A DUAL LOOP ORGANIC RANKINE CYCLE UTILIZING BOIL-OFF GAS IN LNG TANKS AND EXHAUST OF MARINE ENGINE

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ABSTRACT

In this study, a dual loop ORC system for LNG carrier is analysed. First ORC system recovers heat from engine exhaust and releases heat to sea water. Second ORC recovers heat from exhaust sequentially and releases heat to boil-off gas. Due to very low-temperature of boil-off gas (around -160 ^{o}C), condensation temperature could be very low. Possible eight different refrigerants are screened and R218 refrigerant is revealed most suitable for the proposed cycle. Thermodynamic analysis shows maximum output is occurred at optimum evaporating temperature.

1. INTRODUCTION

In recent years, international natural gas market is gradually increased. Among fossil fuel, natural gas emits relatively a small amount of pollutants during the combustion (20% lower CO_2, 85-90% lower NO_x, hardly SO_x) compared to petroleum and coal. Natural gas is intensively reserved in specific areas and the producing areas are far from demand area like petroleum. Due to the significant investment cost for pipeline, liquefied form of natural gas is preferred for distances of above 2,000 km to be supplied by using transportations such as LNG carrier (Dobrota *et al.*, 2013).

During LNG supply process, boil-off gas (BoG) including evaporated volatile components (methane and nitrogen) is significantly generated. About 4-6% of the total LNG freight is converted in the form of BoG during typical 20-day voyage, and the generated BoG raises the pressure of the storage tank. In order to prevent potential risks, BoG needs to be removed from the tank properly. BoG is usually burned, re-liquefied, or re-used as fuel on board. Prior to the last-generation diesel engine, BoG was used as fuel for steam propulsion engine with the very low cycle efficiency. Since adopting diesel engine, BoG is re-liquefied with using Claud cycle (Moon *et al.*, 2007) for on board application. Very recently, BoG is re-considered to be used as a fuel in marine engine because (1) natural gas price is relatively competitive, (2) countries and harbors start to restrict the amount of diesel fuel consumption, and (3) dual fuel engine (DF engine) is introduced in market (e.g. Wärtsilä DF engine series) which has fuel flexibility. DF engine is preferentially adopted in LNG carrier where fuel natural gas could be easily obtained. In use of BoG as fuel, BoG of -160 ^{o}C is preheated to 0 ~ 25 ^{o}C using fuel preheater, and thus the useful cold energy is discarded during fuel preheating. The cold energy could be used for additional power generation as heat sink. In land application, available power generation cycles to be coupled with the LNG vaporization process in a Spanish LNG terminal (Querol *et al.*, 2011).

In marine application, the exhaust heat is used for operating turbo charger and steam service. Since released exhaust gas temperature is ranged about from 180 to 250 ^{o}C, the potential of useful thermal energy is still abundant for additional power generation. Hence, waste heat recovery of internal

combustion engine has been widely studied (Bombarda and Invernizzi, 2010, Vaja and Gambarotta, 2010, Wang *et al.*, 2011, Choi and Kim, 2013) with adopting ORC system.
In this study, dual loop ORC system is proposed for utilizing both BoG cold energy and exhaust heat. Engine exhaust is sequentially utilized and each ORC systems use sea water and BoG as heat sink. High-temperature ORC (HT-ORC) loop is designed for use of R245fa refrigerant as working fluid, and low-temperature ORC (LT-ORC) loop is simulated with various working fluids.

2. DUAL LOOP ORC SYSTEM

2.1 System Description

Cycle design and simulation were conducted for Wärtsilä DF50 engine at 100% load. The key operating condition for cycle simulation is shown in Table 1. Marine engine exhaust gas is used for turbo charger and steam generation. In this study, a DF engine, a turbocharger and a steam generator are considered as single engine unit, and the exhaust gas temperature from the engine unit is assumed as 230 ^{o}C.
Although BoG generation rate and temperature depends on the controlled tank pressure, those are assumed as 0.7 kg/s (engine fuel consumption rate) and -160 ^{o}C under the BoG pressure of 1 bar. The BoG is assumed as pure methane, because 99% of BoG is composed of methane.
Figure 1 shows the schematic of proposed dual loop ORC system. HT-ORC operates using engine exhaust gas of 230 ^{o}C as heat source and sea water as heat sink. For LT-ORC operation, BoG and engine exhaust gas passed through HT-ORC are used as heat sink and heat source, respectively. The exhaust gas with the temperature of 75 ^{o}C is introduced into the evaporator of LT-ORC sequentially. Then, the rest of the thermal energy in the exhaust gas is used for raising the BoG temperature up to 25 ^{o}C because the appropriate temperature of the supply fuel is required for the operation of DF engine. The heat exchanger (HX1) in LT-ORC loop is acting as the BoG preheater and the condenser which is utilizing cold energy of BoG. Second heat exchanger (HX2) is an additional preheater which uses the rest of the heat in the exhaust gas passed through LT-ORC evaporator.

