IMPROVING THE OFF-DESIGN EFFICIENCY OF ORGANIC RANKINE BOTTOMING CYCLE BY VARIABLE AREA NOZZLE TURBINE TECHNOLOGY

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ABSTRACT

Organic Rankine cycles (ORCs) are expected to operate more at varied working conditions, especially when rapid growth in renewable energies put power generation technologies in variable load operation. A power cycle works with lower efficiency in part loads than expected in full load condition. Here, an organic Rankine bottoming cycle with a control strategy based on variable geometry turbine technology is proposed to boost part load efficiency of combined cycles offshore. The Variable Area Nozzle (VAN) turbine is selected to control cycle mass flow rate and pressure ratio independently. A design methodology is presented, and the performance of proposed design and developed control strategy is assessed by an in-house developed tool. With the suggested solution, part load ORC efficiency is kept close to design values while clearly outperforming the reference case with sliding pressure operation. The proposed solution assures efficient operation in a wide range of loads. The combined cycle efficiency showed a clear improvement compared to the reference case, resulting in thousands of tonnes of annual CO_2 (carbon dioxide) emission reduction. Compactness, autonomous operation, and acceptable technology readiness level for VAN turbine solution facilitate application of presented design to offshore oil and gas installations.

1 INTRODUCTION

The increasing awareness of global warming made it generally desirable to reduce the CO_2 emissions per unit of produced oil or natural gas. This has led to the introduction of an offshore CO_2 tax in Norway (Walnum et al. 2013). Harvesting gas turbine (GT) waste heat is a potential solution to reduce CO_2 emitted per MW in offshore oil and gas installations. Organic Rankine cycle (ORC) is one of several options for gas turbine waste heat recovery in offshore oil platforms (Omar et al. 2019). Many authors investigated applicability and competency of ORCs for offshore applications. Pierobon *et al.*, (2014) conducted a case study on an offshore platform in the North Sea to find the optimum bottoming cycle solution based on yearly CO_2 emissions, weight, and economic revenue. They found organic Rankine cycle technology presenting better performance compared to steam Rankine cycle and air bottoming cycle units. ORC is more compact and can operate autonomously in offshore applications. Moreover, design of ORC is not limit to a specific working fluid. This can provide design flexibility to cover a wide range of needs in variable circumstances of offshore installations. Thus, employment of ORC as a bottoming cycle can improve power generation capacity with higher efficiency and less CO_2 emission offshore.

Offshore gas turbines operate mostly in part load. Primary reason is to assure reliability in power generation of offshore platforms. Power load of a gas turbine is typically 50% in offshore applications and may falls down to 20% in some conditions (Pierobon *et al.*, 2013 and Riboldi and Nord, 2017). The other reason is rapid growth in renewable energies such as offshore wind parks and power from shore. They put gas turbines in intermittent operation. Thermal efficiency of an ORC decays as gas turbine load decreases. With lower power demand from the gas turbine, less heat will be available to the bottoming cycle. It can, for example, be 70% of on-design exhaust heat at 50% load for a gas turbine (Mazzetti et al. 2014), or 50% of on-design exhaust heat at 20% loads for other gas turbine types (Nord and Montañés 2018). The value depends on the gas turbine type, but it typically falls near mentioned values. Due to variability of the exhaust gas flow rate over time, finding a proper design and control strategy of the bottoming cycle are essential to avoid poor off-design performance (Nguyen et al. 2016). Various control strategies have been proposed in the literature to make an efficient part load operation of ORCs. Among them, sliding pressure (SP) operation and superheating are commonly used (Cao and Dai 2017). Superheating of the bottoming cycle is not practical offshore as lots of space is needed for the superheater. On the other hand, sliding pressure control strategy will end up with considerable efficiency losses at part load, in which flow rate must be decreased to cope with reduced heat available. Thus, ORC turbine operates at lower pressure ratios due to reduced mass flow rate. This leads to a loss of cycle efficiency at part loads. Fu *et al.* (2014) showed how high the efficiency drop with sliding pressure can be during part load operation. Compensating this efficiency loss will be crucial to reducing the CO_2 emissions as offshore ORCs and gas turbines are expected to spend most of their operational lifetime at part load.

