Two-Phase Expander Model for Energy Recovery Applications

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Abstract: The present study considers a new approach in improving the cost-benefit analysis of waste energy recovery systems by using two-phase expanders. Two-phase expanders have the potential to increase the overall conversion efficiency of organic Rankine cycles (by limiting superheat generation requirement during off-design conditions) and liquefied natural gas plants (by offering an alternative approach to the Joule-Thompson valve). A semi-empirical vapour expander model was adapted for a two-phase piston expander by considering: inlet pressure drop, heat transfer losses (Shah correlation), condensation during expansion and friction losses (Chenn-Flynn). Engine exhaust heat recovery using the organic Rankine cycle was considered. At the design point, the superheated expander produced 9 kW of net power, amounting to 5.7% of engine crankshaft power. The expander was examined with a reducing inlet vapour quality at the typical and the extreme engine speed-load conditions. The two-phase expander at the extreme off-design condition offered a 16% increase in the net power output compared to the reference superheated state. By selecting the suitable two-phase and superheated expander inlet conditions, efficiency was maintained between 58-86% over a wide operating range, hence, improving the cost-benefit analysis of heat recovery systems.

Keywords: Waste heat recovery; Piston expander, Two-phase, Heat transfer, Friction,

Nomenclature

$2p$	Two-phase
A_{in}	Inlet cross section [m^2]
BVR	Built-in Volume Ratio
$FMEP$	Friction Mean Effective Pressure [bar]
N	Speed [rpm]
ORC	Organic Rankine Cycle
p^*	Reduced Pressure
P_{int}	Net Power [W]
P_{heat}	Heat Loss [W]
P_{leak}	Leakage Loss [W]
$P_{fricCF,2p}$	Adapted Chenn-Flynn Friction Loss [W]
T_{wall}	Expander wall temperature [°C]
V_{sw}	Displacement Volume [m^3]

1. INTRODUCTION

The emerging need for combustion engine efficiency improvements has been motivated by intensifying greenhouse gases, fuel consumption regulations and diminishing fossil fuel supplies. Research has been oriented towards three main strategies: alternative fuels, powertrain efficiency enhancements and new engine architectures [1]. Heat-to-power conversion is considered as a vital feature in improving the overall powertrain efficiency and reducing emissions. Organic Rankine Cycles (ORC) are being adopted as a potential solution for converting exhaust heat into useful mechanical or electrical power [2].

The key component in the ORC is the vapour expander, as it is solely responsible for the useful work generation. Two-phase expanders can further improve the cost-benefit analysis for such heat recovery systems by reducing the challenges relating to superheat generation during transient off-design conditions [3]. Other applications in which two-phase expanders are being investigated include: liquefied natural gas plants, by substituting the inefficient Joule-Thompson valve [4]; and the trilateral flash cycle, by increasing the overall energy conversion in low grade heat recovery [5].
This work presents adaptations in the vapour expander model, to consider operation under liquid-vapour state, for assessing the potential for exhaust heat recovery. The semi-empirical base model, considered as the starting point of this investigation, was originally developed for single phase working fluid by Lemort et al. using scroll expander [6]. Recently, Giuffrida has extended this model by improving the mechanical losses for single-screw expander [7].
Firstly, the sub-model adaptations in the inlet valve pressure drop, heat transfer losses and friction losses for a piston expander are presented. Secondly, using water and 1-propanol fluid blend, the design point expander power is obtained in co-simulation with Aspen HYSYS© v10. Finally, two distinctive heat recovery cases are considered: two-phase operation near the design point condition and two-phase operation under the extreme off-design condition.

2. EXPANDER MATHEMATICAL MODEL

This section presents the two-phase expander model, which considers: inlet pressure losses, heat transfer losses through the casing, condensation during expansion, leakage mass flow rate for off-design and piston friction losses. The mathematical modelling chosen methodology was the lumped parameter approach [8], where the physical expander was discretized into nodes using valid assumptions. Hence, the complexity in incorporating physical properties for a 3-D geometry is avoided to obtain reasonably accurate results for a feasibility study.
The key modelling assumptions were: inlet valve was considered as a throttling valve (isenthalpic transformation); pressure drop at the outlet was neglected; leakages were adiabatic; additives to the working fluid were ignored; and expander envelope was isothermal. Figure 1 illustrates the expander sub-models considered, as following: (0-1) pressure drop through the supply valve; (1-2) heat transfer lost with the expander envelope during inlet; (2-3) isentropic expansion; (3-4) isochoric expansion; (4-5) reunion of nominal mass flow rate $\dot{m}_{in}$ with leakages $\dot{m}_{leak}$; and (5-6) heat transfer loss with expander envelope during outlet. Here, P_{int} is the net useful work produced by the expander taking only the mass flow subjected to expansion (2-4); while accounting for, P_{fric} and P_{leak}, i.e. the friction and leakage losses.

