PUMP DEVELOPMENT FOR AN EXHAUST HEAT RECOVERY BOX ON HEAVY DUTY TRUCKS

F. Albergucci, S. Watts, J. Pouponnot, A. Darmedru and R. Daccord*

*EXOES
6 avenue de la grande lande
33170 Gradignan
info@exoes.com

* Corresponding Author

ABSTRACT

Nearly 30 percent of the fuel energy in an internal combustion engine is lost as waste heat in the form of hot exhaust gases. Nowadays it seems clear that the heavy duty manufacturers will implement bottoming Rankine cycles to recover the exhaust heat on their long haul trucks in the 2020s as an answer to future stringent regulations and the still increasing customer pressure for reductions in operating costs.

The Exoès Company developed several components for an organic Rankine system packaged in a box, to be installed in a truck tractor close to the exhaust pipe. This paper is focusing on the pump development results.

A pump prototype and a dedicated test rig were developed in order to assess its performance and to perform endurance tests. The key challenges for such a product development are to keep a good volumetric efficiency with high pressure difference between outlet and inlet around 30bars and with corrosive and low viscosity fluids such as ethanol, pumped in saturated liquid state at quite low flows. Leakage paths are then critical to the pump efficiency as well as the electric actuator design to match the pump torque and speed requirements with high efficiency. Another challenge is to maintain this leakage rate under control over time despite wear. The choices made for the pump architecture and materials will be presented as well as test results.

1. INTRODUCTION

Recovering waste heat on vehicles consists in applying a thermo mechanical converter on one or several heat losses, such as exhaust gas, coolant, exhaust gas recirculation loop, inlet air intercooler, etc. In this paper we consider using a vapor cycle (so called ORC for Organic Rankine Cycle) on the exhaust gas heat only. The pump of the ORC has to circulate the hardly subcooled working fluid at rather low pressure – the closest possible to the atmospheric pressure – towards the evaporator at higher pressure at a controlled flow rate. The target working fluid is a mixture of ethanol, water, denaturant and lubricant as a choice widely spread among OEMs. This mixture enables to have high cycle efficiency in the vehicle but it implies difficulties especially for the pump. The optimized pressure ratio is high ~20 whereas the flow rate is low as well as the viscosity.