Some efforts have been made by researchers to improve ORC part load efficiency. Casartelli *et al.* (2015) proposed a partial admission turbine to adjust ORC turbine capacity independent of suction pressure. However, it is not an efficient way to reduce turbine capacity and will bring considerable pressure losses in the turbine throughflow passage. Cao and Dai (2017) used variable area nozzle (VAN) turbine technology to keep turbine inlet pressure constant and improved ORC part load efficiency. However, a fixed pressure strategy may not necessarily give the best efficiency possible. Muñoz De Escalona *et al.* (2012) designed a constant efficiency part load strategy by putting turbine suction temperature far away from its optimum value in the design point. An acceptable part load performance is achieved with an on-design performance penalty. Consequently, a considerable power generation potential is left unused at full load and a higher exergy loss is detected in the design point. This logic is helpful but will end up in an overdesigned heavy system for the same amount of power demand. It may be beneficial onshore as it improves part load efficiency in expense of more space occupied; but space is limited in offshore installations. Thus, provided design by Muñoz De Escalona et al. (2012) does not suit offshore applications.

This paper will present and evaluate an operational strategy based on Variable Area Nozzle (VAN) turbine technology to improve the part load efficiency of ORC in offshore oil and gas installations. Presented solution is compact, autonomous and can keep the ORC thermal efficiency close to the design point value in a wide range of power loads.

2 MATHEMATICAL MODELS AND METHOD

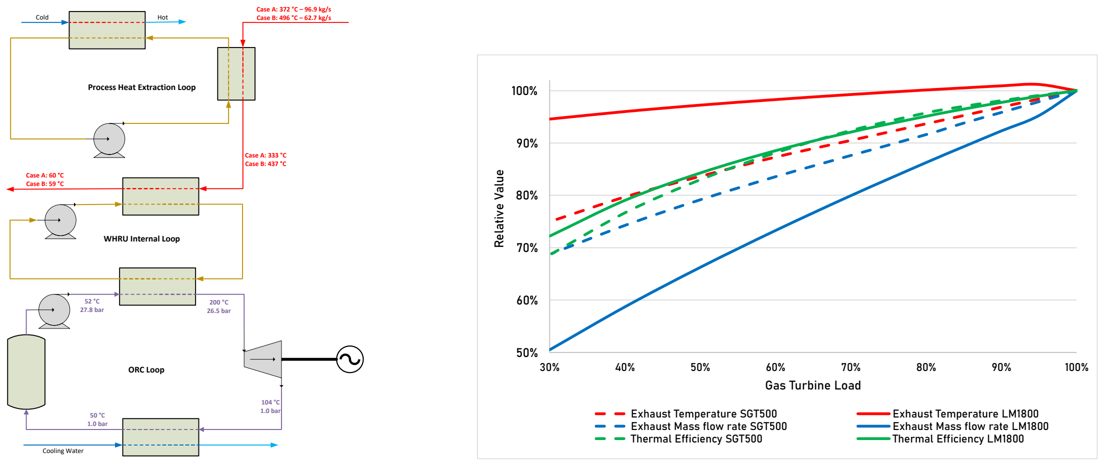
The methodology to design the bottoming cycle and development of the control strategy is described in this section. It is explained how the cycle performance is simulated and how each component is modelled. The developed tool for design and analysis is described.

2.1 System configuration selection

A Cascade architecture is used in this work to harvest GT exhaust heat (Figure 1a). Nami *et al.* (2018) conducted a study on cascade and series configuration layout in ORC bottoming cycles. they concluded that at a fixed source (gas turbine exhaust) temperature, changing heat demand in the cascade system has no effect on the produced power, because heat absorption (evaporator) and rejection in this system occur at fixed temperature profiles. The process heat required to the platform is extracted just at gas turbine discharge and before entering a non-recuperated ORC. It is crucial to guarantee process heat availability in an oil platform. So that, requested amount of heat is always available in all gas turbine loads. Heat transfer is carried out within an internal loop to reduce risk of fire in case of organic fluid leakage. ORC turbine is a fixed-speed radial inflow expander. Turbine stators are pivoted to provide variable area between vanes. Cooling water is assumed to always be available from sea at 30°C or lower in the condensation section. Cooling water temperature may be raised to desired values using waste heat from the plant. The bottoming cycle works with a variable-speed centrifugal pump. Figure 1a illustrates the selected system configuration and some design point thermodynamic states.

