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2014

The Benefit of Variable-Speed Turbine Operation for Low Temperature Thermal Energy Power Recovery

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ABSTRACT

This paper analyzes, given the large variation in turbine discharge pressure with changing ambient temperatures, whether variable-speed radial-inflow turbine operation has a similar benefit for Organic Rankine Cycle (ORC) power recovery systems as variable-speed centrifugal compression has for chiller applications. The benefit of variable-speed centrifugal compression over fixed-speed operation is a reduction in annual electricity consumption of almost 40 %.

Air-conditioning systems are by necessity always designed for the highest possible ambient temperature. This is not necessary for ORC systems. Lower cost ORC systems are obtained when the design point is selected at moderate ambient temperatures. However, these systems show reduced power output at higher ambient temperatures. The more expensive ORC systems designed to achieve full power at higher ambient temperatures will produce constant power independent of ambient temperature but consume more heat and require a control mechanism to prevent overloading the turbine at lower ambient temperatures. The benefit of variable-speed ORC turbine operation over fixed-speed is an increase in annually averaged ORC power output and/or a reduction in annually averaged thermal heat input demand. However, the variable-speed benefit is, as will be explained in the paper, inherently smaller for ORC systems than for centrifugal chillers.

1. INTRODUCTION

Low-temperature thermal energy power recovery through the use of Organic Rankine Cycle (ORC) systems has received considerable interest for green power generation. The low temperature thermal energy could come from renewable energy sources such as geothermal wells or low-cost solar collectors. It could also come from low temperature waste heat sources such as engine coolant fluids and exhaust gases.

The pressure ratio experienced by the ORC turbine/expander is controlled by the temperature of the waste heat stream that sets the boiler saturation temperature (and therefore the turbine inlet pressure) and the ambient temperature that controls the condenser saturation temperature (and therefore the turbine exit pressure). As a result of the variation in ambient temperature the pressure drop over the turbine can vary substantially given the relatively small difference in saturation temperatures between boiler and condenser for low temperature waste heat power recovery systems. Air-cooled ORC systems experience an even larger pressure drop fluctuation than water-cooled systems due to the larger variation in sensible ambient temperature compared to the changes in wet-bulb temperature.

The performance of air-conditioning and refrigeration equipment, which works with even smaller differences in saturation temperature between its evaporator and condenser, is also known to be strongly affected by changes in ambient conditions.

Both the thermal efficiency of an ORC system and the coefficient of performance (COP) of a refrigeration system improve at lower ambient temperatures. Lower ambient temperatures reduce the head the compressor of a refrigeration system has to deliver minimizing the required work input per unit of cooling, thus increasing its COP. Lower ambient temperatures enlarge the head available for the turbine/expander allowing an increase in the power being generated by the turbine for the same thermal input, thus increasing the thermal efficiency of the ORC.

Figure 1 shows side-by-side the schematic of a power consuming vapor compression refrigeration system and a power generating Organic Rankine Cycle system. There are many similarities between these two systems: they both use refrigerants as working fluids and can share heat exchanger components with minor modifications. A change in refrigerant from the medium pressure chiller refrigerant R134a to one of the lower pressure refrigerants R245fa or R1233zd allows the use of centrifugal chiller compressors as ORC radial inflow turbines at similar speed and equal generator/motor size.

