Initial Design of an Optical-Access Piston Expansion Chamber for Wet-Expansion

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Abstract

High efficiency expanders which can cope with fluid-vapour mixtures (i.e wet expansion) at the inlet and during expansion have the potential to increase the power output from thermodynamic power cycles. Volumetric expanders are considered suitable, yet experimental results are scarce and there is no model that can predict the performance of the expansion process. This is mainly due to the knowledge gap on the fundamental aspects of two-phase expansion and the non-equilibrium effects. Therefore, in this work, a variable volume piston expansion chamber is designed. The influence of the main design parameters are investigated by means of a simple deterministic model. Based on initial figures of merit, the component selection and technical layout is presented. The optical access in a later stage will be crucial to allow investigating the mechanistic process of the nucleation and the interfacial effects. This understanding is essential for generalization of the results to other working fluids and geometries.

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Keywords: two-phase expansion, piston expander, flash evaporation

1. Introduction

In the last decade, the organic Rankine cycle (ORC) has become a mature technology to convert low temperature heat to electricity. However these commercially available ORCs are all of the subcritical type. Yet, studies in scientific literature clearly show the potential for increased performance with alternative cycle architectures [1, 2]. The trilateral cycle (TLC) is one of the most promising modifications and can boost the power output up to 35%.

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The reason for this can be found in the lower exergy destruction during heat transfer from hot fluid to ORC working fluid. The main challenge however, is the development of high efficiency expanders which can cope with fluid-vapour mixtures (i.e. two-phase mixtures) at the inlet and during wet-expansion.

Nomenclature

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>A</td>
<td>area, m²</td>
</tr>
<tr>
<td>F</td>
<td>force, N</td>
</tr>
<tr>
<td>h_{lg}</td>
<td>latent heat, J/kg</td>
</tr>
<tr>
<td>L</td>
<td>length, m</td>
</tr>
<tr>
<td>m</td>
<td>mass, kg</td>
</tr>
<tr>
<td>ṁ</td>
<td>mass flow rate, kg/s</td>
</tr>
<tr>
<td>p</td>
<td>pressure, Pa</td>
</tr>
<tr>
<td>\dot{q}</td>
<td>net heat flux, W/m²</td>
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</table>

Greek symbols

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>\rho</td>
<td>density, kg/m³</td>
</tr>
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</table>

Subscripts and superscripts

<table>
<thead>
<tr>
<th>Subscript</th>
<th>Description</th>
</tr>
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<tbody>
<tr>
<td>0</td>
<td>initial starting values</td>
</tr>
<tr>
<td>cyl</td>
<td>cylinder</td>
</tr>
<tr>
<td>e</td>
<td>equilibrium</td>
</tr>
<tr>
<td>i</td>
<td>interface liquid vapour</td>
</tr>
<tr>
<td>sgl</td>
<td>saturated vapour</td>
</tr>
<tr>
<td>sl</td>
<td>saturated liquid</td>
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Volumetric expanders are considered suitable, yet few results are presented up to now. Experimental results are scarce and there is no model that can predict the performance of the expansion process. This is mainly due to the knowledge gap on the fundamental aspects of two-phase expansion and the non-equilibrium effects. Mainly for twin screw-expanders some experimental results are available. Smith et al. [3], performed measurements on double screw expanders with inlet vapour qualities from 50% to 25%. They showed adiabatic efficiencies ranging between 40% and 80%. A trend, indicating reduced adiabatic efficiencies under two-phase expansion for high efficiency single phase double screw machines and vice versa, was shown by Öhman and Lundqvist [4]. However, they state that the available test data in literature is scarce and that the actual behaviour of the two-phase expansion is unknown. Smith et al. [3] demonstrated that a model with the assumption of a homogeneous one-dimensional flow gives a good prediction of the p-V diagram. This simplification is valid according to the authors because of the considerable rotational component. Considering a homogeneous flow, the hypothesis is that the liquid part is broken up in sufficient small components which promote a sufficiently mixed flow. The small particles will flash fast to vapour and give rise to an equilibrium temperature between the gas and the liquid phase.