Presented solution is studied with two gas turbine cases to cover different types of gas turbine control logics. Case A is a *Siemens SGT500A2* gas turbine that has decreasing exhaust temperature in part loads. The low exhaust temperature makes SGT500 a favorable choice for working with organic fluid bottoming cycles. However, combined cycle efficiency decays rapidly at part loads due to fall in GT exhaust temperature. Case B is a *GE LM1800eHP* which is an aeroderivative gas turbine adapted from GE LM2500 family. The GE LM1800e HP has higher exhaust temperature than case A but an almost constant exhaust temperature pattern enables it to have better part load efficiency (Figure 1b).

A variable area nozzle radial inflow turbine is used to extract work in the VAN turbine control strategy, while a conventional non-variable geometry radial inflow turbine is used in the baseline plant. Radial inflow turbines are advantageous since they can be used with minor modifications for organic Rankine cycles (Marcuccilli 2008). The radial inflow turbine works at a constant speed in all working conditions. So, it could be coupled with the electrical generator easily. the only different component between sliding pressure and VAN turbine system is the radial turbine.



a) Flow diagram b) Gas Turbine Performance

Figure 1 a) Combined cycle flow diagram b) Gas turbine part load performance

2.2 Component Modelling

A preparative step to develop the control strategy is to select suitable models for simulation of each component performance. Models used for performance analysis of the components are presented here. Gas turbine performance data is calculated with GT MASTER 29 commercial software from Thermoflow Inc.(GT MASTER 2020). The software provides performance data from numerous gas turbines in operation. However one should make sure validity of software data with field or experimental data. gas turbines performance is considered in steady state operation and ISO (15°C, 1.013 bar) ambient conditions. Performance data include GT efficiency, exhaust mass flow rate and exhaust temperature in different power load demands.

The pump pressurizing the working fluid before entering the preheater is a multi-stage variable speed centrifugal pump. Hu, Li, *et al.* (2015) presented the correlation between head rise and efficiency with volume flow rate for a multistage centrifugal pump in power plant application. Their model is used in this work as it is generalized by using relative-to-design values. It calculates head rise and isentropic efficiency based on the flow rate in off-design conditions.

A reliable and accurate model for variable area nozzle turbine performance prediction is necessary to have a solid assessment of plant performance. Performance of a radial inflow turbine is modelled in off-design conditions by Meitner and Glassman (1980) and is used here. Their model includes a wide range of throat opening in pivoted stator VAN turbines (from 20% to 144% opening). They showed that capacity of a radial inflow turbine can be widened to a range of 20-130% of design point flow rate. Moreover, they showed that pressure ratio of a VAN turbine can be extended to 75-115% of design value. The model developed by Meitner and Glassman (1980) is formulated in the form of following correlation equation in which η_{VANT} is isentropic efficiency of the VAN turbine at an arbitrary working condition.

$$\eta_{VANT} = \eta_{dp}[-0.8452\dot{m}_{c,rel}^2 + 1.5807\dot{m}_{c,rel} + 0.2622] \qquad (1)$$

η_{dp} is efficiency of the turbine in design point. $\dot{m}_{c,rel}$ is relative corrected mass flow rate through ORC turbine and is defined in the following equation.

$$\dot{m}_{c,rel} = \left(\frac{\dot{m}}{\dot{m}_{dp}}\right)\left(\frac{T_0}{T_{0\,dp}}\right)\left(\frac{P_0}{P_{0\,dp}}\right)^{-1} \qquad (2)$$

Here *dp* denotes for design point values and $\dot{m}$ denotes for mass flow rate. T_0 and P_0 are turbine inlet total temperature and total pressure, respectively.