Table 1: Engine operating conditions

Engine output	17.1 MW
Air flow rate at 100% engine load	27.5 kg/s
Exhaust gas flow rate at 100% load	28.2 kg/s
Temperature after engine unit at 100% load	230 ^{o}C
BoG flow rate at 100% engine load	0.7 kg/s
BoG temperature	-160 ^{o}C
BoG pressure	1 bar

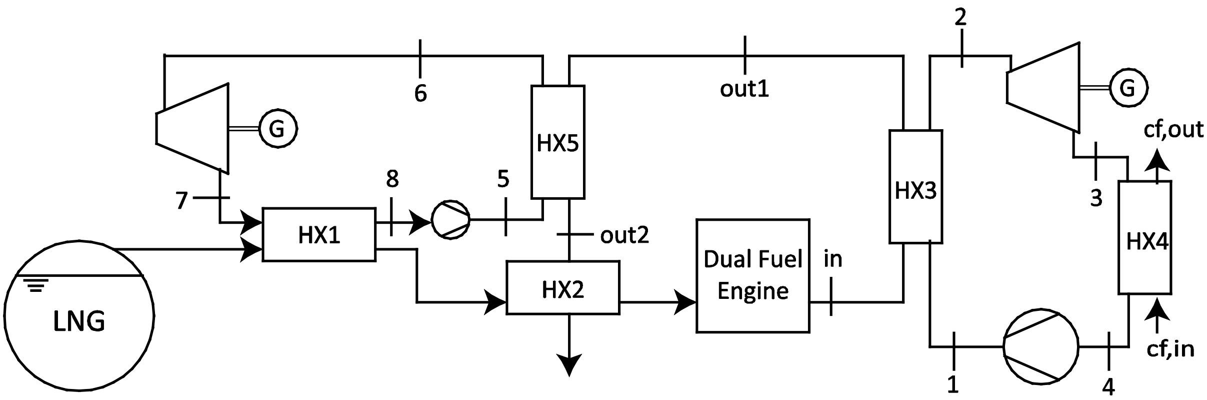


Figure 1: Dual loop ORC system for LNG carrier

2.2 System modeling

2.2.1. Turbine

$\dot{m}_{ORC}$ is working fluid mass flow rate and h is the enthalpy. The power output $\dot{W}_t$ and isentropic turbine efficiency η_t are

$$\dot{W}_{turb1} = \dot{m}_{o1}(h_3 - h_4) \tag{1}$$
$$\dot{W}_{turb2} = \dot{m}_{o2}(h_7 - h_8) \tag{2}$$
$$\eta_{turb1} = (h_3 - h_4)/(h_3 - h_{4s}) \tag{3}$$
$$\eta_{turb2} = (h_7 - h_8)/(h_7 - h_{8s}) \tag{4}$$

2.2.2. Condenser

The condenser heat duty $\dot{Q}_c$ are

$$\dot{Q}_{cd1} = \dot{m}_{o1}(h_4 - h_1) = \dot{m}_{cf}\left(h_{cf,in} - h_{cf,out}\right) \tag{5}$$
$$\dot{Q}_{cd2} = \dot{m}_{o2}(h_8 - h_7) = \dot{m}_{BoG}\left(h_{BoG,in} - h_{BoG,out}\right) \tag{6}$$

2.2.3. Pump

The consumed power $\dot{W}_p$ and isentropic efficiency η_p are

$$\dot{W}_{pp1} = \dot{m}_{ORC1}(h_2 - h_1) \tag{7}$$
$$\dot{W}_{pp2} = \dot{m}_{ORC2}(h_6 - h_5) \tag{8}$$
$$\eta_{pp1} = (h_{2s} - h_1)/(h_2 - h_1) \tag{9}$$
$$\eta_{pp2} = (h_{6s} - h_5)/(h_6 - h_5) \tag{10}$$