An alternative to the twin-screw expander would be a piston expander. Piston expanders allow for larger built-in volume ratios (between 6 and 14) than twin-screw expanders (between 2 and 8) and are thus more adapted to two-phase expansion. The simple geometry furthermore allows for a reduced construction cost. As such, piston type expanders for two-phase flow are readily found in CO₂ refrigeration and CO₂ power systems [5]. A disadvantage of the reciprocating expander is the inherent dead volume that results in some pre-expansion losses. There is also the complexity associated to actuating and timing the inlet and outlet valves. As there is no direct rotational component, it is furthermore unclear whether the assumption of an equilibrium homogeneous flow is still valid. When looking for example to static flashing, see further in section 1.1, there are clearly non-equilibrium effects. Fortunately, the geometry of the piston expanders lends itself to in-situ flow measurements. The process of the two-phase expansion can thus be experimentally examined and mechanistic models can be validated. To the authors’ knowledge there is, alas, no prior research on this topic.

In this work, an experimental test-rig is presented to investigate flashing under changing volume of the flashing chamber during expansion. A possible modelling approach based on the homogeneous relaxation model (HRM) with the extension of flow regime maps is proposed. Finally, an approximate design model is presented and the results are discussed.

1.1. Two-phase flashing fundamentals

The two-phase expansion process consists of a non-equilibrium flashing into a liquid-vapour mixture. This system is in a non-equilibrium state because of the difference in temperature and velocity between the two phases [3, 6]. The vapour fraction can be considered to enter as saturated vapour and during expansion, the vapour becomes partly subcooled before being condensed. The liquid fraction on the other hand partly evaporates during expansion. The rate of evaporation changes according to the number of nuclei during the boiling and the superheat of the liquid. However, the behaviour is more complex as one can expect an interaction between the gaseous and the liquid state.
because of the good thermal contact between the two-phases [4]. This interaction ultimately results in a net vapour
generation rate during flashing that can be expressed by [7]:

$$m_{ev} = \frac{A_l \dot{q}_l}{h_{lg}}$$  \hspace{1cm} (1)

With $A_l$ the liquid-vapour interfacial area, $\dot{q}_l$ the net heat flux to the interface and $h_{lg}$ the latent heat of
evaporation. Flash evaporation has mostly been studied for constant volume chambers (desalination, refrigeration,
chemical engineering). This type of flashing is typically called static flash. The characterizing results are the vapour
generation rate and the flashing time. The important external parameters are the initial liquid level and the
depressurization rate [8]. Phenomenological correlations have been developed to predict $m_{ev}$ [9-11]. These
 correlations take the initial conditions of liquid superheat, depressurization rate and liquid height as input
parameters. Fitting coefficients are then matched to the experimental datasets.

Besides static flashing, there is also circulatory flashing. In this instance the liquid film has a nonzero initial
velocity at the start of flashing. An important application is found in multi-stage flash (MSF) distillation plants [12,
13]. Again, phenomenological correlations are used to predict the mass of liquid evaporated. Junjie et al. [14]
considers the concept of an overall heat transfer coefficient to model the heat transfer between the liquid and the
vapour phase. This heat transfer coefficient is a time dependent function.

Finally, there are systems where the volume of the flashing chamber changes during time. Obvious examples are
two-phase volumetric expanders. The volumetric displacement is described by the equations of motion. The change
in volume will influence the pressure in the system and affect the dynamics of the flashing. Kanno et al. [7] is one of
the few authors who investigate flashing in a variable volume chamber. An experimental setup mimicking a two-
phase reciprocating expander was built. Instead of working with an actual high pressure inlet, the flashing is induced
under vacuum pressure by the work of a linear actuator. Their developed model was greatly simplified by lumping
the non-equilibrium effects in the boiling phenomena with an agitation factor $\beta$. The factor $\beta$ essentially replaces
the parameter $A_l$ in eq. 1. This agitation factor is fitted to experimental data to predict the pressure in the system. This
gives good results but this tells us nothing about the physical behaviour at the interfacial area and the actual
distribution of the vapour and liquid phase.

