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SCREW EXPANDERS IN ORC APPLICATIONS, REVIEW AND A NEW PERSPECTIVE

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ABSTRACT

Performance of Organic Rankine Cycles is sensitive not only to the entry temperature ratio between heat source and heat sink but also to the temperature degradation of the heat source flow, caused by the heat transfer to the process in pre-heater, evaporator and super heater. In order to adopt the cycle to the great variety of heat sources a multitude of fluids are required. Alternatively fluid mixtures, trans-critical or supercritical fluid conditions can be used to match the process temperatures with the heat source. Screw expanders offer an alternative, new approach to the matching problem of ORC's as they allow for flexible multi-phase expansion. Hereby the vapour fraction at the expander entry can be used to partially match the temperatures of the process to a particular heat source. To provide a perspective on the use of such screw expanders in ORC-systems previous experimental and commercial experience have been reviewed and discussed.

Screw expanders are versatile machines used for the production of mechanical work in power ranges from 3kW to 1.5MW. As the functional characteristics differ significantly from dynamic expanders the explanatory models used to generalise results are different. Plenty of research has resulted in well generalized explanatory models for dynamic expander analysis. For screw expanders similar explanatory models exist mainly in commercially confidential environments. A few public sources disclose test data. In the few cases data has been investigated the analyses tend to rely on thermodynamic models suitable for dry gas expansion. Typically that leads to reasonable replication of test results but seldom to models suitable for detailed understanding of the process. In applications with 2-phase expansion the theories used to simulate functional characteristics is entirely insufficient. The main reason for the scarcity of work in this field is probably the empirical difficulties in obtaining good measuring data in multi-phase conditions.

This paper describes a review of multi-phase screw expander experiences and explains why a unique theory is required to model its characteristics. In the absence of such a unique theory a correlation based in empirical data is presented. This allow for estimations of screw expander efficiency in multi-phase conditions. Measured efficiency with dry expansion, or such efficiency simulated, can be used to estimate adiabatic efficiency with expansion entry vapor fractions ranging from 0 to 1 by using this correlation. Hence estimating expansion efficiency during multi-phase expansion is simplified, allowing for better optimisation of the ORC-systems. This way a new perspective of screw expander potential in ORC system integration can be presented.

1. INTRODUCTION

Historical background, experiences with screw expanders in ORC's, 2-phase expansion and the implications of using 2-phase expansion in ORC's are explained to create a perspective of the future potentials in combining the two.

Helical, rotating piston-type, fluid expansion machines have been around as long as their equivalent fluid compression machines. The ideas for developing such devices constitute natural progressions of the Archimedes Screw, known by the ancients. Heinrich Krigar patented a twin rotor helical liquid pump in 1878, with an alternative use as *presse*. The latter term would be similar to the modern term *blower*, i.e. a positive displacement fan with no internal compression. In the 1920's jet engine designer Alf Lysholm started the development of a series of twin screw compressors for the application of combustion chamber air compression in airborne jet engines and turbo-props, see Figure 1. Lysholm and Boestad (1935) describe a fully industrialized twin screw compressor, sold on five continents only two years later. After that an impressive evolution of designs and applications for twin screw compressor technology emerged. They have revolutionized technical systems architecture in combustion engines, refrigeration and air conditioning plants, industrial gas cleaning and pressurized air systems.

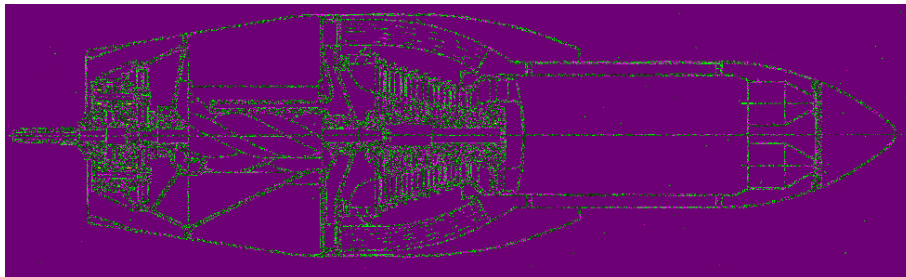


Figure 1. Early integration of twin screw machine in Gas Turbine aircraft engine.