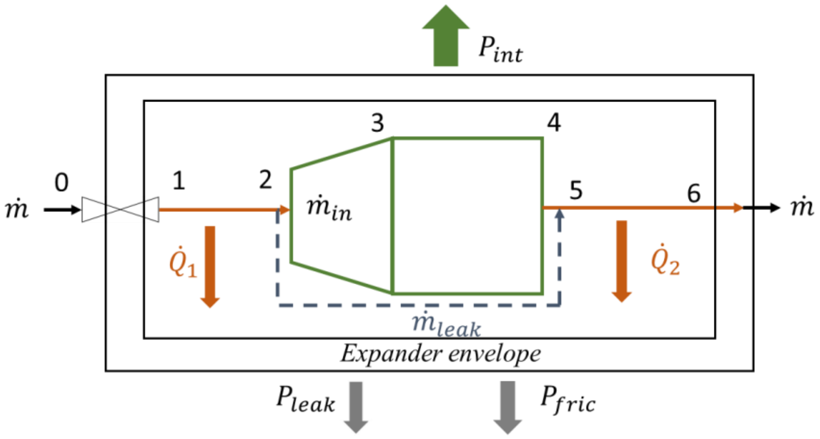

Figure 1 - Expander representation using lumped parameter approach, accounting for pressure, heat, leakage and friction losses

Table 1 presents the processes, descriptions and equations used to characterise the two-phase expander nodes with reference to Figure 1. The sub-model adaptations included: accounting for the increased inlet pressure drop using density proportionality (equation 3); calculating the heat transfer coefficient using Shah correlation (equation 5) [9]; and estimating increased piston friction losses using viscosity proportionality in the Chenn-Flynn equation (equation 11) [10]. The expander mathematical model was co-simulated with Aspen HYSYS, in order to calculate at every node, the fluid properties such as density, specific enthalpy, specific heat at constant pressure, thermal conductivity, viscosity and entropy.

Table 1 - Processes, descriptions and equations utilized in modelling the two-phase expander. Items in bold indicate the sub-model adaptations considered in this work. Refer to the nomenclature for the less common variables

Process	Description	Equation	
	Generic Two-Phase Property	$\varepsilon_{2p} = x\varepsilon_v + (1-x)\varepsilon_l$	*(1)*
Suction 0-1	Isentropic Nozzle	$h_{1,is} = h_0 - 0.5\left(\frac{\dot{m}}{\rho A_{in}}\right)^2$	*(2)*
	Pressure Drop Inlet	$\Delta p_{inlet} = \frac{\rho_{2p}}{\rho_v}(p_0 - p_1)$	*(3)*
Heat Exchange 1-2 / 5-6	Heat Loss	$Q = h_c A(T_1 - T_{wall})$	*(4)*
	Shah Correlation	$h_c = \alpha_f\left[(1-x)^{0.8} + \frac{3.8x^{0.76}(1-x)^{0.04}}{p^{*0.38}}\right]$	*(5)*
	Vapour Quality Variation	$x_{new} = \frac{h_{new} - h_l}{h_v - h_l}$	*(6)*
Isentropic Expansion 2-3	Leakages	$\dot{m}_{leak} = \dot{m} - \frac{\rho_2 N V_{sw}}{BVR}$	*(7)*
	Expansion	$v_3 = v_2 BVR$	*(8)*
Isochoric Expansion 3-4	Enthalpy Variation	$h_{4,2p} = h_{3,2p} - v_{3,2p}(p_3 - p_4)$	*(9)*
Flows Reunion 4-5	Enthalpy Variation	$h_5 = \frac{h_{4,2p}\dot{m}_4 + h_{2,2p}\dot{m}_{leak}}{(\dot{m}_4 + \dot{m}_{leak})}$	*(10)*
	Two-Phase Friction	$P_{fricCF,2p} = \frac{\mu_{2p}}{\mu_v}\frac{FMEP \cdot V_{sw} \cdot N}{2}$	*(11)*
	Net Expander Power	$P_{int} = \dot{m}_{int}(h_2 - h_4) - P_{fric} - P_{heat}$	*(12)*

3. EXHAUST HEAT RECOVERY

This section presents the design point piston expander for exhaust heat recovery in a 10L Diesel engine model. Table 2 summarises the relevant engine, exhaust and heat recovery parameters. Two distinctive engine conditions are presented, a typical speed-load condition, as the expander design point; and a high speed-load condition, as the extreme off-design point. To estimate the off-design heat recovery by the exhaust heat exchanger, the overall heat transfer coefficient multiplied by the heat transfer area was kept fixed.