2.2.4. Evaporator

The heat duty of evaporator $\dot{Q}_e$ are

$$\dot{Q}_{ev1} = \dot{m}_{o1}(h_3 - h_2) = \dot{m}_{in}c_{p,in}(T_{in} - T_{out1}) \tag{11}$$
$$\dot{Q}_{ev2} = \dot{m}_{o2}(h_7 - h_6) = \dot{m}_{in}c_{p,in}(T_{out1} - T_{out2}) \tag{12}$$

$c_{p,in}$ is heat source specific heat.

2.3 System efficiency modeling

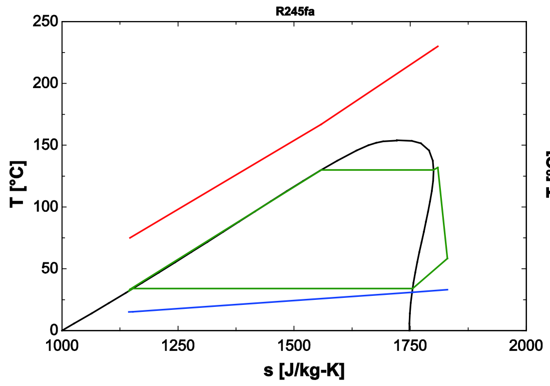
Thermal efficiency can be defined as

$$\eta_{cyc1} = (\dot{W}_{t1} - \dot{W}_{p1})/\dot{Q}_{e1} \tag{13}$$
$$\eta_{cyc2} = (\dot{W}_{t2} - \dot{W}_{p2})/\dot{Q}_{e2} \tag{14}$$

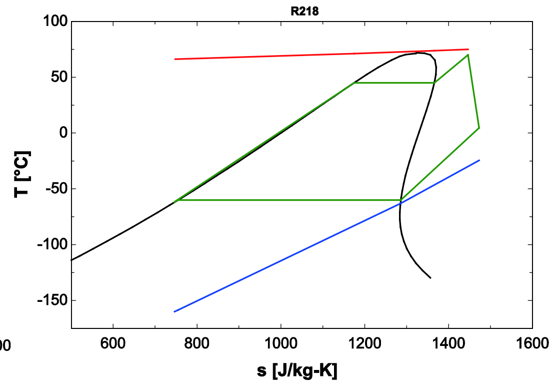
3. RESULTS AND DISCUSSION

3.1 Cycle analysis

Figure 2 shows T-s diagram for the proposed dual loop ORC system. R245fa refrigerant is adopted as working fluid for HT-ORC which gives moderate performance characteristics and has proper material safety. HT-ORC specifications are shown in Table 2. Cycle maximum pressure is set to 20 bar due to high pressure gas regulation. HT-ORC produces 651.8 kW power output with 14.6 % thermal efficiency.



(a)



(b)

Figure 2: T-s diagram of (a) HT-ORC and (b) LT-ORC

Table 2: HT-ORC specifications

Refrigerant	R245fa
Evaporating temperature	130 °C
Condensing temperature	34 °C
Pressure ratio	11.5
Mass flow rate	18.1 kg/s
Power output	651.8 kW
Turbine efficiency	0.85
Pump efficiency	0.70
Cycle efficiency	0.146
Cooling source	Sea water
Cooling source temperature	15 °C
Cooling source mass flow rate	50.7 kg/s

3.2 Fluid selection for LT-ORC

Table 3 shows the physical properties, safety data and environmental data of eight screened refrigerants. All screened fluids has no ozone depleting potential, whereas three fluids (R143a, R125 and R218) shows relatively high global warming potential. Table 4 shows simulation condition for LT-ORC loop. The thermodynamic simulation results for LT-ORC system are shown in Table 5. Evaporating and condensing conditions, mass flow rate, volume flow rate, quality at turbine exit, power output, thermal efficiency and pressure ratio are shown in Table 5. LT-ORC with R218 refrigerant produces power output of 49.8 kW and recovers heat of 277 kW used in fuel preheating. Following factors are considered for suitable refrigerant selection for LT-ORC:

(1) The system power output shall be as large as possible.
(2) Dry expansion is preferred.
(3) The system volumetric flow rate shall be as small as possible.
(4) The system mass flow rate shall be as small as possible.
(5) Pressure ratio be within acceptable range.