The opportunity of reverse use, i.e. twin screw expanders, was obvious to anyone involved. However markets for supply of shaft power always tended to be smaller than for supply of compressed gases. Therefore commercial volumes, and consequently the variations of designs, are significantly less for screw expanders than its reversely operating sibling. From the same reasons scientific investigation of twin screw expanders is limited and many industrial records have been cleaned out.

One branch of screw expander applications is steam expansion with dry entry. Such applications were common before the advent of affordable, variable speed, electric drives and have seen a revival in modern small scale distributed power generation. Significant knowledge exists in industry and science and that branch will not be specifically addressed in this paper.

With modern developments towards local small scale power generation, as well as widespread availability of thermodynamic computational tools, development of various ORC systems has increased. The use of screw expanders in such applications is still in its infancy and requires more scientific study. Furthermore the screw expanders currently used tend to be limited to designs based on screw compressor design optimizations, leaving room for significant improvements of the machine design itself.

One of the perhaps most interesting possibilities to improve the technology of ORC's is implementing the unique characteristic of allowing efficient expansion of mixtures of gas and liquid from the beginning of the expansion process. Sample effects on ORC system efficiency and specific cost were demonstrated in Öhman and Lundqvist (2014). Using standard practice thermodynamic analysis tools economic optimization of cycle design, and cost allocation, becomes trivial provided that the efficiency of the mixed phase expansion process can be estimated.

Little information on 2-phase expansion efficiency of screw expanders is available. Part of the historic experience of using screw expanders is therefore presented below as well as state-of-the-art on performance prediction and its implications on ORC system optimization.

2. PRIOR FIELD EXPERIENCES WITH ORC'S

Very little of industrial knowledge prior to the digital revolution is traceable without having access to proprietary archives. Also such information is hard to come by as most companies involved has ceased to exist over time and their records scattered. Traces however can be found, often tracked in interviews with elderly engineers and scientists, or in technical periodicals.

One cannot write about twin screw machines without briefly mentioning the late engineering company ALÅ/SRM, 1908-2011. *Aktiebolaget Ljungströms Ångturbin, later renamed Svenska Rotor Maskiner AB*. As described in Timuska (2008) and Svenningsson et al (2010) SRM dominated the systematic development of twin screw machines and by licensing the technology to most suppliers on the market the impact of their development cannot be overestimated. SRM's archives originally contained design calculations and performance data from worldwide screw expander power generation systems from 1932 to 2011. Unfortunately that archive is only partly available and could be expected to be partly shattered. The information in this paper is based on interviews with experienced engineers and available publications.

ORC's using screw expanders was well known in the 1960's. Minto (1967) describe an automotive version of an ORC and Behrendt (1970) cites Minto implementing it in 2 different full scale cars, one of them using a 112mm, A-profile screw expander and R114 as seen in Figure 2.

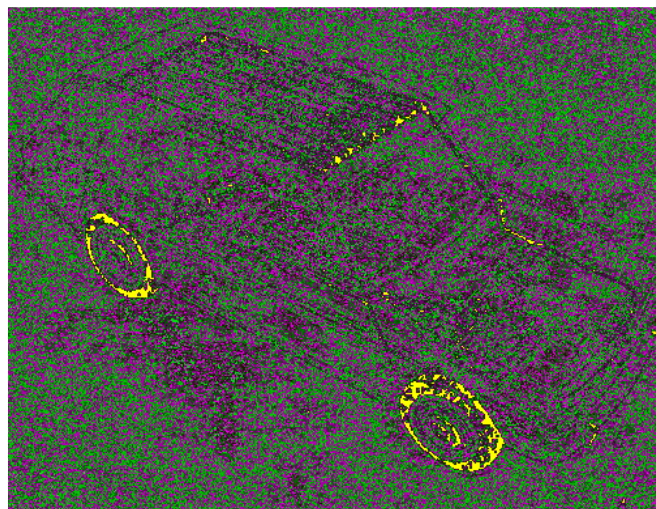


Figure 2. Schematic image of a Datsun station wagon equipped with a Minto ORC drive train.

