SCREW EXPANDER CHARACTERISTICS FOR ORC SYSTEM DESIGN

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ABSTRACT

Dynamic expanders, axial or radial, used to be the common solution in designing ORC's though the use of screw expanders is continuously increasing. In the power range of 5 kW to 1 MW a number of screw expander ORC's are operating in the field. Dynamic expanders and screw expanders show some fundamental differences in characteristics which are important for understanding differences between, and potentials for, the ORC systems in which they are integrated. This paper describes typical characteristics of screw expanders and relate to design optimization opportunities for the ORC.
Screw expander design functions, like multi-phase expansion, side-loading, stall speed and post-compression, are described and the corresponding opportunities for cycle-optimization are presented.
Screw expander performance characteristics, and the conventional methodology for expressing screw expander performance, is explained as well as some related potentials for advanced ORC control strategies. Typical mechanical limitations of the screw expander and the implications on ORC system design freedom is discussed.
Scope is limited to twin screw expanders with synchronized rotors though large parts of the explanations are also relevant for triple screw expanders and screw expanders with direct rotor contact. The major differences related to such technologies are commented.
To explain performance characteristics properly novel efficiency data is presented based on testing of a commercially available twin screw expander.

1 INTRODUCTION

For applications in ORC's we distinguish between three types of screw expanders; synchronized, oil flooded, and oil reduced. Synchronized implicates that a synchronizing mechanism is used dethatched from the expansion process, typically a set of high precision gear wheels controls the relative position of the two helical rotors. Oil flooded means that oil is injected to bearings and expansion chamber to lubricate and re-heat the expansion process. After the expansion the oil is separated from working fluid and pumped/heated to be again injected. Oil reduced means that a small proportion of oil is mixed into the working fluid and circulated through the entire ORC. The circulated oil is sufficient to lubricate rotors, bearings and fluid pump while providing very limited re-heat to the expansion process. This paper mainly discusses synchronized twin screw expanders with cylindrical rotor bodies.
Note that there are screw expanders with three or more rotors as well as with conical/spherical rotor bodies. Their principal behavior follows similar principles but also show specific variations. Also, the expansion process with oil present is quite different from the expansion process without oil present.
The unique feature of screw expanders in ORC's is the large degree of freedom in process architecture without adding large cots/complexity to the drive train. Admission and exit ports can be easily integrated in the expander without changing the expensive basic design, such as rotors, bearings, seals etc. These features can be used to better adapt the ORC process to fit to characteristics of heat source and heat sink. Waste heat applications with multiple heat/cold streams, like ICE's, can be significantly improved by combined cycle processes. To utilize the design freedom of the ORC-

system, performance of the screw expander must be estimated in a wide operating window. One often needs to do such work lacking access to advanced expansion process models. Therefore we need, and find, some general characteristics only in older publications, such as(Schibbye and Wagenius 1953), (Fairchild-Hiller 1957), (Brown 1984) and (Kaneko and Hirayama 1985). Such old data is often difficult to correctly interpret due to gradual changes is terminology and methods. A few newer publications show test data of some basic design screw expander, often tested in a minor part of its operating window, making generalization difficult.
Advanced, well correlated, numerical simulation tools could be used to create performance characteristics. Unfortunately, that also requires access to test data from representative screw expander designs, in very wide operating windows, to limit model artifacts. Advanced performance simulation models therefore provide surprisingly little help as they tend to show the characteristics of the model instead of the characteristics of the expander. (Öhman 2016) provides more background on modelling development of screw expander performance.
Novel performance data, combined with already published information, and a systematic performance representation model is shown below. With the presented method of representing performance characteristics only few simple assumptions are required to have access to a rough model suitable for ORC process analysis trade-offs. Characteristic impact of changing process fluid is also briefly explained.
Comparison with performance characteristics of fundamentally different types of expansion machines is not part of this paper.

