

CASCADED ORGANIC RANKINE CYCLE SYSTEM

BACKGROUND

The present disclosure relates generally to Organic Rankine Cycle (ORC) systems and, more particularly, to a cascaded organic Rankine cycle.

The Organic Rankine Cycle (ORC) is a vapor power cycle with an organic fluid refrigerant instead of water/steam as the working fluid. The working fluid is heated in an "evaporator/boiler" by a source of waste or low quality heat. The fluid starts as a liquid and ends up as a vapor.

The high-pressure refrigerant vapor expands in the turbine to produce power. The low-pressure vapor exhausted from the turbine is condensed then sent back to the pump to restart the cycle.

The simple rankine cycle used for power generation follows the process order: 1) Adiabatic pressure rise through a pump; 2) Isobaric heat addition in a preheater, evaporator and superheater; 3) Adiabatic expansion in a turbine; and 4) Isobaric heat rejection in a condenser, although other cycle modifications are possible such as the addition of a vapor-to-liquid recuperator.

Typical cascaded organic Rankine cycle systems are disclosed in WO2009045196 and WO2011005374 whilst US20100122534 discloses a Rankine cycle system including a recuperator intended to increase system efficiency.

Furthermore, it is known from US 4085593 that the transitioning of fluids into superheated phases is desirable for power generation when the working fluid is expanded through the turbine.

Such thermal energy generation need not necessarily come from conventional engines but may be derived from solar power, (see for example the disclosure contained within US20100295306).

A main thermodynamic irreversibility in organic Rankine cycles is caused by the large temperature difference in the evaporator between the temperature of the waste heat stream and the boiling refrigerant. The higher the waste heat stream temperature the greater this irreversibility becomes.

One way to reduce this loss is to cascade two thermodynamic cycles together where a cycle operating at higher temperatures rejects heat to a cycle operating at lower temperatures.

Other systems employ 'counter flow' heat exchangers such as the one disclosed in WO2010080861 whereby streams of fluid flow in opposite directions through the heat exchanger thus causing it to operate at a higher efficiency than systems with heat exchangers wherein the streams flow in the same direction.

5 Other losses such as those caused by mechanical stress may also be reduced through the use of a counter flow heat exchanger (US6241953) and greater overall heat transfer efficiency may be achieved through the use of a saturated heating fluid (US7487955).

10 Although the aforementioned cited prior art addresses some of the efficiency issues associated with systems incorporating the Rankine cycle, increased efficiency of such systems is still desirable.

SUMMARY

15 A cascaded Organic Rankine Cycle (ORC) system according to an exemplary aspect of the present disclosure includes a bottoming cycle in thermal communication with a topping cycle through a condenser/evaporator in which a bottoming cycle working fluid is first evaporated and then superheated and a topping cycle working fluid is first desuperheated and then condensed such that a percentage
20 of total heat transfer from the topping cycle fluid that occurs during a saturated condensation is equal to or less than a percentage of total heat transfer to the bottoming cycle fluid that occurs during a saturated evaporation.

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A method of operating a cascaded Organic Rankine Cycle (ORC) system in which a bottoming cycle is in thermal communication with a topping cycle according to an exemplary aspect of the present disclosure which includes maintaining a percent saturation for a fluid in the topping cycle at less than a 40 percent saturation for a fluid in the bottoming cycle.

BRIEF DESCRIPTION OF THE DRAWINGS

Various features will become apparent to those skilled in the art from the following detailed description of the disclosed non-limiting embodiment. The drawings that accompany the detailed description can be briefly described as follows:

Figure 1 is a schematic diagram of a cascaded organic rankine cycle with a topping cycle and a bottoming cycle;

Figure 2 is a TS-diagram for the bottoming cycle;

Figure 3 is a TS-diagram for the topping cycle; and

Figure 4 is a plot of temperature profiles in the counter-flow heat exchangers of the de-superheating and then condensing topping fluid (Siloxane MM), and the evaporating and then superheating bottoming fluid (R245fa).