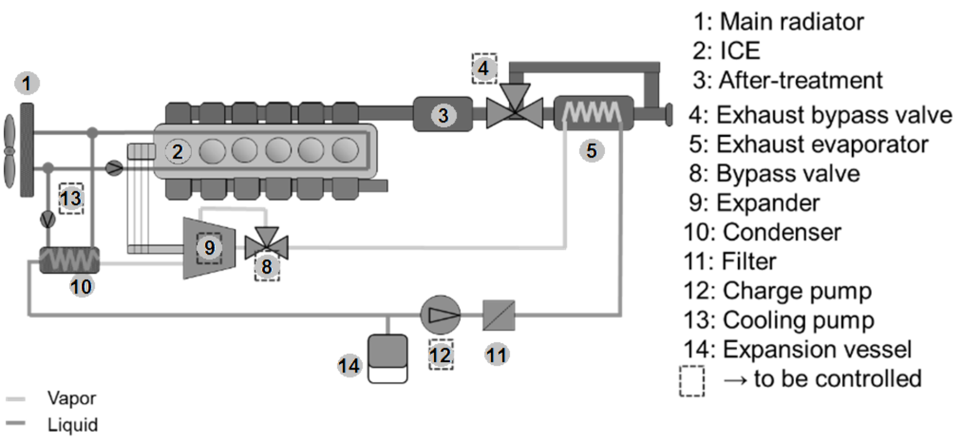

Figure 1: Organic Rankine Cycle system for a heavy truck

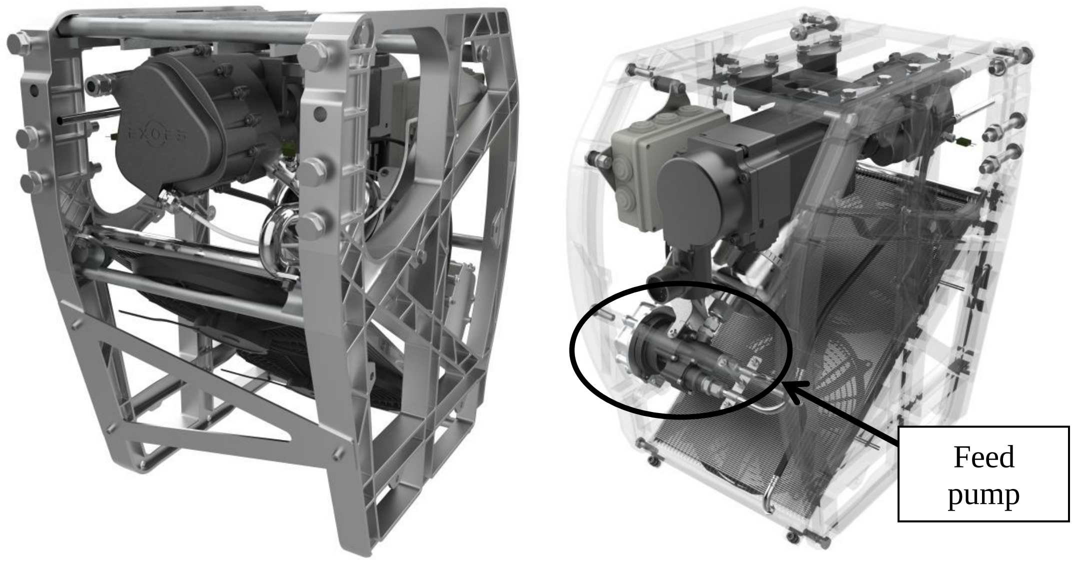

Figure 2: Rankine Cycle packaged in a box for a heavy truck

Among all the pump types available (piston pump, diaphragm pump, etc), external gear pump architecture was chosen for its compactness, low vibrations level and cost-effectiveness []. This type of pump consists in two gears, one being driven by the other. The working fluid is led by the gears from the Low Pressure Area (LPA) to the High Pressure Area (HPA).

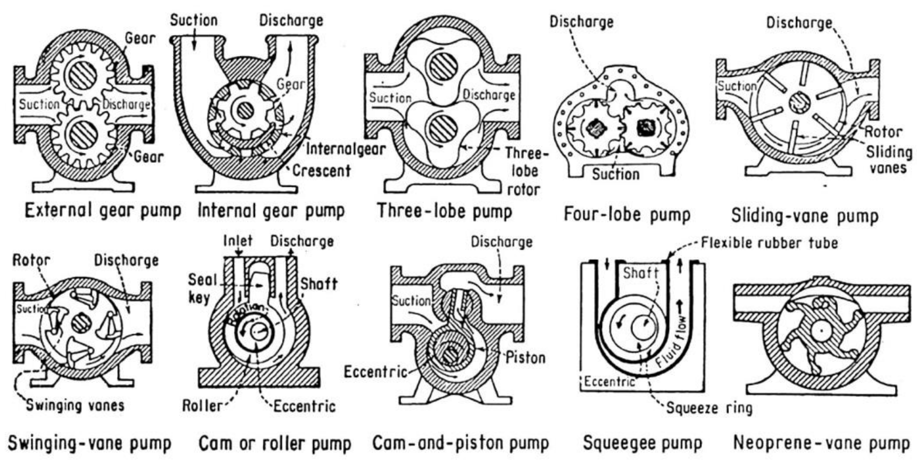

Figure 3: types of displacements pumps. Suction area is the LPA, whereas discharge is HPA.

Internal leakages are critical to the pump efficiency as well as the electric actuator design to match the pump requirements with high efficiency.

The authors found little bibliography, and no comprehensive scientific publication in open literature about the design of gear pump for Rankine automotive systems [9], [10]. Among the documents listed in reference, some mention the use of an industrial pumps for Rankine demonstrator [2]; while others describe alternative pump technologies like sliding vane [1] or reciprocating piston [5].
Additionally, generic to gear pump design is detailed in various aspects beneficial to this study in [3], [4], [6], [7], [8].

2. INCREASING PUMP PERFORMANCE

Since a low-viscosity fluid is used as working fluid, minimizing leakages between HPA and LPA in the pump is crucial in order to maintain acceptable performances. In this paper, we focused on the volumetric efficiency, which is defined as followed:

$$\eta_{vol} = \frac{\dot{V}_{vol}}{N_{rpm} \cdot V_s} \qquad (1)$$

With $\dot{V}_{vol}$ the volumetric flow, N_{rpm} the pump rotation speed, V_s the swept volume

The leakage sources are described in the paragraphs below, together with the solutions implemented on this pump to limit these leaks. In an external gear pump, the main leak paths are on the gearing teeth periphery, the gears flanks and the gears interface.