2 ORC ARCHITECTUAL FREEDOM

ORC's architectures can be varied infinitely but the cost of tailormade expander drivetrain is often prohibitive to implementing solutions suitable in thermodynamic terms. With screw expanders the very same drive train can often be used while changing only limited parts of housing geometry to add opportunities. *This is well known and utilized with screw compressors but much less so with screw expanders.* Figure 1 shows a sample of porting arrangements in screw expanders and how they could be implemented for ORC design. Porting arrangements are shown in the form of volume curves, technical connections are shown as process schemes and thermodynamics is indicated in generalized state diagrams. V*olume curves follow convention of positive direction for compressors and negative for expanders. Also, for most profiles, $V > 0$ at an angle of zero since the angle of zero is defined by profile design and not by the volume curve.* Such cycle modifications are used to arrange transfer of heat in and out of the process at the most beneficial temperatures. Many waste heat applications provide heat streams at different temperatures and different apparent heat capacities combined. *One example of that is ICE's with high temperature exhaust gas, medium temperature jacket cooling water and low temperature lubrication oil. Another example is cement plants with high temperature flue gas and medium temperature product residual heat.* Current trend like (Carstensen, Horn et al. 2019) is to use separate ORC's for different heat streams and optimize each ORC accordingly. This practice leads to high efficiency but also to excessive cost as many functions in the separate ORC's are needed twice or more. Hybridization of technologies allows for a different approach where adaptation to multiple heat streams can be made in one single heat engine. Complexity appears to become larger but mainly affect intellectual activities, not practical operation of the machine.