2.3 Cycle design point

The bottoming cycle design point properties are selected with a parametric study on evaporation temperatures, condensing temperature and mass flow rate. In a Rankine cycle with negligible superheating, selecting evaporating and condensing temperatures will specify the pressures of the thermodynamic cycle. Afterwards, mass flow rate can be selected such that heat exchanger approach temperatures are kept within allowed range. Design choice is made to attain best values for figure of merits (Higher thermal efficiency and higher specific power) while watching the constraints (cycle pressure ration and heat exchangers pinch point). Table 1 shows the input to design process of the bottoming cycle. The parametric study is carried out by an in-house tool, developed in MATLAB framework. Cycle performance is simulated with a so-called zero-dimensional approach in which thermodynamic properties at a component exhaust is calculated based on inlet conditions and a generalized performance correlation available in open literature. Thermodynamic properties library is established by linking CoolProp database to the developed tool. CoolProp is a thermodynamic calculator coupling with Python and is available free (Bell *et al.* 2014). Heat exchangers is designed by assuming 97% effectiveness. Heat exchanger discharge temperatures and mass flow rate are then calculated based on inlet conditions and energy balance.

Table 1: Design point performance and conditions

Gas Turbine	Case A	Case B	Bottoming Cycle	
Power Output [MW]	18.60	17.53	Hex Pinch Point Limit [°C]	10
Efficiency [%]	33.24	34.35	Pump Isentropic Efficiency [%]	70
Exhaust Flow Rate [kg/s]	96.9	62.7	Turbine Isentropic Efficiency [%]	80
Exhaust Temperature [°C]	372	496	Condenser Pressure Loss [%]	5
Process Heat Extraction [MW]	4	4	Evaporator Pressure Loss [%]	5

2.4 Part Load Strategy Development

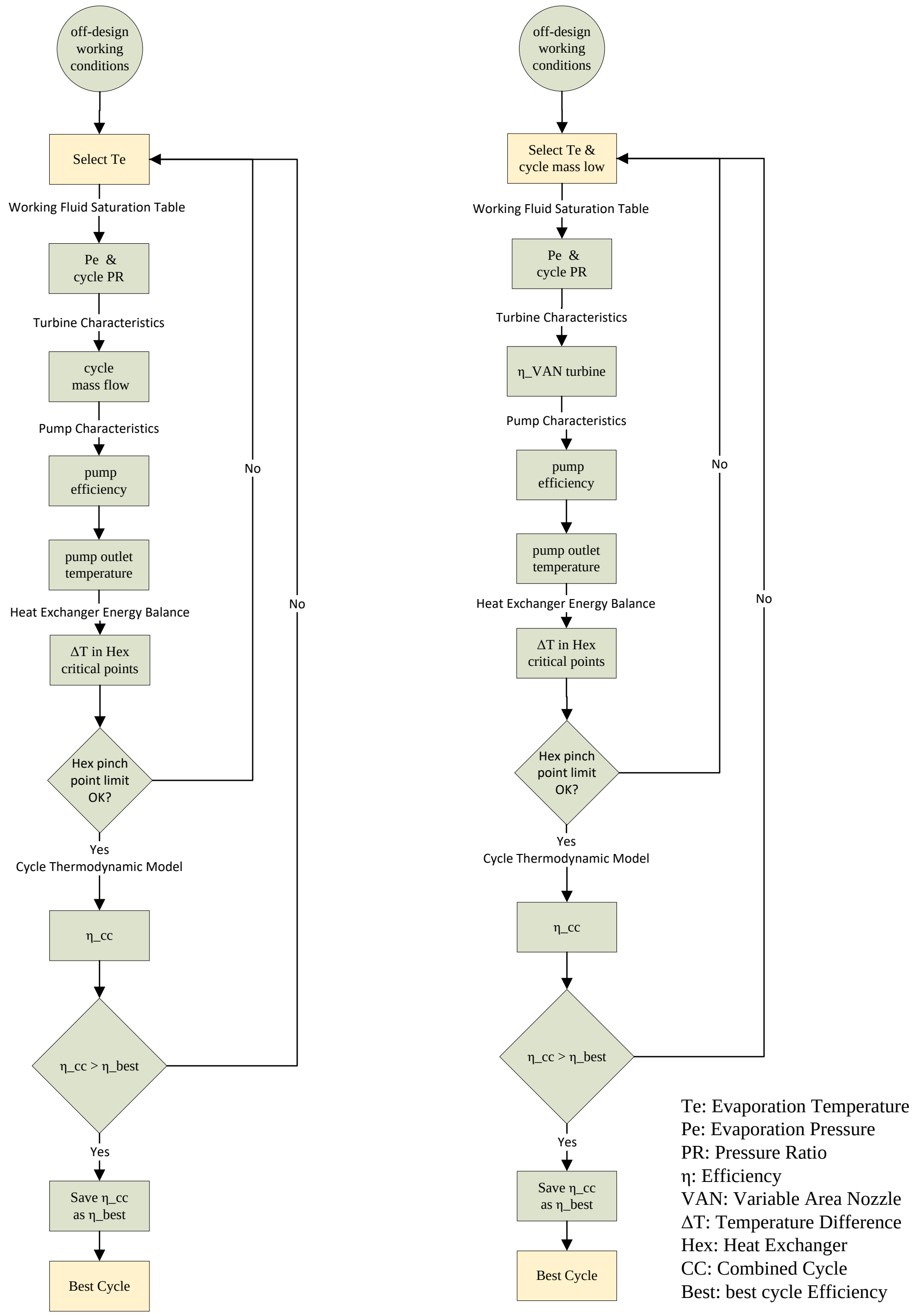
Two control logics are developed and simulated in this work, An improved version of sliding pressure (SP) logic and a VAN turbine strategy. The SP logic is considered as baseline so the improvements made by the proposed VAN logic can be compared with it. However, available SP cases in the literature were not showing maximum possible part load efficiency. Thus, it was decided to develop and optimize a baseline SP logic first, then the proposed VAN strategy is to be proposed. The sliding pressure controller manipulates pump speed and cooling water temperature to control cycle power output according to waste heat available from GT while keeping heat exchanger pinch point limitations in the permitted range. Pump speed governs evaporation pressure in the cycle while condensing pressure is regulated by adjusting condensing temperature. In the improved version of sliding pressure, a thorough search is conducted through all input parameters possibilities (mass flow rate and evaporation pressures) to find the pair with maximum cycle efficiency. Figure 2a illustrates flow chart for search and selecting best cycle in sliding pressure part load strategy. A simple plane search is used as small possibility domain does not necessitate a complex optimization method.