Before Mintos automotive ORC, Linde created so called MK-systems integrating an ORC-system with an air conditioning system, also using R114. 5 sizes of such machines were developed in the 1960's, using integrated screw expander-screw compressor-liquid pump units as described by Öhman (1997). CIT-ALCATEL sold ORC-units for solar energy pumping stations in the 1970's, developed in the 1960's. They used a 112mm, A-profile, screw expander and R114, see Figure 3, similar to at least one of Mintos ORC-driven prototype vehicles. Thermofrost developed a 10kW waste heat driven ORC with a screw expander, followed by a similar 350kW ORC, with screw expanders reengineered from standard screw compressors of Japanese making. One of the latter was using waste heat from Korsnäs Pulp factory in mid-Sweden. A German supplier built a handful of ORC screw expanders for waste heat recovery purposes in the 1980's and 1990's as well as one 60kW, iso-butane ORC-expander in Biburg, Germany as of 2004.

Rumors of several sizes of field applications with ORC's using screw expanders from Japanese suppliers echoes from 1970's to 1990's though that information has not been verified. In 2003 ITRI installed a field pilot of a 50kW ORC in a waste heat recovery plant in Taiwan. They used a 113mm, semi-hermetic, screw expander using R141b.

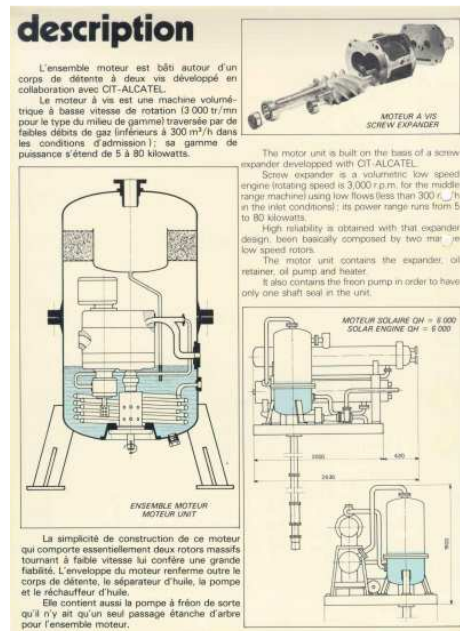


Figure 3. Detail of commercial leaflet on Solar driven ORC. *Mid 1970's, not dated.*

Between 2005 and 2012 a Swedish supplier commissioned 4 waste heat recovery ORC units with 254mm screw expanders; 350kW/R410a, 750kW/NH₃ described in Öhman and Lundqvist (2012), 650kW/R410a and 600kW/R236fa. The latter was installed for jacket cooling water waste heat recovery on a 14MW marine diesel in a commercial freight ship. Allegedly they are still in operation. During the same time period a UK supplier commissioned a significant number of 65kW ORC units using R245fa. Data for those screw expanders are yet unavailable to the authors. An Australian supplier commissioned a 450kW screw expander ORC for waste heat recovery on a gas turbine in 2014. Biederman and Brasz (2014) describe four parallel 1MW/R245fa geothermal ORC expanders in Lightning Dock, New Mexico. Reports are coming in on some recent Chinese pilot installations of screw expander ORC's. This list of experiences is not conclusive but indicates that significant amount of industrial experience is available though scarcely seen in scientific literature.