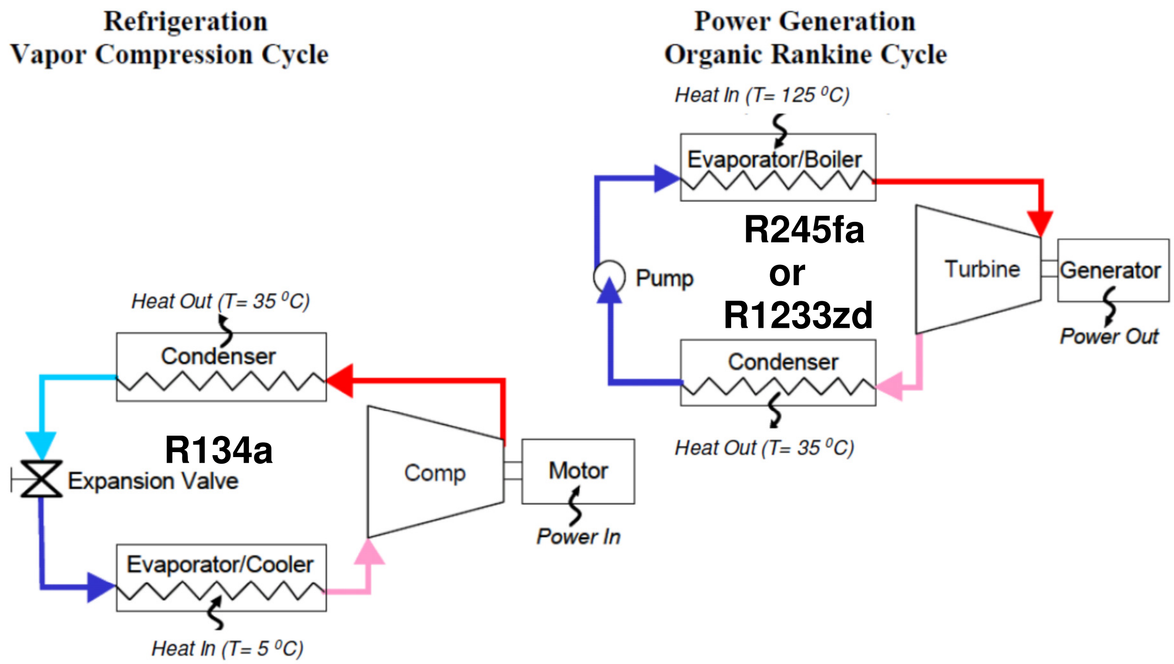

Figure 1: Side-by-side comparison between the refrigeration vapor compression cycle and the power generation organic Rankine cycle

Before the availability of variable frequency drive technologies the electrically driven centrifugal chillers had fixed speed compressor operation. The switch to inverter-driven variable-speed centrifugal compressors has dramatically increased part-load compressor efficiency resulting in a reduction in annual energy consumption of up to 40%.
Most ORC systems currently on the market have fixed-speed turbines designed for a specific design pressure ratio corresponding to a given difference in saturation temperature between the evaporator and the condenser. Given the off-design performance improvement of variable speed operation for centrifugal chillers one might wonder about the benefit of variable speed for off-design performance of ORC systems. The object of this paper is to quantify the off-design performance benefit for ORC systems when switching from fixed-speed turbine operation to variable speed and compare this benefit to the known benefit of variable speed for centrifugal chillers.

2. THE BENEFIT OF VARIABLE-SPEED CENTRIFUGAL COMPRESSOR OPERATION FOR CHILLERS

Air-conditioning systems are by necessity designed for the warmest days of the year when their compressors run continuously delivering maximum flow (cooling capacity) and head (~ the difference in condenser and evaporator saturation temperature). Power consumption peaks under those full-load design conditions. These conditions define the electrical demand charges imposed by the utilities as well as necessary electrical infrastructure and the capacity of the heat rejection equipment.

Annual energy consumption is only indirectly affected by full-load efficiency. The majority of the operating hours require less cooling capacity and lower head. Chillers should be designed to handle these off-design conditions as efficiently as possible to minimize annual energy consumption. Refrigeration cycle efficiency improves at lower ambient temperatures resulting in reduced centrifugal chiller power consumption even at lower off-design compressor efficiency. The evaporator saturation temperature of chillers used for air conditioning stays more or less constant independent of load for dehumidification purposes. The condenser saturation temperature tends to reduce with load when ambient temperatures drop. The American Heating and Refrigeration Institute (AHRI) has defined chiller operating points at four capacities and has given weighting factors for each of these points simulating average part-load operating conditions. This allows a comparison of the annual energy consumption of different centrifugal chillers. The four chiller operating points defined by AHRI to compare annual energy consumption correspond to four points on the compressor performance map. Figures 2 show the location of these points on the normalized performance map of a fixed-speed centrifugal compressor with inlet guide vanes as its capacity control mechanism. Note the drop in relative efficiency for the lower capacity points. Figure 3 shows these same IPLV points on the map of a variable-speed centrifugal compressor. It becomes clear from comparing Figures 2 and 3 that chiller part-load efficiency and therefore its annual energy consumption can be improved dramatically.