In the VAN turbine control strategy, pump speed, cooling water temperature and turbine vanes setting angle are manipulated to control cycle power output and to maximize cycle efficiency. A VAN turbine adds a degree of freedom to the manipulating parameters which enables controller to regulate cycle mass flow rate independent of evaporation pressure. Thus, evaporation pressure can be adjusted to have maximum cycle efficiency in different gas turbine loads while mass flow rate is being regulated to watch for heat exchangers pinch point temperature limitation. Condensing pressure is controlled by adjusting cooling water temperature. As condensation takes place in the two-phase saturation region, condensing pressure is the saturation pressure in cooling temperature. Working fluid leaves the condenser toward receiving tank in liquid saturation state. A thorough search is conducted on all possible evaporation pressure and mass flow rates to find best efficiency point for each gas turbine load. One should note that the optimum evaporation pressure in part load is not necessarily equal to the design value since turbine isentropic efficiency decays as vane setting angel changes. Figure 2b shows flow chart diagram for searching and selecting best cycle in VAN turbine part load strategy. A simple plane search is used since small search region do not require advance optimization methods.

Temperature difference between cold and hot stream in the heat exchangers are considered as a constraint that limits range of operation. Cold and hot stream temperature difference in the preheater cold side, preheater hot side (evaporator cold side) and the superheater hot side is checked to watch heat transfer limitations. Heat exchanger performance is calculated based on effectiveness coefficient and pressure loss. The effectiveness and pressure loss changes in off design conditions, but calculation of their exact value can be omitted here. The reason is that off-design operation of ORC in this context includes just part loads. In which, mass flow rate and heat transferred in the heat exchangers are always lower than design values. Thus, heat exchanger part load performance coefficients are substituted with design values which are the worst case of operation. This will simplify modeling and reduces calculations.

3 RESULTS AND DISCUSSION

An organic Rankine bottoming cycle is designed for two cases of gas turbine topping cycle (cases A and B). An ORC is designed and assessed for both topping cycle cases. Off-design performance of ORC and combined cycle are investigated with the baseline (SP) and proposed (VAN turbine) control strategies. Improvement made through presented VAN turbine logic is then compared with the baseline.



a) Sliding pressure control logic   b) VAN turbine control logic

Figure 2: Flow chart of selecting optimum cycle

3.1 Design point

On-design characteristics of the gas turbine topping cycle were tabulated for the selected gas turbines in Table 1. Bottoming cycle is designed according to requirements and practical limitations offshore. Figure 3 shows results of the parametric study conducted on condensing and evaporation temperatures to find the best possible cycle efficiency while not exceeding design limitations. Larger evaporation to condensing temperature ratio makes better efficiency for the cycle (Figure 3 b), but it requires higher pressure ratios across the turbine which increases turbine weight and size.