3. TWO PHASE EXPANDERS

A screw expander consists of two rotating, helical bodies creating “chambers” with certain characteristics between them. Each chamber starts from a volume of zero, is expanded to a maximal volume and thereafter compressed to a volume of zero. The typical cycle time for one chamber process is about 700 degrees rotation of the Male rotor. Figure 4 indicates that during the first part of volume expansion the chamber is in contact with a suction line, feeding fluid into the chamber at almost constant pressure. During the second part of chamber expansion the chamber is isolated, forcing the pressure to decline. After passing the apex of volume the chamber comes into contact with a discharge line, emptying the expanded fluid at almost constant pressure. Work is produced during the expansion by a combination of rotational speed and distribution of fluid forces onto the two rotors. The rotors can be separated by lubrication liquid or external synchronization gear wheels depending on type of design. Rotor combinations can be designed in multiple ways with two, three or more rotors interacting. Twin screw expanders are the most common and will be discussed further under the name *screw expanders*. Triple screw machines, with one Male rotor and two Gate rotors also called

single screw machines, and multiple screw design are mechanically complex and therefore less suitable in 2-phase expansion applications.

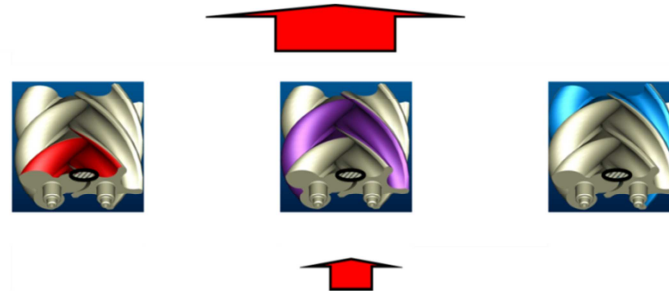


Figure 4. Schematic image of screw expander rotors indicated entry port and chamber volume development.

Screw expander rotors can be designed in multiple manners. The most common is two parallel rotors with constant diameter and constant helix. Olofsson (1989) showed a variant with non-constant helix rotors and non-parallel shaft centers. *Herring bone* arrangements, with minimal axial bearing forces as in Figure 5, were popular in the 1960's but disappeared from the market, most likely due to difficulties of manufacture.

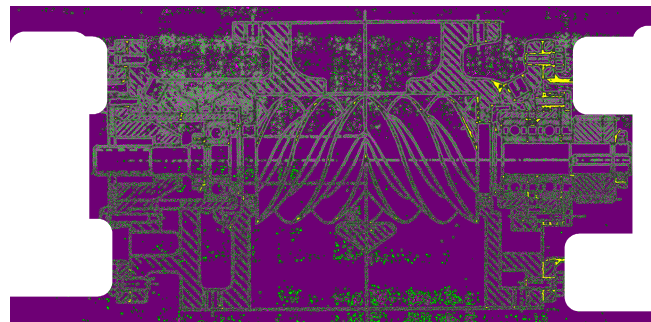


Figure 5. Example of a Herring bone type screw expander from the Japanese market.

Regardless of macro-design parameters, such as diameters, lengths, helix and shaft angles all screw expander rotors need a suitable profile. It is defined by micro-design parameters controlling the seals between chambers, port flow velocities, bending, vibrational stability and substrate erosion. Rotor profiles are typically defined as hypo-cycloids or epi-cycloids alternatively generating mating geometries with a predefined distribution of clearances to allow machining tolerances and material deflections.

During the 1950's and 1960's a series of design rules for screw compressors were issued by some of the manufactures. Also design rules for screw expanders were produced. They all have in common being based on three design handbooks from SRM, popularly called the *Compressor Bibles* and the *Expander Bible*. Such documents are of course currently replaced by computer models. Interestingly enough designs according to the *Expander Bible* are still competitive to most available designs on the market. The *Bibles* were distributed among screw compressor manufacturers but due to their age few can be expected to be still available. As determining the geometrical entities of screw expanders is known this paper focus on the significantly more difficult issue of predicting, and consequently optimizing, functional efficiency.

3.1 Experience from testing of 2-phase expansion in screw expanders

Wagenius (1958) show test data from a 40mm screw expander with a mixture of air and oil. Weiss et al (1975), Steidel et al (1977), Steidel et al (1981), McKay (1982) and Kauder and Kliem (2003) reported on 2-phase expansion with steam. Kliem (2005) tested improved filling solutions with 2-phase steam expansion. Schibbye (1959) reports test data from 3-phase expansion, gas/liquid/solid, of air and humidity. Sprankle (1973), Brasz (2003) and Öhman (2004) discuss the expansion of flashing liquid in screw expanders, also displaying some test data. Merigaux and Pocard (1978), Smith et al (1994) and Öhman and Lundqvist (2013) present data on 2-phase expansion of refrigerants.