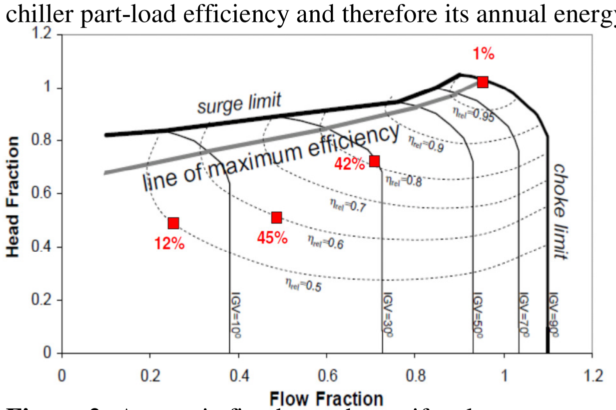

Figure 2: A generic fixed-speed centrifugal compressor performance map with the location of the four operating points and their weighting factors according to AHRI

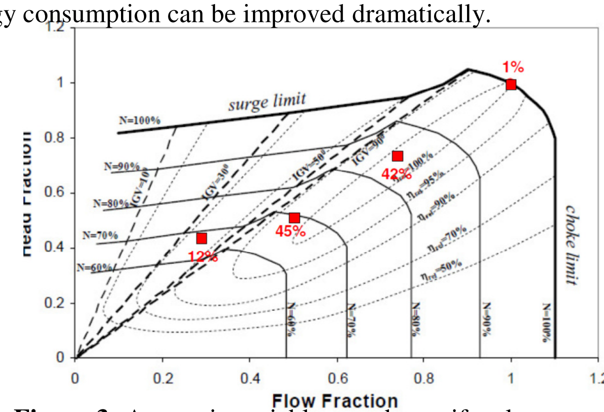

Figure 3: A generic variable-speed centrifugal compressor performance map with the location of the four operating points and their weighting factors according to AHRI

3. IS THERE A SIMILAR BENEFIT OF VARIABLE-SPEED RADIAL-INFLOW TURBINE OPERATION FOR ORC SYSTEMS?

3.1 Similarities and Differences between Centrifugal Compressors and Radial-Inflow Turbines

Refrigeration cycle performance deteriorates with increased head while ORC thermal efficiency increases. For ORC systems the heat source temperature that controls the saturation pressure of the boiler depends on the type of heat source being employed. This temperature is more or less constant for a given heat source creating a fixed turbine inlet pressure. The variable speed performance maps of compressors and radial inflow turbine are quite different as shown by Dixon and Hall 2010. When the pressure ratio seen by the turbine varies from 3 to 12 depending on ambient conditions the turbine flow is choked in the nozzle of the radial-inflow turbine and its flow rate is therefore constant. Turbine efficiency depends on both pressure ratio and speed. Since the speed lines lay on top of each other at higher pressure ratios the turbine efficiency cannot be read from a pressure ratio versus flow map as is possible for centrifugal compressors where higher compressor speed always results in increased capacity even under choked flow conditions.