2.1 Leaks at gears teeth periphery

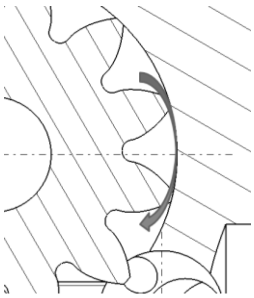

Figure 4: Leak path at gears teeth periphery

Many strategies exist in order to reduce this leak. In our case, we chose to use gearing in fiber charged PEEK, which has a higher thermal expansion coefficient than steel. Consequently, the gears expand more than the casing when temperature rises. Then, the play between the gears and the casing has been set so that the gears can be mounted at 20°C, but are in interference with the casing at 100°C. As a honing process, an initial running-in period is then necessary at 100°C so that the gears match perfectly the casing, the gearing being “machined” against the casing.

2.2 Leaks at gears flanks

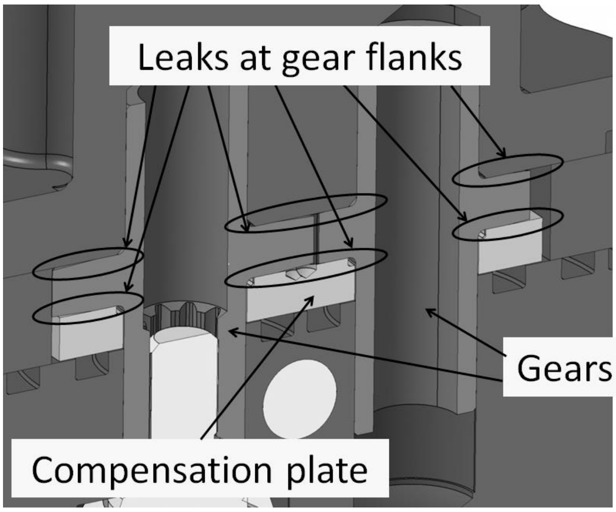

Figure 5: Leak path at gears flanks

In order to minimize this leak, a floating part called compensation plate was used. This part acts just as a spring, pushing the gearing on the casing in axial direction. The initial pressure of the compensation plate against the gearing is done through an O-Ring, which stiffness should be wisely chosen (if this O-Ring is too stiff, the gearing will be too strongly pushed against the casing, leading to exaggerated friction and wear). This O-Ring also isolates the LPA from the HPA on the compensation plate, casing side (see Figure 6).

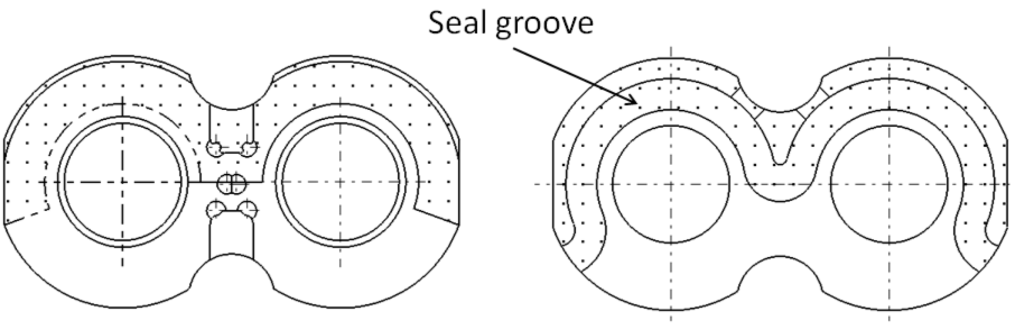

Figure 6: floating compensation plate design.
Left: gear side. Right: casing side. HPA is highlighted by stippling

The O-Ring groove on the compensation plate needs therefore to be designed so that resulting pressure on casing side is a bit higher than pressure on gearing side. If this is properly done, the compensation plate is “closing” the circuit, i.e. is pushing the gears against the casing. If not, the circuit is “opened” and flow is lost, as shown in Figure 7. Since the pressure distribution on the compensation plate is hard to predict, several prototypes iterations were necessary in order to find the good compromise.