Table 2 - Diesel engine, exhaust gases and heat recovery parameters (design point / extreme off-design point)

Engine		Exhaust		Heat recovery	
$\dot{W}_{crankshaft}$	158 / 316 kW	$T_{exhaust\ max}$	423 / 493 °C	$UA_{EXH\ HEX}$	1000 W/°C
Speed	1440 / 1720 rpm	$\dot{m}_{exhaust}$	0.212 / 0.408 kg/s	$T_{evaporation}$	202 °C
				$T_{condensing}$	115 °C

The expander model required limited input data, which included: boundary conditions (e.g. initial pressure, mass flow rate, desired pressure ratio); expander geometry (e.g. inlet radius, displacement volume, rpm); and fluid properties (i.e. co-simulation with Aspen HYSYS). Following a working fluid selection study [11], the chosen ORC fluid was an azeotripic blend of 27% water and 73% 1-propoanol by mass. Hence, by fixing the expander design point Built-in Volume Ratio (BVR) and utilising the inlet working fluid condition, the expander was optimised by varying the inlet radius and the displacement dimensions. Table 3 summarises the expander design point parameters and the temperature/pressure/state evolution along the different nodes.

The design point piston expander with a BVR of 8:1 was 86% efficient, due to zero leakages and negligible heat transfer losses. The expander produced 9 kW, which amounted to 5.7% of additional engine crankshaft power. The expander was considered geared 1:1 to the engine crankshaft. The auxiliary power consumption by the ORC pump was insignificant, and furthermore, the ORC condensing temperature of 115°C was achieved using the existing engine coolant.

Table 3 - Expander design point parameters and thermo-physical property evolution at the nodes

Design-point expander		Node evolution (*T[°C] / P[bar] / state*)	
Inlet fluid condition	50°C superheat, 0.047 kg/s	0	250 / 23.5 / superheated
Pressure Ratio	10:1	1	250 / 23.42 / superheated
Built-in Volume Ratio	8:1	2	249.9 / 23.41 / superheated
Inlet radius	0.005 m	3	117.2 / 2.4 / superheated
Displacement volume	0.0007 m^3	4	117 / 2.35 / superheated
Expander efficiency	86%	5	117 / 2.35 / superheated
Expander power	9 kW	6	116.9 / 2.34 / superheated

4. OFF-DESIGN EXHAUST HEAT RECOVERY

This section elaborates the vapour and two-phase expansion results under typical and high engine speed-load conditions for the optimised expander from Table 3. Figure 2 presents the results when the design point expander is operated with reducing inlet vapour quality at the typical engine speed-load condition for exhaust heat recovery. The presentation and discussion of results include: ideal power using isentropic expansion; heat transfer losses using Shah correlation; piston friction losses using adapted Chenn-Flynn equation; leakage losses for varying mass flow rate; and inlet loss using adapted valve pressure losses.

At the 50°C superheated design point, the expander generated 9 kW of net power, with zero leakages, negligible inlet pressure losses and negligible heat transfer losses. The reduction in power compared to the ideal expansion was primarily attributed to the 1.2 kW of piston frictional losses. The constants considered in the Chenn-Flynn equation were representative of comparable automotive piston engines.

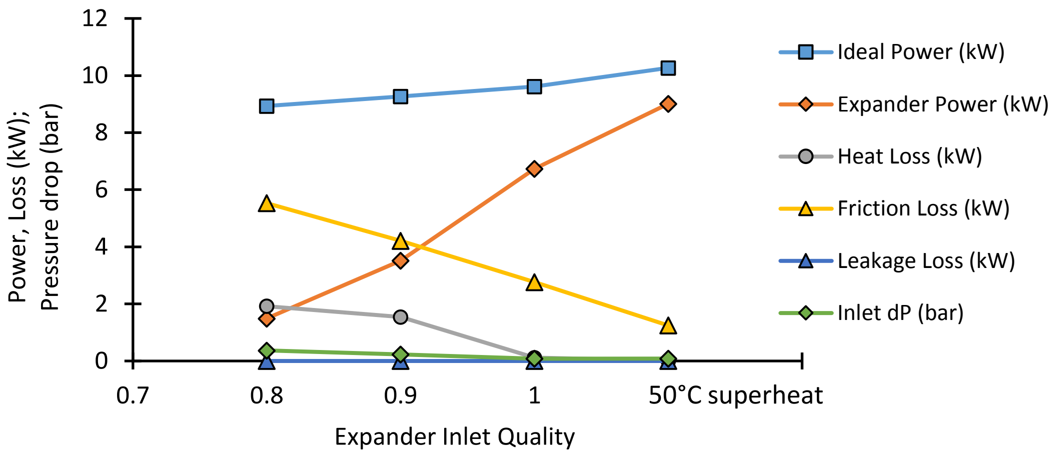

Figure 2 - Expander power and different losses with varying inlet vapour quality at the typical engine condition