DETAILED DESCRIPTION

Figure 1 schematically illustrates a cascaded Organic Rankine Cycle (ORC) system 20. The cascaded ORC system 20 includes at least two Rankine cycles, where a relatively hotter topping cycle 22 is cascaded with a relatively cooler bottoming cycle 24. In the disclosed non-limiting embodiment, the topping cycle 22 uses Siloxane MM as the working fluid while the bottoming cycle 24 uses R245fa. It should be appreciated, however, that additional cycles and other working fluids may additionally be utilized.

The topping cycle 22 generally includes a power producing turbine 26 which is driven by the working fluid to drive a generator 28 that produces power. A refrigerant pump 30 increases the pressure of the working fluid from a condenser/evaporator 32. The heat exchanger group that transfers heat from the topping cycle 22 to the bottoming cycle 24 is referred to herein as the “condenser/evaporator” 32, although it should be understood that it may also include

desuperheating and subcooling of the working fluid in the topping cycle 22, and preheating and superheating of the working fluid in the bottoming cycle 24.

An evaporator 34 such as a boiler receives a significant heat input from, for example, an oil circuit 36 to vaporize the Siloxane MM working fluid with the vapor thereof passed through to the turbine 26 to provide motive power. Upon leaving the turbine 26, the relatively lower pressure working fluid vapor passes to the condenser/evaporator 32 and is condensed by way of a heat exchange relationship with the bottoming cycle 24 such that the condenser/evaporator 32 operates as a condenser in the topping cycle 22 as well as an evaporator in the bottoming cycle 24.

In the disclosed non-limiting embodiment, the turbine 26 is a radial inflow turbine that expands the topping cycle working fluid vapor down to a lower pressure and generates power by the extraction of work from this expansion process. The vapor is still superheated so that its heat potential is utilized in the condenser/evaporator 32. The condenser/evaporator 32 actually de-superheats the working fluid and ultimately condenses the working fluid back to liquid for communication through the pump 30. The condensed working fluid is then circulated to the evaporator 34 by the pump 30 to complete the topping cycle 22.

The bottoming cycle 24 generally includes a power producing turbine 36 which is driven by the working fluid in the bottoming cycle and in turn drives a generator 38 that produces power. A refrigerant pump 40 increases the pressure of the working fluid from a recuperator 46. The bottom cycle working fluid is in thermal communication with a cooling system such as a water circuit 42 through a water cooled condenser 44.

By the nature of the proposed cycle, the vapor entering and leaving turbine 36 is highly superheated. The energy potential of the superheated vapor at the turbine exit is not wasted, but is fed into a recuperator 46. The recuperator 46 transfers heat from the low-pressure hot vapor from the turbine exit to the high pressure liquid at the pump exit.

The recuperator 46 uses this superheat to preheat the liquid working fluid downstream of the pump 40. That is, if a cycle is driven to high turbine inlet superheat, then turbine outlet superheat will be high. The availability of this heat is thereby captured to maintain cycle efficiency as the recuperator 46 is an internal heat exchanger. When the low pressure side of the topping cycle 22 is de-superheated, it is essentially recuperated into the bottoming cycle 24 which is where high superheat

is achieved. Matching of the working fluids and the pressures thereof facilitates this interaction.

The recuperator 46 is only in the bottoming cycle 24. As the topping cycle 22 is not recuperated, its waste heat is captured by the condenser/evaporator 32. Both cycles are highly superheated yet avoid heat-exchanger pinches to minimize the heat-transfer temperature difference and minimize process irreversibility

Figure 2 shows a TS diagram for the bottoming cycle 24. The condenser/evaporator 32 receives nearly saturated liquid (a temperature that is close to boiling) from the recuperator 46. The condenser/evaporator 32 boils then heats the refrigerant from state 6 to 1. The state 1 condition is highly superheated. The exit state from the turbine 36, state 2, is also highly superheated. The recuperator 46 uses this heat (state 2 to 3) to heat the high pressure working fluid (state 5 to 6). Sizing of the recuperator 46 affects state 6. A smaller recuperator 46, for example, results in less heat transferred and therefore a cooler more subcooled state at 6 which results in more heat transfer required from the condenser/evaporator 32, and a larger percentage of that heat in the preheating and evaporating regimes.