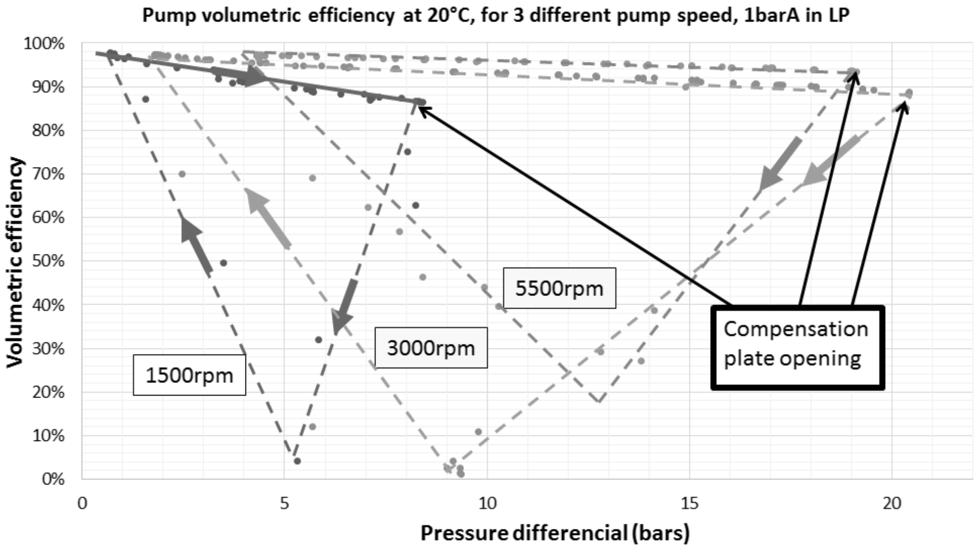

Figure 7: example of a compensation plate which opens the circuit (HPA too small on casing side)

This floating compensation plate is also a way to ensure that the performance remains constant despite gears wearing during pump lifetime.

2.3 Leaks at gears interface

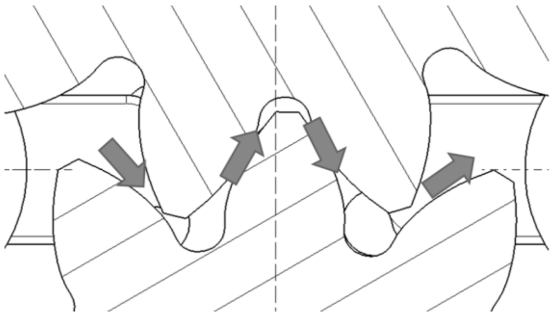

Figure 8: Leak path at gears interface

Contrary to the other leaks, fight against this leak path is not necessarily useful in our case. Indeed, this area presents some pockets in which volume is crushed and expanded in a very high speed. Since liquid cannot be compressed, low leak on this area would mean very high pump torque.
In our pump, small cavities were added on the housing in order to increase these leaks during critical compression and expansion phases. This was done in order avoid suction, and therefore reduce the NPSH (see Section 3.1) even if volumetric efficiency is reduced.

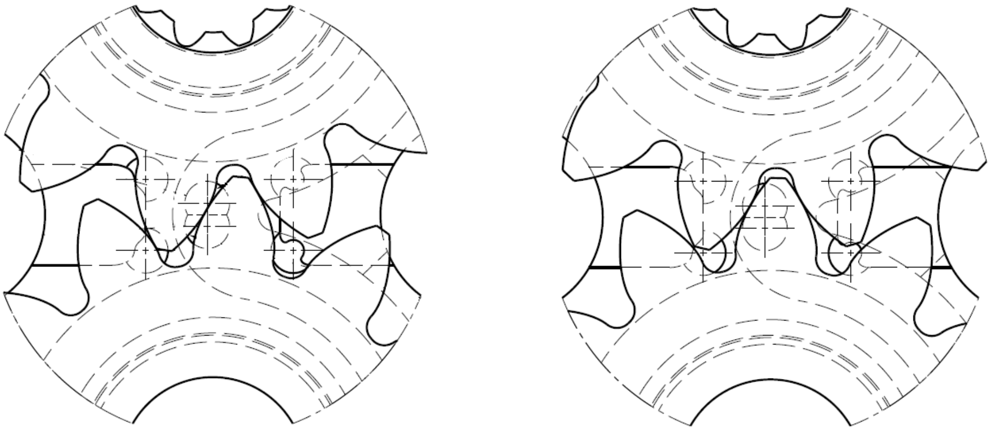

Figure 9: gears interface cavities for 2 gearing positions

2.4 Influence of pump rotation speed and pressure on volumetric efficiency

The influence of the leaks on the performances decreases when the pump is running at high speeds. The following graph shows the volumetric efficiency for several pump speeds, tested with an increasing pressure differential between LPA and HPA. A volumetric efficiency equals to 100% means there is no leak at all.