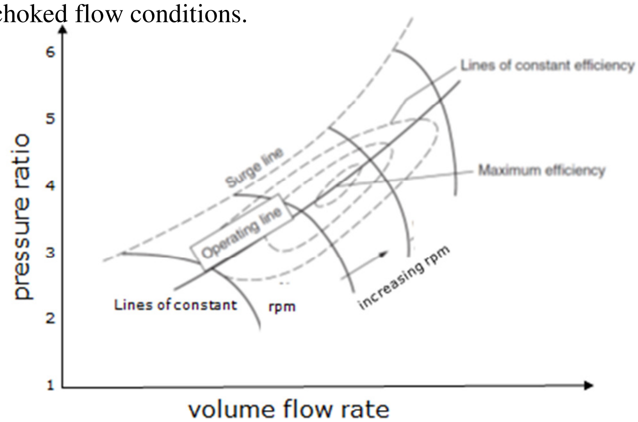

Figure 4: Variable-speed centrifugal compressor map

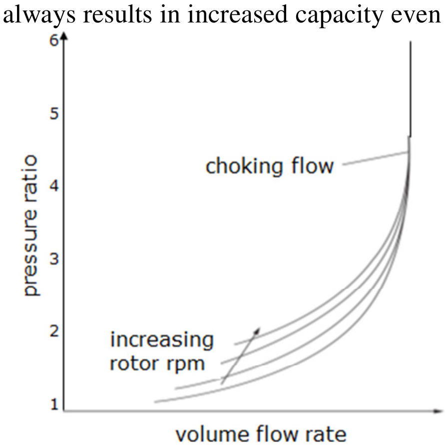

Figure 5: Variable-speed radial-inflow turbine map

3.2 Similarities and Differences between Centrifugal Chillers and ORC Systems

Contrary to the variation in cooling load which centrifugal chillers have to accommodate, ORC systems always produce the maximum amount of power possible. Hence, they do not necessarily require variable geometry such as rotatable inlet nozzles to reduce output power, similar to the inlet guide vanes reduce centrifugal chiller load.
Both centrifugal chillers and low-temperature ORC systems are affected strongly by changes in ambient conditions. Given the relatively small difference in saturation temperatures between boiler and condenser the pressure ratio experienced by the turbine can vary substantially with changes in ambient temperature. Air-cooled refrigeration and ORC systems experience this even more than water-cooled systems due to the larger variation in sensible ambient temperature compared to the changes in wet-bulb temperature.

3.3 Turbine Efficiency Correlation

The effect of changes in pressure ratio on turbine efficiency cannot be determined from its map as shown in section 3.1. However, as shown by Balje 1980 radial inflow turbine efficiency $\eta_{turbine}$ changes at off-design conditions as a function of the ratio between rotor tip speed u and spouting velocity c_0

$$\eta_{turbine} = f(\frac{u}{c_0}) \tag{1}$$

where the spouting velocity c_0 is defined as:

$$c_0 = \sqrt{2\Delta h_{is}} \tag{2}$$

with $h_{is} = h(P_{in}, T_{in}) - h(P_{out}, s_{in})$ and $s_{in} = s(P_{in}, T_{in})$

The efficiency of a radial-inflow turbine reaches its peak efficiency when the ratio between the rotor inlet tip speed u and the spouting velocity c_0 is 0.7. While the absolute peak efficiency of radial inflow turbines might vary based on differences in size, pressure ratio and design, the relative efficiency drop for ratios deviating from its optimum u/c_0 value has been similar for different radial inflow turbines. It follows a curve as shown in Figure 6. Eventually, if the tip speed of the turbine rotor becomes large enough the turbine rotor start adding energy to the fluid as opposed to extracting energy from it: the turbine becomes a compressor crossing the zero efficiency line. Similarly, if the tip speed becomes lower less energy transfer takes place between the fluid and the rotor. Zero energy transfer takes place when the rotor tip speed $u = 0$, resulting in a turbine efficiency of zero.

The rotor tip speed u does not change for a constant-speed turbine. Changes in the available head (isentropic enthalpy drop) as a result of changes in ambient conditions will increase or decrease the u/c_0 ratio and cause a drop in efficiency for those turbines. Changing turbine speed and therefore the rotor tip speed u allows continuous operation at turbine peak efficiency. The points on the turbine efficiency curve will be discussed below.

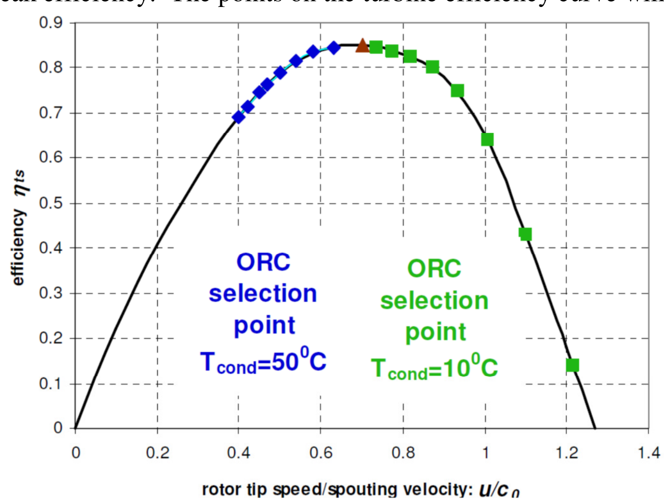

Figure 6: Radial inflow turbine rotor the rotor efficiency as a function of u/c_0

3.4 Differences in Full-Load Design Point Selection for Chillers versus ORC Systems

As mentioned earlier, the design point of a chiller is for the hottest day when it encounters its maximum cooling demand and the highest ambient temperature.