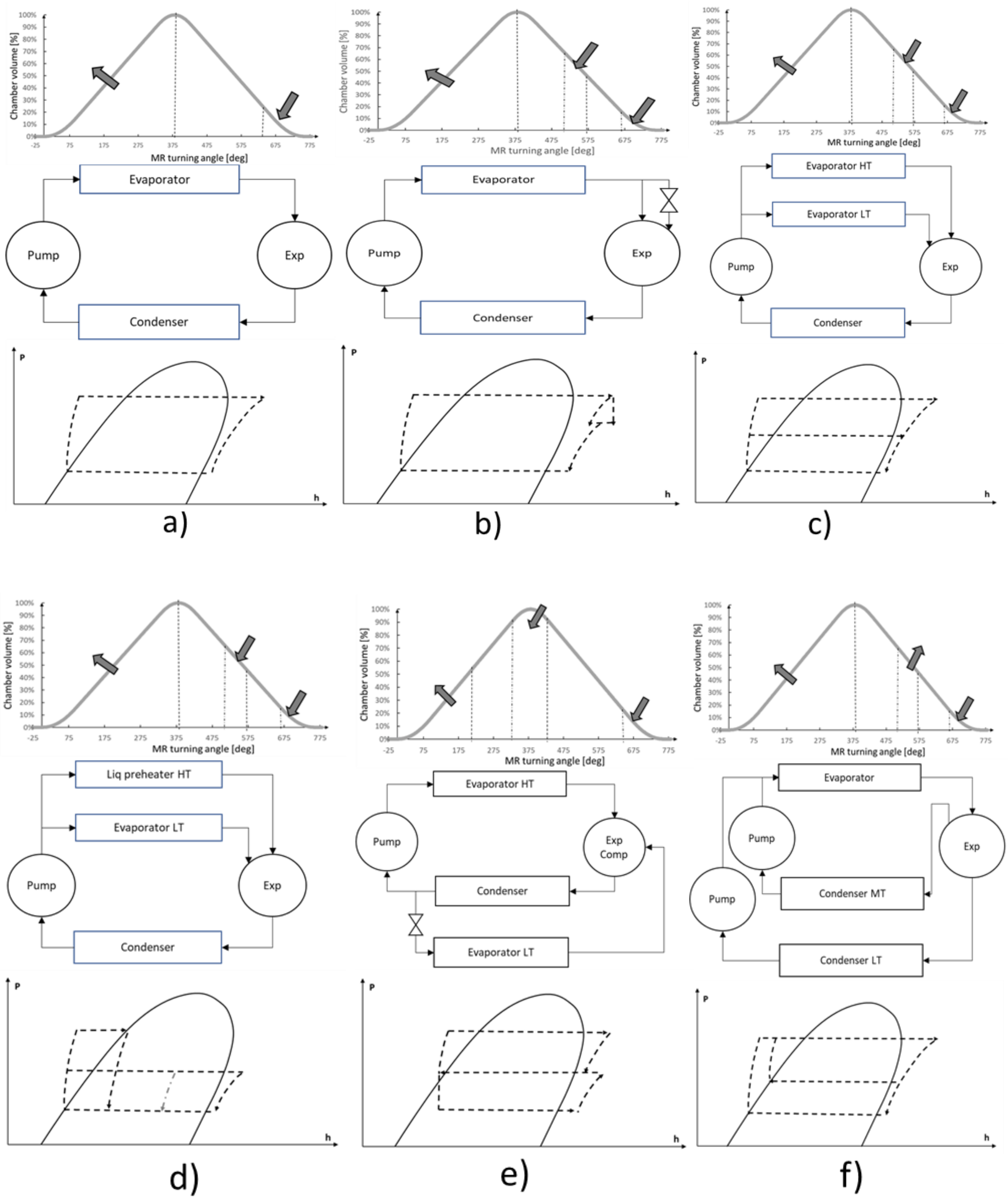

Figure 1: a) b) c) d) e). Examples of port arrangements and ORC system architecture for a screw expander with identical drive train. a) basic ORC (Brasz and Biederman 2014) b) continuous evaporator temperature control (Öhman 2005) c) dual evaporator temp ORC *ex. in ICE's* d) hybrid ORC/TFC *ex. cement plant* (Öhman 2004) e) Cogeneration of power and cooling (Olofsson 1993) f) High temp ORC with warm water take-off

Modelling and analysis of ORC architectures such as exemplified in Figure 1 is well known if performance characteristics of its components is available. Literature provide an abundance of component performance data for heat exchangers, fluids, pumps, piping, valves, generators etc. However, for screw expanders available performance data is often lacking. Without understanding of screw expander characteristics optimization of advanced cycle architectures becomes inaccurate and inconclusive. The below method is specific for synchronized screw expanders though using it for other types is possible albeit with reduced accuracy.

3 PERFORMANCE CHARACTERIZATION

Generalized performance characteristics of synchronized screw expanders, when operating in saturated or superheated fluid conditions, is relatively easy to characterize using systematic data representation. *For applications with wet expansion, 2 or 3-phase, it becomes more challenging due to lack of systematic data. For oil injected and oil reduced screw expanders the challenge is also availability of useful test data.* Designations, geometric definitions and conventions used in this paper are made according to the most common industrial practices (Schibbye and Wagenius 1953) and (SRM 1984). Note that most available performance data for synchronized screw expanders stem from tests with air expansion. This is partly because of cost reasons. Testing with air is significantly easier and less costly than testing with advanced fluids. Partly it is also due to confidentiality as organizations investing heavily in testing with advanced fluids regard the data as precious knowledge. Few organizations can afford to test on multiple fluids meaning that comparisons of performance for the same machine using different fluids is unavailable. Thereby also correlation of advanced simulation tools, such as CFD, for this purpose becomes inaccurate and inconclusive. Data in this paper is only showing performance with air. *Systematic transformation of performance characteristics from air to other fluids deserves a paper of its own. Some general guidelines are however provided below*. It is sufficient to characterize filling factor (flow capacity) and adiabatic efficiency for the needs of ORC process optimization.

Performance data emanates from the specified references as well from novel test results from a commercial screw expander, *Atlas Copco EZ2*. Table 1 provides an overview of geometric data, test ranges and use in this paper for six different screw expanders.

Table 1: Geometric data, and ranges for six sources. Only data from EZ2 is use for characterizing adiabatic efficiency. *C and D is of identical design except different admission port/Vi.*

Screw Expander	**ODM**	**L/D**	**Vdp**	**Vi**	**Pin/Pout**	v_{tip}	**Use**	**#**
EZ2	113	1,6	1,05/Vi	2	1,5–21	10-110	η_{ad}, f	A
(Kaneko and Hirayama 1985)	81,5	1,45	-------	1,98	1,6–2,9	30-91	f	B
(Öhman and Österberg 1999)	100	2	0,98/Vi	2,5	1,5–5,5	5-63	f	C
[Öhman and Österberg 1999)	100	2	0,98/Vi	1,7	2–6	21-85	f	D
(Wagenius 1957 a)	63	1,11	0,128/Vi	2,2	1,7-4,5	30-100	f	E
(Wagenius 1957 b)	63	1,11	0,128/Vi	3,1	3-7,8	67-115	f	F

