Thermo-economic evaluation of ORCs for various working fluids

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Abstract

An inclusive component-level technical and economic assessment procedure for the general design and operating strategy of Organic Rankine Cycles (ORC) for use across major application categories (waste heat recovery, solar thermal, geothermal) and sub-MW scales can be an important tool for leveraging the cost-effective deployment of low and medium temperature power cycles. Previous analyses and design approaches tended to focus on thermodynamic efficiency rather than financial performance. To bridge this gap, a general thermo-economic optimization of sub 500 kWe ORCs is developed using a 7-dimensional design space with minimum investment cost per unit of nameplate electricity production as an objective function. Parameters used include working fluid, heat source temperature, pinch in condenser, boiler (HEX) and regenerator, expander inlet pressure and air cooled condenser area. Optimized power block configurations are presented for the application of ORCs with waste or “free” heat sources and solar heat input for power scales of 5, 50 and 500 kWe to facilitate rapid selection of design parameters across a wide range of thermal regimes. While R152a yields the lowest cost ORCs in the case of the former, isopentane is found to be more cost effective in the latter case for heat source temperatures between 125 and 275 °C.

25 *Keywords:* Organic Rankine cycles, thermo-economics, waste heat application, solar heat
26 application

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28 **Nomenclature**

29 c cost USD

30 p pressure bar

31 1 to 6 states in ORC

32

33 *Subscript*

34 sp specific

35 wf working fluid

36

37 *Superscript*

38 ' states on real thermodynamic cycle

39

40 *Abbreviations*

41 ACC Air Cooled Condenser

42 ASHRAE American Society of Heating, Refrigerating, and Air-Conditioning Engineers

43 DNI Direct Normal Irradiance (Kwh)

44 HTF Heat Transfer Fluid

45 INR Indian national Rupee

46 ODP Ozone Depletion Potential

47 ORC Organic Rankine Cycle

48 USD United States Dollar

49 VFD Variable Frequency Drive

50

1. Introduction

Increasing electricity demand necessitates the utilization of diverse energy resources, including the exergy potential of low temperature ($< 300\text{ }^{\circ}\text{C}$) heat streams such as geothermal, waste heat from industrial processes or low concentration factor (< 100) solar collector systems. At sub steam-Rankine temperatures, the organic Rankine cycle (ORC) is a scalable thermodynamic solution for converting thermal resources [1–4] that can be configured to meet variable project demands. These resources are widely distributed in space, and consequently investment decisions for such inherently distributed generation projects require costly site and application-specific engineering and economic analysis. The design of ORC equipment optimized for unique resources and applications also involves a number of trade-offs and adds cost to projects [5].

When designing an ORC to project specific requirements, the working fluid is a key consideration. Various working fluids have been proposed in the literature with optimized selections based on thermodynamic performance using the definition of either the first law [6–11] or the second law [12–15] of thermodynamics. However, when the strategic objective is minimized investment cost for a specific level of power output, the corresponding configuration and design operating conditions need not occur at the point of maximum efficiency as indicated by Quoilin et al. [16]. Work in [17–21] supports the above observation and underscores the need for a detailed thermo-economic evaluation of various ORC design parameters for comparison on an equivalent basis and generalizable across a wide range of ORC applications. In general, the thermo-economic approach used in these studies consists of low-order lumped parameter models that fail to capture the effect of the working fluid and operating conditions on the specific investment costs in ORCs. To address this deficiency a higher order thermo-economic model is necessary wherein the cost contribution of individual

components is coupled with detailed thermal models with implicit operational control strategies modulated for their impact on system level economic performance.

This work presents such a study wherein the key design parameters influencing the specific cost of power from an air-cooled ORC are identified and used to formulate a 7-dimensional space to search for the minimum investment costs for applications with a) waste heat and b) solar sources. Corresponding maps of operating parameters are generated to facilitate ORC engineers in the design of economic systems within constraints such as available heat source temperatures, maximum expander inlet pressures imposed, etc. Further, the effect of power scaling on these specific costs is evaluated for ORC capacities between 5 and 500 kW_e.

