the market are not suited to a waste heat recovery application on a passenger car gasoline engine.

The design methodology has been successfully applied and the sized expander has been manufactured and tested. The preliminary tests show a rather low isentropic efficiency, mainly affected by a huge internal leakage and, secondarily, by the friction losses. Additional investigations regarding the leakage clearance and the tip seal configuration will be conducted in order to increase the isentropic efficiency.

An Oil-free machine was manufactured and tested without mechanical issues, helping in simplifying the Rankine loop for vehicle integration.

The compactness of the machine is very good and the expander is currently similar in size as a compressor for air conditioning applications in automotive and commercial vehicle applications.

The experiments showed that high temperatures could be achieved and a maximum temperature of 240°C at the inlet of the expander has been reached.

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