3.1 Flow capacity and control, filling factor

Viewed from the “ORC perspective” the purpose of an expander is to control evaporator pressure to maximize power output. Evaporator pressure is controlled via admitted flow to the expander. This applies also to intermediate pressure ports and their connected process functions. As seen in Equation 1 admitted flow is dependent on geometry, speed and filling factor. The latter, Equation 2, is independent of pressure ratio and pressure level but very dependent on porting geometry, rotating speed and fluid inertia. *At very low tip speeds, <20 m/s in the synchronized case, f is dependent on pressure ratio, but the impact is small and usually ignored. It is acceptable to ignore this since economic use of an expander leads to running at higher tip speed and since accuracy of f becomes* *worse as tip speed becomes very low. One must also be aware that in the case of intermediate pressure inlet ports there is an apparent risk for under- or overexpansion of the expansion process of the HP fluid. Such effects of course dramatically affect the flow capacity on top of the estimated filling factor.*

$$\dot{V}_{in} = 1000 * \frac{V_{dp}}{V_i} * \frac{N}{60} * f \quad (1)$$

$$f = \dot{V}_{in} * V_i * \frac{60}{V_{dp} * N} \quad (2)$$

$$f = C1 + \frac{C2}{\left(1+\left(\frac{v_{tip}}{C3}\right)^{C4}\right)} \quad (3)$$

Filling factor from the six sources and the new measurements can be seen in Figure 2. Eq. 3 shows the approximation format where C1, C2, C3 and C4 are correlation constants. Results in the different references show similar characteristic but different amplitude. The characteristic of Equation 3 is very practical to use when estimating flow characteristics as guessing is limited to amplitude and tip speed.

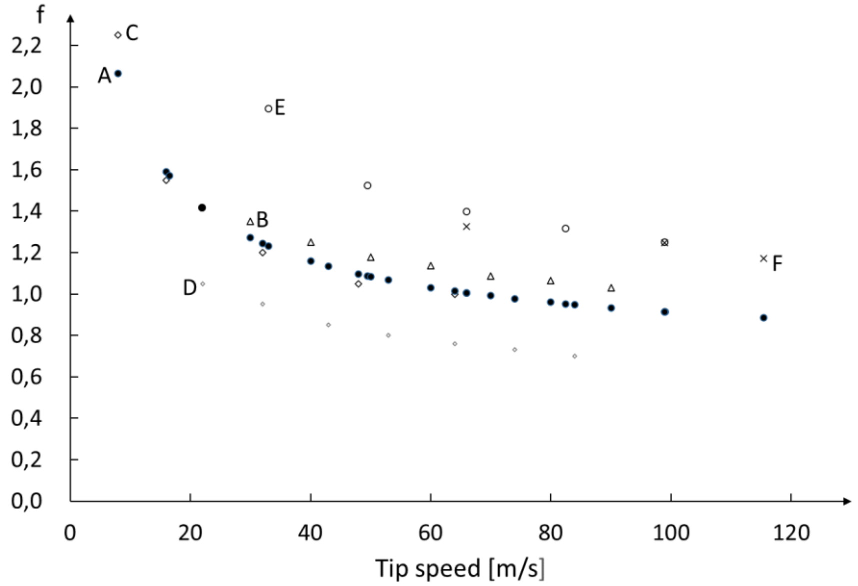

Figure 2: Filling factor vs tip speed for six synchronized screw expanders, see Table 1.

Controlling the flow admitted from an ORC evaporator is a common manner to optimize the temperature in the evaporator. Synchronized screw expanders allow speed regulation from 5 m/s to 180 m/s. Therefore, limitations in generators or power electronics typically define real turn-down ratio in combination with the filling factor. As power electronics is very costly, 100-200 €/kW(electric) fixed speed expanders with variable swept volume (Wagenius 1960), are sometimes cost efficient. A less complicated method is described in (Öhman 2005) and seen in Figure 1 b). From the perspective of flow control, it is irrelevant whether the intermediate port is used as in Figure 1 b), c), d), e) or f). *Filling factor is dependent mainly on molecular weight of the fluid though comparative test data is not available. When simulating a process with a different fluid one must make a dedicated assumption on the effect of molecular weight ratio on Filling factor.*

3.2 Adiabatic efficiency

The term "adiabatic efficiency" is used to distinguish between methods of measurement. Isentropic efficiency measured as temperature difference across an expansion process is common with turboexpanders. This is logic as very little heat transfer between expansion process and ambient occurs. For screw expanders however there is considerable heat transfer between expansion process and ambient. Hence isentropic efficiency measured as temperature difference across a screw expander is named "temperature efficiency" while isentropic efficiency measured as mechanical shaft power is named "adiabatic efficiency". Adiabatic efficiency of a screw expander is therefore always defined according to Equation 4.