2. Cycle details

The physical system considered in this paper consists of an ORC energized by a heat transfer fluid (HTF) that could be heated in two ways, forming Case A and B as follows:

Case A: The HTF gains enthalpy from industrial waste heat or similarly relatively inexpensive sources (Fig. 1). Here, the minimization of costs for a specific power output (i.e. \$/kW_e) are driven by the ORC capital expenditure at its performance envelope. This case can also be extended to ORCs connected to geothermal sources if the cost associated with them is negligible compared to the ORC power block (i.e. near-surface sources such as hot springs or previously developed and amortized wells).

Case B: An ORC is coupled to a thermo-solar collector field via a thermal storage system (Fig. 2) that maintains steady output in the ORC throughout the day. Here, the thermal source is a parabolic trough collector with significant cost. The optimization strategy in this case would be to configure the ORC to minimize the total system cost for a specific power output.

Fig.1. Schematic of an ORC connected to a waste heat recovery source

Fig. 2. Schematic of an ORC connected to a solar source

2.1. ORC component details

The major components in a typical ORC include expander, heat exchangers (boiler, regenerator), air-cooled condenser (ACC), pumps, and variable frequency drives. For sub-MWe ORCs, positive displacement expanders are appropriate at the smaller scales where turbo-expanders suffer from high rotational speeds and lower isentropic efficiencies [22]. For the working fluid pump, a plunger type positive displacement device suitable for sub-MWe ORCs is coupled to a variable frequency drive to achieve the designed mass flow rates. The regenerator and boiler are modeled as plate heat exchangers which are limited to 60 bar and hence, the high side cycle pressure ($p_{3,wf}$) is restricted to 60 bar. These heat exchangers are modeled for a pressure drop of 1.5 % of their pressures at the inlets on both cold and hot side along the length of heat exchanger. Heat transfer coefficient and pressure drop calculations are performed using correlations from literature sources ([23] for single phase heat transfer, [24] for two phase heat transfer and [25] for heat transfer in supercritical regime) for sub- as well as trans-critical cycles. 1-D numerical approach is employed to model these exchangers, accounting for the variation in the thermo-physical properties of the working fluid during heat transfer. Temperature profile of the HTF in the boiler is determined by a procedure based on minimization of entropy generation in the boiler [26]. The expander chosen is scroll (connected to an electric generator) for which the overall isentropic efficiency is assumed to be 65% from thermal to electrical [27]. The cost function for scroll expanders is generated using a database of existing scroll compressors for HVAC applications. In the case of cycle volume ratio > 2.89 , two or more scrolls are used in series. For power scales above $> 50 \text{ kW}_e$, scroll expanders are used in parallel. The condenser is air cooled and designed in such a way that the overall system cost is optimized taking into account its fan power. Commercially available condenser unit sizes are used in the analysis with the smallest and the largest surficial area ranging from 0.5 to 9.2 m². In case of higher power levels requiring larger

131 surficial areas ($> 9.2 \text{ m}^2$), parallel units of condenser are considered. Modeling details of the
132 condenser can be found in the Masters thesis of Monifa Fela Wright [28]. Major instruments
133 considered in the power block include temperature and pressure sensors, flow meter, current
134 transducers, data acquisition system, data processing and control systems. Details of the cost
135 functions used in this analysis can be found in appendix A. Costs pertaining to the process
136 pipes and other accessories such as liquid receiver, lubricants and safety equipment are not
137 considered in this paper. We note the majority of the cost functions used in this study were
138 derived from commercial sources in the United States and Europe; the actual costs incurred in
139 an ORC plant may differ from the one presented in this paper. Although factors including
140 Engineering Procurement and Construction (EPC) and owners costs (land, permitting,
141 insurance, labour, taxes) depend on the project location, they are independent from the cost
142 implications of the various choices of ORC parameters considered here. Hence, we limit the
143 scope of this paper to capture the important cost trends and design and control variables
144 encountered in the thermal space that produce systemic cost differences.

146 2.2. Solar-ORC details

147 In the solar heat source case, the ORC described above remains identical with
148 additional system components added including thermal energy storage (TES) system and a
149 solar collector field. The storage type selected is pebble bed (efficiency assumed to be 85%)
150 which can be economical for ORC systems [29]. Modeling details of such a TES system can
151 be found in our previous work [30,31]. Parabolic trough collector efficiency is modeled as a
152 function of operating HTF temperatures [32]. Solar calculations are performed for Vernal
153 Equinox at the Tropic of Cancer (roughly the mean of extreme Indian latitudes) using the
154 ASHRAE clear sky model. The clearness factor is selected to achieve an annual average DNI
155 of $5.3 \text{ kWh/m}^2\text{-day}$ [33]. Annual average ambient temperature on such a location is $30 \text{ }^\circ\text{C}$.