From Table 5, R218 has the highest power output, whereas, ammonia and R152a have the highest cycle efficiency. Ammonia also has the smallest mass flow rate and R41 has the smallest volume flow rate and pressure ratio. Ammonia, R152a and R134a shows wet expansion and higher pressure ratio compared to other refrigerants. It is obvious that there is no suitable refrigerant satisfying every design considerations at the same time.

LT-ORC loop with R41 produces much less power output. LT-ORC with ammonia, R152a and R134a has high power output, however pressure ratio is very high and wet expansion is required. LT-ORC with propane, R14a and R125 produce lower power output, however volume flow rate and pressure ratio are low. R218 is the most suitable for LT-ORC due to the highest power output and dry expansion characteristics with not bad pressure ratio and volumetric flow rate.

Table 3: Physical, safety and environmental data for the working fluids.

	T_{bp} (°C)	T_{fr} (°C)	T_c (°C)	P_c (bar)	$T_{sat@20bar}$ (°C)	Safety data	ODP	GWP
Propane	-42.11	-187.62	96.74	42.51	57.26	A3	0	~ 20
R143a	-47.24	-111.81	72.707	37.61	43.75	A3	0	3800
R125	-48.09	-100.63	66.023	36.18	39.82	A1	0	2800
R41	-78.31	-143.33	44.13	58.97	-0.79	-	0	97
Ammonia	-33.33	-77.65	132.25	113.33	49.35	B2	0	<1
R152a	-24.02	-118.59	113.26	45.17	72.65	A2	0	140
R134a	-26.07	-103.3	101.06	40.59	67.48	A1	0	<1
R218	-36.79	-147.7	71.87	26.4	59.31	A1	0	8600

Table 4: LT-ORC simulation conditions

Turbine Efficiency	0.85
Pump Efficiency	0.7
Turbine inlet temperature	70 °C
Condensation temperature	-60 °C
Ambient temperature	25 °C

Table 5: Thermodynamic results for the LT-ORC system

	T_{cd} (°C)	P_{cd} (bar)	T_{ev} (°C)	P_{ev} (bar)	M (kg/s)	V (m^3/s)	x	W_{net} (kW)	η_{cyc}	r_p
Propane	-60	0.43	57.26	20.0	0.322	0.305	1.00	46.0	0.239	46.9
R143a	-60	0.53	43.75	20.0	0.606	0.255	1.00	44.0	0.223	37.7
R125	-60	0.54	39.82	20.0	0.836	0.262	1.00	44.8	0.214	36.8
R41	-60	2.52	-0.7909	20.1	0.310	0.064	1.00	32.3	0.177	7.96
Ammonia	-60	0.22	49.35	20.0	0.118	0.467	0.84	47.9	0.250	91.3
R152a	-60	0.15	65	16.9	0.430	0.701	0.92	48.1	0.250	112.4
R134a	-60	0.16	65	18.9	0.609	0.646	0.98	45.9	0.241	118.7
R218	-60	0.31	59.31	19.8	1.253	0.473	1.00	49.8	0.207	64.1

3.3 Thermodynamic analysis for LT-ORC system

In this chapter, thermodynamic analysis of LT-ORC for different evaporation temperature and condensation temperature is presented. Propane, R143a, R125 and R218 refrigerants are used as working fluids.

Figure 3 and Figure 4 show the change in the power output and cycle efficiency according to change of evaporation temperature. Turbine outlet pressure and turbine inlet pressure are fixed during the simulation. In all refrigerants, maximum power output is occurred which indicate optimum operating condition exist. On the other hand, maximum thermal efficiency is occurred only for R125. Thermal efficiency of the other refrigerants are linearly increased as evaporating temperature is increased. In all cases, the point of maximum thermal efficiency and maximum power output is not matched.

Figure 5 and Figure 6 show the power output and cycle efficiency of the variation of the condensation temperature at the optimum evaporation temperature. Increase in condensation temperature results in linear decrease in power output and thermal efficiency.