$$\eta_{ad} = \frac{\dot{W}}{\dot{m} * \Delta h_{is}} \tag{4}$$

$$\eta_{S-I} = \frac{\left(1 - PR_i^{\left(\frac{1-\kappa}{\kappa}\right)}\right) + (\kappa - 1) * \left(1 - \frac{PR_i^{\frac{1}{\kappa}}}{PR}\right)}{\kappa * \left(1 - PR^{\left(\frac{1-\kappa}{\kappa}\right)}\right)} \tag{5}$$

Where $PR_i = {V_i}^{\kappa}$ according to standard conventions for screw machines. (6)

$$\eta_{ad} = \eta_{ad-peak} * \frac{\left(1 - PR_{dyn}^{\left(\frac{1-\kappa}{\kappa}\right)}\right) + (\kappa - 1) * \left(1 - \frac{PR_{dyn}^{\frac{1}{\kappa}}}{PR}\right)}{\kappa * \left(1 - PR^{\left(\frac{1-\kappa}{\kappa}\right)}\right)} \tag{7}$$

The characterization of adiabatic efficiency in screw expanders traditionally uses Equation 5, the semi-ideal screw expander adiabatic efficiency. Though used extensively in industry (Kaneko and Hirayama 1985) is believed to be the first publishing it. *Note that this representation of efficiency works very well for synchronized screw expanders but two- or three-phase and oil injected screw expanders fit less well. This is caused by a very different set of loss mechanisms for the latter.* With two simple corrections, peak efficiency and "dynamic V_i", Equation 5 closely approximates measured data. V_i is according to convention purely a geometric entity assuming absence of any dynamic effects. In reality a "dynamic V_i" can always be found by identifying $PR_{dyn} = f(PR)$ by observing where $\frac{d\eta_{ad}}{dPR} = 0$ at constant tip speed, as of Equation 6. By replacing PR_i with PR_{dyn} in Equation 5 and multiplying with peak adiabatic efficiency we come to Equation 7 which can be easy tabularized by using input from test data analysis or advanced simulation output. As PR_{dyn} and $\eta_{ad-peak}$ are both dependent on dynamic effects dictating losses, we can reach very close correlation with test data and easily extrapolate the results as shown in Figures 3 and 4. Figure 3 shows an example of characteristics of adiabatic efficiency of the screw expander EZ2 based on test data with air. The method for defining "dynamic Vi" and peak efficiency is indicated for the curve of adiabatic efficiency at 90 m/s tip speed. Once Equation 7 has been estimated it is trivial to use this in the optimization of the ORC-process. Sometimes it is also useful to make 2-dimensional graphs indicating efficiency of the screw expander for the human eye. Figure 4 shows iso-efficiency curves for EZ2 at three different tip speeds.