The nature of 2-phase screw expander applications is such that pressure level, nominal power, temperatures, fluid and machine tip speed often differs, making comparison difficult. Furthermore the design of each screw expander is unique. Oil injected machines are to be compared to synchronized, sleeve bearing designs to anti friction bearing designs and stiff rotors to weak rotors. Also the built in volume ratio is not always optimal for the test conditions and seldom explicitly stated. Öhman and Lundqvist (2013) explain the logical complexity of simulating 2-phase expansion in a screw machine and yet, to date, there are no physics based models claiming to predict adiabatic efficiency of such expansion with any accuracy. In the same article a method according to Equation (1) is established using information on peak efficiency and efficiency at saturated gas conditions in order to determine adiabatic efficiency as a function of expansion entry vapor fraction.

$$\eta_{ad}(X_{entry}) = \eta_{ad,sat} + \psi_{2phase} \cdot (1 - X_{entry}) \cdot 10 \quad (1)$$

The correlation is defined by Equation (2), as of Öhman and Lundqvist (2013)

$$\psi_{2phase} = \frac{d\eta_{ad}}{dX_{entry}} = -0.15 \cdot \eta_{ad,peak} + 0.09 \quad (2)$$

Note that the sign of the first term is derived from table data and graph in the reference. There is an obvious typographic error regarding the sign in the equation in the reference.

In reality adiabatic efficiency of screw expanders is relatively insensitive to the degree of superheat at expansion entry why Equation (3) is more practical to use. *Note that the correlation only allows us to estimate the effect of variations in adiabatic efficiency. The magnitude needs to be determined by physical tests or simulations using dry expansion entry conditions.*

$$\eta_{ad}(X_{entry}) = \eta_{ad,peak} + \psi_{2phase} \cdot (1 - X_{entry}) \cdot 10 \quad (3)$$

As can be seen in Figure 6 the correlation predicts a linear relation between vapor fraction and adiabatic efficiency. *Seemingly too simple the sources support the conclusion of a linear realtion, see for example Figure 2-6, p. 2-27 in McKay (1982) or Figure 8, p. 1217 in Öhman and Lundqvist (2013).* Hereby cycle simulations can be performed as discussed in Chapter 4.

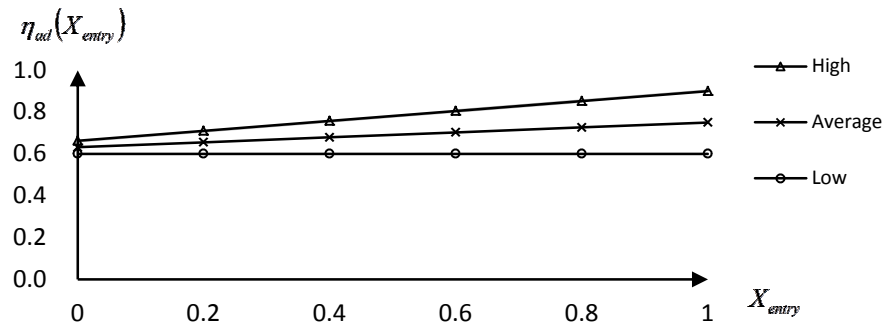


Figure 6. Adiabatic efficiency with 2-phase expansion entry vs. vapor fraction. Peak efficiencies High (90%), Average (75%), Low (60%) as of Öhman and Lundqvist (2014)

3.2 On the difficulty of modeling performance of a 2-phase screw expander

A 2-phase mixture of working fluid needs to be in equilibrium to be stable. Equations of state, used to determine the energy of a fluid, assume that such equilibrium exists, *quasi-static conditions*. The problem with a 2-phase screw expander is that no quasi-static conditions will exist. Therefore any modeling using conventional thermodynamic entities of the fluid will be susceptible to errors by definition.