The thermal energy storage capacity and solar field are sized to ensure steady output of the plant at an ORC capacity factor of 0.5 throughout the year. Whenever the HTF operating temperature is less than 180 °C, Ethylene Glycol is used as an HTF; otherwise Therminol VP1 is used.

3. Methodology

Detailed component level models are developed and integrated in Matlab to simulate the systems described in cases A and B. For the given power capacity, investment cost of these systems is highly dependent on the operating conditions in their power blocks. Optimization proceeds by varying these operating conditions within the 7-dimensional space formed by working fluid type, HTF inlet temperature (heat source temperature), pinch in condenser, regenerator and HEX (boiler), expander inlet pressure and condenser area. The objective function is to minimize the specific cost of the system (C_{sp}) taking into account parasitic losses of working fluid and HTF pumping and fan power in condenser.

$$C_{sp} = \frac{\text{Total investment cost}}{\text{ORC output} - \text{parasitic losses}} \quad (1)$$

The algorithm to optimize the above power block is given in Fig. 3. The parametric sweep was performed for 3 different power scales, 5, 50 and 500 kW_e. Above 500 kW_e, the expected technical setup changes, (for example, positive displacement expanders could be replaced by turbines and plunger pumps by centrifugal, etc.), hence this study is confined to small scale ORCs ~ 500 kW_e or below.

Fig. 3. Algorithm to optimize the ORC system on thermo-economic basis

4. Results and discussions

Optimum C_{sp} values are calculated for 19 zero-ODP and positive condenser pressure (also equal to saturation pressure corresponding to 45 °C) working fluids for HTF supply temperatures between 125 and 275 °C in steps of 25 °C for the power capacities of 5, 50 and 500 kWe. These working fluids are listed in Appendix A. To facilitate visualization of the data, only the 8 highest performing fluids are presented here (including isopentane and R245fa, two widely used fluids in ORCs). For some fluids, equations of state in the REFPROP [34] property data base were extrapolated beyond their specified limits to enable a

one to one comparison for the given HTF temperature, however, operating temperatures considered are well below the fluid thermal stability temperatures as suggested by Morgan and Munday [35], Angelino and Invernizzi [36], Hidaka et al. [37], Moon et al. [38] and Ito et al. [39].

4.1 ORCs connected to waste heat sources (Case A)

For a 5 kWe ORC, the optimum C_{sp} values are presented in Fig. 4 against HTF_{in} temperature for different working fluids. At any temperature, C_{sp} is found to be a strong function of working fluid type with a larger variance at lower temperatures. Isopentane yields the highest C_{sp} whereas R152a and R161 compete at the low end. The trend for C_{sp} among different fluids is consistent at different temperatures. Certain fluids exhibit a sudden increase in cost with increase in HTF_{in} temperature due to transitioning from sub- to trans- critical cycle or due to excess heat duties of boiler and condenser in cases where regenerator is not employed. While there is a significant potential for reducing C_{sp} with temperature in the case of isopentane, the same is not true in the case of R152a or R161 where the reduction in cost is just 10 % with increase in HTF temperature from 125 to 275 °C. Further, Fig. 4 illustrates that an increase in efficiency is not always economically advantageous.

Fig. 4. C_{sp} and cycle efficiency for various working fluids against $T_{HTF,in}$ temperatures for a 5 kWe ORC.

The corresponding map of optimum operating conditions is given in Fig. 5 wherein the volumetric expansion ratio (v_4/v_3), expander inlet pressure ($p_{3,wf}$), pinch in condenser and HEX, condenser area and regenerator fractional cost are plotted. It is noteworthy that the lowest C_{sp} values are observed at the lowest volumetric expansion ratios in the expander at higher expander outlet pressures. Volume ratios as high as 20 are observed, requiring a multi stage expansion process or non-conventional scroll expanders [40]. Also, higher pinch temperatures can be tolerated at higher $T_{HTF,in}$ temperatures. Optimized condensers converge on the smallest size possible in the domain of standardly available air-cooled condensers considered. Further, improvement in efficiency via incorporating a regenerator is not economically justified for 5 kWe scale ORCs and hence, most of the optimized ORCs do not incorporate the regenerators. The rationale behind the exclusion of a regenerator at 5 kWe is justified by comparing the best cases with and without regenerator for minimum C_{sp} as shown in Fig. 6. Also, decreasing volumetric flow rate at the expander exhaust is found to be a key parameter for lower C_{sp} in cases of high condenser pressure fluids with the exception of R134a which should have yielded a cost equivalent to that of R152a. In case of R134a, the