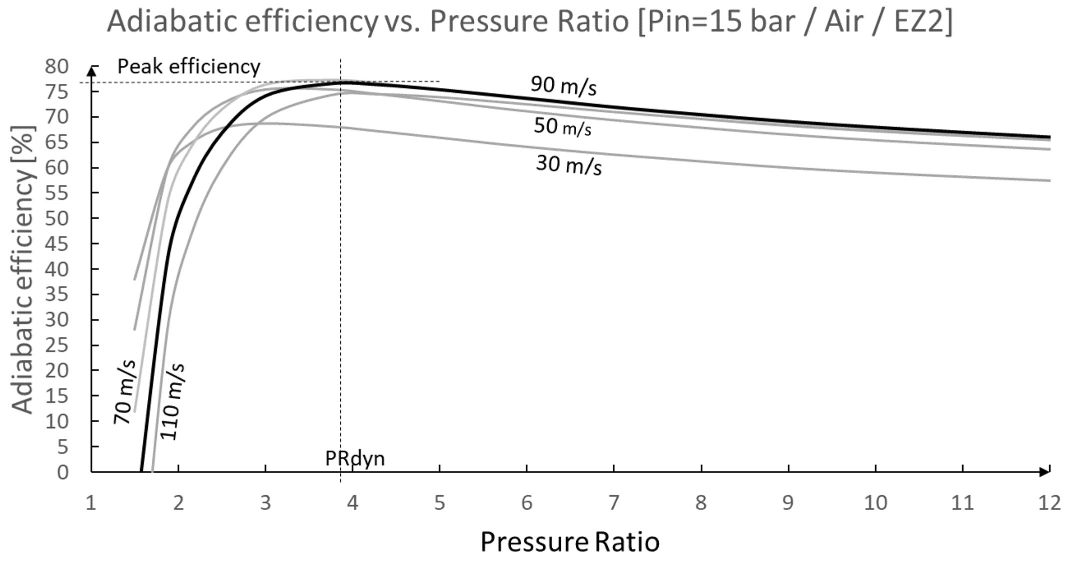

Figure 3: Adiabatic efficiency for screw expander EZ2 at admission pressure of 15 bara and Air. *"PR(Dynamic Vi)" and Peak efficiency indicated for 90 m/s.*

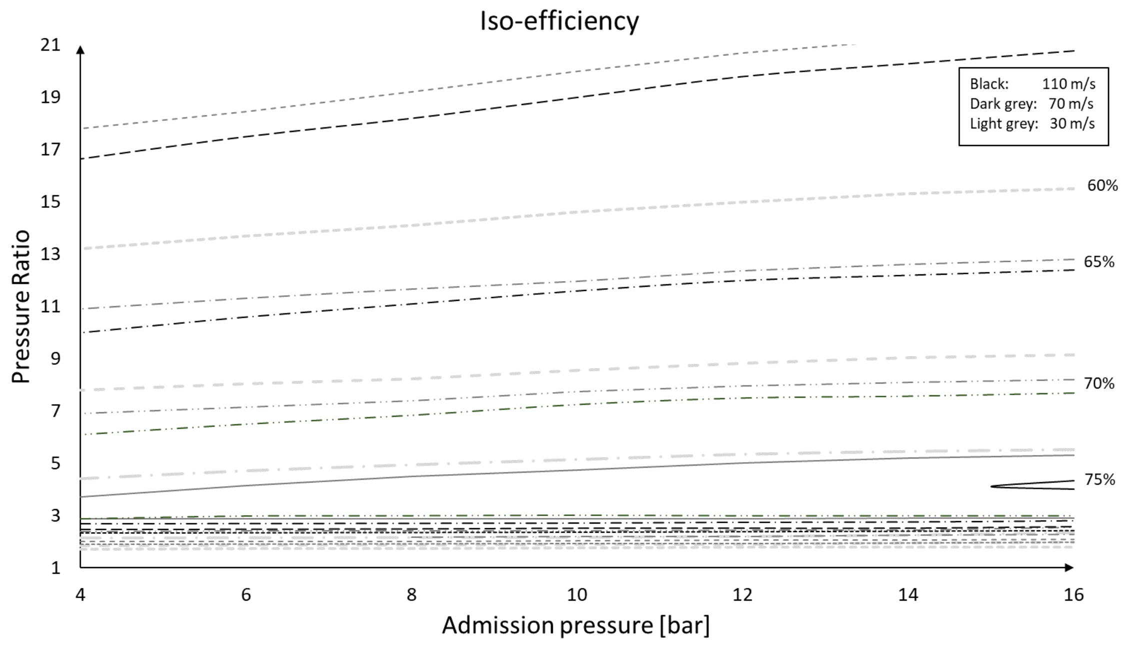

Figure 4: Adiabatic efficiency for screw expander EZ2 with air.

Characteristics of adiabatic efficiency will differ if another fluid is assumed. The natural characteristic relative to pressure ratio, as of Figure 3, is easily transformed by using the corresponding κ in equations 5 – 7. Considering the current lack of published information, the factor $\eta_{ad-peak}$ in equation 7 must be assumed/simulated separately.

3.3 Mechanical limitations and general guidelines on adiabatic efficiency

Typical operating limitations for screw expanders can be found in Table 2. However, any specific product will display more narrow operating windows due to different design choices. Typically, pressure difference is limited by bearing design, inlet pressure by casing design, inlet/outlet temperature by material integrity and torque by rotor design. Speed is either limited by bearing life or unacceptable performance degradation. When integrating screw expanders into ORC's oil separation/circulation, fluid vacuum, fluid side volume and molecular stability add further limitations. General guidelines for adiabatic efficiency are either too narrow to be useful or too detailed to be possible to validate with test data. Therefore Figure 5 is constructed from the general experience of the Author and a set of colleagues in the screw expander industry. It is to be used only for guidance in making simplified screw expander performance characteristics as described in this paper.