Contrary to HVAC equipment, the full-load design point for ORC systems can be selected for different ambient temperatures. If selected for the hottest day (like an HVAC system), the turbine generator set reaches its maximum design capacity already at that hottest day when the available head and the ORC cycle efficiency is lowest. At lower ambient temperatures when the ORC cycle efficiency improves as a result of the available higher head the turbine-generator set will be undersized. The refrigerant mass flow rate entering the turbine should then be reduced to protect the turbine generator from overload. This can be accomplished in two ways:

1. Variable-geometry nozzles (rotatable nozzle vanes or a movable nozzle side wall) will reduce the throat area of the nozzles which is the smallest cross-sectional flow area where the flow velocity reaches its maximum value of Mach 1, the speed of sound of the working fluid at prevailing temperature and pressure. A smaller throat area will result in a lower mass flow rate. The increase in turbine head combined with the smaller turbine mass flow rate can maintain turbine operation at design power output. The turbine output control methodology reduces the amount of heat required and improves the ORC thermal efficiency at lower ambient temperatures.

2. Turbine mass flow rate reduction can also be achieved by adjusting the ORC pump speed. A lower ORC pump speed will reduce the boiler and turbine inlet pressure. Lower turbine inlet pressure will result in lower density refrigerant which will reduce the mass flow rate at choked nozzle flow conditions. Compared to the variable nozzle turbine there is less gain in thermal efficiency as a result of the lower turbine inlet pressure at lower ambient conditions. However, this turbine output control method corresponds to a much simpler and more reliable turbine design (no internally moving parts besides the rotor). If the amount of heat required at the highest ambient temperature is freely available at lower ambient temperatures the preferred solution is pump speed variation with a fixed geometry nozzle arrangement.

Both control methods guarantee turbine protection against overloading. The higher cycle efficiency at lower ambient temperatures does not result in additional power (the turbine capacity is maxed out) but in lower thermal heat input. Demand for electrical power is highest at high ambient temperatures. ORC design point selection at these conditions, although resulting in more expensive equipment, guarantees maximum power production at high ambient temperatures. This type of ORC system is beneficial for cases when the thermal heat that is not needed at lower ambient temperatures can be saved or stored.

Variable speed turbine operation allows adjustment of the rotor tip speed u to maintain optimum turbine efficiency at ambient conditions that deviate from the design conditions allowing even more savings in thermal heat input. To quantify the benefit of variable speed operation for an ORC system designed for high ambient temperatures we will consider a low temperature ORC system with 75 ^{0}C as the evaporator saturation temperature. Using R245fa as our working fluid this corresponds to a turbine inlet pressure of 695 kPa. We consider an air-cooled condenser for this ORC unit with condenser saturation temperature of 50 ^{0}C at maximum ambient temperature, corresponding to a condenser pressure of 345 kPa. The ORC system is designed to deliver full power at these conditions. Isentropic enthalpy drop over the turbine equals 13.3 kJ/kg resulting in a spouting velocity c_0 = 163 m/s. A turbine designed for these conditions, has a rotor tip speed is 0.7*163 = 114 m/s as shown in the 7th row of Table 1. The earlier rows in Table 1 show the reduction in condenser pressure at lower condenser saturation temperatures. The isentropic enthalpy drop over the turbine increases accordingly from 13.35 kJ/kg when $T_{sat,cond}$ is 50 ^{0}C to 38.80 kJ/kg when $T_{sat,cond}$ is 10 ^{0}C. Turbine mass flow control through the use of variable nozzle geometry or pump speed variation is therefore required to prevent exceeding the power limit of the turbine designed for 50 ^{0}C at the off-design conditions with lower saturation temperatures. The end result is that constant power is being generated with smaller amounts of heat input at lower ambient temperatures.

Equation (2) allows the calculation of the spouting velocity c_0 at different condenser saturation temperatures. Combined with the rotor tip speed u at the $T_{sat,cond}$ = 50 ^{0}C turbine design condition, u/c_0 can be calculated. The turbine efficiency at off-design conditions can now be determined using the curve of Figure 6. Turbine efficiency drops from 84% when $T_{sat,cond}$ = 50 ^{0}C to 70% when $T_{sat,cond}$ = 10 ^{0}C as shown on the points to the left in Figure 6.