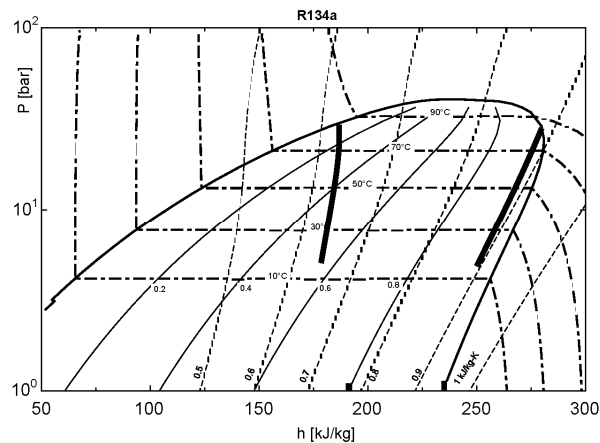


Figure 7. Schematic indication of non-reversible expansion of left side: saturated liquid, right side: saturated gas, R134a

One way to explain the process is to use the simplified Figure 7. The mixture can be schematically separated into; expansion of liquid, left side of the diagram and of gas, right side of the diagram.

Expansion of the pure liquid leads to immediate flashing of liquid, a process well handled in refrigeration condensate valve technology and conventional Trilateral Flash Cycles. The evaporated gas tends to counteract the pressure reduction in the chamber meaning that we need very detailed information on the flashing in order to predict chamber pressure. Flashing also absorbs heat from the liquid, a process also requiring detailed simulation in order to predict the consequential chamber pressure. Expansion of the saturated gas, as in a conventional ORC without superheat, would quasi-statically lead to an immediate condensation of some fluid. The condensation is however not immediate, something well known to engineers designing screw expanders for ice production. Instead the gas can exist in sub-cooled conditions for a remarkably long time. Theories for sub cooled gas behavior are established in steam turbine expansion thermodynamics. However, in screw expanders the two expansion processes co-exist in the same chamber with undefined thermal contact, interacting with each other. This disqualifies any existing theory for dynamic expansion.

Commonly used models, using this logic of Figure 7, assume local equilibrium to simulate fluid states. To the knowledge of the authors no evidence of accuracy suitable to explain details of the expansion process, nor to predict performance for practical use, has been published. Research on flashing of liquid is ongoing, see Polanco et al (2010), but the methodologies are not yet well adapted to flash expansion in volumetric machines.

Physically modeling the expansion seems complicated but not impossible. However, on top of the basic thermodynamics we need to add strong centrifugal effects within the chamber, leakage of hot/cold gas/liquid of non-quasi-static conditions as well as the existence of lubrication oil. Solubility of fluid in oil is a function of pressure and temperature. Further increasing complexity by using fluid mixtures limits the probability designing a theoretical model to predict performance.

With the perspective of the above the approach of Equations (1) and (3) seems as a better short term alternative to predict performance and thereby implementing the unique characteristics of 2-phase expansion in screw expanders.

4. A NEW PERSPECTIVE ON ORC CYCLE DESIGN

In optimizing ORC performance adaption of cycle temperature to that of the heat source has critical impact on the results. Of course a balanced approach is required due to the trade-off between high cycle efficiencies and high cost of heat exchanger area. Saleh et al (2007), Pierobon et al (2014) and many others have developed methodologies to choose an appropriate working fluid for any particular application. Woodland et al (2013) showed that the combination of mixed flow expansion and fluid blends can provide opportunities to improve ORC design. *The downside of choosing the appropriate working fluid/fluid blend for every application is obviously the lack of standardization and the difficulty of moving one ORC unit from one application to another, or shifting its operating conditions.*

Öhman and Lundqvist (2014) showed that when ORC's can be optimized on expansion entry vapor fraction then the sensitivity to choice of working fluid is significantly reduced. Thus an ORC unit, designed for variable vapor fraction, can more easily be adapted to various heat sources and achieve better industrial cost efficiency. In the comparison a marine diesel engine cooling water waste heat recovery application was investigated assuming 3 radically different working fluids, R717/R134a/R245fa. To estimate the effects of component efficiencies three grouped efficiency classes were used, see reference for details.