Table 2: Approximate mechanical limitations for three different types of twin screw expanders.

Type	Pin(max)	Pout(min)	Tin [°C]	Tout [°C]	ODM	v_{tip}	Allowed phases
Synchronized	< 40	< 1	<1400	>-160	<1000	5-180	Dry/Wet/Solid-mix
Oil Flooded	< 110	>1	<150	>-60	<510	2-45	Superheated
Oil Reduced	< 20	>1	<150	>0	<130	5-70	Dry/Wet-mix

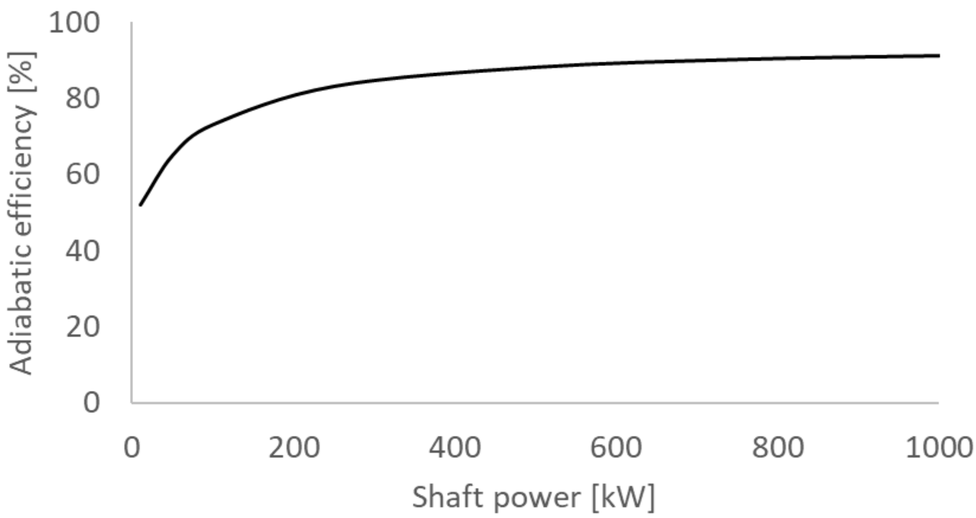

Figure 5: General estimation of Peak adiabatic efficiency vs. shaft power with synchronized screw expander of good quality.

4 CONCLUSIONS

- Examples are provided on how innovative ORC architectures can be supported by the design flexibility offered by screw expanders while limiting drive train cost increase.
- Data from a series of screw expanders have been found to exhibit similar flow capacity characteristics and a suitable equation is presented.
- The traditional equation for characterizing adiabatic efficiency of a screw expander has been implemented and shown for one synchronized screw expander using novel data.
- A general estimation on peak adiabatic efficiency has been presented.
- The characteristic shown can be used to represent expander performance when optimizing ORC process performance.

NOMENCLATURE

ORC	Organic Rankine Cycle	
Tin	admission temperature (at flange)	(°C)
Tout	exit temperature (at flange)	(°C)
TFC	trilateral flash cycle	
ICE	internal combustion engine	
ODM	male rotor outer diameter	(mm)
L/D	male rotor length to diameter ratio	-
Vdp	maximum displacement volume of rotor pair	(l/rev)
Vi	internal volume ratio	-
Pin	admission pressure (at flange)	(bar)
Pout	exit pressure (at flange)	(bar)
v_{tip}	male rotor tip velocity	(m/s)
η_{ad}	adiabatic efficiency (measured as work)	-
f	filling factor	-
$\dot{V}_{in}$	admission volume flow	(m3/s)
N	rotational speed	(rpm)
$\dot{W}$	shaft power	(W)
$\dot{m}$	admission mass flow	(kg/s)
Δh_{is}	isentropic enthalpy difference	(J/kg)
η_{S-I}	semi-ideal screw expander isentropic efficiency	-
κ	Cp/Cv	-
PR_i	internal pressure ratio, f(fluid and Vi)	-
PR	admission/exit pressure ratio	-
PR_{dyn}	“dynamic” internal pressure ratio	-

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