Figure 3 shows a TS diagram for the topping cycle 22. The exit state of the topping cycle turbine 26 is highly superheated, but a recuperator is not used. Instead, the low pressure working fluid vapor is de-superheated as the bottoming cycle high-pressure working fluid is superheated. The choice of a heavy molecule such as Siloxane for the topping cycle 22 results in the highly angled saturation dome. As a result, the inlet state to turbine 26 is only slightly superheated.

Figure 4 represents an idealized counter-flow heat exchanger. The x-axis is normalized enthalpy change of each fluid, and the y-axis is temperature. The x-axis is based on the First Law of Thermodynamics which can be written for a heat exchanger as:

$$\dot{m}_A (h_{A in} - h_{A out}) = \dot{m}_B (h_{B out} - h_{B in})$$

Where the subscripts *A* and *B* refer to streams A and B respectively, *m* is the mass flow rate, and *h* is the enthalpy of the fluid.

In Figure 4, the warmer fluid (A) is shown to travel from right to left, and the colder fluid (B) to travel from left to right through the heat exchanger. Heat transfers from fluid A to fluid B; therefore, fluid A's enthalpy decreases while fluid B's enthalpy increases. For each section of the heat exchanger the above equation

must be true. For example, the first 10% reduction in enthalpy of Fluid A must equal the last 10% increase of enthalpy of fluid B. If the fluids were simple fluids with constant specific heat, then each temperature profile would be a straight line. When the fluids are refrigerants, the temperature profiles have various non-linear shapes.

5 When a fluid is saturated there is no change in temperature with change in enthalpy. The change in temperature with enthalpy is generally different for a fluid as a liquid than as a vapor; therefore, the choice of fluid and operational temperatures affect the shape of these curves. Furthermore, the choice of other system components will affect their shape. Specifically the choice of and the size of the recuperator 46 in the
10 proposed cycle affects the starting enthalpy (and therefore temperature) of stream B.

Figure 4 shows how each temperature profile relates to the other at each physical location along the heat exchanger. In order for heat to flow from Fluid A to Fluid B, Fluid A must always be warmer than Fluid B. If A gets too close to B this is referred to as a temperature “pinch” condition. This is undesirable because a large
15 heat exchange area is required to exchange the enthalpy in this region. In fact, the entire size of a heat exchanger may be defined by a “pinch” condition. Where the temperature difference is large, the thermodynamic cycle will be less efficient since more entropy is generated by heat exchange through larger temperature differences. An ideal arrangement is when the temperature difference throughout the heat
20 exchanger remains relatively constant. Since vapor heat exchange usually has a lower heat transfer rate than saturated, it may be desirable to maintain a somewhat higher temperature difference in this region, typically up to or equal to 1 to 2 times. For the ORC system 20, the condenser/evaporator 32 heat exchanger has two major regions. The first (on the left in Figure 4) is saturated for both fluids and the
25 temperature profiles are flat. This section covers about 40 percent of the total heat transfer in the disclosed non-limiting embodiment. The second (on the right in Figure 4) is superheated and temperature increases with enthalpy. That is, a percent saturation for a fluid in the topping cycle 22 is maintained at 38 percent saturation compared to a 40 percent saturation for the working fluid in the bottoming cycle 24.

30 The point where the temperature profile transitions from flat (saturated) to increasing (vapor) will be identified herein as the “knee.” For the above goals to be achieved, the “knee” of fluid A must lie equal to or slightly to the left of the “knee” of fluid B in the normalized enthalpy plot. If the “knee” lies far to the left then the saturated section may have a good heat transfer difference (typically 5 to 15F; 3 to

8C), but the heat transfer difference of the vapor section will be too large. If the "knee" lies too far to the right then a "pinch" condition will be created between the two fluids. Practically the temperature difference will increase and the saturated temperature difference will be too high.