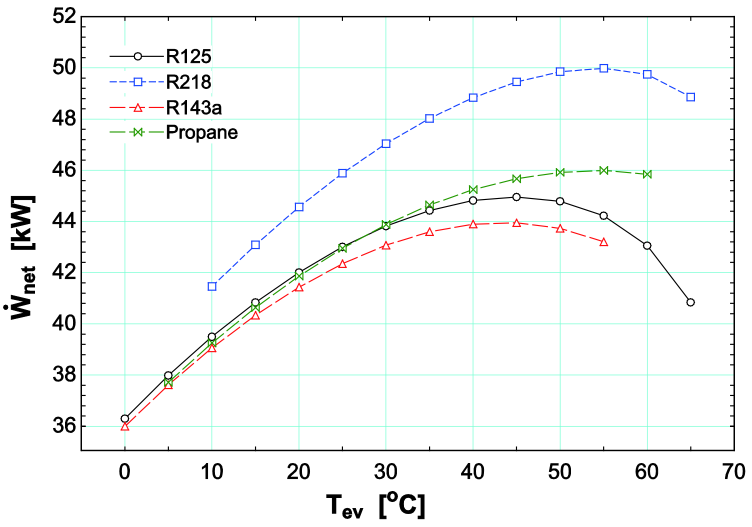


Figure 3: Power output of LT-ORC corresponding to evaporation temperature

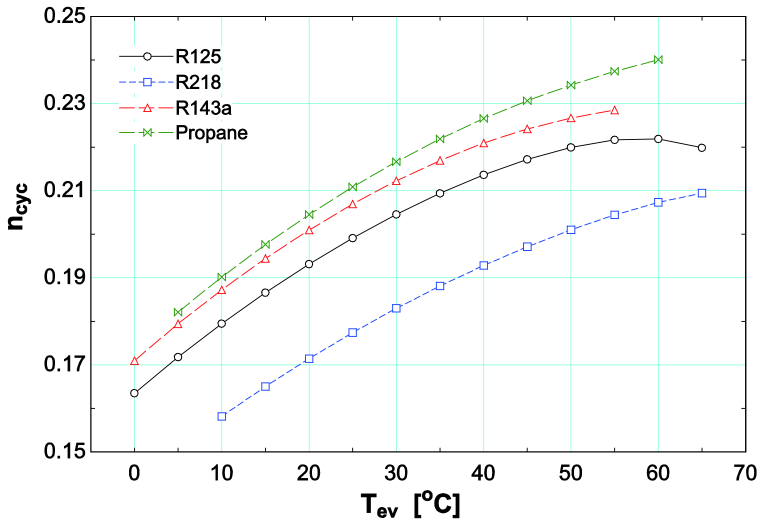


Figure 4: Cycle efficiency of LT-ORC corresponding to evaporation temperature

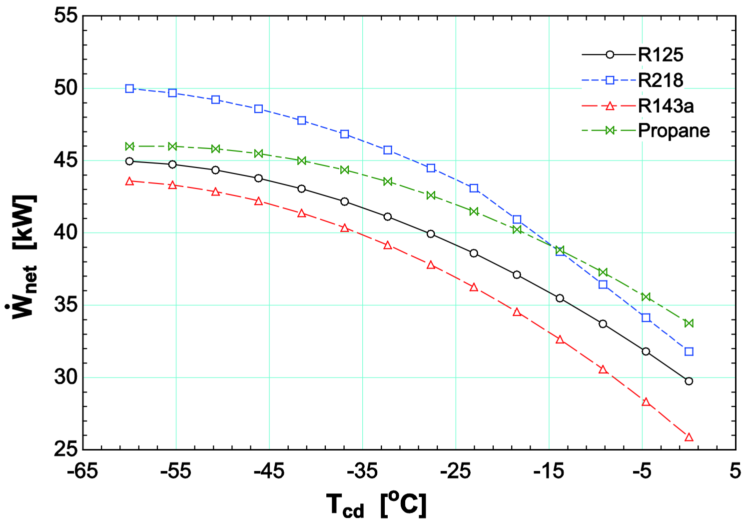


Figure 5: Power output of LT-ORC corresponding to condensing temperature

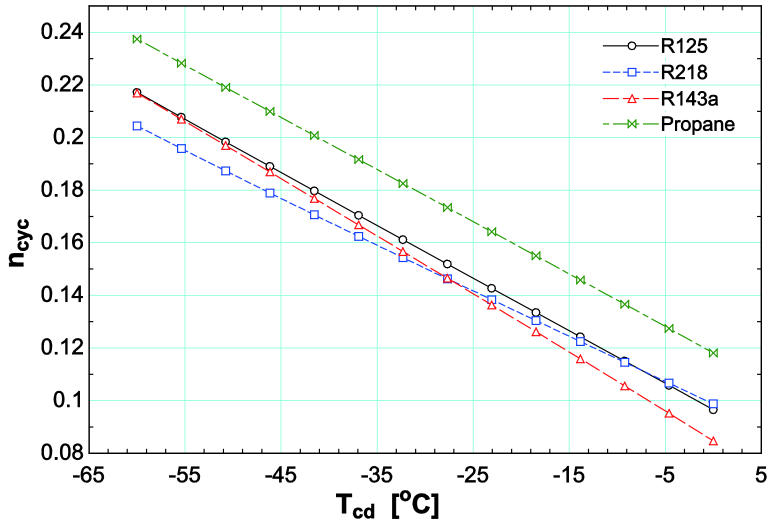


Figure 6: Cycle efficiency of LT-ORC corresponding to condensing temperature

4. CONCLUSIONS

A novel dual loop ORC system for waste heat recovery of LNG carrier has been described and analyzed. The proposed ORC system is intended to use the waste heat in engine exhaust gas and the cold energy in BoG. In order to investigate suitable working fluid and optimum operating condition for LT-ORC system, eight refrigerants including R218, R143, R125, propane were simulated with varying evaporation and condensation temperature. By using R218 refrigerant in LT-ORC loop, the highest power output of 49.8 kW was achieved from the waste heat in the exhaust gas passed through the HT-ORC loop. Also, the amount of additional waste heat recovery of 277 kW is expected during fuel preheating process.

NOMENCLATURE

BOG	boil-off Gas	(–)
c_p	specific heat	(kJ/K)
GWP	global warming potential	(–)
η	efficiency	(–)
M	mass flow rate	(kg/s)
ODP	ozone depleting potential	(–)
P	pressure	(bar)
Q	heat	(kW)
r_p	pressure ratio	(–)
T	temperature	(°C)
V	volumetric flow rate	(m^3/s)
W	work	(kW)
x	quality	(–)

Subscript

1	HT-ORC
2	LT-ORC
c	critical
cf	cooling fluid
cyc	cycle
cd	condenser
ev	evaporator
fr	freezing

in	inlet