Table 1: Calculation of u/c_0 and $\eta_{turbine}$ of an ORC turbine designed for high ambient temperatures

$T_{sat,cond}$	$P_{sat,cond}$	$T_{sat,evap}$	$P_{sat,evap}$	Δh_{is}	c_0	u	u/c_0	$\eta_{turbine}$
^{0}C	kPa	^{0}C	kPa	kJ/kg	m/s	m/s	-	-
10	83	75	695	38.80	279	114	0.41	0.70
15	102	75	695	35.17	265	114	0.43	0.74
20	124	75	695	31.65	252	114	0.45	0.77
25	149	75	695	28.25	238	114	0.48	0.79
30	179	75	695	24.97	223	114	0.51	0.81
35	213	75	695	21.78	209	114	0.55	0.82
40	252	75	695	18.71	193	114	0.59	0.83
45	296	75	695	15.74	177	114	0.64	0.84
50	345	75	695	13.35	163	114	0.70	0.84

3.5 Full-Load ORC Design Point Selected for Low Ambient Temperature

The full-load design point of the low temperature ORC unit can also be selected for a much lower ambient temperature. This reduces the amount of heat input per kW of power produced substantially, similar to the controlled reduction in heat input for the ORC turbine selected for a high ambient temperature to prevent overloading of the compressor at lower ambient temperatures. The difference is that an ORC system that is designed for a lower ambient temperature will see a reduction in power output at higher ambient temperatures. The system will be lower in cost since it is designed for smaller flow rates but the absolute total annual amount of electricity produced will be lower. Where to select the design point of an ORC system requires an economic optimization.

The ORC system designed to deliver full power at 10 ^{0}C will now be analyzed. Isentropic enthalpy drop over the turbine equals 38.3 kJ/kg resulting in a spouting velocity c_0 = 279 m/s. A turbine designed for these conditions has a rotor tip speed is 0.7*279 = 195 m/s as shown in the first row of Table 2. The later rows in Table 2 show the increase in condenser pressure at higher condenser saturation temperatures. The isentropic enthalpy drop over the turbine diminishes accordingly from 38.80 kJ/kg when $T_{sat,cond}$ is 10 ^{0}C to 13.45 kJ/kg when $T_{sat,cond}$ is 50 ^{0}C. For a fixed turbine mass flow rate the turbine power output drops at higher condenser saturation temperatures because of lower cycle efficiency as well as lower turbine efficiency in case of a fixed speed turbine. Equation (2) allows the calculation of the spouting velocity c_0 at different condenser saturation temperatures. Combined with the rotor tip speed u at the $T_{sat,cond}$ = 10 ^{0}C turbine design condition u/c_0 can be calculated. The turbine efficiency at off-design conditions can now be determined using the curve of Figure 6. Turbine efficiency drops from 84% when $T_{sat,cond}$ = 10 ^{0}C to 22% when $T_{sat,cond}$ = 50 ^{0}C as shown on the points to the right in Figure 6.

Table 2: Calculation of u/c_0 and $\eta_{turbine}$ of an ORC turbine designed for low ambient temperatures

$T_{sat,cond}$	$P_{sat,cond}$	$T_{sat,evap}$	$P_{sat,evap}$	Δh_{is}	c_0	u	u/c_0	$\eta_{turbine}$
^{0}C	kPa	^{0}C	kPa	kJ/kg	m/s	m/s	-	-
10	83	75	695	38.80	279	195	0.70	0.84
15	102	75	695	35.17	265	195	0.74	0.83
20	124	75	695	31.65	252	195	0.78	0.82
25	149	75	695	28.25	238	195	0.82	0.81
30	179	75	695	24.97	223	195	0.87	0.8
35	213	75	695	21.78	209	195	0.93	0.74
40	252	75	695	18.71	193	195	1.01	0.63
45	296	75	695	15.74	177	195	1.10	0.45
50	345	75	695	13.35	163	195	1.19	0.22

3.6 Quantifying the Benefit of Variable-Speed Operation for ORC Turbines

Variable speed operation would allow constant turbine efficiency for all condenser saturation temperatures. The results presented in Tables 1 and 2 suggest that variable speed has more of an advantage over fixed-speed turbines for designs based on low condenser saturation temperatures since that design point selection results in the largest drop in fixed-speed turbine efficiency at off-design conditions. However, ORC systems tend to produce more power at lower ambient conditions when the turbine experiences a larger isentropic enthalpy drop. As a result we have to introduce a weighting factor at the various condenser saturation temperatures proportional to the isentropic enthalpy rise. Combined with a second weighting factor for the relative probability of ambient conditions resulting in the condenser saturation temperatures the real benefit of variable speed can be assessed. The product of these two weighting factors $f_{ambient}$ and $f_{\Delta h,is}$ are normalized in column 6 so that the sum of their frequencies equals 1.