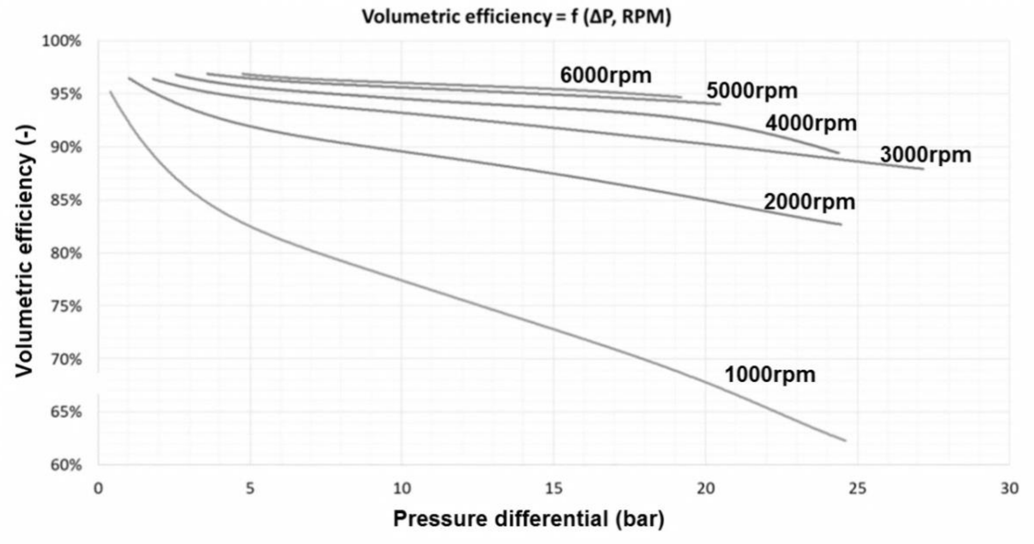

Figure 10: influence of the pump speed and pressure differential on the volumetric efficiency

It is therefore more efficient to use this pump at high speed, i.e. with a high fluid flow, even if at some point this will be balanced by additional friction losses as well as electric drive efficiency.

2.5 Influence of working fluid viscosity on volumetric efficiency

In a Rankine cycle system, oil is mixed to the working fluid in order to lubricate mechanical components such as the expander, which is the "motor" of the system. As shown in Figure 11, the oil percentage in the working fluid has a strong influence on the kinematic viscosity, as well as the temperature. Since a fluid with a high viscosity is less sensitive to leaks, volumetric efficiency should decrease at higher temperature and lower oil percentage.

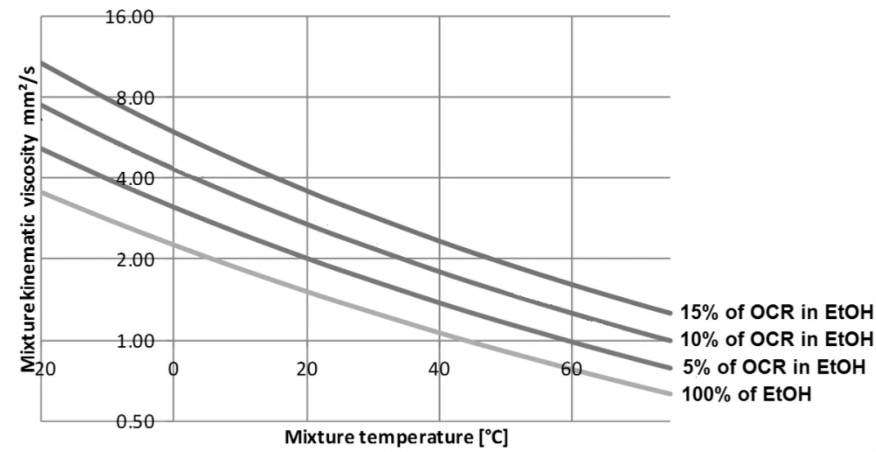

Figure 11: kinematic viscosity of ethanol with several different oil percentages

3. DECREASING PUMP NPSH

3.1 NPSH – definition

Generally, a fluid enters a pump by suction. As a consequence, its pressure decreases. If the fluid enters close to its saturation pressure, the risk of cavitation is high.