penalty for lack of regenerator is relatively higher than for R152a and R161, where expansion significantly reduces the potential for heat recovery in the regenerator.

The above analysis is extended to power scales of 50 and 500 kW_e and is represented in Figs. 7 to 10. Unexpectedly, the lowest C_{sp} values exhibit a jump at HTF temperature of 175 °C. At this temperature, power blocks without regenerator demand higher heat duties in HEX and condenser and hence, optimum power blocks begin to employ a regenerator, resulting in higher C_{sp} .

Fig. 5. Corresponding map of optimum operating parameters for various working fluids against HTF,in temperatures for a 5 kWe ORC.

Fig. 6. a) Comparison of the best cases with and without regenerator for minimum C_{sp} for a 5 kWe R152a ORC, b) C_{sp} against the volumetric flow rate at the expander exhaust.

Fig. 7. C_{sp} and cycle efficiency for various working fluids against HTF,in temperatures for a 50 kWe ORC.

Fig. 8. Corresponding map of optimum operating parameters for various working fluids against HTF,in temperatures for a 50 kWe ORC.

Fig. 9. C_{sp} and cycle efficiency for various working fluids against HTF,in temperatures for a 500 kWe ORC.

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Fig. 10. Corresponding map of optimum operating parameters for various working fluids against HTF,in temperatures for a 500 kWe ORC.

A cost break-down of the lowest C_{sp} ORC configurations is presented in Fig. 11 for the power scales of 5, 50 and 500 kWe. At 5 kWe, instrumentation, DAQ system and controls represents a proportionally high fraction of total system costs, followed by ACC, scroll expander, pumps and HEX. The trend is reversed as the power scale goes up. While the absolute cost of instrumentation remains almost constant, the cost of volumetric machinery (pumps and scroll expander) scales linearly.

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268 Fig. 11. Cost break up for ORCs with lowest specific costs at various HTF operating
 269 temperatures. a) 5 kWe, b) 50 kWe and c) 500 kWe. Pump costs are inclusive of working
 270 fluid, HTF pumps and their variable frequency drives.

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4.1 ORCs connected to solar heat source (Case B)

The initial investment cost for a solar-ORC per unit power produced is presented in Fig. 12 for the different working fluids against HTF_{in} temperature. At lower HTF operating temperatures, these costs are an order of magnitude higher than in ORCs with “free” thermal sources (Case A) but decrease by a factor of 4 across the range of HTF_{in} temperatures, indicating the advantage of moving to higher operating temperatures. The trend among the working fluids is reversed with isopentane offering the lowest C_{sp} values and R161 or R152a the highest. Further, at higher HTF temperatures, there exist two distinct clusters of working fluids, at higher and lower costs levels. This trend can be mapped on the efficiency plot in Fig. 12, emphasizing the fact that lower C_{sp} values are the direct result of higher cycle efficiencies, which in turn minimize the solar field area required and its associated cost. However, even at elevated temperatures (275 °C), investment costs remain high relative to PV at 5 \$/We, which may contribute to a low installation rate for solar ORCs (fewer than 1 % of ORC deployed capacity) [41]. The potential for lower cost energy storage in form of heat and the relatively higher capacity factors achievable in solar-ORCs compared with photovoltaic systems makes for an interesting competitive analysis between solar ORC with TES and PV with batteries in applications that require load-following and dispatch-ability. A detailed application-based thermo-economic comparison for this scenario under variable storage capacity assumptions is outside of the scope of this work, but is highlighted as a useful research direction for future component level cost optimization.

Fig. 12. C_{sp} and cycle efficiency for various working fluids against HTF,in temperatures for a 5 kWe solar-ORC.