Note that in all the above cases, the modelling is based on phenomenological correlations which neglect the
specific structure of the flow during flashing. Saury and Siroux [11] state it would be interesting to analyse
fundamental aspects of the bubble formation in static flashing in order to predict in a mechanistic way the
evaporated mass. For circulatory flashing, the modelling approach by Saha et al. [15] is analogous to the current
state of the art in solving two-phase flows in heat exchangers with the help of flow maps [16]. Yet, limited experimental
research is available on the void fraction profile and the flow maps during flashing [17]. For variable
volume flashing, there is absolutely no flow regime dependent data available according to the authors’ knowledge.
Optical access to the piston would provide possibilities to construct these flow regime dependent correlations.

1.2. Optical accessible piston designs

The idea of an optical accessible expansion chamber is taken from research on internal combustion engines
(ICE). The cylinder of a working ICE is adapted with a transparent liner and a transparent piston crown. As such,
optical access is possible on two perpendicular planes. One of the first implementations of such a research engine
was presented by Bowditch [18]. In his concept, the piston was elongated so that there was place for a mirror to
visualize the combustion chamber through the piston crown. Key in these test-rigs is the choice of materials. The set-
up of Bowditch [18] used quartz liners and a quart piston crown that was bonded to the lower metallic piston with a
high-temperature epoxy. Richman and Reynolds [19] developed an optical accessible engine with a sapphire
cylinder lining. The single-crystal sapphire was custom made. A 0.5 mm clearance was provided between piston and
cylinder and the piston was centred by alignment bushings. The sealing between piston and cylinder lining was
made with carbon filled Teflon rings. A main challenge is the fouling of the windows by the oil and the combustion
products. Hard graphite rings between piston and cylinder are successful in cleaning the window lining but also
provide low friction sealing [20]. Fouling of the windows is however less likely in a piston expander. Also the
pressures and temperatures in the system are distinctly lower than in ICEs.
2. Description of the experimental test-rig

The experimental setup should be able to give insight in both the injection process (i.e. intake) and the expansion process. As such, flexibility in both the injection timing and the volumetric expansion profile is necessary. The layout of the experimental system is portrayed in Figure 1. The main component is the optical accessible piston cylinder. The displacement of the plunger is controlled by an electromechanical linear actuator. The injection is controlled by a fast acting solenoid valve. The component technologies are selected to have an optimal match. However there are still important technical constraints which should be taken into account. Solenoid valves are for example constrained by their time lag. In order to reduce throttling losses a large orifice size is preferred while in order to decrease pre-expansion losses a small enough response time is needed. A faster opening and successive closing will also lead to larger volumetric expansion ratios which are beneficial for wet expansion. However the response time of solenoid valves increases with orifice size. Baek et al. [5] used solenoid valves in a high pressure CO₂ expander. For their orifice size of 1.19mm diameter they report opening and closing times without load of 10 ms and 50 ms respectively.

![Fig. 1. Layout of the Optical-Access Piston Expansion Chamber (OPEC) test-rig.](image)

<table>
<thead>
<tr>
<th>Table 1. Nominal design of the OPEC test-rig.</th>
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<tbody>
<tr>
<td><strong>Design parameter</strong></td>
</tr>
<tr>
<td>Piston bore</td>
</tr>
<tr>
<td>Piston maximum stroke</td>
</tr>
<tr>
<td>Minimum valve response time</td>
</tr>
<tr>
<td>Valve orifice</td>
</tr>
<tr>
<td>Maximum force on piston</td>
</tr>
<tr>
<td>Material piston</td>
</tr>
<tr>
<td>Material cylinder</td>
</tr>
</tbody>
</table>

The nominal design parameters are given in Table 1. In this work, a solenoid inlet valve is chosen which limits the possible sizes as discussed in the previous paragraph. The stroke of the piston is constrained by the external load (max. 400mm). Furthermore, the piston stroke to bore ratio should preferably be low to reduce friction and heat loss.
Note that this is a free piston concept; the actual stroke length will be determined by the operating conditions (expander load, expander inlet conditions and valve timing). Finally there is the bore of the piston. For a given inlet valve, the diameter of the piston will determine the pre-expansion losses. An optimal (i.e. small enough) diameter reduces these losses. The piston should however be large enough to accommodate all the measuring equipment. Therefore a compromise has been made with a diameter of 30mm.