o	organic Rankine cycle
out	outlet
pp	pump
s	isentropic
sat	saturation
turb	turbine

REFERENCES

Bombarda, P., Invernizzi, C. M., Pietra, C., 2010, Heat recovery from Diesel engines: A thermodynamic comparison between Kalina and ORC cycles, *Appl. Therm. Eng.*, vol. *30, no.* 2: p. 212-219.

Choi, B. C., Kim, Y. M., 2013, Thermodynamic analysis of a dual loop heat recovery system with trilateral cycle applied to exhaust gases of internal combustion engine for propulsion of the 6800 TEU container ship, *Energy*, vol. *58*: p. 404-416.

Dobrota, Đ., Lalić, B., Komar, I., 2013, Problem of boil-off in LNG supply chain. *Trans. Maritime Science*, vol. *2, no.* 02: p. 91-100.

International Energy Agency, 2013, *World Energy Outlook*

Moon J, Lee YP, Jin Y, Hong E, Chang H., 2007, Cryogenic refrigeration cycle for re-liquefaction of LNG boil-off gas, *In.International Cryocooler Conference, Cryostar The Cryostar Magazine*, p. 629-635.

Querol, E., Gonzalez-Regueral, B., García-Torrent, J., Ramos, A., 2011, Available power generation cycles to be coupled with the liquid natural gas (lng) vaporization process in a Spanish LNG terminal, *Applied Energy*, vol. 88: p. 2382-2390.

Vaja, I., Gambarotta, A., 2010, Internal combustion engine (ICE) bottoming with organic Rankine cycles (ORCs), *Energy*, vol. *35, no.* 2: p. 1084-1093.

Wang, E. H., Zhang, H. G., Fan, B. Y., Ouyang, M. G., Zhao, Y., Mu, Q. H., 2011, Study of working fluid selection of organic Rankine cycle (ORC) for engine waste heat recovery. *Energy*, vol. *36, no.* 5: p. 3406-3418.

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