The corresponding map of optimum operating parameters is given in Fig. 13. In solar-ORCs, this parameter space adjusts itself to maximize the cycle efficiency and hence, pinch temperatures in condenser, regenerator and HEX tend to be minimized and ORCs tend to be regenerative. This is in contrary to the trend observed for the ORCs coupled to waste heat sources. Similar exercises have been repeated for different power scales and are presented in Figs. 14-17.

Fig. 13. Corresponding map of optimum operating parameters for various working fluids against HTF,in temperatures for a 5 kWe solar-ORC.

Fig. 14. C_{sp} and cycle efficiency for various working fluids against HTF,in temperatures for a 50 kWe solar-ORC.

Fig. 15. Corresponding map of optimum operating parameters for various working fluids against HTF,in temperatures for a 50 kWe solar-ORC.

Fig. 16. C_{sp} and cycle efficiency for various working fluids against HTF,in temperatures for a 500 kWe solar-ORC.

Fig. 17. Corresponding map of optimum operating parameters for various working fluids against HTF,in temperatures for a 500 kWe solar-ORC.

Cost break-down for solar-ORCs is plotted in Fig. 18 which indicates the solar field cost to be highest followed by the power block and then storage system. Potential for further decrease in specific cost lies in the realization of higher temperature ORCs.

Fig. 18. Cost break down of solar-ORCs with lowest specific costs at various HTF operating temperatures. Cases for 5 and 50 kWe (Figs. 18 a) and b) respectively) are shown and 500 kWe are omitted due to their close visual similarity. Fluid machinery includes pumps and expander, and heat exchangers include regenerator, HEX and condenser.

5. Conclusions

The variable space formed by HTF supply temperature, pinch in condenser, regenerator and boiler, expander inlet pressure, condenser area and working fluid affects the ORC investment cost in a significant way. A tool developed to analyze the design space is applied for two major applications of ORCs, namely waste heat and solar. The conclusions drawn from these two studies are well distinguished and summarized as follows:

ORC coupled to waste heat source

- i. At any given temperature, R152a and R161 are found to yield the lowest investment cost. This cost pattern is attributed to their higher condenser pressures (10-15 bar). High condenser pressures in turn imply lower volumetric flow rates at the expander exhaust (state of maximum volumetric flow rate) and result in compact fluid machinery.
- ii. Some smaller ORCs may optimize their specific costs without incorporating a regenerator. Instrumentation costs are proportionally higher for smaller ORCs.
- iii. Increase in HTF supply temperature does not yield a significant decrease in the lowest specific cost indicating diminishing returns with increased operating temperature.

ORC coupled to solar heat source

- i. Isopentane emerges as the best working fluid in solar ORCs at low HTF temperatures. However, at elevated temperatures, isopentane, R245fa and R245ca costs converge.
- ii. The regenerator cost is justified even at lower temperatures and lower power scales owing to higher cycle efficiencies and lower solar field costs.

- iii. The lowest cost of power generation at 5, 50 and 500 kWe are observed to be 5.9, 4.72 and 4.71 \$/We, respectively indicating asymptotic behavior of cost beyond 50 kWe.
- iv. Storage costs (2.73 hours of storage, theoretically, ensuring a capacity factor of 50 %) associated with a TES system designed for damping the solar insolation fluctuations on the power block side are found to be ~ 75 % of the power block. This cost is justified as it ensures a reliable power supply with higher capacity factors.
- v. Solar field cost remains a major contributing factor towards the total solar-ORC cost, thus providing motivation to search for high temperature and hence more efficient ORCs requiring lower solar field area.

A general finding from this research is that minimizing the volumetric flow rate at the expander exhaust correlates with cost savings in the power block, while the detailed application-oriented results of thermo-economic comparison of fluids and temperature regimes provide a framework that can facilitate the engineering design and operation of cost effective ORCs.

6. Future work

Future work involves the evaluation of the levelized cost of energy and appropriate business strategies for target ORC applications with comparative analysis of strategic alternatives in the market. In the light of the fact that many of the pure organic fluids are prone to either high GWP or issues like flammability, this analysis will be extended to the ORCs using mixtures of hydrocarbon and hydrofluorocarbon that exhibit low volumetric flow rate at the expander exhaust and are non-flammable and low GWP working fluids.