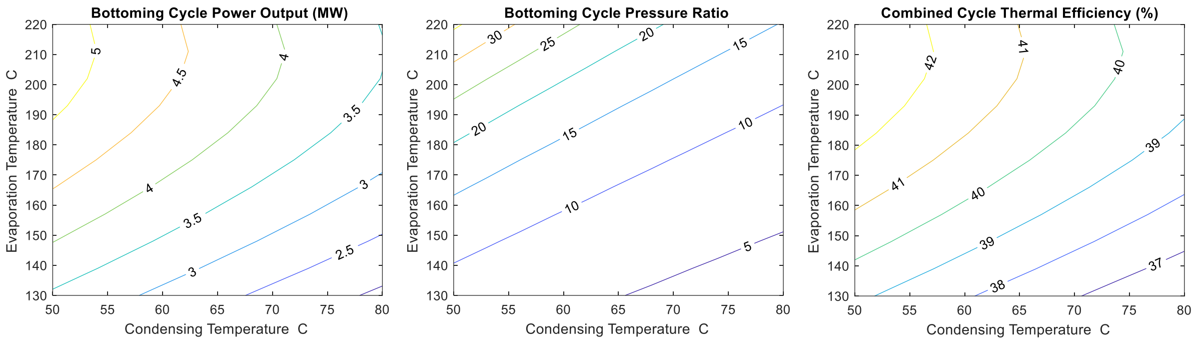
Figure 3 shows that for a specific condensing temperature, net power output and efficiency increases with evaporation temperature. However, the evaporation temperature is limited to the working fluid thermal decomposition temperature in a Rankine cycle (Rajabloo, Bonalumi, and Iora 2017). Turbine pressure ratio assigned to each design choice is depicted in Figure 3b. In practice, a radial inflow turbine design may have a pressure ratio of 9 or 27 with two or three stages, respectively. Although radial inflow turbines are suggested by many authors for working with organic fluids in offshore applications, but axial turbines can also be used if chosen. One should note that too high pressures require heavier facilities and reduces safety level of the system. On the other hand, too low pressure-ratios give poor performance to the cycle. The pressure ratio between condensation and evaporation should be practical so that could be handled by the turbomachinery components.

Setting heat exchanger pinch point limit to 10°C, optimum ORC mass flow rate is selected 54.0 kg/s and 51.5 kg/s for case A and B, respectively. Higher specific power (power per mass flow rate) means more compact designs for a required power or more power available in an available space. So, the mass flow rates are selected as low as possible for the specific power output. Mass flow rates that lead to approach temperatures less than allowed heat exchanger pinch point cannot be selected though.

Cyclopentane is selected as working fluid due to favorable pressure values in temperature working range, less hazardous level, and less environmental impact. Cyclopentane properties are tabulated in Table 2. A 5℃ superheating is applied to evaporator discharge to assure pure vapor phase entering the turbine. Superheating occurs in a gas-gas heat exchanger and occupies a large space which is not desired in offshore applications. Design point properties of the designed cycle are presented in the Table 2.

3.2 Part load operation

Figure 4 illustrates optimized ORC turbine inlet temperature and pressure in SP and VAN turbine control logics. The temperature and pressure are selected in each GT load such that maximum possible cycle efficiency is achieved. The optimized values for both GT cases are illustrated in Figure 4.



a) ORC power b) ORC pressure ratio c) Combined cycle efficiency

Figure 3: Parametric study of a)ORC power, b)ORC pressure ratio, c)Combined cycle efficiency

Table 2: On-design properties of designed ORC and the working fluid
(Cyclopentane specifications are adapted from (Leonardo Pierobon et al. 2013).)