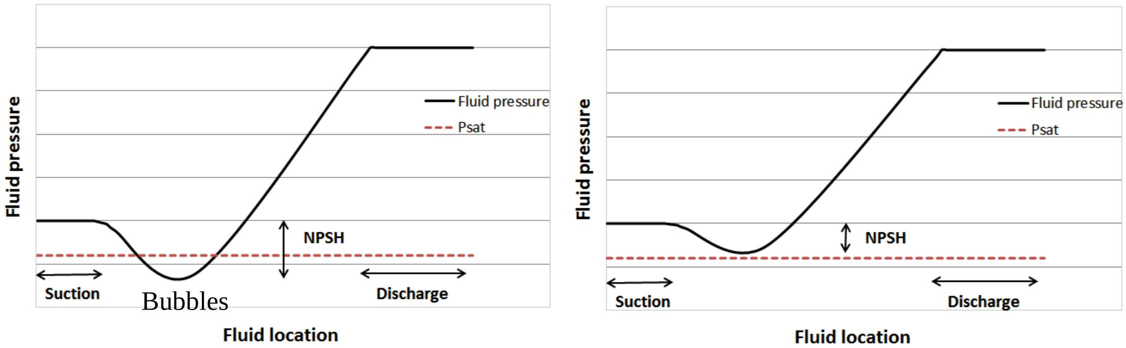

Figure 12: Left graph: fluid pressure goes below P_{sat} (=saturation pressure), cavitation occurs.
Right: NPSH is lower, fluid pressure remains above P_{sat}. No cavitation occurs.

3.2 Influence of the NPSH / cavitation

In a Rankine cycle system, it is a clear advantage for the pump to be able to run with a working fluid in liquid phase close to its boiling temperature without cavitation. Indeed, sub cooling the working fluid is a direct loss in the ideal Rankine cycle, as shown in the graph below. Using a pump with a low NPSH is therefore a way to improve the global efficiency of the Rankine system.

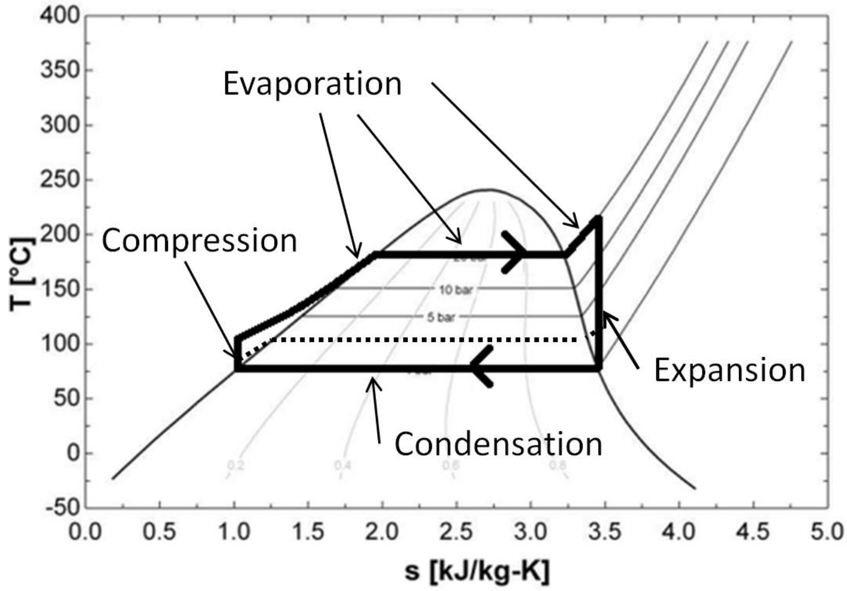

Figure 13: T-S diagram for a typical Rankine cycle (plain).
Diagram with high NPSH pump is shown in dashed, requiring more pressure at exhaust to avoid cavitation.
Therefore, less power is produced (diagram area is lower)

Moreover, running close to the boiling point allows downsizing the condenser, which is a heavy component in the system. Furthermore, the working fluid can be condensed thanks to a cold source which may be already at around 60°C / 80°C (for instance, working fluid may be cooled down thanks to the vehicle engine coolant circuit), or the environment of the Rankine system might be already hot if implemented close to the ICE.