However, with the reduction in the vapour quality from x=1 to 0.8, the expander net power decreased significantly from 6.7 to 1.5 kW. This was due to marginal increases in inlet pressure losses (0.1 to 0.4 bar), noticeable increases in heat transfer losses (0.1 to 1.9 kW), and significant increases in piston friction losses (2.8 to 5.5 kW). These higher losses were attributed to increases in key thermo-physical properties of the two-phase state, including, density, thermal conductivity and viscosity.

Figure 3 presents the results when the design point expander is operated with reducing inlet vapour quality at the extreme engine speed-load condition for exhaust heat recovery. This extreme engine condition is characterised by nearly doubling of the exhaust gas flow rate and a 70°C increase in the exhaust gas quality compared to the typical engine condition. As a result, the flow rate of the working fluid exiting the exhaust heat exchanger and entering the expander increased significantly (0.047 to 0.1 kg/s).

At the 50°C superheated state, the expander generated 10.6 kW of net power, a total reduction of 11.1 kW of power compared to the ideal expansion. The increased mass flow rate caused the expander to operate under significant leakage losses, amounting to 9.5 kW. The inlet pressure loss and the friction loss were marginal at 0.4 bar and 1.5 kW.

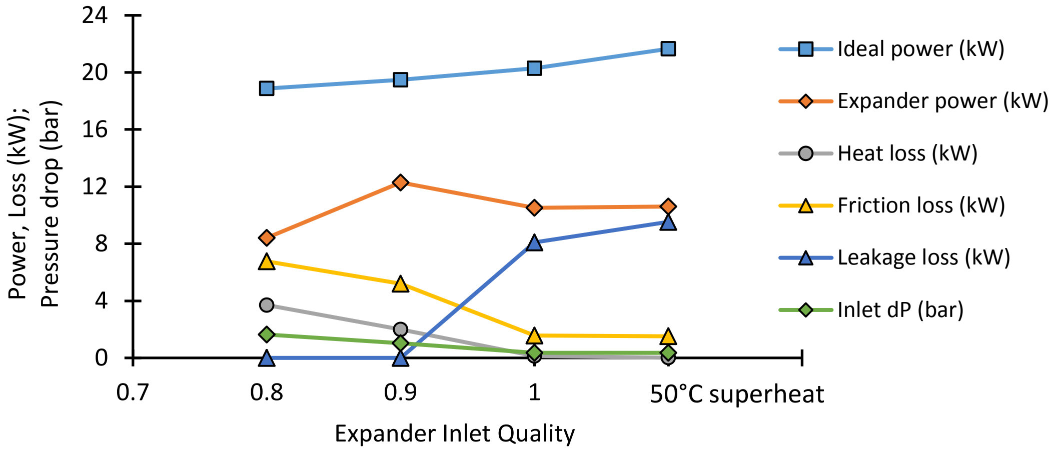

Figure 3 - Expander power and different losses with varying inlet vapour quality at the extreme engine condition

However, with the reduction in the vapour quality from x=1 to 0.8, the expander presented a counter intuitive behaviour. Rather than the reduction in the expander power with two-phase operation as detailed in Figure 2; Figure 3 shows that at x=0.9, a maximum in expander output is achieved. At this condition, the expander generated 12.3 kW of net power by maintaining an acceptable efficiency of 58%. This amounted to a 16% increase in power output from the reference superheated state, hence, improving the cost-benefit analysis of heat recovery systems. The increased expander power is attributed to the sudden nullification of the leakage losses. This was since, the two-phase state density is much higher than the vapour state, and as a result, greater amounts of working fluid were admitted into the expander for power production.

Reducing the vapour fraction further to x=0.8, the expander net power decreased noticeably to 8.4 kW, due to a threefold reason. Firstly, due to the significant increase in the estimated piston friction losses linked to the viscosity proportionality in Chenn-Flynn equation (5.2 to 6.8 kW). Secondly, due to the noticeable increase in the heat transfer losses linked to the increment in the convective heat transfer coefficient (2 to 3.7 kW). Finally, due to a moderate increase in the inlet pressure drop linked to the density proportionality (1 to 1.6 bar). Hence, despite the zero leakage condition, the above three losses remained dominant with reducing vapour quality.