Cyclopentane		Cycle Performance			Case A	Case B
Health/ physical/ fire hazard	2/3/1	Evaporation Temperature [°C]	200	Power Output [MW]	5.37	4.93
GWP100* [kg (CO_2EQ)/kg]	3	Condensing Temperature [°C]	50	Evaporation Approach Temperature [°C]	20	81
T_c [K]	511.7	Pressure Ratio	27	Mass Flow Rate [kg/s]	56	51.5
P_c [kPa]	4515	Superheating [°C]	5	Air Exhaust Temperature [°C]	60	59
$T_{auto\ ignition}$ [K]	580	Thermal Efficiency [%]	19.29	Condensation Approach Temperature [°C]	10	9
ρ_{oc} [mol/L]	70.13			Combined Cycle Efficiency [%]	44.76	45.68

* The global warming potential is calculated considering a time interval of 100 years.

Part load performance of the gas turbine is illustrated in Figure 1b for both cases. As shown, exhaust temperature of LM1800 case is less decreased in part loads relative to SGT 500 case. This is a favorable characteristic for a gas turbine working with bottoming cycles. Lower GT Exhaust temperature in part loads may degrade capacity or efficiency of the bottoming cycle since it forces the ORC to work far from optimum operational point due to approach temperature limitation in the heat exchangers.

Figure 5a and Figure 6a show the organic Rankine cycle efficiency in part loads. While efficiency drops to 75% of design value at 40% load with SP control strategy, it remains above 92% of design efficiency with the suggested VAN turbine strategy until 40% loads. This means 30% (or 70%) compensation of bottoming cycle efficiency loss in GT load of 30% for case A (or B). Reduced heat available from gas turbine at loads lower than 50% requires cycle mass flow rate to be reduced in the bottoming cycle. Consequently, evaporation pressure decays with reduction of mass flow in the SP approach because of radial inflow turbine characteristics. In contrast to this, the VAN turbine allows reduction of mass flow rate without considerable change in the turbine inlet pressure. Part load pressures and temperatures of the cycle remain close to design values, thus allowing for having an almost constant cycle efficiency for the bottoming cycle at part loads.

Figure 5b and Figure 6b show plant efficiency in part loads with three cycle configurations: simple cycle gas turbine, combined cycle with SP ORC and combined cycle with VAN turbine ORC. The bottoming cycle has improved plant efficiency at 50% load from 26% (simple cycle gas turbine) to 33.5% (SP controlled combined cycle) or 35% (VAN turbine controlled combined cycle). The improvement in design point full load is 10 efficiency points with the presented combined cycle. Assuming 250 g/kWh CO_2 emission, this means a 10-kiloton reduction of CO_2 emission yearly for a single 15 MW gas turbine working at full load yearly. Proposed VAN turbine logic improved 1.5 efficiency points for the combined cycle in comparison to the sliding pressure controller at 30% GT load (case B). The outcome is 4% relative reduction in CO_2 emission yearly in a combined gas turbine cycle which is mostly working at loads equal or lower than 50%. The VAN turbine part load strategy shows 35% and 20% relative higher bottoming cycle power than SP in 50% load for case A and B, respectively. (Figure 5c and Figure 6c). The increment is almost the same for both control logics.

Although bottoming cycle has the same design point efficiency value with both gas turbines in cases A and B, but part load efficiency decreases more rapidly for SGT500, Case A. The reason is that GT exhaust temperature decays in part loads in SGT500. Consequently, exhaust heat is delivered in lower temperatures to the bottoming cycle. In such cases, conventional SP controller limits evaporation temperature and put plant in a less efficient thermodynamic cycle.

4 CONCLUSION

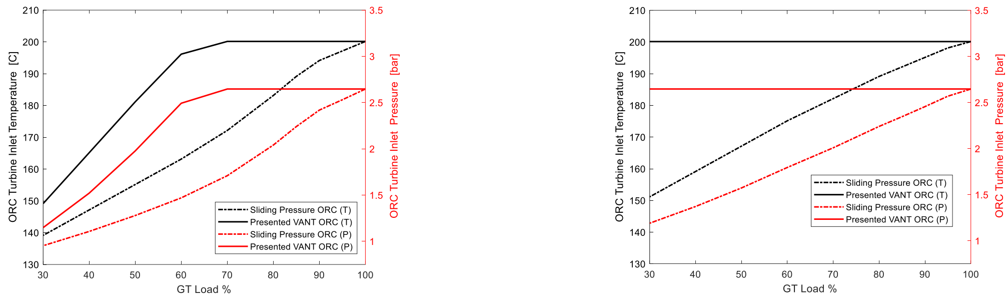
An organic Rankine bottoming cycle is designed for harvesting gas turbine waste heat and improving power generation efficiency in offshore oil platforms.