Reducing the NPSH to its minimum is the best way to avoid cavitation when close to saturation temperature. Cavitation shall be avoided, since it decreases dramatically the volumetric ratio and may damage the mechanical parts over time.

NPSH is directly linked to the pressure drop in the pump suction area, which shall be as "direct" as possible. Since it was necessary to implement a filter before the pump (to avoid any particle to go through the gearing), keeping the NPSH as low as possible required another design trick. This was done by implementing the buffer storage tank (expansion vessel) directly on the gears suction area, after the filter so that its pressure drop has no influence on pumping process.

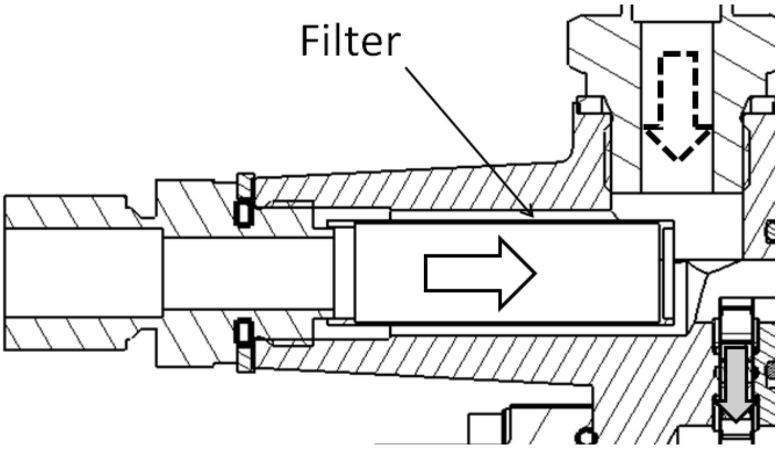

Figure 14: plain arrow is the flow coming from the condenser, and dash arrow is the fluid controlled in pressure thanks to the expansion vessel located just above.

Thanks to this buffer location, and other design features such as the cavities presented in Section 2.3, the NPSH was measured at a level lower than 300 mbar.

3.3 Influence of the NPSH on the global Rankine efficiency

In order to illustrate the loss of efficiency of a Rankine system when the subcooling needed by the pump is high, a Rankine cycle was simulated with several NPSH levels for the pump.
In this example, the cycle considered is static: steady expander speed, steady exhaust temperature, etc… The working fluid considered is ethanol.
In order to avoid cavitation in the pump when fluid is at cold source temperature, the condenser pressure needs to be modified using expansion vessel. As a consequence, when the NPSH is high, the pressure at expander outlet needs to be high also, and expander gives less power (Pressure ratio between inlet and outlet of the expander is reduced).
As shown in the graph below (Figure 15), the power produced by the expander decreases when the NPSH of the pump increases.

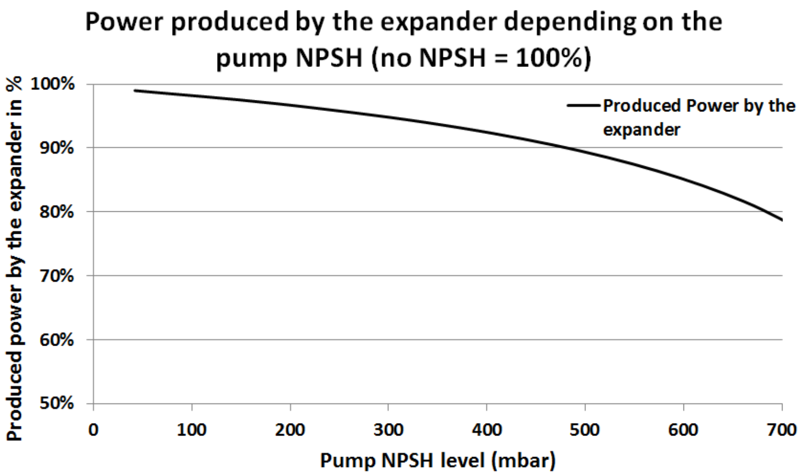

Figure 15: expander relative performance (in %) depending on the pump NPSH level.