The new Optical-Access Piston Expansion Chamber (OPEC) test-rig will be built at the Applied Thermodynamics & Heat Transfer (ATHT) lab. The setup will be based on the one of Kanno et al. [7]. However, there are several key adaptations to reflect real operational regimes. In contrast, the new test rig is designed to work under high temperatures (<160 °C) and pressures (<15 bar). Furthermore the design allows working with established ORC fluids.

2.1. Model reduction

The processing of the data will be done using an adapted version of the homogeneous relaxation model (HRM) [21]. The HRM provides a mathematical straightforward alternative to model two-phase flows. In this framework one assumes that the velocity, pressure and temperature are given for one averaged pseudo-fluid. For this fluid, the mass, energy and momentum balance is solved. The non-equilibrium effects are modelled with the help of one additional constitutive equation. This relaxation equation effectively models the vapor generation rate during flashing and can be written as:

\[ \frac{\partial \rho}{\partial t} \frac{\partial x}{\partial t} = - \frac{x-x_{eq}}{\theta}. \]  

In this equation x is the vapor fraction of the fluid and \( \theta \) is the relaxation time. The relaxation time is derived from empirical correlations and typically includes terms related to the instantaneous pressure and vapor fraction. The equations in this system are thus highly coupled. Simple phenomenological correlations are available in literature which roughly predicts circulatory flashing for water [14] and \( \text{CO}_2 \) [22]. Instead of working with a single phenomenological correlation to determine the relaxation time \( \theta \), the idea is to work with flow-regime-dependent correlations to improve accuracy. The flow regimes are categorized based on the flow characterization from optical observations.

3. Sizing model and results

In order to size and assess the technical feasibility of the test-rig, a thermodynamic model of the expansion process is made. The HRM model is implemented but with the assumption of instantaneous flashing. This model will thus give an over assumption of the pressures in the cylinder. Considering that this is a design model and that there is no previous research available to accurately implement the non-equilibrium effects this is deemed acceptable.

3.1. Model equations and assumptions

The dynamics of the piston motion is described by Newton’s second law:

\[ m_p \left( \frac{d^2 x}{dt^2} \right) = \frac{\pi D^2}{4} (p_{cyl} - p_{BP}) - F_r - F_{load}. \]  

with \( p_{cyl} \) the instantaneous pressure inside the cylinder and \( p_{BP} \) the pressure at the outside of the piston which is equal to atmospheric pressure. \( F_r \) is the friction force between piston and cylinder wall and \( F_{load} \) the external load from the electromechanical linear actuator. The expansion process is modelled as a sequence of small equilibrium steps: an isentropic expansion, isochoric heating by friction and heat transfer from or to the expander walls. The heat transfer from the walls is expressed as:

\[ Q_{cyl} = h_{wall} \cdot A_{wall} \cdot (T_{cyl} - T_{wall}). \]  

For the cylinder surface temperature \( T_{wall} \), the average between inlet and outlet conditions is taken as:

\[ T_{wall} = \frac{T_{in} - T_{out}}{2}. \]
For this type of application, the Woschni [23] heat transfer correlation is chosen as suggested by Gusev et al. [24]:

$$h = 3.26D^{-0.2}p_{cyl}^{0.8}T_{cyl}^{-0.55}\left(\frac{dx}{dt}\right)^{0.8}. \quad (6)$$