A VAN turbine control strategy for the ORC is proposed to enhance the part load efficiency. With suggested VAN turbine part load strategy, the bottoming cycle efficiency is kept above 95% of design value in the range of 40% to 100% GT load (case B). This is achieved while it falls to 80% of design value with the SP control strategy. As offshore gas turbines and combined cycles are expected to work mostly in 50% load or lower, this part load improvement can get more attention in application.

Two gas turbine cases were studied in combination with proposed solutions. The GT with constant exhaust temperature pattern showed better plant efficiency in part loads than GT with decreasing exhaust temperature pattern. For SGT 500, combined cycle efficiency falls to 66% of design value in

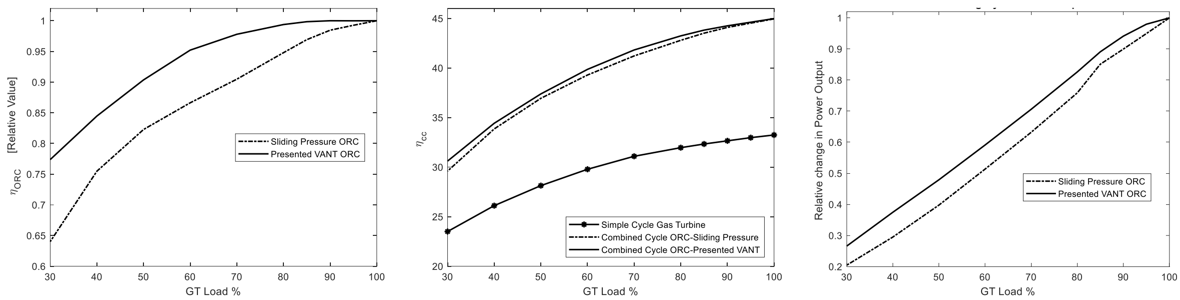
30% gas turbine load. While for GE LM1800, combined cycle efficiency falls to 77% of design value in 30% load. This suggests that gas turbines with constant part load exhaust temperature are a better choice for offshore ORC combined cycle applications in the sense of part load efficiencies.

The ORC bottoming cycle technology for offshore oil and gas installations is not fully commercially available yet. However massive efforts are seen in on-going developments. Design and assessment of the presented control strategy will contribute to achieve an up to date understanding of ORC applicability based on latest requirements in offshore oil and gas industry. However, detail design development and study of dynamic behavior of the control logic are left for future works.



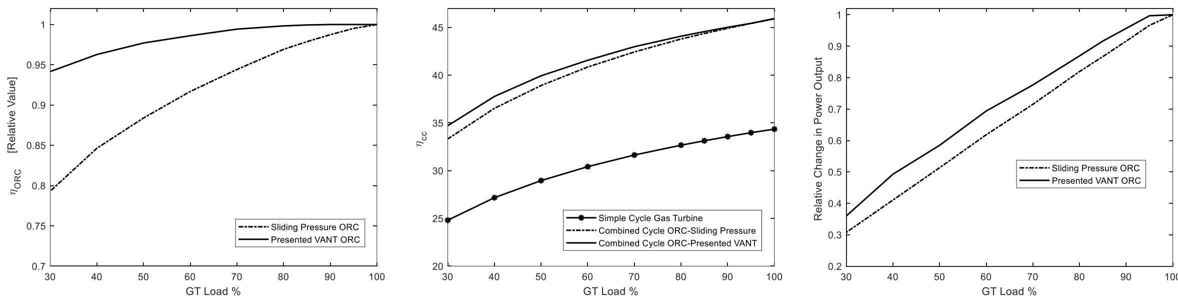
a) Case A GT SGT500 b) Case B GT LM1800

Figure 4: ORC turbine inlet temperature and pressure in off-design conditions



a) ORC efficiency b) Combined cycle efficiency c) ORC power output

Figure 5: Part load performance case A



a) ORC efficiency b) Combined cycle efficiency c) ORC power output

Figure 6: Part load performance case B

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