To give an example: let us assume that the cold source has a temperature equal to 68°C and that we use a working fluid with a boiling point equal to 78°C at 1barA. The inlet pressure of the expander is set to 12 barA. In this case, if a pump with a NPSH equal to 600 mbar is used, then the pressure at

expander outlet needs to be at 1,5 barA, and the expander relative performance is around 84% to the ideal case where the pump could circulate saturated liquid.
Now, if the pump presented in this document is used (NPSH = 250 mbars), then the fluid pressure at the outlet of the expander can be decreased to 0,87 barA. The expander relative performance is then equal to 96%.
This means that, for this example, the Rankine cycle efficiency is improved by 12% when a pump with a low NPSH is used.

4. CONCLUSIONS

Leakage paths are critical to the pump in order to maintain a good volumetric efficiency. But, in the case of a pump used in a Rankine cycle, design priority needs to be done on decreasing NPSH.
Since this design philosophy was followed for the prototype, the measurements done on the pump confirmed that it shows a good trade-off between mechanical and volumetric efficiency as well as NPSH required, in order to reach the best cycle efficiency. This is therefore an ideal candidate for an application in Rankine cycles for heavy duty trucks.

Figure 16: pump prototype, shown with its expansion vessel.

NOMENCLATURE

EtOH	Ethanol
HPA	High Pressure Area
ICE	Internal Combustion Engine
LPA	Low Pressure Area
NPSH	Net Positive Suction Head
OCR	Oil Circulation Ratio
OEM	Original Equipment Manufacturer
PEEK	Polyether Ether Ketone

REFERENCES

[1] Giuseppe Bianchi, Fabio Fatigati, Stefano Murgia, Roberto Cipollone, *Design and analysis of a sliding vane pump for waste heat to power conversion systems using organic fluids*, Applied Thermal Engineering 124 (2017) 1038–1048

[2] Dickes, Rémi; Dumont, Olivier; Declaye, Sébastien; Quoilin, Sylvain; Bell, Ian; and Lemort, Vincent, *Experimental Investigation of an ORC System For A Micro-Solar Power Plant* (2014). International Compressor Engineering Conference. Paper 2372. https://docs.lib.purdue.edu/icec/2372

[3] P. J. Gamez-Montero 1R. Castilla and E. Codina, *Methodology based on best practice rules to design a new-born trochoidal-gear pump*

[4] Saleem Ahmed M. // Tikrit Journal of Eng. Sciences. - Mosul : [s.n.], December 2009, *Effect of Tooth Geometry on Gear Pump Performance* - 4 : Vol. 16. - pp. 20-27.

[5] Arnaud Landelle, Nicolas Tauveron, Rémi Revellin, Philippe Haberschill, Stephane Colasson, *Performance investigation of reciprocating pump running with organic fluid for organic Rankine cycle*. Applied Thermal Engineering, Elsevier, 2017, 113, pp.962 - 969. 10.1016/j.applthermaleng.2016.11.096. hal-01457279

[6] E. Mucchi, G.Dalpiaz, A.Fernàndez del Rincòn, *Elasto-dynamic analysis of a gear pump–Part I: pressure distribution and gear eccentricity,* Mech. Syst. Sign. Process. 24(2010) 2160-2179

[7] Yu Murai Masayuki Tsuchiya, *Development of a lightweight and low-friction resin materials for a oil pump rotor*

[8] Chen Liping Zhao Yan, Zhou Fanli, Zhao Jianjun, Tian Xianzhao // Proceedings 8th Modelica conference. - Dresden : [s.n.], 2011, *Modeling and simulation of gear pumps based on modelia/MWorks*

[9] M.Weigand, A.Merz, J.Branczeisz, Toni Jankowski - Patent Application *External gear unit* DE102017213467A1 2017-08-03

[10] G.Bredenfeld, J.Branczeisz, M.Riedle - Patent Application *Fluid pump for a waste heat recovery system* DE102016225847A1 2016-12-21

ACKNOWLEDGEMENT

We would like to express our special thanks to Xavier Durand, who contributed a lot to the design of the pump.