5. CONCLUSION

This work has considered the improvement in cost-benefit analysis of waste energy recovery systems. A new approach using two-phase expanders is presented to increase the overall conversion efficiency in; Organic Rankine Cycles (ORC), by limiting superheat generation requirement during off-design conditions; and liquefied natural gas plants, by offering an alternative approach to the Joule-Thompson valve.

A semi-empirical vapour expander model was adapted for a two-phase piston expander, accounting for increased inlet pressure drop using density proportionality, higher heat transfer coefficient using Shah correlation, condensation during expansion, and increased friction losses using viscosity proportionality in the Chenn-Flynn equation. Two cases were analysed, with the expander being operated with reducing inlet vapour quality at the typical (i.e. design point) and at the extreme engine speed-load condition for exhaust heat recovery. The chosen ORC working fluid was an azeotropic blend of 27% water and 73% 1-propoanol by mass.

The design point 50°C superheated expander with a build-in volume ratio of 8:1 was 86% efficient, producing 9 kW of net power which amounted to 5.7% of additional engine crankshaft power. Reducing the vapour quality near the design point condition decreased the expander power significantly, due to increased valve pressure, heat transfer and friction losses in the two-phase state.

However, reducing the vapour quality at the extreme engine speed-load condition presented a counter intuitive behaviour. Rather than the reduction in the expander power with two-phase operation, at x=0.9 a maximum in expander output was achieved. The expander generated 12.3 kW of net power, maintaining an acceptable efficiency of 58%. The increased expander power was attributed to the sudden nullification of the leakage losses, due to greater amounts of working fluid being admitted into the expander for power production, despite the increasing valve pressure, heat transfer and frictional losses. This net power amounted to a 16% increase in the output compared to the reference superheated state, hence, improving the cost-benefit analysis of heat recovery systems.

6. ACKNOWLEDGEMENT

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7. REFERENCES

[1] Automotive Council and Advanced Propulsion Centre. The Roadmap Report - Towards 2040: a Guide To Automotive Propulsion Technologies. 2018.

[2] Converting Waste Heat into Electrical Energy: BorgWarner’s Organic Rankine Cycle. Auburn Hills: 2018.

[3] Tchanche BF, Lambrinos G, Frangoudakis A, Papadakis G. Low-grade heat conversion into power using organic Rankine cycles - A review of various applications. Renewable and Sustainable Energy Reviews 2011;15:3963–79. doi:10.1016/j.rser.2011.07.024.

[4] Qyyum MA, Qadeer K, Lee S, Lee M. Innovative propane-nitrogen two-phase expander refrigeration cycle for energy-efficient and low-global warming potential LNG production. Applied Thermal Engineering 2018;139:157–65. doi:10.1016/j.applthermaleng.2018.04.105.

[5] Bianchi G, Kennedy S, Zaher O, Tassou SA, Miller J, Jouhara H. Two-phase chamber modeling of a twin-screw expander for Trilateral Flash Cycle applications. Energy Procedia 2017;129:347–54. doi:10.1016/j.ijrefrig.2018.02.001.

[6] Lemort V, Quoilin S, Cuevas C, Lebrun J. Testing and modeling a scroll expander integrated into an Organic Rankine Cycle. Applied Thermal Engineering 2009;29:3094–102. doi:10.1016/j.applthermaleng.2009.04.013.

[7] Giuffrida A. Improving the semi-empirical modelling of a single-screw expander for small organic Rankine cycles. Applied Energy 2017;193:356–68. doi:10.1016/j.apenergy.2017.02.015.

[8] Bernagozzi M, Charmer S, Georgoulas A, Malavasi I, Michè N, Marengo M. Lumped parameter network simulation of a Loop Heat Pipe for energy management systems in full electric vehicles. Applied Thermal Engineering 2018;141:617–29. doi:10.1016/j.applthermaleng.2018.06.013.

[9] Shah MM. An Improved and Extended General Correlation for Heat Transfer During Condensation. Hvac&R Research 2009;15:37–41. doi:10.1080/10789669.2009.10390871.

[10] Chen SK, Flynn PF. Development of a Single Cylinder Compression Ignition Research Engine. National Powerplant and Transportation Meetings, SAE International; 1965. doi:https://doi.org/10.4271/650733.

[11] Panesar AS. An innovative organic Rankine cycle approach for high temperature applications. Energy 2016;115:1436–50. doi:10.1016/j.energy.2016.05.135.