For an ORC system with its design point at a condenser saturation temperature of 10 ^{0}C the weighted average turbine efficiency of a fixed speed turbine becomes 0.765 as opposed to a constant turbine efficiency of 0.84 in case of a variable speed turbine (see Table 3). Variable speed turbine operation improves net ORC power output by 9.1%.

Table 3: Calculation of the seasonal efficiency of a fixed-speed radial-inflow ORC turbine with its design point at a condenser saturation temperature of 10 ^{0}C using weighting factors $f_{ambient}$ for ambient temperature occurrence.

$T_{sat,cond}$	$f_{ambient}$	Δh_{is}	$f_{\Delta his}$	$f_{ambient}$ * $f_{\Delta h,is}$	$f_{normalized}$	$\eta_{turbine}$	$f_{normalized}$ * $\eta_{turbine}$
^{0}C	-	kJ/kg	-	-	-	-	
10	0.02	38.80	0.170	0.003	0.031	0.84	0.026
15	0.06	35.17	0.154	0.009	0.084	0.83	0.070
20	0.12	31.65	0.139	0.017	0.151	0.82	0.124
25	0.18	28.25	0.124	0.022	0.202	0.81	0.164
30	0.24	24.97	0.109	0.026	0.238	0.80	0.191
35	0.18	21.78	0.095	0.017	0.156	0.74	0.115
40	0.12	18.71	0.082	0.010	0.089	0.63	0.056
45	0.06	15.74	0.069	0.004	0.038	0.45	0.017
50	0.02	13.35	0.058	0.001	0.011	0.22	0.002
Total	1.00	228.42	1.000	0.110	1.000		0.765

For an ORC system with its design point at a condenser saturation temperature of 50 ^{0}C the weighted average turbine efficiency of a fixed speed turbine becomes 0.782 as opposed to a turbine efficiency of 0.84 in case of a variable speed turbine (see Table 4). Variable speed turbine operation reduces the annual net ORC heat input requirement by 7.0%.

Table 4: Calculation of the seasonal efficiency of a fixed-speed radial-inflow ORC turbine with its design point at a condenser saturation temperature of 50 ^{0}C using weighting factors $f_{ambient}$ for ambient temperature occurrence.

$T_{sat,cond}$	$f_{ambient}$	Δh_{is}	$f_{\Delta his}$	$f_{ambient}$ * $f_{\Delta h,is}$	$f_{normalized}$	$\eta_{turbine}$	$f_{normalized}$ * $\eta_{turbine}$
^{0}C	-	kJ/kg	-	-	-	-	
10	0.02	38.80	0.170	0.003	0.031	0.69	0.021
15	0.06	35.17	0.154	0.009	0.084	0.72	0.060
20	0.12	31.65	0.139	0.017	0.151	0.75	0.113
25	0.18	28.25	0.124	0.022	0.202	0.77	0.156
30	0.24	24.97	0.109	0.026	0.238	0.79	0.188
35	0.18	21.78	0.095	0.017	0.156	0.82	0.128
40	0.12	18.71	0.082	0.010	0.089	0.83	0.074
45	0.06	15.74	0.069	0.004	0.038	0.84	0.032
50	0.02	13.35	0.058	0.001	0.011	0.84	0.009
Total	1.00	228.42	1.000	0.110	1.000		0.782

4. CONCLUSIONS

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- The change in cycle efficiency with ambient conditions has a large impact on both the annual energy consumption of air-conditioning equipment and annual energy production of low temperature ORC equipment.
- The benefit of variable-speed centrifugal compressor operation for air-conditioning equipment (water- and air-cooled chillers) over fixed-speed operation on annual power consumption is known to be almost 40%.
- Variable speed turbine operation can improve the annual electricity production of low temperature ORC systems by up to 10% relative to fixed speed turbine operation, allowing maximum utilization of low temperature heat sources for ORC systems that have their full-load selection point at lower ambient temperatures.
- Variable speed turbine operation can reduce the annual heat input requirement of low temperature ORC systems by up to 9% for ORC systems that have their full-load selection point at high ambient temperatures.
- The benefit of variable speed turbine operation for ORC systems is less than that of variable speed centrifugal compressor operation for refrigeration systems as a result of the larger head variation experienced by air-conditioning equipment and the stronger sensitivity of compressor efficiency for the same relative variation in head.
- The benefit of variable speed ORC turbines diminishes for higher temperature ORC systems when the relative changes in head with variations ambient temperature become smaller.

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