The friction force $F_{fr}$ is modelled assuming the behaviour of pneumatic cylinders. As described by Tran and Yanada [25] an adapted LuGre model is implemented. The working fluid enters the expander through an orifice in the solenoid valve. The mass flow rate through this valve is modelled by assuming steady-state, incompressible flow:

$$\dot{m} = C.A_{or.}\sqrt{2\rho_{in}(p_{in} - p_{cyl})} \quad (7)$$

The flow coefficient C of 0.430 and the orifice diameter $A_{or}$ of 4.67 mm is taken from the data sheet of the manufacturer [26]. At this point we should not that flashing in the valve could have an important effect on the mass flow rate entering the expander. The effect of this is again difficult to predict due to the limited experimental results. The current model will likely over predict the mass flow rate entering the expander. This will be safe for sizing the expander load but will not necessarily predict the physical reality. Therefore we will also do tests where we first slowly fill a pressurized expansion chamber and after the filling start the expansion process.

### 3.2. Simulation results

In this section, the simulations results are discussed according to the boundary conditions of Table 2. Instead of working with water, which is typically used when investigating flashing, R245fa is used. This gives a lower volume increase over the expander and thus large under-expansion losses are avoided. In addition, no vacuum pressure is needed in the condenser to optimize the power output.

<table>
<thead>
<tr>
<th>Description</th>
<th>Value</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>Working fluid</td>
<td>R245fa</td>
<td></td>
</tr>
<tr>
<td>Inlet temperature</td>
<td>100</td>
<td>°C</td>
</tr>
<tr>
<td>Inlet vapour quality</td>
<td>1</td>
<td>-</td>
</tr>
<tr>
<td>Inlet pressure</td>
<td>12.64</td>
<td>bar</td>
</tr>
<tr>
<td>Clearance volume</td>
<td>5</td>
<td>% of the total displacement volume</td>
</tr>
<tr>
<td>Load profile</td>
<td>50. $V^2$</td>
<td>N</td>
</tr>
<tr>
<td>Piston weight</td>
<td>2</td>
<td>kg</td>
</tr>
<tr>
<td>Valve response (open and close)</td>
<td>20</td>
<td>ms</td>
</tr>
</tbody>
</table>

The results of the simulation are given in Figure 2. Note that the electromechanical actuator pushes out the working fluid, therefore it is assumed that at the start of the intake process the clearance volume has a pressure equal to the inlet conditions. The complete expansion takes about 0.40 seconds, during this time, the working fluid volume increases with a ratio of 8.93. The averaged power output from a single stroke is 229 W.

Due to the relative slow closing and opening of the solenoid valve and the small orifice, there is already a pressure reduction in the cylinder during filling. This is clearly visible in Figure 2b. The piston stroke to bore ratio should preferably be low to reduce friction and heat losses. A fast intake process, resulting in a low intake mass, is thus beneficial considering the high specific volume increase during expansion. Going to a larger orifice thus necessitates a faster valve to reduce the inlet mass. However, a larger orifice and a faster response time lead to a larger electrical coil for actuating the valve. Unfortunately, this type of equipment is not commercially available.

Figure 2c and 2d show respectively the velocity and the acceleration of the piston. Important for the design, is the confirmation that both the maximum velocity and the maximum acceleration fall in the operating range of an electromechanical linear actuator.
In this work, the challenges of modelling wet-expansion in piston-expanders are discussed. This topic is almost not investigated in scientific literature but there are analogous processes. The concepts in static and circulatory flashing can be used to model the phenomena seen during the expansion process. The current models however provide limited information because fundamental aspects of the heat transfer at the liquid-vapour interface are neglected. A homogeneous relaxation (HRM) model with the extension of flow regime maps could provide an alternative and more accurate modelling approach. To acquire the necessary experimental data, optical access to the piston expander is however crucial. Previous research on internal combustion engines has shown that these concepts are feasible. Based on this information, an initial design of an optical-access piston expander test-rig was made. The results of a simplified HRM model (assuming instantaneous evaporation) confirm the feasibility. The average predicted power output for a single expansion stroke is 229 W with as working fluid R245fa. It is however important to note the need for fast acting inlet valves. Future research will focus on the dissemination of the experimental results. Also the effect of different working fluids and the flashing during the intake process (i.e. flashing liquid jet) deserves further attention.

Acknowledgements

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References