5 The effect of the recuperator 46 on the condenser/evaporator 32 in the proposed cycle is to change the inlet enthalpy, and therefore temperature, of the colder fluid, B. By increasing the size of the recuperator 46, the enthalpy of the inlet of B increases by recovering heat from the turbine exit. This results in a smaller percentage of the total heat transfer for Fluid B occurring to the left of the knee, shifts the knee of B to the left and results in a pinch condition. Conversely, if the
10 recuperator heat exchange is reduced or eliminated, this shifts knee of B to the right and therefore increases the temperature difference in the vapor section. That is, a percentage of total heat transfer from the working fluid in the topping cycle 22 that occurs during a saturated condensation is equal to or slightly less (within 10%) than a
15 percentage of total heat transfer to the working fluid in the bottoming cycle 24 that occurs during a saturated evaporation.

 The selection of a high superheat cascaded cycle with a condenser/evaporator heat exchanger transferring heat from the topping cycle to the bottoming cycle, and the selection of refrigerants for the topping and bottoming cycles and recuperator in
20 the bottoming cycle allows for optimized heat exchanger temperature profiles.

 It should be understood that relative positional terms such as "forward," "aft," "upper," "lower," "above," "below," and the like are with reference to the normal operational attitude of the vehicle and should not be considered otherwise limiting.

 It should be understood that like reference numerals identify corresponding or similar elements throughout the several drawings. It should also be understood that
25 although a particular component arrangement is disclosed in the illustrated embodiment, other arrangements will benefit herefrom.

 Although particular step sequences are shown, described, and claimed, it should be understood that steps may be performed in any order, separated or
30 combined unless otherwise indicated and will still benefit from the present disclosure.

 The foregoing description is exemplary rather than defined by the limitations within. Various non-limiting embodiments are disclosed herein, however, one of ordinary skill in the art would recognize that various modifications and variations in

light of the above teachings will fall within the scope of the appended claims. It is therefore to be understood that within the scope of the appended claims, the disclosure may be practiced other than as specifically described. For that reason the appended claims should be studied to determine true scope and content.

CLAIMS

What is claimed is:

1. A method of operating a cascaded Organic Rankine Cycle (ORC) system in
5 which a bottoming cycle is in thermal communication with a topping cycle
comprising:
 - superheating a working fluid in the topping cycle;
 - expanding the working fluid over a turbine in the topping cycle such that the
working fluid leaving the turbine is still superheated;
 - 10 circulating the working fluid from the turbine in the topping cycle into a
condenser/evaporator;
 - circulating a working fluid in the bottoming cycle into the
condenser/evaporator for heat exchange with the working fluid of the topping cycle,
the working fluid in the bottoming cycle being superheated in the
15 condenser/evaporator;
 - during the heat exchange in the condenser/evaporator, maintaining a percent
of total heat transfer from the working fluid in the topping cycle that occurs during
saturated condensation to be less than a percent of total heat transfer to the working
fluid in the bottoming cycle that occurs during saturated evaporation.
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2. The method as recited in claim 1, further comprising:
 - utilizing Siloxane MM as the working fluid in the topping cycle; and
 - utilizing R245fa as the working fluid in the bottoming cycle.
- 25 3. The method as recited in claim 1, comprising expanding the working fluid in
the bottoming cycle over a turbine in the bottoming cycle and circulating the working
fluid in the bottoming cycle from the turbine into a recuperator.
4. The method as recited in claim 3, comprising using the working fluid that is
30 circulated into the recuperator from the turbine in the bottoming cycle to preheat the
working fluid in the bottoming cycle at a location downstream from a pump in the
bottoming cycle and prior to circulation into the condenser/evaporator such that the

percent total heat transfer from the working fluid in the topping cycle that occurs during saturated condensation is within 10% of the percent of total heat transfer to the working fluid in the bottoming cycle that occurs during saturated evaporation.

5 5. The method as recited in claim 1, comprising superheating the working fluid in the topping cycle using boiler in the topping cycle.

6. The method as recited in claim 5, comprising superheating the working fluid in the topping cycle in the boiler using an oil circuit.

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