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510

511 APPENDIX A

512 Cost functions (in USD) used in the thermo-economic analysis are listed below. Some
 513 of these cost functions are generated using the quotations from vendors of the specific
 514 components. Since most of the quotations received were in the USD, it is chosen as a default
 515 currency to represent the results. An exchange rate of 60 INR against 1 USD can be used for
 516 interpreting the results in INR.

517

$$518 \quad cost_{PHE} = 492.76 + 609.6A_{heat-transfer} \quad (A.1)$$

$$519 \quad cost_{ACC} = NPU(1000A_{fan} + 173.3) \quad (A.2)$$

$$520 \quad cost_{scroll} = (266.2 + 170976VFR_{outlet}) \left(0.6858 + 0.2732 \ln \left(\frac{VFR_{outlet}}{VFR_{inlet}} \right) \right) \quad (A.3)$$

$$521 \quad cost_{tank} = 130V + 1140 \quad (A.4)$$

$$522 \quad cost_{pump} = 19.7VFR_{inlet} \quad (A.5)$$

$$523 \quad cost_{VFD} = 93.6\dot{W} + 110.1 \quad (A.6)$$

$$524 \quad cost_{alternator} = 56.5\dot{W} + 8.9 \quad (A.7)$$

$$525 \quad cost_{solar-field-per-sq.meter} = 245 \quad (A.8)$$

526

527 Cost per kg of the working fluids and HTFs considered is as follows:

$$528 \quad cost_{isopentane} = 2.85 \quad (A.9)$$

$$529 \quad cost_{R152a} = 3.51 \quad (A.10)$$

$$530 \quad cost_{R134a} = 5.85 \quad (A.11)$$

$$531 \quad cost_{isobutane} = 4.68 \quad (A.12)$$

$$532 \quad cost_{R245ca} = 13.6 \quad (A.13)$$

$$533 \quad cost_{butane} = 3.4 \quad (A.14)$$

$$534 \quad cost_{R245fa} = 12.4 \quad (A.15)$$

$$535 \quad cost_{R161} = 12.36 \quad (A.16)$$

$$536 \quad cost_{R236ea} = 19.6 \quad (A.17)$$

$$537 \quad cost_{R236fa} = 7.5 \quad (A.18)$$

$$538 \quad cost_{R218} = 17 \quad (A.19)$$

$$539 \quad cost_{R227ea} = 11.2 \quad (A.20)$$

$$540 \quad cost_{R1234ze} = 26 \quad (A.21)$$

$$541 \quad cost_{R1234yf} = 4.5 \quad (A.22)$$

$$542 \quad cost_{R143a} = 4.0 \quad (A.23)$$

$$543 \quad cost_{R125} = 3.0 \quad (A.24)$$

$$544 \quad cost_{R32} = 2.5 \quad (A.25)$$

$$545 \quad cost_{RC318} = 9.0 \quad (A.26)$$

$$546 \quad cost_{propane} = 3.0 \quad (A.27)$$

$$547 \quad cost_{EG,per,kg} = 1.6 \quad (A.28)$$

$$548 \quad cost_{Therminol-VP1,per,kg} = 5.8 \quad (A.29)$$

549 Cost of instrumentation is as follows:

$$550 \quad cost_{pressure-sensor} = 200 \quad (A.30)$$

$$551 \quad cost_{temperature-gauge} = 50 \quad (A.31)$$

Table A.2. Working fluids considered in the analysis

Working fluids	$T_{critical}$ (°C)	$P_{critical}$ (bar)	$P_{saturation}$ (bar) at 45 °C
isopentane	187.2	33.78	1.76
R245ca	174.42	39.25	2.05
R245fa	154.01	36.51	2.94
R236ea	139.29	35.02	3.93
butane	151.98	37.76	4.34
R236fa	124.92	32	5.06
RC318	115.2	27.77	5.65
isobutane	134.6	36.3	6.04
R227ea	101.75	29.25	8.04
R1234ze	109.37	36.3	8.76
R152a	113.26	45.16	10.36
R134a	101.06	40.6	11.59
R1234yf	94.7	33.8	11.53
R218	71.87	26.4	14.38
R161	102.15	50.91	15.42
propane	96.7	42.5	15.34
R143a	72.7	37.6	20.58
R125	66.023	36.2	22.60
R32	78.105	57.82	27.94