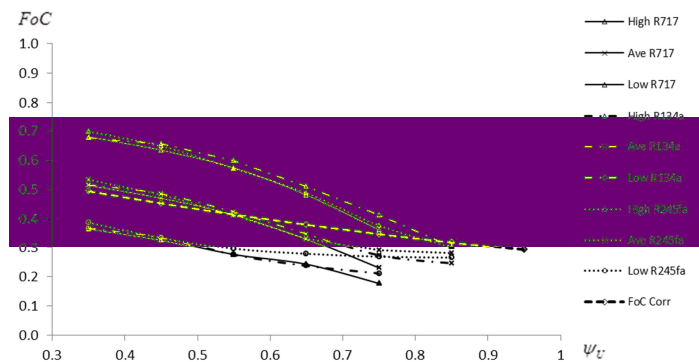


Figure 8. Fraction of Carnot vs. Utilization, both non-dimensional, in a marine diesel waste heat recovery ORC optimized on vapor fraction using R717, R134a and R245fa. *High, Ave and Low indicate grouped component efficiency class levels as of Öhman and Lundqvist (2014). FoC Corr is an estimation of current market product performance for the same conditions, same reference.*

As seen in Figure 8 the optimized thermal efficiency of the three different fluids was found very similar, with concern taken to different component efficiency classes. *If the ORC cycles would have been limited to saturated, or superheated, expansion entry conditions the outcome of the three different working fluids would have been significantly more different.* Another observation from the comparison was that there are specific cost minima with significant effects on plant architecture. Those specific cost minima are strongly dependent on the vapor fraction.

5. DISCUSSION

As far as is known by the Authors very few publications addressing variable vapor fraction ORC system optimization exists. The work in Öhman and Lundqvist (2014) only demonstrates the effects and the subject matter needs further investigations in order to establish limitations of opportunities. However, the work supports the assumption that by using screw expanders to allow variable vapor fraction expansion entry conditions opportunities exist to significantly increase cost efficiency of ORC systems compared to current practice.

An obvious weak point in the theoretical simulations is the correlation of adiabatic expansion efficiency explained in Chapter 3.1. The proposed correlation in Equation (2) is based on a limited

amount of test data with many unknown parameters. Further refinement of that correlation is however time consuming and costly. One could argue that a wiser use of such resources might be to focus more on simulation of the physical processes inside the chamber.

A reflection on the work is that there is an apparent need to rethink our assumptions on what an ORC is and reconsider our approaches to choice of working fluid. The Authors look forward to further improvements in the methodologies for design choice of working fluid in the perspective of the above. Equation (2) means that fundamentally new modeling tools are not immediately required.

6. CONCLUSIONS

This review shows that a multitude of experience of using twin screw expanders in ORC applications exists. However such experience seems to be mostly industrial and not well explained scientifically. The review also shows that multi-phase expansion in screw machines is a well proven technology. Also, in determining expansion efficiency, while having mixed flow at expansion entry, a correlation can be used. A final conclusion is that the above findings allows for a new perspective on optimization of ORC's to its heat sources and significantly affects opportunities to standardize choices of working fluid.

NOMENCLATURE

X_{entry}	Vapor fraction at expansion entry	(-)
η_{ad}	Adiabatic efficiency (isentropic and adiabatic)	(-)
$\eta_{ad,sat}$	Adiabatic efficiency, with saturated gas entry	(-)
$\eta_{ad,peak}$	Adiabatic efficiency, at peak efficiency entry condition	(-)
ψ_{2phase}	Correlation factor due to vapor fraction	(-)
FoC	Fraction of Carnot, ratio of real-to-reversible thermal efficiency	(-)
ψ_U	Utilization of available heat transport from source to sink	(-)

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